Muskeget Channel tidal energy test facility

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MUSKEGET CHANNEL TIDAL ENERGY TEST FACILITY

BY

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BS, Mechanical Engineering, Cedarville University, 2009

THESIS

Submitted to the University of New Hampshire
in Partial Fulfillment of
the Requirements for the Degree of

Masters of Science
in
Ocean Engineering

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ABSTRACT

MUSKEGET CHANNEL TIDAL ENERGY TEST FACILITY

by

Toby Dewhurst

University of New Hampshire, May, 2013

Conceptual designs were investigated for a tidal hydrokinetic device test facility at Muskeget Channel, MA. Six platform concepts were investigated for devices of various sizes: A floating platform, a submerged buoyant platform, a fixed bottom-mounted gravity foundation, a telescoping bottom-mounted gravity foundation, a fixed four-pile group foundation, and a two-pile surface-piercing structure that could raise and lower a device. A natural berth option was also considered. Designs for each concept were developed for structural soundness, dynamic response, vibration, scour, corrosion, bio-fouling, electrical connection, monitoring, operating limits, ease of turbine installation and access, and cost. The floating platform and two-pile platform were found to be the most practical. A floating platform would require less installation work and would be easier to remove at the end of its service life, but would need to be towed to port for extreme weather. A two-pile, surface-piercing platform would constitute a more significant infrastructure investment.
CHAPTER 1

INTRODUCTION

Background and Previous Work

Interest in tidal energy has increased significantly in the past decade. The desire to extract power from the tides while having minimal environmental impact has led many developers to pursue hydrokinetic turbines. These devices operate in high currents at low pressure head, much like wind turbines operate in air. Thus, they allow for energy generation without the need for dams or other high-impact infrastructure.

The majority of hydrokinetic technologies are still under development, and new concepts are continually emerging (Musial, 2008). These technologies must be tested as they are developed, but deploying devices in the ocean is expensive and extremely time consuming (Sterne et al., 2008). Therefore, accessible and cost-effective test sites are necessary for the industry to grow. However, very few facilities of this type exist.

The European Marine Energy Centre (EMEC) in Scotland’s Orkney Islands is the only facility that has successfully demonstrated itself as a commercial test site for hydrokinetic devices. It has tested numerous devices at its eight tidal test berths—sections of seafloor at depths ranging from 12m (39 ft.) to 50m (164 ft.) with currents up to almost 4 m/s (8 knots) and grid connected power take-off equipment (European Marine Energy Centre Ltd., 2012). It has also begun testing devices at its scaled sites—locations with large anchoring systems provided in maximum currents of 2 m/s (4 knots) in depths of 21 m (69 ft.) to 25 m (82 ft.), which are not connected to the electrical grid. EMEC’s approach to testing has been quite successful but is very
expensive and is not conducive to technologies in the early stages of development. And, of course, the prospect of testing overseas raises a host of logistical challenges for developers in North America.

The Fundy Ocean Research Center for Energy (FORCE), located in the Bay of Fundy, Nova Scotia, employs a test model similar to that of EMEC. It is developing four grid-connected test beds in depths up to 45 m (148 ft.) with maximum velocities approaching 5 m/s (10 knots) and tested its first device during 2009 and 2010 (Fundy Ocean Research Center for Energy, 2012). Its goal is to provide the “ultimate test” for tidal developers who have already demonstrated their technology at milder sites and are ready to prove their devices in the harsh conditions of the Bay of Fundy.

In the United States, test options are extremely limited. One test site is under development by the Northwest Marine Renewable Energy Center in Snohomish County, WA (University of Washington, 2011). The proposed site would test devices in depths of 20 m (66 ft.) to 50 m (164 ft.), with currents reaching 2.5 m/s (5 knots) (Polagye, 2010). The University of Florida is also developing a test location for hydrokinetic devices, although in the Gulf Stream rather than in tidal currents (Mueller et al., 2009). Neither of these sites was operational when this document was written.

The University of New Hampshire (UNH) Center for Ocean Renewable Energy (CORE) has successfully tested multiple hydrokinetic devices in a tidal estuary site, shown in Figure 1, which has currents that reach a maximum of 2.5 m/s (5 knots) in a depth of 8 m (24 ft.) at mean lower low water.
To date, three turbines—one 1 m by 1.25 m (3 ft. by 4 ft.) cross-flow axis device, one 1 m by 2.5 m (3 ft. by 8 ft.) cross-flow axis device, and one 0.9 m (3 ft.) diameter in-stream axis device—have been deployed from a moored 10.7 m (35 ft.) floating platform, as described by Dutile et al. (2009), Wosnik et al., (2009) and Rowell (2013). A larger floating platform is under development which will be capable of testing turbines up to the sizes shown in Table 1. Larger turbines cannot be reasonably tested at this site because of the limited depth of the channel.

<table>
<thead>
<tr>
<th>Turbine type</th>
<th>Height</th>
<th>Width</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ducted in-stream axis</td>
<td>4 m</td>
<td>4 m (13 ft.)</td>
</tr>
<tr>
<td>Vertical cross-flow axis</td>
<td>3 m</td>
<td>2 m (7 ft.)</td>
</tr>
<tr>
<td>Horizontal cross-flow axis</td>
<td>3 m</td>
<td>5 m (16 ft.)</td>
</tr>
</tbody>
</table>

The need for an accessible test site for tidal energy technologies in the U.S. has led the North East Marine Renewable Energy Center (NE-MREC) to investigate the Muskeget Channel near Edgartown, MA, shown in Figure 2.
Figure 2. Muskeget Channel Tidal Energy Test Site. Inset: Contours of detailed bathymetry data taken by Howes et al. are overlaid on a nautical chart. The proposed test site lies in 100 ft. (30 m) of water. Images from Google, NOAA, EarthNC, Howes et al. (2009), Harris, Miller, Miller, and Hanson (2010).

The site is also being considered for a commercial tidal energy plant, which provides a unique opportunity for sharing the costs of permitting, site investigation, cabling, and monitoring. A preliminary permit was obtained from the Federal Energy Regulatory Commission (FERC) and a careful oceanographic, environmental, and logistical investigation of the site is ongoing (Barrett, 2010). Studies include Howes et al. (2009), Coastal Systems Program, University of Massachusetts-Dartmouth (2011), and Schlezinger (2012). It has been found that this site experiences maximum velocities of about 2.5 m/s (5 knots), with depths up to 43 m (143 ft.). Thus, this facility would complement the UNH CORE Tidal Energy Test Site by having more depth to accommodate larger full scale systems in an exposed ocean environment. In this sense, the UNH CORE site could serve as a "nursery site" for testing turbines of limited size in a sheltered environment, and the Muskeget Channel site could provide a full-scale test site as the next step in the scale-up process.
Objectives

The goal of this work was to develop a conceptual design for a test facility at Muskeget Channel for the testing of tidal hydrokinetic devices. The specific objectives for each design alternative considered were to:

- Identify design alternatives using different mounting structure approaches.
- Establish fundamental dimensions required for testing turbines of the desired sizes and identify suitable materials and equipment.
- Perform basic engineering calculations to demonstrate functionality.
- Estimate construction and installation costs.
- Compare alternatives and select the most suitable option(s).

Approach

Design criteria were formulated based on the expected needs of turbine developers. In this connection, a range of maximum size turbines to be tested was identified and the loading forces associated with each size were determined. Six design alternatives were generated and basic engineering calculations completed for each alternative. Designs include provision for mounting vertical and horizontal cross-flow axis turbines, as well as turbines with axes parallel to the flow. Costs for fabrication and installation of each concept for each maximum turbine size were estimated. Features of the natural berth concept were also documented. The positive and negative aspects of the concepts are discussed along with considerations regarding development of the site and long-term sustainability. A recommendation is made regarding the best approach for facility infrastructure.
Design Criteria

Site

The specific test site within the Muskeget Channel lies in SMAST Survey Transect 6 (shown on Figure 3), whose velocity cross-section is shown at its spring tide maximum in Figure 4 (Howes et al., 2009). It should be noted that the peak velocities occur near the surface.

Figure 3. Bathymetry of Muskeget Channel. Contours of detailed bathymetry data taken by Howes et al. are overlaid on a nautical chart. The survey track of SMAST transect 6 is shown in red. The proposed test site lies in 100 ft. (30 m) of water. From Harris, Miller, Miller, and Hanson (2010).

Figure 4. Example velocity profile along Transect 6 at the proposed Muskeget Channel test site at maximum flood tide (Howes et al., 2009). Maximum velocities occur near the surface.
This site has a similar maximum velocity environment as the UNH CORE site, but more than 3 times the depth, as compared in Table 2.

<table>
<thead>
<tr>
<th></th>
<th>Muskeget Channel</th>
<th>UNH CORE Site</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water Depth</td>
<td>30 m 100 ft.</td>
<td>8 m 26 ft.</td>
</tr>
<tr>
<td>Max. Current</td>
<td>2.5 m/s 5 kts</td>
<td>2.5 m/s 5.0 kts</td>
</tr>
<tr>
<td>Min. Height from Seafloor</td>
<td>15 m 62 ft.</td>
<td>~</td>
</tr>
</tbody>
</table>

**Devices**

The practical and financial feasibility of test platform concepts were investigated for testing of turbines with maximum diameters from 4.4 m (14 ft.) to 17.5 m (57 ft.). This would allow the Muskeget Channel platform to accommodate turbines up to about U.S. Department of Energy Technology Readiness Level (DOE TRL) 8 and the U.K. Department of Energy and Climate Change (DECC) Stage 4. (See Table 3 and Table 4.)

<table>
<thead>
<tr>
<th>Tidal-current Protocol</th>
<th>Protocol Stage</th>
</tr>
</thead>
<tbody>
<tr>
<td>Production</td>
<td></td>
</tr>
<tr>
<td>Commercial demonstrator tested at sea for an extended period.</td>
<td>5</td>
</tr>
<tr>
<td>(Scope of Protocol ends here)</td>
<td></td>
</tr>
<tr>
<td>Full-scale prototype tested at sea</td>
<td>4</td>
</tr>
<tr>
<td>Subsystem testing at large scale</td>
<td>3</td>
</tr>
<tr>
<td>Subsystem testing at intermediate scale. Computational Fluid Dynamics. Finite Element Analysis. Dynamic analysis.</td>
<td>2</td>
</tr>
<tr>
<td>Tidal-current energy conversion concept formulated</td>
<td>1</td>
</tr>
<tr>
<td>(Scope of Protocol begins here)</td>
<td></td>
</tr>
</tbody>
</table>

Four different turbine sizes were considered, and engineering analysis and costing were completed for the corresponding four sizes of each design alternative.

The smallest maximum turbine size corresponded to the largest size that can be tested at the UNH CORE Tidal Energy Test Site. The other three had length scales 2, 3, and 4 times larger. This resulted in the largest maximum size having a diameter over half the Muskeget Channel depth, as detailed below.

The test platform needs to accommodate several types of turbine. Since the Muskeget site would complement the UNH CORE site, design criteria for the smallest maximum turbine size were chosen to correspond to the maximum size turbines that could be tested at the UNH CORE site. The weights and drag forces of these turbines are shown in Table 5.

Table 5. Turbine specifications for the smallest maximum turbine size considered. Drag forces shown are for the design flow speed of 2.5 m/s (5 knots).

<table>
<thead>
<tr>
<th>Area</th>
<th>Weight</th>
<th>$C_d$</th>
<th>Turbine Drag</th>
<th>Total Design Drag</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ducted, in-stream axis</td>
<td>12.6</td>
<td>135.3</td>
<td>~ 31,138</td>
<td>62,275 14,000</td>
</tr>
<tr>
<td>Vertical cross-flow axis</td>
<td>6.0</td>
<td>435</td>
<td>960 0.9</td>
<td>48,418 10,885</td>
</tr>
<tr>
<td>Horizontal cross-flow axis</td>
<td>15.0</td>
<td>2300</td>
<td>5100 0.9</td>
<td>74,338 16,712</td>
</tr>
</tbody>
</table>

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The drag force on the cross-flow axis turbines was taken to be

\[ D = \frac{1}{2} \rho C_d A V^2. \]  

(1)

Here \( \rho \) is the density of seawater, \( A \) is the projected area of the turbine, \( V \) is the fluid velocity, and \( C_d \) is the turbine drag coefficient. Here, a value for cross-flow axis turbines was used, as acquired from tow-tank testing by Bachant (2010). Other device types might have much higher drag coefficients. In these cases, the maximum allowable turbine size would be smaller than shown in Table 6. Weights were estimated to be proportional to volume, scaled from an existing 45 kg (100 lb.) helical turbine and doubled to allow for the weight of support structure. Drag on the support structure was estimated from tow tank testing of a Froude-scaled model of the UNH CORE test site platform conducted by Byrne (2013). It was found that the drag on a structure capable of supporting any of the turbines in Table 5 was 31 kN (7,000 lbf.) at 2.5 m/s (5 knots). It was assumed that the size of this support structure would scale with the size of the turbine. Thus, since the drag force on the structure is proportional to its projected area, this force was taken to be proportional to the projected area of the largest turbine to be tested.

Of the turbines listed in Table 5, the horizontal axis helical turbine represents the greatest size, weight, and drag force. For convenience, an in-stream axis turbine with weight and drag characteristics equal to those of the horizontal axis turbine was chosen as the design device for each scale. The design criteria for each possible maximum turbine size are shown in Table 6, along with a scale-up factor that is the ratio of each turbine size to the maximum size that could be tested at the UNH CORE site.
Table 6. Parameters of representative in-stream axis turbine for each possible maximum turbine size.

<table>
<thead>
<tr>
<th>Diameter</th>
<th>Diameter/Depth</th>
<th>Mass, kg</th>
<th>Weight, lb.</th>
<th>Drag</th>
<th>Scale-up factor from UNH CORE site</th>
</tr>
</thead>
<tbody>
<tr>
<td>m</td>
<td>ft.</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>4.4</td>
<td>14 1/7</td>
<td>2,300</td>
<td>5,100</td>
<td>74,000</td>
<td>17,000</td>
</tr>
<tr>
<td>8.7</td>
<td>29 1/3</td>
<td>14,200</td>
<td>31,000</td>
<td>297,000</td>
<td>67,000</td>
</tr>
<tr>
<td>13.1</td>
<td>43 2/5</td>
<td>43,000</td>
<td>94,800</td>
<td>669,000</td>
<td>150,000</td>
</tr>
<tr>
<td>17.5</td>
<td>57 3/5</td>
<td>96,000</td>
<td>211,700</td>
<td>1,189,000</td>
<td>267,000</td>
</tr>
</tbody>
</table>

**Design Alternatives**

Six platform concepts were investigated for testing hydrokinetic turbines of the specified parameters:

1. A floating platform. The platform will consist of a catamaran-type hull-deck structure with a deck opening to lower and raise test turbines. The platform will be moored on station during testing and be towed to a shore base during storms and between test programs.

2. A submerged buoyant platform. The submerged platform will be held in place using a flexible mooring system. The platform may be brought to the surface for mounting and recovering test turbines.

3. A gravity foundation fixed at mid-depth. A large concrete block supports a framework for attaching turbines at mid-depth.

4. A gravity foundation with telescoping piles. The extendable framework allows changing turbines at the surface, while the test position is at mid-depth.

5. A four-pile foundation fixed at mid-depth. A mid-depth platform on top of four piles serves as a permanent base for mounting turbines.

6. A two-pile, surface piercing pile foundation. A horizontal platform between two vertical piles can be moved vertically. Testing is normally done at mid-depth, while attaching and removing test turbines is done at the surface.
A Natural Berth option was considered in addition to these platform concepts. This option would supply the developer with a section of seafloor on which to install a turbine. All options would include instrumentation and power take-off.
A floating platform, illustrated in Figure 5, was considered for the following advantages:

- The platform could be towed to harbor for repair, maintenance, and turbine operations, and also in the threat of extreme storms, etc.
- Turbines would be tested in the high-velocity region near the surface.

Disadvantages include the following:

- The rough seas in the Muskeget Channel are adverse to a moored surface platform.
- Marine traffic would need to avoid the surface presence.
- A surface presence could raise objections over alterations to the existing viewscape.
- The surface structure could become a target for vandalism.
Specific Design Criteria

- The Muskeget platform must not tip more than 1° under steady-state design loading, to maintain adequate freeboard at the up-current end.
- The platform must not allow more than one water-deck contact event per hour when operating in waves.
- Accelerations must remain below normal thresholds for crew operations.

Initial cost estimates for a floating platform were obtained by scaling up existing plans for the UNH CORE Tidal Energy Site platform (Byrne, 2013). Scaling and costing were conducted under the following assumptions:

- Critical forces are buoyancy, weight, and drag. Drag is in an asymptotic range due to high Reynolds number. Thus, Froude scaling is applicable.
- Cost is proportional to the weight of material, and that is proportional to the volume of the body.

Platform Hydrostatics

Governing Equations

The forces and dimensions relative to the hydrostatic analysis governing the surface platform are shown in Figure 6, and the associated variables are explained in Table 7.
Figure 6. Floating platform Free Body Diagram. Current is from right to left. Two mooring lines are used, each attached to the outside of the platform.

Table 7. Floating Platform hydrostatics variables.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$r_t$</td>
<td>Turbine Hub from Surface</td>
</tr>
<tr>
<td>$\theta$</td>
<td>Bow-down Angle</td>
</tr>
<tr>
<td>$W_p$</td>
<td>Platform Weight</td>
</tr>
<tr>
<td>$W_t$</td>
<td>Turbine Weight</td>
</tr>
<tr>
<td>$D_p$</td>
<td>Platform Drag</td>
</tr>
<tr>
<td>$D_t$</td>
<td>Turbine Drag</td>
</tr>
<tr>
<td>$r_i$</td>
<td>Distance from CG to Turbine Drag</td>
</tr>
<tr>
<td>$r_i$</td>
<td>Distance from CG to Mooring Attachments</td>
</tr>
<tr>
<td>$\beta$</td>
<td>Mooring Line Angle from Vertical</td>
</tr>
<tr>
<td>$T_i$</td>
<td>Tension in a Single Mooring Line</td>
</tr>
<tr>
<td>$B_p$</td>
<td>Buoyant Force</td>
</tr>
<tr>
<td>$d_p$</td>
<td>Platform Draft</td>
</tr>
<tr>
<td>$L$</td>
<td>Platform Length (At waterline)</td>
</tr>
<tr>
<td>$M_R$</td>
<td>Righting Moment</td>
</tr>
</tbody>
</table>

Satisfying Newton’s second law in the horizontal direction relates the mooring line tension to platform drag, turbine drag, and mooring angle, so that

$$2T_i \sin \beta = D_p + D_t. \quad (2)$$

Vertical equilibrium yields the buoyant force in terms of platform and turbine weight, mooring line angle, and mooring line tension,

$$B_p = W_p + W_t + 2T_i \cos \beta. \quad (3)$$

Moment equilibrium about the center of gravity requires the righting moment to balance the turbine drag and mooring line moments, so that

$$-D_t r_t - 2T_i \cos(\beta) r_i + B_p \frac{g}{m} \theta = 0 \quad (4)$$
where $FB\bar{m}\theta$ is the righting moment when tipped a small angle, $\theta$. Furthermore, from submerged volume considerations, the metacenter height, $\bar{m}$, is given by

$$\bar{m} \approx \frac{l}{V} = \frac{(2w_h)L^3}{12d_p(2w_h)L} = \frac{L^2}{12d_p} \quad (5)$$

where $w_h$ is the width of each platform hull.

**Design Process**

The drag force and weight of the turbine and support structure were taken from Table 6. The platform dimensions were taken to be proportional to those of the UNH CORE Tidal Energy Test Site 64 ft. (19.5 m) platform. Thus, since drag is proportional to projected area,

$$D_p = D_{po} \left( \frac{L}{L_0} \right)^2 \quad (6)$$

where $L_0 = 64$ ft. (19.5 m) and $D_{po}$ is the drag on the 64 ft. platform at the design current speed of 2.5 m/s (5 knots), as determined from tow tank testing of a Froude scaled model by Byrne (2013). Similarly, the weight of the platform was taken to be

$$W_p = W_{po} \left( \frac{L}{L_0} \right)^3 \quad (7)$$

where $W_{po}$ is the weight of the 64 ft. platform, equal to 88,000 lbf (390 kN).

A mooring line length-to-water-depth ratio (scope) of 7:1 was used, which is standard for use with embedment anchors. The mooring line was assumed to be straight in all cases, so the 7:1 scope results in a mooring angle of $\beta = 82$ degrees (1.4 radians).

For each maximum turbine size to be tested, a maximum allowable draft for the platform hulls was chosen to be proportional to that of the 64 ft. UNH CORE Tidal Energy Test Site platform, so that
\[ d_p = d_{p0} \left( \frac{L}{L_0} \right) \] (8)

where \( d_{p0} \) is the draft of the 64 ft. platform, equal to 1.4 ft. (0.4 m).

The distance between the platform's center of gravity and the mooring line attachment points, \( r_1 \), was taken to be proportional to the size of the maximum turbine size to be tested. These values are shown in Table 8.

For each maximum turbine size, the equations above were solved iteratively—using a Generalized Reduction Gradient (GRG) nonlinear forward difference solver in the Microsoft Excel® Solver package—to find the platform length, \( L \), that resulted in a tipping angle of 1 degree (0.17 radians) at the design current speed of 2.5 m/s (5 knots). The design inputs and results of this analysis are shown in Table 8 for each maximum turbine size investigated.

<table>
<thead>
<tr>
<th>Table 8. Floating Platform Parameters.</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Symbol</strong></td>
</tr>
<tr>
<td><strong>Turbine Size</strong></td>
</tr>
<tr>
<td>Platform Weight</td>
</tr>
<tr>
<td>Distance from CG to Turbine Drag</td>
</tr>
<tr>
<td>Distance from CG to Mooring Attachments</td>
</tr>
<tr>
<td>Mooring Scope</td>
</tr>
<tr>
<td>Tension in Single Mooring Line</td>
</tr>
<tr>
<td>Platform Draft</td>
</tr>
<tr>
<td>Tipping Angle</td>
</tr>
<tr>
<td>Platform Length (at waterline)</td>
</tr>
<tr>
<td>Total Platform Width</td>
</tr>
<tr>
<td>Symbol</td>
</tr>
<tr>
<td>--------</td>
</tr>
<tr>
<td>Platform Weight $W_p$</td>
</tr>
<tr>
<td>Distance from CG to Turbine $r_t$</td>
</tr>
<tr>
<td>Drag</td>
</tr>
<tr>
<td>Distance from CG to Mooring $r_m$</td>
</tr>
<tr>
<td>Attachments</td>
</tr>
<tr>
<td>Mooring Scope</td>
</tr>
<tr>
<td>Tension in Single Mooring Line $T_i$</td>
</tr>
<tr>
<td>Platform Draft $d_p$</td>
</tr>
<tr>
<td>Tipping Angle $\theta$</td>
</tr>
<tr>
<td>Platform Length (at waterline) $L$</td>
</tr>
<tr>
<td>Total Platform Width</td>
</tr>
</tbody>
</table>

**Mooring System**

The platform would be held in place by four mooring lines. Each would connect to an embedment anchor via a length of heavy chain. During each tidal cycle the aft pair of lines would be slack. Thus, the platform would not pivot to match the tidal cycle. However, the moorings would be laid out such that the platform would align with the dominant current direction on both the ebb and flood tides, which are approximately 20 degrees off of a perfect 180 degree alignment (Howes et al., 2009).

Once the mooring line tension was determined for each platform size, a mooring line was chosen that would have a safety factor greater than two for even the largest platform investigated. Plasma 12 strand rope was chosen for its low stretch, low creep, ease of handling, easy splicing, neutral buoyancy in water, and the fact that it does not torque when loaded. Once a suitable mooring line was chosen, studlink chain with a similar breaking strength was selected. The properties of the selected rope and chain are shown in Table 9.
Table 9. Rope and chain specifications.

<table>
<thead>
<tr>
<th>Material</th>
<th>Nominal Diameter</th>
<th>Breaking Strength</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Rope</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Plasma 12 Strand, 28 mm</td>
<td>28 in 1.1</td>
<td>653,900 lbf 296,600</td>
</tr>
<tr>
<td><strong>Chain</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>36mm U3 Stud Link Chain</td>
<td>36 in 1.42</td>
<td>731,826 lbf 332,000</td>
</tr>
</tbody>
</table>

Stingray embedment anchors were chosen for their high ratio of holding power to weight. A safety factor of 5 was required (partially because the holding power of the anchor is specified for sand, and the seafloor in the Muskeget Channel is sand-gravel). As with the mooring line, one anchor size was chosen which would be sufficient for each maximum turbine size to be tested. The properties of Stingray anchors of several sizes are shown in Table 10.

Table 10. Stingray anchor specifications. The selected anchor is shown in bold.

<table>
<thead>
<tr>
<th>Weight</th>
<th>Holding Power in Sand</th>
</tr>
</thead>
<tbody>
<tr>
<td>kg</td>
<td>tonne</td>
</tr>
<tr>
<td>250</td>
<td>30.9</td>
</tr>
<tr>
<td>375</td>
<td>42.6</td>
</tr>
<tr>
<td>500</td>
<td>53.6</td>
</tr>
<tr>
<td>750</td>
<td>74.0</td>
</tr>
<tr>
<td>1000</td>
<td>93.0</td>
</tr>
</tbody>
</table>

**Platform Dynamics**

The initial design criteria for the Floating Platform concept, addressing static stability, were expanded to include the platform’s dynamic behavior in the Muskeget Channel environment. The objective was to design a floating platform with minimal operational downtime due to the sea state.

A coordinate system for this seakeeping analysis is shown in Figure 7. Nomenclature is consistent with that of Faltinsen (1990) and SNAME (1988), which is often employed for seakeeping analysis.
Here $\eta_1$ is the surge displacement (positive in the direction of mean current flow), $\eta_3$ is the heave (vertical) displacement, and $\eta_5$ is the angular displacement in pitch, where a positive value of $\eta_5$ corresponds to a lowered bow.

Traditional seakeeping analysis assumes linear buoyancy, damping and acceleration forces. This allows for a superposition approach which separates the forcing and response into contributors—the platform oscillating in still water and the platform held steady in an incident wave field. Thus, ignoring mooring forces, the equations of motion take the form

$$\sum_{k=1}^{6} [(M_{jk} + A_{jk})\dot{\eta}_k + B_{jk}\eta_k + C_{jk}\eta_k] = F_j \quad (j = 1, ..., 6)$$

(9)

as described by, for example, Faltinsen (1990) and Berteaux (1991). Here $A$, $B$, and $C$ are the added mass, damping (associated with viscous and wave-generating forces from the platform moving in still water), and hydrostatic coefficient matrices, and $F$ is the time-dependent wave forcing on the platform. For the floating platform analysis,
however, the dynamic equations were developed from first principles, leading to less coupling but having nonlinear contributions.

A platform for deploying hydrokinetic devices requires special considerations that require a departure from this standard approach. Specifically, the drag force on the device and support structure, which generally involve bluff body components, are taken to be proportional to the square of relative fluid velocity, i.e.

$$F_D = \frac{1}{2} C_D \rho A U^2$$  \hfill (10)

where $C_D$ is a drag coefficient, $\rho$ is fluid density, $A$ is projected area, and $U$ is the relative fluid velocity. Thus, the drag force is non-linear in $U$. Furthermore, the relationship between platform displacement and mooring line tension is non-linear. Because of these non-linearities, superposition is no longer valid and new equations of motion must be developed. To this end a free-body diagram of the surface platform is shown in Figure 8.

Figure 8. Free-body diagram of Surface Platform with wave and current forcing.

Because of the number of distinct variables necessitated by the complexity of the problem, the following naming convention was used:

- All forces and moments are labeled with a capital "F".
• Capital subscripts refer to the type of force or moment. (These are “W” for weight, “B” for buoyancy, “D” for drag or damping, “A” for fluid acceleration, “T” for tension).

• Lowercase subscripts on forces and moments refer to the bodies on which they act. (These are “p” for the platform, “t” for the turbine, and “l” for the mooring line.) Forces or moments with no lowercase subscript act on the entire system.

• Numerical subscripts indicate the direction in which forces act. (The subscript “5” indicates a moment about the negative y-axis.) These are only shown when necessary for clarity.

Thus, for example, \( F_{D3} \) is the drag/damping force on the total system (platform plus turbine) in the 3- (vertical) direction, \( F_{Ap5} \) is the added mass moment on the platform (in pitch), and so forth. Also shown in Figure 8 are the pontoon diameter, \( Dp \), the platform draft, \( DRp \), the distance from the center of gravity to the turbine’s center of drag, \( r_d \), and the horizontal distance from the center of gravity line of action of the mooring force, \( r_f \).

**Assumptions**

The following assumptions were made when developing this model:

• The hull shape at the waterline changes negligibly during the analysis.

• Interaction between hulls is negligible. (However, some of the hull interaction was captured in the experimentally-derived damping terms in heave and pitch.)

• The effect of forward speed/current on the damping and added mass coefficients can be neglected. According to Smith (1967) and Salvesen et al. (1970), heave and pitch added mass and damping coefficients can show better
agreement with experimental data when forward speed is included in their
determination. However, the current speeds considered here are considerably
slower than typical ship speeds and this effect is negligible.

- The entire hull-turbine structure is rigid.

Equations of Motion

From Figure 8, Newton's second law applied in the horizontal (surge, 1-, and
z-direction) yields the following equation of motion:

$$F_{Dp1} + F_{Dt1} + F_{Ap1} + F_{At} + F_{T1} \cos \phi = \ddot{\eta}_1 m.$$  \hspace{1cm} (11)

Here $F_{Dp1}$ is the drag force on the platform, $F_{Dt1}$ is the force of drag on the turbine,
$F_{Ap1}$ is the fluid acceleration force on the platform, and $F_{At}$ is the fluid acceleration
force on the turbine, and $F_{T1}$ is the mooring line tension. Furthermore, $\ddot{\eta}_1$ is the
acceleration of the platform in surge and $m$ is the mass of the system.

The equation of motion in the vertical (heave, 3-, and z-direction) is

$$F_B - F_W + F_{D3} + F_{A3} - F_{T1} \sin \phi = \ddot{\eta}_3 m.$$  \hspace{1cm} (12)

Here $F_B$ is the buoyant force on the system, $F_W$ is the weight of the system, $F_{D3}$ is the
heave damping force on the system (including wave and viscous damping), and $F_{A3}$ is the
heave added mass force on the system. Furthermore, $\ddot{\eta}_3$ is the acceleration of the
platform in heave.

Consideration of moments about the center of gravity (CG), in the pitch,
negative $y$, and $5$-direction results in

$$-F_{Bp5} + F_{Dp5} + F_{Ap5} + (F_{At} + F_{Dt})r_t + r_f F_{T1} \sin \phi = \ddot{\eta}_5 l_5.$$  \hspace{1cm} (13)

Here $F_{Bp5}$ is the buoyancy moment applied to the platform, $F_{Dp5}$ is the damping
moment applied to the platform, $F_{Ap5}$ is the added mass moment applied to the
platform, \( r \) is the vertical distance from the center of gravity to the turbine hub, \( r_f \) is the horizontal distance from the center of gravity to the mooring attachment point. Furthermore, \( \eta_5 \) is the angular acceleration of the platform in pitch, and \( I_3 \) is the mass moment of inertia of the system about the \( y \)-axis.

**Hydrostatic Forces**

In equilibrium, the hydrostatic force in the heave direction, \( F_B \), is equal to the weight of the system. When the water level changes relative to the system’s center of gravity, C.G., an incremental change in the buoyant force is induced so that the total buoyant force in heave can be written

\[
F_B = mg + \rho g A_{wp} (\zeta_{3eff} - \eta_3),
\]

where \( m \) is the mass of the system, \( \eta_3 \) is the heave displacement of the system, \( A_{wp} \) is the planform area of the platform at the waterline, and \( \zeta_{3eff} \) is the effective surface elevation in heave, found by averaging the surface elevation over the length of the platform, \( L \). For a linear surface wave, this is

\[
\zeta_{3eff} = \frac{1}{L} \int_{-L/2+\eta_1}^{L/2+\eta_1} \frac{H}{2} \cos(kx - \sigma_e t) \, dx
\]

\[
= \sin\left(\frac{H}{2}\right) \frac{H}{2} \cos(k\eta_1 - \sigma_e t),
\]

where \( k = 2\pi/\lambda \) is the wavenumber; \( \lambda \) is wavelength, \( \sigma_e \) is the wave radian encounter frequency (wave frequency modified by advection of the wave field at the current velocity); \( H \) is wave height; and \( \eta_1 \) is platform surge displacement.

The hydrostatic moment on the platform is the sum of the hydrostatic restoring moment and the wave forcing moment. That is,
\[ F_{BBS} = (mg) \bar{m}(-\eta_5) + \int A \rho g \zeta_3(x) x dx, \]  

(16)

where \( \bar{m} \) is the metacentric height, the product \( mg \) is the buoyant force, and \( \eta_5 \) is the angle the platform is tipped, and \( \zeta_3 \) is the surface elevation. (This equation approximates the instantaneous buoyant force with the equilibrium buoyant force, \( mg \).) Using linear wave theory for \( \zeta_3 \), this becomes

\[ F_{BBS} = (mg) \bar{m}(-\eta_5) + \int_{L/2+\eta_1}^{L/2} \rho g \cos(kx + k\eta_1 - \sigma t)xdx. \]  

(17)

Here the planform area of the platform at the waterplane is treated as a rectangle of length \( L \) and constant width \( b \). The result of this integration is

\[ F_{BBS} = (mg) \bar{m}(-\eta_5) + \rho gb \left[ -\frac{L \cos(kx/k^2)}{k^2} + \frac{2 \sin(kx/k^2)}{k^2} \right] \frac{H}{2} \sin(\sigma_0 t - k\eta_1) + \rho gb \left[ \eta_1 L \frac{\sin(kL/2)}{kL} \right] \frac{H}{2} \cos(k\eta_1 - \sigma_0 t). \]  

(18)

**Drag and Damping Forces**

Horizontal forces on the platform and turbine associated with relative fluid velocity are taken to be proportional to the square of velocity (see Equation (10)), in keeping with the standard approach to drag forces. In the vertical (heave) direction, however, velocity-dependent forcing is taken to be proportional to the velocity, as in the traditional wave and viscous damping approach common to basic seakeeping methods. Thus, in the pitch mode of motion the linear and non-linear contributions must be handled carefully, as described below.

The drag forces in surge are drag on the turbine,
\[ F_{Dt} = \frac{1}{2} \rho C_{Dt} A_t U_{hub} |U_{hub}|, \]  

(19)

and drag on the platform,

\[ F_{Dp} = \frac{1}{2} \rho (C_{Dp} A_p + C_{Dmb} A_{mb}) U_{surf} |U_{surf}|. \]  

(20)

Here the \( A_t, A_p, \) and \( A_{mb} \) are the submerged projected areas of the turbine, platform, and mooring ball, respectively. If no mooring ball is used, then \( A_{mb} = 0. \)

Furthermore, the drag coefficients, \( C_{Dt} \) and \( C_{Dp}, \) are determined experimentally. \( C_{Dmb} \) is taken as the drag coefficient of a sphere at the appropriate Reynolds number.

Relative fluid velocity at the hub is

\[ U_{hub} = (U_{cur} + \zeta_1 - \eta_1 - \eta_3); \]  

(21)

at the surface,

\[ U_{surf} = (U_{cur} + \zeta_1 - \eta_1), \]  

(22)

where \( U_{cur} \) is the current velocity and \( \zeta_1 \) is the wave induced velocity, calculated at the appropriate depth using linear wave theory.

In heave, the damping forces are taken to be linear, so those due to the wave velocities and platform velocities can be summed such that

\[ F_{D3} = B^*_{p33} \dot{\eta}_3^{eff} - B_{p33} \dot{\eta}_3 + B^*_{t33} \dot{\eta}_3 - B_{t33} \dot{\eta}_3, \]  

(23)

where \( B_p \) refers to damping coefficients for the platform, and \( B_t \) refers to damping coefficients for the turbine; all were determined experimentally. Here \( B \) refers to damping coefficients for motion in still water and \( B^* \) refers to damping coefficients for fluid motion past the stationary body. In this model the still water damping
coefficient in heave, $B_{33}$, is assumed to be equal to the wave velocity damping coefficient in heave, $B^*_{33}$. This approach is comparable to that taken by Korvin-Kroukovsky and Jacobs (1957) and Salvesen et al. (1970). Also, the vertical wave-induced fluid velocity, $\zeta_3$, is calculated at the turbine hub depth using linear wave theory and $\zeta_{3\text{eff}}$ is the averaged vertical fluid velocity at the surface averaged over the length of the platform. The derivation of this is similar to that of the average surface elevation, and the result is

$$\zeta_{3\text{eff}} = \frac{\sin(h/2)}{h} \zeta_3. \quad (24)$$

In this case, $\zeta_3$ is evaluated at the surface. Since there is no significant wave-induced angular velocity in the fluid (linear wave theory is irrotational), the damping moment applied to the platform in pitch is simply

$$F_{Dps} = B_{pss} \eta_s, \quad (25)$$

where $B_{pss}$ is the damping coefficient in pitch and $\eta_s$ is the angular velocity of the platform. The turbine’s contribution to the drag/damping pitch moment can be simplified by assuming that the pitching of the platform has a negligible effect on the drag/thrust coefficient of the turbine, as demonstrated by Bahaj et al. (2006). Thus, that contribution is

$$F_{Dts} = F_{Dt} r_t, \quad (26)$$

where $r_t$ is the distance from the system’s center of gravity to the turbine hub and, again, $F_{Dt}$ is the drag force on the turbine in surge.
Acceleration Forces

The horizontal force on the platform associated with platform and fluid acceleration, $F_{Ap1}$, is defined as

$$F_{Ap1} = A^*_p \ddot{\xi}_1 - A_{p11} \ddot{\eta}_1,$$  \hspace{1cm} (27)

where, as with $B$, $A$ is the matrix of added mass coefficients for motion in still water; $A^*$ is the added mass coefficient matrix for fluid motion past the stationary body; $\ddot{\xi}_1$ is the horizontal fluid acceleration; $\ddot{\eta}_1$ is the platform acceleration in surge. According to Berteaux (1991), $A$ and $A^*$ are related by

$$A^* = A + m.$$  \hspace{1cm} (28)

The horizontal acceleration force on the turbine takes a similar form to that on the platform. If the turbine is modeled as a flat plate, $m$, the mass of the fluid displaced by the body, is zero, so $A^* = A$ and the force is

$$F_{At} = A_{111} (\ddot{\xi}_1 - \ddot{\eta}_1),$$  \hspace{1cm} (29)

where the added mass is a function of the turbine diameter,

$$A_{111} = \frac{8}{3} \rho \pi L_{Dr}^3.$$  \hspace{1cm} (30)

Here $L_{Dr}$ is the diameter of the turbine. In the vertical direction, the same method can be used. Thus, the acceleration force is

$$F_{A3} = A^*_3 \ddot{\xi}_{3eff} - A_{33} \ddot{\eta}_3.$$  \hspace{1cm} (31)

Here, again, $\ddot{\xi}_3$ is the wave-induced fluid acceleration in the vertical direction, averaged over the length of the platform; $\ddot{\eta}_3$ is the platform acceleration in heave. Furthermore, $A_{33}$ is the added mass of the system when heaving in still water.
Mooring Forces

The mooring system modeled includes an embedment anchor (assumed fixed), a length of heavy chain, and a fiber rope extending to the platform (see Figure 9). A mooring ball can also be included, as shown Figure 10. In this model the primary forces are assumed to be weight and tension. Fluid forces on the line and chain, the weight of the line, and the inertia of the mooring system are neglected. Thus, the line is assumed to always be straight (but elasticity is included) and the mooring forces are governed by the catenary equations.

Figure 9. Catenary mooring system in equilibrium.
Figure 10. Catenary mooring system, with mooring ball, in equilibrium.

For steady current loading (no waves), if some chain lies on the seafloor and the line
is attached to the platform at the waterline, Berteaux (1991) derives the following
governing equations:

\[ T_0 = T_l \cos(\phi), \]  
\[ T_l = T_0 \cosh \left( \frac{pX_r}{T_0} \right), \]  
\[ (\eta_3 - L \sin(\eta_3)) + h + d_{CG} = Y_r + L_{ml} \sin(\phi), \]  
\[ Y_r = \frac{T_0}{p} \left( \cosh \left( \frac{pX_r}{T_0} \right) - 1 \right), \]  
\[ S_{up} = \frac{T_0}{p} \sinh \left( \frac{pX_r}{T_0} \right). \]

Here Equation (34) has been modified to account for platform displacement due to
waves and the vertical location of the center of gravity. The nomenclature used here is
shown in Table 11.
Table 1. Mooring system nomenclature.

<table>
<thead>
<tr>
<th>Name</th>
<th>Description</th>
<th>In steady case:</th>
<th>In unsteady case:</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_0$</td>
<td>Horizontal component of tension in the mooring line</td>
<td>Known</td>
<td>Unknown</td>
</tr>
<tr>
<td>$T_i$</td>
<td>Total tension in the line</td>
<td>Unknown</td>
<td>Unknown</td>
</tr>
<tr>
<td>$\phi$</td>
<td>Angle between the mooring line and the horizontal</td>
<td>Unknown</td>
<td>Unknown</td>
</tr>
<tr>
<td>$p$</td>
<td>Submerged weight of the chain per unit length</td>
<td>Known</td>
<td>Known</td>
</tr>
<tr>
<td>$L_{ml}$</td>
<td>Length of the mooring line</td>
<td>Unknown</td>
<td>Unknown</td>
</tr>
<tr>
<td>$L_{ml0}$</td>
<td>Mooring line length with zero tension</td>
<td>Known</td>
<td>Known</td>
</tr>
<tr>
<td>$L_{chain}$</td>
<td>Length of chain</td>
<td>Known</td>
<td>Known</td>
</tr>
<tr>
<td>$K$</td>
<td>Effective spring constant of the mooring line</td>
<td>Known</td>
<td>Known</td>
</tr>
<tr>
<td>$X_0$</td>
<td>Equilibrium horizontal distance from anchor to platform</td>
<td>Unknown</td>
<td>Known</td>
</tr>
<tr>
<td>$X_r$</td>
<td>Horizontal distance between where the chain leaves the ground and the raised end of the chain</td>
<td>Unknown</td>
<td>Unknown</td>
</tr>
<tr>
<td>$Y_r$</td>
<td>Vertical distance between the seafloor and the end of the chain</td>
<td>Unknown</td>
<td>Unknown</td>
</tr>
<tr>
<td>$h$</td>
<td>Water depth</td>
<td>Known</td>
<td>Known</td>
</tr>
<tr>
<td>$d_{cg}$</td>
<td>Height of the center of gravity from the waterline</td>
<td>Known</td>
<td>Known</td>
</tr>
</tbody>
</table>

In the case which includes a mooring ball, it was assumed that primary force balance is between line tension and drag. Thus, the above equations hold true, but $T_0$ includes the drag on the mooring ball. Furthermore, the mooring tension contribution to the dynamics in heave, as governed by Equation (12), goes to zero.

Allowing for elasticity in the straight mooring line introduces the additional equation,

$$L_{ml} = L_{ml0} + \frac{T_i}{K}.$$  \hspace{1cm} (37)

For the case of steady current and no waves (equilibrium case), the horizontal tension, $T_0$, is known, and the system of Equations (32)—(35) and (37) are sufficient for describing the mooring system. For the unsteady case, that is, with current and waves, $T_0$ is unknown, so another equation is required. From Figure 9 and Equation (35) it is evident that
At equilibrium, this equation can be solved for $X_0$. In the unsteady case, this is the final equation necessary to completely describe the mooring forces. These equations, coupled with the equations of motion in each degree of freedom, can then be solved at each time step.

**Model Implementation**

The equations of motion were solved in the time domain using a "marching solution" approach implemented in a MATLAB program. Initial conditions for the platform's three degrees of freedom and their derivatives (i.e. the initial position and velocity in three degrees of freedom) were specified and forces on the system were calculated at each time step.

**Empirical Constants**

A number of empirically-derived inputs were required for this model, including drag, damping, and added mass coefficients. These could have been obtained in various ways; the following describes how physical experiments were used.

Added mass and damping coefficients for the platform-turbine system in heave and pitch were obtained by measuring the platform's response to an initial perturbation in still water, i.e. a "free-release test."

When no wave, current, or mooring loads are present, Equation (12) simplifies to

\[
\eta_3 \left( m + A_{33} \right) + B_{33} \eta_3 + C_{33} \eta_3 = 0,
\]

which describes a harmonic oscillating system. Here, again, $A_{33}$ is the added mass of
the system in heave, and $B_{33}$ is the system damping in heave. $C_{33}$ is given by

$$C_{33} = \rho g A_{wp},$$

(40)

Where, again, $A_{wp}$ is the planform area of the platform at the waterline. Dividing Equation (39) by $(m + A_{33})$ yields the standard form,

$$\ddot{\eta}_3 + 2\zeta \omega_0 \dot{\eta}_3 + \omega_0^2 \eta_3 = 0,$$

(41)

Where $\omega_0$ is the undamped natural frequency of the system in heave, which is related to the damped natural frequency, $\omega_d$, by

$$\omega_0 = \frac{\omega_d}{\sqrt{1 - \zeta^2}},$$

(42)

where $\zeta$ is the damping ratio, defined as

$$\zeta = \frac{B_{33}}{2\omega_0(m + A_{33})}.$$

(43)

Substituting into Equation (39) yields

$$\omega_0^2 = \frac{C_{33}}{(m + A_{33})},$$

(44)

By measuring the results of a free-release test, the damped natural period, $T_d$, can be observed directly as the time between two local maxima in vertical position. This gives a value for the damped natural frequency according to

$$\omega_d = \frac{2\pi}{T_d}. $$

(45)

The last equation that must be solved is the log-decrement equation, which states that the ratio, $R$, of two consecutive local maxima in position must follow the relationship,

$$R = e^{-\zeta \omega_0 T_d}.$$

(46)

A 1:9 Froude-scaled model of the 35 ft. UNH V1 Tidal Energy Test Platform and the 34 in. FloDesign tidal turbine was constructed, and its response to heave and
pitch perturbations in still water were measured by UNH's Optical Positioning, Instrumentation, and Evaluation (OPIE) system, which is described by Michelin and Stott (1996). Figure 11 shows, from a typical heave free-release test:

- The optically-tracked vertical position of the physical model.
- The locations of (negative) peaks used to find natural period and damping ratio. The averaged time between peaks (or zero crossings) was taken to be the heave natural period and the damping ratio, $\zeta$, was found from the formula

$$\zeta = \frac{\ln\left(\frac{1}{R}\right)}{\omega_o T_d} \quad (47)$$

where $\zeta$ is the damping ratio, $R$ is the ratio of one peak value to the successive peak (of the same sign), $\omega_o$ is the undamped natural frequency, and $T_d$ is the measured period of oscillation.

- The measured damped natural period of oscillation, $T_d$, for the test case shown.
- The time-decaying oscillation amplitude for the test case shown.
- The results of simulating a free-release test in the mathematical model, using the average of the hydrodynamic coefficients computed from each free-release test.
This free-release method was repeated to find the pitch damping and added mass coefficients. Drag coefficients (in the surge direction) for both the platform and turbine were found from full-scale physical experiments. Theoretical added mass coefficients were used for both the platform and the turbine in surge.

Coding

The mathematical model was implemented in a MATLAB® program, shown in Appendix A. What follows is a general description of this program, but is not intended to be a complete user guide.

Platform and turbine parameters must be supplied to the model, and wave and current environment must be specified. Current is input as a scalar and is assumed to
be uniform with depth (a reasonable approximation near the surface). The wave environment can be specified by any one of the following:

- An amplitude and wave frequency for a single wave.
- A range of wave frequencies (with a single amplitude or a range of amplitudes) to be analyzed sequentially. (This is useful for comparing model predictions to wave tank results, and for generating RAOs from single-frequency waves.)
- A significant wave height and a dominant period, in which case a randomized sea having a Bretschneider spectrum, \( S \), will be generated according to the equation

\[
S = \frac{5}{16} \frac{H_{1/3}^2}{f_{pk}} \left( \frac{\sigma_0}{\sigma_{pk}} \right)^{-5} \exp \left( -\frac{5}{4} \left( \frac{\sigma_0}{\sigma_{pk}} \right)^{-4} \right).
\]

where, \( H_{1/3} \) is the significant wave height, \( f_{pk} \) is the peak frequency in Hz, \( \sigma_0 \) is the wave frequency in radians, and \( \sigma_{pk} \) is the peak wave frequency in radians.
- A user-supplied wave spectrum. This is useful for comparing with measured ocean data or predicting performance in an area where the wave spectra have been measured.

If a spectrum is specified, the wave height at each frequency is determined by the equation

\[
H_n = 2\sqrt{2S_n \Delta \sigma_0}.
\]

For each specified wave frequency the wave number, \( k_n \), is computed using the (exact) dispersion relation,
\[ \sigma_{en}^2 = g k_n \tanh k_n h, \]  

(50)

where \( h \) is the water depth. For the case of a steady current with speed \( U_{cur} \), the wave frequency, \( \sigma_n \), is replaced by the frequency of encounter, \( \sigma_e \), such that

\[ \sigma_e = \sigma_o \pm u_{cur} k, \]  

(51)

where the \( \pm \) depends on the wave direction, and is positive when the waves propagate in the direction of the current (head seas) and negative when the waves propagate against the current (following seas).

With the platform and turbine parameters (including drag coefficients and projected areas) provided, the equilibrium drag force on the system (which depends only on the current and includes additional components such as a mooring ball) can be computed. This allows the initial catenary equations to be solved (as described previously) for equilibrium values, using the built-in MATLAB® function, \( \text{fsolve} \).

Since \( \text{fsolve} \) uses an iterative algorithm to solve the set of nonlinear catenary equations, initial guesses for each unknown must be provided. These initial guesses for each unknown are provided as in Table 12.
Table 12. Initial guesses for equilibrium catenary solution.

<table>
<thead>
<tr>
<th>Unknown</th>
<th>Initial Guess</th>
</tr>
</thead>
<tbody>
<tr>
<td>Line tension</td>
<td>$T_i$</td>
</tr>
<tr>
<td>Mooring angle</td>
<td>$\phi_m$</td>
</tr>
<tr>
<td>Horizontal position of chain end (from where the chain leaves the seafloor)</td>
<td>$x_r$</td>
</tr>
<tr>
<td>Vertical position of chain end (from the seafloor)</td>
<td>$y_r$</td>
</tr>
<tr>
<td>Length of mooring line (under load)</td>
<td>$L_{mi}$</td>
</tr>
</tbody>
</table>

Since solving the catenary equations at each time step is computationally expensive, it can be convenient to use a linearized version of the catenary equations. If such a method is desired, the full catenary equations are solved again with modified inputs to obtain force/distance coefficients (or “spring constants”) in each direction. The spring constants are given by

\[ K_{11} = \frac{\Delta T_1}{\Delta \eta_1}, \quad K_{13} = \frac{\Delta T_3}{\Delta \eta_3} \]  
(52)

and

\[ K_{31} = \frac{\Delta T_3}{\Delta \eta_1}, \quad K_{33} = \frac{\Delta T_3}{\Delta \eta_3}. \]  
(53)

Here $\Delta \eta_1$ and $\Delta \eta_3$ are the horizontal and vertical displacements, respectively, and are illustrated in Figure 12. The corresponding changes in the horizontal and vertical components of the mooring line tension are $\Delta T_1$ and $\Delta T_3$, respectively.
Figure 12. Linearized representation of catenary mooring forces. Effective spring constants were found as the ratio of the change in mooring force to the distance the platform was displaced.

The horizontal spring constant is found by replacing the input $T_o$ with $T_o \times \text{mult}$, where $\text{mult}$ is a multiplication factor which produces a horizontal displacement on the order of the wave amplitude. The vertical spring displacement is found by replacing the water depth input, $h$, with $h + H/2$, where $H$ is the wave height. (When using linearized catenary equation in a random sea simulation, $H_{1/3}$ is used.) Using this approach, the horizontal and vertical components of mooring line tension are

$$T_{l_1} = K_{11}\eta_1 + K_{13}\eta_3$$  \hspace{1cm} (54)

and

$$T_{l_3} = K_{31}\eta_1 + K_{33}\eta_3.$$  \hspace{1cm} (55)

It was found that using the linearized catenary equations generally yielded overall results that agreed well with those using exact catenary solutions. However, when mooring loads are of specific interest, the exact catenary equations should be used.

Initial conditions supplied to the solver were generally zero, although some small horizontal position and velocity perturbations were used to help the simulation reach equilibrium more quickly in regular waves. Wave loading is increased linearly from zero to its full value over the first second to avoid discontinuous forces.
**Solver:** At each time step the ordinary differential equation solver *ODE45* calls a user-defined function containing the equations of motion described previously. This function computes the forces on the platform system (as described below). These forces are then used to solve the equations of motion to provide the accelerations at the given time, and these accelerations are returned to the solver, which then computes the velocities (and positions) at the subsequent time step.

For each time step, fluid velocity and acceleration in each direction (horizontal and vertical) is computed at the turbine hub depth for the specified wave using linear wave theory (assuming an undisturbed wave). These accelerations and velocities are then used to calculate the forces on the turbine. If random seas are being simulated then the accelerations and velocities are computed at the turbine hub depth for each wave contribution and the results are summed before being applied to the turbine. The wave loading on the platform is calculated the same way, except that the wave velocities and accelerations are calculated at an approximate “center-of-drag” depth, for example, half of the platform draft.

When an exact catenary solution is used, the mooring forces are calculated at each time step using initial guesses from the previous solution, which reduces solution time. If the platform’s surge position is such that the entire length of mooring chain rests on the seafloor, there is no mooring force applied to the platform. The overall solution process is illustrated in Figure 13.
Inputs:
- Platform, turbine parameters: Geometry (platform shape, cross-sections, and lengths), mass, etc.
- Empirical constants (drag, damping, added mass coefficients)
- Wave, current environment (spectrum or single frequency and amplitude)

Hydrodynamic Coefficients (input) → Mooring forces: Solve catenary equations

ODE45

ICs → Time → Wave forcing → Positions, velocities → Forces on system → Fluid velocity, acceleration

System accelerations → Primary outputs:
System kinematics
Forces on platform, mooring, turbine

Figure 13. High-level flow diagram of model. The equations of motion were implemented in a MATLAB program that calls several subroutines.

Post processing

The mathematical model finds the forces on the turbine-platform system and the system kinematics as time series. Key results that can be computed include platform accelerations, mooring loads, turbine loads, and Response Amplitude Operators (RAOs)—the ratio of platform response to wave forcing in each degree of freedom.
**Platform Accelerations:** Root-mean-squared (RMS) accelerations are important for quantifying the effect of platform motion on crew comfort (International Standards Organization, 1997). Maximum vertical accelerations occur at either the bow or the stern of the vessel, so the model generates a time series of vertical acceleration at both locations by summing the effect of heave and pitch acceleration and calculates the RMS value. This can then be compared to, for example, ISO standards for crew comfort.

**Mooring Loads:** Since the model calculates the mooring load on the platform at each time step, it can be used to (iteratively) design mooring systems. If an exact catenary solution is used then the model also reveals whether the mooring chain remained on the seafloor for the duration of the simulation. (If the chain is lifted off of the floor, the holding power of embedment or deadweight anchors is compromised.)

**Turbine Loads:** The loading on the turbine hub can be used to set operable sea state/current limits. This force is returned as a time series of scalars representing the magnitude of the fluid forcing on the turbine, centered at the hub, in the axial direction. The vertical force on the turbine can be extracted in the same way. (In addition, since the power generated by the device is a function of the relative fluid velocity, this model could also be used to estimate the effect of platform and wave motion on turbine performance.)
**Response Amplitude Operators**: The kinematics of the platform can be represented by Response Amplitude Operators (RAOs), defined in general as the ratio of platform response amplitude to wave forcing amplitude in each degree of freedom over a range of frequencies (Tupper, 2004). The definitions used to compute the RAOs using single-frequency waves and random seas are shown in Table 13.

<table>
<thead>
<tr>
<th>Mode of motion</th>
<th>Wave type</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>Surge</td>
<td>Single-frequency</td>
<td>( \frac{\eta_{1\text{max}} - \eta_{1\text{min}}}{H} )</td>
</tr>
<tr>
<td></td>
<td>Random</td>
<td>( \frac{\sqrt{S_{\text{platform 1}}}}{S_{\zeta}} )</td>
</tr>
<tr>
<td>Heave</td>
<td>( \frac{\eta_{3\text{max}} - \eta_{3\text{min}}}{H} )</td>
<td></td>
</tr>
<tr>
<td></td>
<td>( \frac{\sqrt{S_{\text{platform 3}}}}{S_{\zeta}} )</td>
<td></td>
</tr>
<tr>
<td>Pitch</td>
<td>( \frac{\eta_{5\text{max}} - \eta_{5\text{min}}}{Hk} )</td>
<td></td>
</tr>
<tr>
<td></td>
<td>( \frac{\sqrt{S_{\text{platform 5}}}}{k^2S_{\zeta}} )</td>
<td></td>
</tr>
</tbody>
</table>

Here \( \eta_1, \eta_3, \) and \( \eta_5 \) are the surge, heave, and pitch displacements respectively; \( S_{\text{platform 1}}, S_{\text{platform 3}}, \) and \( S_{\text{platform 5}} \) are the spectral energy density of the platform response in the surge, heave, and pitch directions respectively. Furthermore, \( S_{\zeta} \) is the spectral energy density of the surface elevation, \( H \) is the wave height, and \( k \) is the wave number at each frequency.

In certain situations, computing RAOs requires special care. Specifically, in following seas, a problem arises because \( \sigma_e \) is indeterminate in \( \sigma_o \), as is apparent from Equation (51). This problem is discussed in depth by, for example, (Korsmeyer, 1995). When computing RAOs in following seas using the single-frequency approach, the model dealt with this problem by summing the responses to both values of \( \sigma_o \) that contribute to the total response at a given \( \sigma_e \). The phase lag between platform response and wave forcing, \( \epsilon_j \), was found for each contribution \( (j=1 \text{ and } j=2) \) by computing the lagged cross-correlation between the wave forcing and the response for each degree of
freedom. The time lag that resulted in the maximum cross-correlation, \( t_{\text{lag}M} \) is expressed as a phase angle by equation,

\[
\epsilon_j = \frac{t_{\text{lag}Mj}}{T} 2\pi
\]

where \( T \) is the wave period. This approach yielded responses of the form,

\[
C_1 \cos(\sigma_{o1}t + \epsilon_1)
\]

\[
C_2 \cos(\sigma_{o2}t + \epsilon_2)
\]

where \( \sigma_{o1} \) is the first wave frequency which has an encounter frequency of \( \sigma_\phi \) due to advection by the current and \( \sigma_{o2} \) is the second; \( C_j \) is the magnitude of the response to the \( j^{th} \) wave frequency; and \( \epsilon_j \) is the phase lag of the response to the \( j^{th} \) wave frequency. The magnitude of the response at the given encounter frequency, \( D \), is

\[
D = \sqrt{(A_1 + A_2)^2 + (B_1 + B_2)^2}
\]

where

\[
A_j = C_j \cos(\epsilon_j)
\]

\[
B_j = -C_j \sin(\epsilon_j)
\]

**Model Validation**

The model was first run for a range of single frequency waves and compared to wave tank results using a Froude-scaled model physical model of the existing UNH CORE Tidal Energy Test Platform. The model was then further validated using the existing full-scale UNH CORE 35-ft. platform. The mathematical model was used to simulate environmental conditions experienced during a full-scale ocean deployment and the outputs were compared to the measured platform response.
Wave Tank

The mathematical model was used to simulate the system response to single frequency waves ranging from 3 s to 9 s periods (full-scale) and 0.5 m to 4.0 m wave heights (full-scale), using input parameters as described previously. A 1:9 Froude-scaled model of the platform-turbine-mooring system (shown in Figure 14) was tested in the UNH wave tank for the same range of conditions. The mooring system consisted of chain (Froude-scaled by length and weight) connecting a fixed anchor and lightweight, low-stretch monofilament line, which was attached near the bow of the model using a bridle configuration.

![Figure 14. Froude-scaled model of UNH CORE Tidal Energy Test Platform in wave tank.](image)

The response of the physical model to the wave forcing was measured with UNH's Optical Positioning, Instrumentation, and Evaluation (OPIE) system, which is described by Michelin and Stott (1996).

Testing the physical model over a range of single-frequency waves and analyzing the results yielded a preliminary set of RAOs. While the data contained a large amount of scatter, the results confirmed the model's predictions that no resonant response occurs within the range of wave periods (3-9 s) analyzed.

The model was then used to predict the dynamic response of UNH CORE 35-
ft. platform to expected conditions in Massachusetts' Muskeget Channel. Mooring loads, turbine loading, and kinematic response were predicted for a range of sea states and the operable range of the platform-turbine system was established. Table 14 shows some predicted loads on the 35 inch diameter FloDesign turbine under expected Muskeget conditions.

Table 14. Predicting loading on FloDesign turbine at various sea states, with a mean current of 2 m/s (4 knots). Significant wave heights used are mean values as per Faltinsen (1990).

<table>
<thead>
<tr>
<th>Sea State</th>
<th>Significant Wave Height</th>
<th>Predicted Maximum Load on Turbine</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sea state 2</td>
<td>0.30 m (1 ft.)</td>
<td>2500 N (560 lb)</td>
</tr>
<tr>
<td>Sea state 3</td>
<td>0.88 m (2.9 ft.)</td>
<td>3400 N (760 lb)</td>
</tr>
</tbody>
</table>

Ocean Deployment

UNH's 35-foot (10.67 m) tidal energy test platform was used to test a FloDesign hydrokinetic turbine in Muskeget Channel on in July 2012. Details of each day of testing are summarized in Table 15.

Table 15. Muskeget Channel deployment details. Significant wave heights used are mean values as per Faltinsen (1990).

<table>
<thead>
<tr>
<th>Date</th>
<th>Slack Time</th>
<th>Max Current</th>
<th>Tidal Stage</th>
<th>Max. Significant Wave Height</th>
<th>Max. Sea State</th>
</tr>
</thead>
<tbody>
<tr>
<td>July 15</td>
<td>9:15 AM</td>
<td>1.3 m/s (2.6 kts.)</td>
<td>Ebb</td>
<td>0.8 m (2.6 ft.)</td>
<td>3</td>
</tr>
<tr>
<td>July 16</td>
<td>10:00 AM</td>
<td>1.6 m/s (3.1 kts.)</td>
<td>Ebb</td>
<td>1.0 m (3.3 ft.)</td>
<td>3</td>
</tr>
<tr>
<td>July 19</td>
<td>5:45 PM</td>
<td>1.9 m/s (3.8 kts.)</td>
<td>Flood</td>
<td>0.5 m (1.7 ft.)</td>
<td>2</td>
</tr>
</tbody>
</table>

The water depth at the site was approximately 20 m (65 ft.). The platform and the deployment are described in further detail by Rowell (2013). The energetic environment and the instrumentation onboard the platform made for a good full scale validation case.
**Instrumentation:** In the full scale deployment the platform instrumentation included a high-end Inertial Measurement Unit (IMU), a wave staff mounted on the bow of the platform, load cells, and flow sensors, listed in Table 16. Load cells were used to measure the mooring loads on the platform. The wave staff, corrected for platform motion with the IMU, was used to determine the wave forcing spectrum. The IMU was also used to find the platform response spectra, which were used to compute the system Response Amplitude Operators (RAOs).

<table>
<thead>
<tr>
<th>Instrument</th>
<th>Model</th>
<th>Manufacturer</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wave Staff</td>
<td>OSSI-010-002E</td>
<td>Ocean Sensor Systems</td>
</tr>
<tr>
<td>Acoustic Doppler Velocimeter (ADV)</td>
<td>Vector Cable Probe Standard</td>
<td>Nortek USA</td>
</tr>
<tr>
<td>Inertial Motion Unit (IMU)</td>
<td>402225 DMS 3</td>
<td>Teledyne TSS</td>
</tr>
<tr>
<td>Load Cells</td>
<td>SS 20000</td>
<td>Sensing Systems</td>
</tr>
</tbody>
</table>

**Mooring Setup:** On each day of testing the platform was moored on a single-point mooring, shown in Figure 15.

![Figure 15. Mooring system for ocean deployment.](image-url)
The platform was deployed during two ebb tides and one flood, each on a different
day. This mooring system allowed the test platform to align itself with the current,
allowing for a wide radius of potential locations. Thus, the water depth at the platform
location was recorded on each day of testing so that the wave environment could be
modeled accurately. Since the anchor was not reset when the current reversed, it
dragged on several occasions during the third day of testing. Data from those events
were not used for model validation.

**Data Acquisition and Processing:** Platform motion data were recorded continuously
and saved in 10 segments at 10 Hz for the duration of the testing. Wave, current, and
platform data were analyzed in 20 minute segments, so platform motion data was
spliced together for processing. The wave staff recorded a single file for each day of
testing and also sampled at 10 Hz. The Vector ADV also recorded a single file for
each day of testing, sampling at 32 Hz.

The location of the system's Center of Gravity (C.G.) was found using a
detailed Solidworks® model of the platform-turbine system, accounting for the mass
distribution of the crew. Knowing the geometric relation of the IMU to the C.G., the
platform motion at/about the C.G. in each degree of freedom was found as,

\[ \eta_1 = \eta_1^* - R_{IMU} \sin \eta_5, \]  \hspace{1cm} (62)

\[ \eta_3 = \eta_3^* - R_{IMU} \sin \eta_5, \]  \hspace{1cm} (63)

where \( \eta_1^* \) and \( \eta_3^* \) are, respectively, the surge and heave displacement measured at the
IMU and \( R_{IMU} \) are, respectively, the horizontal and vertical components
of vector from the C.G. to the IMU, shown in Figure 16. Since the system is treated as
a rigid body, \( \eta_5 \) needs no correction.
The fluid surface elevation time series was found by correcting the elevation measured by the wave staff with the platform displacement as measured by the IMU. The vertical position of the platform bow (on which the wave staff was mounted) was calculated as

$$\eta_{bow} = \eta_3 + \sin(\eta_5)R_{bow}.$$  \hspace{1cm} (64)

The wave spectrum was then computed as the power spectral density of the corrected surface elevation time series, $\eta_{bow}$, using a Hanning window and using 4 ensembles (5-minute samples) and band averaging over 5 adjacent Fourier frequencies, as described by, for example, Bendat and Piersol (2010).

The power spectral density of the platform motion in each degree of freedom was computed using the same method as for the wave spectrum—using a Hanning window and using 4 ensembles (5-minute samples) and band averaging over 5 adjacent Fourier frequencies. The RAO in each degree of freedom was then computed as shown in Table 13.

After the full-scale ocean deployment, the model was run with environmental
conditions matching those measured at sea using the Wave Staff, Inertial Motion Unit, and the Vector Velocimeter. It should be noted that some of the model's approximations and assumptions were not met exactly during testing. Most significantly, while the model assumes colinearity between waves and current, some wave propagation in the transverse direction was present. However, the resulting platform roll was observed to be small compared to the pitching motion. Predicted and measured RAOs compared quite well, as illustrated in Figure 17 and Figure 18.

Figure 17. Comparison of Normalized Response Amplitude Operators in following seas. Current: 1.5 m/s, Significant Wave Height: 0.82 m. (Ocean Data: Ensembles: 2, Bands: 5)
Figure 18. Comparison of Normalized Response Amplitude Operators in head seas. Current: 1.9 m/s, Significant Wave Height: 0.5 m. (Ocean Data: Ensembles: 2, Bands: 5)

Model Application

The mathematical model was used to simulate the response of a Muskeget test facility platform that was larger than, but proportional to, the UNH CORE 35 ft. platform. Thus, the free-release tests on the scale model of the UNH platform could simply be scaled using a new Froude scale factor. The initial length of the platform design was found using hydrostatics, as the minimum length that allowed less than 1 degree tipping at 2.5 m/s current with no waves while deploying a 9 m (29 ft.) turbine (as described in “Floating Platform: Governing Equations-Hydrostatics”). The platform’s operability range was then analyzed using the mathematical model with long-term wave data as described below. Tow design criteria were applied to determine operating limits—loss of crew functionality and wave contact with the platform ends. This was repeated until a design was found that could operate for more than 90% of the days in an average year.
Operating Limits

Wave data was obtained from the Martha’s Vineyard Coastal Observatory (MVCO), roughly six miles west of the southern opening of Muskeget Channel, one mile off of the coast of Martha’s Vineyard in 12 m (39 ft.) water. There are differences between the MVCO site and the Muskeget site (depth, currents, wind patterns, etc.) but this was the most relevant data available at the time of analysis. The observatory calculates wave height spectra for twenty-minute segments by using ADCP instrumentation to measure the fluid velocity and direction near the surface (specifically, at 85% of the distance to the mean free surface, with the full distance calculated from a pressure sensor in the ADCP). That value is then extrapolated to the surface using linear wave theory (Wood's Hole, 2012). These data are plotted in Figure 19 and Figure 20 for a typical winter and summer month, respectively.

![Wave Height Spectra: 2011, Day 1- Day 31](image1)

![Wave Height Spectra: 2011, Day 182- Day 212](image2)

**Figure 19.** Wave height spectra at MVCO, January 2011

**Figure 20.** Wave height spectra at MVCO, July 2011

UNH’s 35-foot (10.67 m) tidal energy test platform was used to test a hydrokinetic turbine in Muskeget Channel on July 15, 16 and 19, 2012 (Dewhurst et al., 2012). During the testing, significant wave height measurements were generally
within 10% of those at the MVCO site. Thus, it was concluded that the historical data from MVCO was sufficiently representative of the wave climate in Muskeget Channel.

The percentage of time during which the platform could operate was found as follows. The historical wave height spectrum data (Figure 17-A) from the Martha’s Vineyard Coastal Observatory was converted to wave heave acceleration spectra using the relationship,

\[ S_a = \sigma_o^4 S_\zeta \]  \hspace{1cm} (65)

where \( S_a \) is the wave acceleration spectra (Figure 17-C), \( \sigma_o \) is the wave frequency in rad/s (Figure 17-B) and \( S_\zeta \) is the wave height spectrum. This results from the relationship of vertical acceleration, \( \ddot{\zeta} \), to elevation at the surface, \( \zeta \), for a linear wave,

\[ \ddot{\zeta} = -\sigma^2 \zeta \]  \hspace{1cm} (66)

The heave Response Amplitude Operator (RAO) for the platform was found using a range of single frequency waves in the mathematical model. Vertical displacement RAOs were found at both the bow and stern of the vessel, and it was found that the RAO at the stern (Figure 17-D) was consistently higher than that at the bow. This is due to the phase relationship between heave and pitch. The wave acceleration spectrum was then multiplied by the square of the stern RAO to find the stern acceleration spectrum, \( S_{\phi \phi} \) (Figure 17-E). This spectrum was numerically integrated over the frequency range to find the variance of the platform acceleration. Finally, the Root-Mean-Squared (RMS) acceleration was found as the square-root of the variance.
Figure 21. Example of the development of the platform acceleration response spectrum beginning with an arbitrarily-selected wave height spectrum. This spectrum was used to compute RMS accelerations (averaged over 20 minute segments) throughout a typical year.

Thus, the maximum RMS acceleration experienced on the platform for any wave height spectrum was found as

\[ a_{\text{RMS}} = \sqrt{\int RAO_{\text{stern}}^2 \sigma^4 S_h \, df}, \]  

(67)

where \( f \) is the wave frequency. This value was calculated for each wave spectrum acquired from the MVCO (20-minute samples) for the year 2011 and compared to a maximum operable RMS acceleration. This limit was taken to be 0.2 g (1.96 m/s\(^2\)), which the International Standards Organization (1997) says is “not tolerable for longer periods” and “quickly causes fatigue” and allows only “light manual work by people adapted to ship motions.” The RMS accelerations that would have been

53
experienced on the platform throughout 2011, in reference to this maximum acceleration limit, are shown in Figure 22. It was found that the accelerations experienced exceeded the limit of 0.2 g for about 1.5% of a typical year.

![Figure 22. RMS accelerations predicted, using 2011 wave data from the MVCO.](image)

In addition to the crew's ability to work on the platform, wave contact and water-on-deck events were also considered. In the mathematical model, a platform design meeting the maximum acceleration criteria was subjected to a range of single frequency waves. The difference between the surface elevation at the bow and the vertical position of the bow was compared to the freeboard of the platform at equilibrium, specified as one tenth the length of the platform. The height of the single-frequency wave in which the freeboard was regularly exceeded was taken to be the maximum significant wave height in which the platform could operate. This maximum significant wave height was 3.4 m (10.4 ft.).

After the maximum single-frequency wave height was found, a similar approach was used with irregular waves. In the mathematical model the design was subjected to a Bretschneider wave spectrum. The significant wave height (and period)
of this spectrum was increased until the frequency of wave contact/water-on-deck events exceeded once per hour. This resulted in a maximum allowable significant wave height of 2.6 m. It was noted in the course of this analysis that the non-linearity in the system’s pitch response makes it particularly vulnerable to storm events. Since this method yielded a lower significant wave height than the single frequency approach, the more conservative value of 2.6 m was used to compute operational limits.

The significant wave height data for the past five years (obtained in 20-minute averages) was examined to calculate the percent time in which wave heights were below the 2.6 m limit. Table 17 shows the percentage of days in each month during which the significant wave height exceeded 2.6 m. These results show that the platform could operate for 90% of the days during a typical year.

<table>
<thead>
<tr>
<th></th>
<th>Jan</th>
<th>Feb</th>
<th>Mar</th>
<th>Apr</th>
<th>May</th>
<th>Jun</th>
<th>Jul</th>
<th>Aug</th>
<th>Sep</th>
<th>Oct</th>
<th>Nov</th>
<th>Dec</th>
<th>Avg.</th>
</tr>
</thead>
<tbody>
<tr>
<td>2007</td>
<td>16%</td>
<td>11%</td>
<td>32%</td>
<td>10%</td>
<td>6%</td>
<td>7%</td>
<td>3%</td>
<td>0%</td>
<td>0%</td>
<td>3%</td>
<td>7%</td>
<td>10%</td>
<td>9%</td>
</tr>
<tr>
<td>2008</td>
<td>19%</td>
<td>32%</td>
<td>19%</td>
<td>27%</td>
<td>0%</td>
<td>0%</td>
<td>0%</td>
<td>0%</td>
<td>3%</td>
<td>13%</td>
<td>17%</td>
<td>35%</td>
<td>13%</td>
</tr>
<tr>
<td>2009</td>
<td>10%</td>
<td>36%</td>
<td>23%</td>
<td>13%</td>
<td>3%</td>
<td>0%</td>
<td>3%</td>
<td>10%</td>
<td>7%</td>
<td>0%</td>
<td>7%</td>
<td>26%</td>
<td>12%</td>
</tr>
<tr>
<td>2010</td>
<td>10%</td>
<td>7%</td>
<td>13%</td>
<td>0%</td>
<td>3%</td>
<td>0%</td>
<td>0%</td>
<td>0%</td>
<td>7%</td>
<td>10%</td>
<td>13%</td>
<td>10%</td>
<td>6%</td>
</tr>
<tr>
<td>2011</td>
<td>3%</td>
<td>18%</td>
<td>6%</td>
<td>23%</td>
<td>0%</td>
<td>0%</td>
<td>0%</td>
<td>6%</td>
<td>3%</td>
<td>0%</td>
<td>0%</td>
<td>19%</td>
<td>7%</td>
</tr>
<tr>
<td>Avg.</td>
<td>12%</td>
<td>21%</td>
<td>19%</td>
<td>15%</td>
<td>2%</td>
<td>1%</td>
<td>1%</td>
<td>3%</td>
<td>4%</td>
<td>5%</td>
<td>9%</td>
<td>20%</td>
<td>9%</td>
</tr>
<tr>
<td>SD.</td>
<td>0.06</td>
<td>0.11</td>
<td>0.09</td>
<td>0.10</td>
<td>0.02</td>
<td>0.03</td>
<td>0.01</td>
<td>0.04</td>
<td>0.03</td>
<td>0.05</td>
<td>0.06</td>
<td>0.10</td>
<td>0.10</td>
</tr>
</tbody>
</table>

It should be noted that non-operating conditions are due to periods of high waves that could reasonably be attributed to major storm events. These could presumably be forecast in advance, allowing the platform to be towed into a safe port.

Final Design

The final iteration of the floating platform was longer than the initial design,
which was based only on hydrostatics. Also, unlike the initial design, cylindrical pontoons are employed. The specifications of a platform capable of deploying a 9 m (29 ft.) turbine for 90% of the days in an average year are shown in Table 18.

<table>
<thead>
<tr>
<th>Table 18. Floating platform specifications for deploying a 9 m (29 ft.) turbine.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pontoon Diameter</td>
</tr>
<tr>
<td>Pontoon Length</td>
</tr>
<tr>
<td>Beam (between centers of pontoons)</td>
</tr>
<tr>
<td>Freeboard (at equilibrium)</td>
</tr>
<tr>
<td>Total (estimated) Mass of Structure</td>
</tr>
<tr>
<td>Draft</td>
</tr>
<tr>
<td>Chain Diameter</td>
</tr>
<tr>
<td>Total Chain Length</td>
</tr>
<tr>
<td>Total Line Length</td>
</tr>
</tbody>
</table>

Legend:
- Iterated values
- Design inputs
- Calculated values

**Costing**

The material and fabrication costs for the floating platform were estimated by prorating quotes obtained for the UNH CORE V2 platform (Byrne, 2013). Quotes for hulls fabricated from A36 steel and coated with marine-grade epoxy were scaled by the cube of the length ratio (the length of the Muskeget platform divided by the length of the V2 platform). The same was done for the deck (grade 50 steel), derrick, and cage structures and a quote for assembling the platform. For the lifting mechanism, quotes were obtained from TWG Lantech (2011) for winches of various sizes. Quotes for the mooring equipment were obtained from Jeyco, of Austalia (2011) and Puget Sound Rope, CT (2011). The cost of installing the mooring grid was estimated as the cost of a 100 ft working vessel hired for seven (7) days. The results of this cost analysis are shown in Table 19.
Table 19. Cost of Floating Platform.

<table>
<thead>
<tr>
<th>Turbo Diameter</th>
<th>Costing</th>
<th>Unit cost</th>
<th>Quantity</th>
<th>Cost</th>
<th>Quantity</th>
<th>Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turbo Diameter</td>
<td>4 m (14 ft)</td>
<td>9 m (29 ft)</td>
<td>Structure</td>
<td>Hulls</td>
<td>$64,000</td>
<td>64'</td>
</tr>
<tr>
<td></td>
<td>Beams, Derrick</td>
<td>$37,000</td>
<td>64'</td>
<td>55 ft</td>
<td>$22,358</td>
<td>97 ft</td>
</tr>
<tr>
<td></td>
<td>Assembly</td>
<td>$50,000</td>
<td>64'</td>
<td>55 ft</td>
<td>$30,214</td>
<td>97 ft</td>
</tr>
<tr>
<td></td>
<td>Mooring</td>
<td>500 Kg Stingray Anchors</td>
<td>$1,458 ea.</td>
<td>4</td>
<td>$5,832</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td></td>
<td>28 mm Plasma Rope</td>
<td>$15 /ft.</td>
<td>2756</td>
<td>$40,320</td>
<td>2756</td>
</tr>
<tr>
<td></td>
<td>Lifting</td>
<td>M18Winch</td>
<td>$23,000 ea.</td>
<td>2</td>
<td>$46,000</td>
<td>2</td>
</tr>
<tr>
<td></td>
<td>Installation</td>
<td>Nobska</td>
<td>$5,000 /day</td>
<td>7</td>
<td>$35,000</td>
<td>7</td>
</tr>
<tr>
<td></td>
<td>Total</td>
<td>$218,398</td>
<td>$938,018</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 19. Cost of Floating Platform (continued).

<table>
<thead>
<tr>
<th>Turbo Diameter</th>
<th>Costing</th>
<th>Unit cost</th>
<th>Quantity</th>
<th>Cost</th>
<th>Quantity</th>
<th>Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turbo Diameter</td>
<td>13 m (43 ft)</td>
<td>17 m (57 ft)</td>
<td>Structure</td>
<td>Hulls</td>
<td>$64,000</td>
<td>64'</td>
</tr>
<tr>
<td></td>
<td>Beams, Derrick</td>
<td>$37,000</td>
<td>64'</td>
<td>119 ft</td>
<td>$234,309</td>
<td>146 ft</td>
</tr>
<tr>
<td></td>
<td>Assembly</td>
<td>$50,000</td>
<td>64'</td>
<td>119 ft</td>
<td>$316,634</td>
<td>146 ft</td>
</tr>
<tr>
<td></td>
<td>Mooring</td>
<td>500 Kg Stingray Anchors</td>
<td>$1,458 ea.</td>
<td>4</td>
<td>$5,832</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td></td>
<td>28 mm Plasma Rope</td>
<td>$15 /ft.</td>
<td>2756</td>
<td>$40,320</td>
<td>2756</td>
</tr>
<tr>
<td></td>
<td>Lifting</td>
<td>M18Winch</td>
<td>$23,000 ea.</td>
<td>2</td>
<td>$350,000</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td>Installation</td>
<td>Nobska</td>
<td>$5,000 /day</td>
<td>7</td>
<td>$35,000</td>
<td>7</td>
</tr>
<tr>
<td></td>
<td>Total</td>
<td>$1,387,387</td>
<td>$2,355,786</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
A Submerged Buoyant platform, illustrated in Figure 23, was considered for the following advantages:

- The platform could be raised to the surface and even towed to a shore facility for ease of turbine installation and repairs, and also platform maintenance.
- The platform would operate below surface activity.

Disadvantages include:

- Multiple moving systems may be required.
- Mooring array may require large seafloor footprint.
The Submerged Buoyant platform would comprise two cylindrical hulls rigidly connected by a truss structure, a derrick for raising and lowering the turbine, and a mooring system. This platform would be towed to and from the site with the device in the “up” position (as shown in the inset of Figure 23). Once on site, the platform would be connected to the mooring system, including a pendant weight. The turbine would then be lowered to the “down” position. The platform would then be submerged by allowing compartments in the bulkheaded pontoons to fill with seawater, until the pendant weight rested on the seafloor. This would keep the platform at the desired depth for the duration of testing. Once testing was completed, the process would be reversed: The platform would be raised to the surface by expelling the seawater from the pontoons using compressed air; the turbine would be raised to the “up” position; the mooring would be disconnected; and the platform would be towed back to shore.

**Specific Design Criteria**

- The platform must be stable at the surface, while submerged, and at all points in between. This means that when the pontoons are on the surface the platform must not tip more that \(1^\circ\) when subjected to any foreseeable load (e.g. strong wind), and while submerged the hydrostatic restoring moment must exceed the overturning moment when tipped any small angle.

Cost estimates for a Submerged Buoyant platform were obtained by designing a steel structure of suitable size, strength, and stability and estimating total expenses. Costs include those for material and labor to construct the platform, variable buoyancy system, the turbine lift system, the mooring line handling system, and the mooring system, including installation.
**Governing Equations-Hydrostatics**

The submerged-buoyant platform was analyzed for pitch, roll, and vertical stability under both submerged and surface conditions.

**Submerged**

The free-body diagram of the platform deploying a turbine at mid depth is shown in Figure 24 and variables therein are identified in Table 20.

![Figure 24. FBD of Submerged Buoyant platform. Current is from right to left.](image)

**Table 20. Submerged Buoyant platform hydrostatics variables.**

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$r_t$</td>
<td>Distance from C.G to Turbine Drag</td>
</tr>
<tr>
<td>$\theta$</td>
<td>Bow-down Angle</td>
</tr>
<tr>
<td>$W_p$</td>
<td>Platform Weight</td>
</tr>
<tr>
<td>$W_t$</td>
<td>Turbine Weight</td>
</tr>
<tr>
<td>$D_p$</td>
<td>Platform Drag</td>
</tr>
<tr>
<td>$D_{Wt}$</td>
<td>Drag from Wind Loading</td>
</tr>
<tr>
<td>$D_t$</td>
<td>Turbine Drag</td>
</tr>
<tr>
<td>$B_p$</td>
<td>Buoyant Force</td>
</tr>
<tr>
<td>$R_t$</td>
<td>Distance from CG to Mooring Attachments</td>
</tr>
<tr>
<td>$\beta$</td>
<td>Mooring Line Angle from Vertical</td>
</tr>
<tr>
<td>$T_1$</td>
<td>Tension in a Single Mooring Line (two used)</td>
</tr>
<tr>
<td>$D_{Op}$</td>
<td>Pontoon Diameter</td>
</tr>
<tr>
<td>$D_{Rp}$</td>
<td>Platform Draft</td>
</tr>
<tr>
<td>$L_b$</td>
<td>Platform Length (At waterline)</td>
</tr>
<tr>
<td>$M_b$</td>
<td>Righting Moment</td>
</tr>
</tbody>
</table>

Note: The tension in pendant lines is assumed to be negligible.

When submerged, the platform must satisfy horizontal, vertical, and rotational equilibrium. In the horizontal direction,

$$2T_1 \sin \beta - D_p - D_t = 0; \quad (68)$$

vertically,
\[-2T_i \cos \beta - W_p - W_t + B_p = 0; \quad (69)\]
in rotation,
\[-rT_i \cos \beta - r_D + bgD_p = 0. \quad (70)\]

Stability for a completely submerged rigid body is achieved when, for any reasonable tipping angle, the righting moment (due to the distance between the center of gravity and the center of buoyancy) exceeds the tipping moment (due to the new angle of attack of the body). This criterion requires,
\[
(W_t + W_p)\overline{b\theta} > C_m A q \frac{L_p q A}{q}, \quad (71)
\]
where \(q\) is the free-stream dynamic pressure, \(1/2\rho U^2\) with \(U\) being the free-stream velocity of the fluid. Furthermore, \(A\) is the area of the base of the body, and \(C_m\) is the pitching moment coefficient, which is a function of \(\theta\). Table 21 gives values of \(C_m\) for a long cylindrical body with a nose cone at a Reynolds number, \(Re\), comparable to that of the flow over the submerged platform. In this case, the \(Re\) is defined by
\[
Re = \frac{LU}{v}, \quad (72)
\]
where \(L\) is the length of the body and \(v\) is the dynamic viscosity of the fluid.

<table>
<thead>
<tr>
<th>Angle of Attack</th>
<th>(C_m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Degrees</td>
<td>Radians</td>
</tr>
<tr>
<td>-4</td>
<td>-0.070</td>
</tr>
<tr>
<td>0</td>
<td>0.000</td>
</tr>
<tr>
<td>5</td>
<td>0.087</td>
</tr>
<tr>
<td>10</td>
<td>0.175</td>
</tr>
<tr>
<td>15</td>
<td>0.262</td>
</tr>
</tbody>
</table>
Surfaced

In addition to remaining stable while submerged, the platform must safely function as a surface vessel while being towed to and from the site. To this end, the hydrostatics in both the vertical direction and the pitch direction were analyzed. The forces present in these analyses are shown in Figure 25.

![Figure 25. FBD of Submerged-Buoyant Platform at Surface](image)

On the surface, the platform must satisfy vertical equilibrium and not pitch or roll more than the maximum allowed angle. Vertical equilibrium mandates

$$V_s \rho g - W_t - W_p = 0.$$  \hspace{1cm} (73)

Here $V_s$ is the submerged volume in the pontoons, found by specifying a draught of 0.4 times the pontoon diameter, so that any tipping increases the platform's area moment of inertia at the waterplane. Pitch stability is found by summing moments about the center of gravity, such that

$$D_t r_t + B_p \bar{g} m \theta = 0,$$  \hspace{1cm} (74)

where $\theta$ is specified to be less than 0.017 rad ($1^\circ$). The stabilizing effect of the mooring is not taken into account because the platform must be stable while being towed to and from the site, independent of the mooring system. (Roll stability is calculated the same way, but the area moment of inertia, and thus the metacentric height, $\bar{g} m$, is always greater in that direction for this platform.) Here the wind drag,
$D_w$ was estimated as

$$D_w = 2\left( \frac{1}{2} \rho_a A_t C_D U^2 \right),$$

(75)

where $\rho_a$ is the density of air, $A_t$ is the projected area of the turbine, $C_D$ is the coefficient of drag of the turbine (1.4) and $U$ is the design wind speed, 15 m/s (29 knots). The drag force on the turbine was doubled to account for surrounding structure.

**Variable Buoyancy**

The submerged floating platform would operate on the principle of variable buoyancy. This method of suspending buoyant structures at fixed depths has been demonstrated extensively in the aquaculture industry (Celikkol et al., 2006).

Variable buoyancy systems can be highly unstable if the air-ballast water chambers include large free surfaces. In this case, a small perturbation will cause a large in the location of both the center of gravity and the center of buoyancy. To prevent this, each pontoon was divided into several chambers by bulkheads, shown in Figure 26.

![Figure 26. Cut-away view of a Submerged Buoyant platform pontoon. Bulkheads increase stability (and structural rigidity). A central space is included for compressed air storage and controls.
When the platform is being lowered, chambers will be filled sequentially. This process of ballasting will start with each of the four corner chambers and then move...](image-url)
to the next furthest chamber from the center of gravity until the total buoyancy is sufficiently reduced. Beginning with the outermost chambers ensures that the platform's mass moment of inertia is always at a maximum, making it less susceptible to impulsive perturbations.

A critical criterion in the design of a variable buoyancy system is that pressure of the stored air must be much greater than the ambient pressure of the seawater to expel the fluid from the ballast tanks. At any given depth \( h \), the absolute air pressure required is given by

\[
P_a = \rho g h + 1 \text{ atmosphere.} \tag{76}
\]

For this application \( P_a \) is approximately 250 kPa (36 psi). Thus, commercially available air-storage systems capable of storage pressure, \( P_s=30 \text{ MPa} \) (4300 psi) are more than sufficient.

The volume of water that must be expelled from the integrated ballast tanks each time the platform is raised is

\[
V_W = V_p1 - V_{p2}. \tag{77}
\]

Here \( V_{p1} \) is the total volume of the pontoons (that required for surface stability), and \( V_{p2} \) is the volume required by hydrostatics in the submerged case. Incorporating the ideal gas law with negligible temperature change, the required volume for storing the compressed air is

\[
V_s = \frac{P_{\text{atm}}}{P_a} V_W. \tag{78}
\]

**Mooriing System**

Variable buoyancy systems can be very difficult to control in the open ocean. To eliminate the need for an exact force balance, a pendant system was incorporated into the Submerged Buoyant platform design. This system, illustrated in Figure 23, would hold the platform at the desired depth. This would be accomplished by leaving...
reserve buoyancy in the platform ballast tanks. Thus, the actual volume of air in the pontoons would always exceed the calculated volume required for vertical equilibrium, $V_p$. To ensure effectiveness of the system, the required vertical force that the pendant system exerted on the platform was calculated as twice the vertical component of the mooring force.

In addition to the pendant system, the platform would be held in place with four mooring lines, each extending to an embedment anchor. During each tidal cycle the aft pair of lines would be slack. Thus, the platform would not pivot to match the tidal cycle. However, the moorings would be laid out such that the platform would align with the dominant current direction on both the ebb and flood tides, which are approximately 20 degrees off of a perfect 180 degree alignment (Howes et al., 2009). Anchors were chosen which provided a pull-out safety factor, $SF_{pull}$, greater than 5, where

$$SF_{pull} = \frac{T_i \cos \beta}{T_{hold}}. \quad (79)$$

Here $T_{hold}$ is the rated holding power of the anchor in sand/gravel, $T_i$ is the tension in a single mooring line, and $\beta$ is the angle between the mooring line and the horizontal, assumed fixed. (Note that a proper mooring system in which a length of heavy chain connects the mooring line to the anchor, would effectively make $\cos \beta = 1$.)

**Solving**

Since both the submerged and surfaced conditions depend on the weight and dimensions of the platform, they cannot be solved independently. Thus, the surface and submerged equations were simultaneously solved numerically under the stability conditions. Equation (72) was then solved for the necessary distance between the center of buoyancy and center of gravity, $\bar{b}g$, for each angle in the above table and the
maximum was used. This value was generally found to be on the order of \( \frac{1}{2} \) the pontoon diameter. Thus the platform can be constructed to be stable independent of the aid of a bridle system, but it will require careful distribution of the platform’s mass. The results of this design work are shown for each turbine size in Table 22.
Table 22. Submerged Buoyant Platform parameters.

<table>
<thead>
<tr>
<th>Turbine Diameter</th>
<th>4 m (14 ft)</th>
<th>9 m (29 ft)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Platform Structure</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Required Pontoon Diameter, Surfac ed</td>
<td>1.89 m 6.2 ft</td>
<td>2.57 m 8.4 ft</td>
</tr>
<tr>
<td>Required Pontoon Diameter, Submerged</td>
<td>1.00 m 3.3 ft</td>
<td>1.35 m 4.4 ft</td>
</tr>
<tr>
<td>Pontoon Length</td>
<td>6.7 m 21.9 ft</td>
<td>9.3 m 30.7 ft</td>
</tr>
<tr>
<td>Beam (between centers of pontoons)</td>
<td>8.7 m 28.4 ft</td>
<td>12.9 m 42.5 ft</td>
</tr>
<tr>
<td>Deck Length (Width)</td>
<td>4.0 m 13.1 ft</td>
<td>4.0 m 13.1 ft</td>
</tr>
<tr>
<td>Pontoon Volume, Submerged</td>
<td>1.0 m³ 371 ft³</td>
<td>1.35 m³ 4,716 ft³</td>
</tr>
<tr>
<td>Platform Wall Thickness</td>
<td>0.0064 m 1/4 in</td>
<td>0.0064 m 1/4 in</td>
</tr>
<tr>
<td>Mass of Pontoon s</td>
<td>3,942 kg 8,691 lbm</td>
<td>10,731 kg 23,658 lbm</td>
</tr>
<tr>
<td>Mass of Truss Members</td>
<td>2,628 kg 5,794 lbm</td>
<td>7,154 kg 15,772 lbm</td>
</tr>
<tr>
<td>Mass of additional items</td>
<td>1,314 kg 2,897 lbm</td>
<td>3,577 kg 7,886 lbm</td>
</tr>
<tr>
<td>Total (estimated) Mass of Structure</td>
<td>7,885 kg 17,383 lbm</td>
<td>21,462 kg 47,316 lbm</td>
</tr>
<tr>
<td><strong>Submerged Mooring</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Scope</strong></td>
<td>1/7</td>
<td>1/7</td>
</tr>
<tr>
<td><strong>Surface Stability</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Draft</td>
<td>0.76 m 2.5 ft</td>
<td>1.03 m 3.4 ft</td>
</tr>
<tr>
<td>Chord of Pontoon at Waterline</td>
<td>1.85 m 6.1 ft</td>
<td>2.52 m 8.3 ft</td>
</tr>
<tr>
<td>Submerged Volume (Total, lower pontoons)</td>
<td>14 m³ 492 ft³</td>
<td>52 m³ 1,824 ft³</td>
</tr>
<tr>
<td>Rolling Angle</td>
<td>0.002 rad 0.1 deg</td>
<td>0.001 rad 0.1 deg</td>
</tr>
<tr>
<td>Pitching Angle</td>
<td>0.010 rad 0.6 deg</td>
<td>0.006 rad 0.3 deg</td>
</tr>
<tr>
<td><strong>Pendant Weight</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pendant Safety Factor</td>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td>Pendant Mass</td>
<td>4,367 kg 9,628 lbm</td>
<td>16,284 kg 35,900 lbm</td>
</tr>
<tr>
<td>Size of one whole cubic Pendant</td>
<td>1.22 m 4.0 ft</td>
<td>1.89 m 6.2 ft</td>
</tr>
<tr>
<td><strong>Submerged Stability</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Required Distance from CG to CB</td>
<td>0.743 m</td>
<td>1.106 m</td>
</tr>
<tr>
<td><strong>Ballast</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Required Storage Volume</td>
<td>0.0528 m³ 53 L</td>
<td>0.1922 m³ 192 L</td>
</tr>
<tr>
<td><strong>Lifting</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Number of Winches</td>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td>Maximum Winch Line Pull Required</td>
<td>53,188 N 11,957 lbf</td>
<td>276,828 N 62,233 lbf</td>
</tr>
<tr>
<td>Winch Selected</td>
<td>M18 LWD3500</td>
<td></td>
</tr>
<tr>
<td><strong>Mooring Equipment</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Desired Working Safety Factor</td>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td>Chain Length</td>
<td>80 m 262 ft</td>
<td>80 m 175.0 ft</td>
</tr>
<tr>
<td>Line Length</td>
<td>1290.5 ft</td>
<td>1290.5 ft</td>
</tr>
<tr>
<td>Stingray</td>
<td>399 m 1,309 ft</td>
<td>893 m 2,929 ft</td>
</tr>
</tbody>
</table>

**Legend:**
- Iterated values
- Design inputs
- Calculated values

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Table 22. Submerged Buoyant Platform parameters.

<table>
<thead>
<tr>
<th>Turbine Diameter</th>
<th>13 m (43 ft)</th>
<th>17 m (57 ft)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Platform Structure</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Required Pontoon Diameter, Surfacd</td>
<td>3.23 m</td>
<td>10.6 ft</td>
</tr>
<tr>
<td>Required Pontoon Diameter, Submerged</td>
<td>1.72 m</td>
<td>5.6 ft</td>
</tr>
<tr>
<td>Pontoon Length,</td>
<td>20.0 m</td>
<td>65.6 ft</td>
</tr>
<tr>
<td>Beam (between centers of pontoons)</td>
<td>26.0 m</td>
<td>85.3 ft</td>
</tr>
<tr>
<td>Deck Length (Width)</td>
<td>12.5 m</td>
<td>39.4 ft</td>
</tr>
<tr>
<td>Pontoon Volume, Submerged</td>
<td>93 m$^3$</td>
<td>3,272 ft$^3$</td>
</tr>
<tr>
<td>Platform Wall Thickness</td>
<td>0.0064 m</td>
<td>1/4 in</td>
</tr>
<tr>
<td>Mass of Pontoons</td>
<td>20,246 kg</td>
<td>44,635 lbm</td>
</tr>
<tr>
<td>Mass of Truss Members</td>
<td>13,497 kg</td>
<td>29,756 lbm</td>
</tr>
<tr>
<td>Mass of additional items</td>
<td>6,749 kg</td>
<td>14,878 lbm</td>
</tr>
<tr>
<td>Total (estimated) Mass of Structure</td>
<td>40,492 kg</td>
<td>89,269 lbm</td>
</tr>
<tr>
<td><strong>Submerged Mooring</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Scope</td>
<td>1/7</td>
<td></td>
</tr>
<tr>
<td><strong>Surface Stability</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Draft</td>
<td>1.29 m</td>
<td>4.2 ft</td>
</tr>
<tr>
<td>Chord of Pontoon at Waterline</td>
<td>3.17 m</td>
<td>10.4 ft</td>
</tr>
<tr>
<td>Submerged Volume (Total, lower pontoons)</td>
<td>123 m$^3$</td>
<td>4,329 ft$^3$</td>
</tr>
<tr>
<td>Rolling Angle</td>
<td>0.001 rad</td>
<td>0.1 deg</td>
</tr>
<tr>
<td>Pitching Angle</td>
<td>0.004 rad</td>
<td>0.2 deg</td>
</tr>
<tr>
<td><strong>Pendant Weight</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pendant Safety Factor</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>Pendent Mass</td>
<td>35,956 kg</td>
<td>79,269 lbm</td>
</tr>
<tr>
<td>Size of one whole cubic Pendent</td>
<td>2.47 m</td>
<td>8.1 ft</td>
</tr>
<tr>
<td><strong>Submerged Stability</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Required Distance from CG to CB</td>
<td>1.290 m</td>
<td></td>
</tr>
<tr>
<td><strong>Ballast</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Required Storage Volume</td>
<td>0.4654 m$^3$</td>
<td>465 L</td>
</tr>
<tr>
<td><strong>Lifting</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Number of Winches</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>Maximum Winch Line Pull Required</td>
<td>767,034 N</td>
<td>172,436 lbf</td>
</tr>
<tr>
<td>Winch Selected</td>
<td>LWD3500</td>
<td>LWD3500</td>
</tr>
<tr>
<td><strong>Mooring Equipment</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Desired Working Safety Factor</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>Chain Length</td>
<td>80 m</td>
<td>262.5 ft</td>
</tr>
<tr>
<td>Line Length</td>
<td>393 m</td>
<td>1115.5 ft</td>
</tr>
<tr>
<td>Stingray</td>
<td>1000 kg</td>
<td>1,102 lbm</td>
</tr>
</tbody>
</table>

**Legend:**
- **Iterated values**
- **Design Inputs**
- **Calculated values**
Costing

The material cost of the pontoons was estimated from a quote from L.B. Foster (2010) for ASTM A252 Gr. 3 steel piles on a per-pound basis. The material cost of the mechanical tubing that constitutes the platform structure was estimated from a quote from American Steel for ASTM A333 Gr. 6 mechanical tubing, also on a per-pound basis. Corrosion protection costs were based on a quote from L.B. Foster for marine-grade epoxy coating over the exterior surface area of the platform. Welding costs were estimated from a quote supplied to Jeff Byrne for his V2 design (2010). The material costs were subtracted from a quote that included deck beams and welded mechanical tubing and the remainder was assumed to be the welding cost, which was reduced to dollars per pound of tubing. (While this is clearly an overestimate of the fraction which is welding cost, it is also worth noting that the quotes used to estimate the welding cost were for A36 steel, which may be easier to weld than ASTM A333 Gr. 6). The cost of final assembly was also taken to be a function of structure weight and was estimated from a quote for the V2 platform. The cost of forming a concrete pendent weight was determined from R.S. Means (2011) and the cost of the required lines was obtained as for the mooring lines, described below.

The cost of the variable-buoyancy system was estimated by using the per-pound cost of ASTM A252 Gr. 3 as the cost of the integrated ballast tanks, the cost of ASTM A333 Gr. 6 for the necessary piping, and the per-pound welding cost as above. The price of twenty (20) stainless steel 2 in. ball valves with remote activation was obtained from Swagelok (2011). The most expensive type was used in order to compensate for other valves, etc. not included in the cost analysis. (Corrosion in these
components will need to be given careful consideration during the detailed design phase because stainless steel acts as the sacrificial anode to most structural steels.)

For the turbine lifting mechanism, quotes were obtained from TWG Lantech (2011) for winches of various sizes. Quotes for the mooring equipment were obtained from Jeyco (2011) and Puget Sound Rope (2011). The cost of installing the mooring grid was estimated as the cost of a 100 ft. working vessel hired for seven (7) days. The results of this cost analysis are shown in Table 23.

<table>
<thead>
<tr>
<th>Table 23. Cost of Submerged Buoyant Platform.</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Turbine Diameter</strong></td>
</tr>
<tr>
<td><strong>Unit Price</strong></td>
</tr>
<tr>
<td><strong>Pontoons</strong></td>
</tr>
<tr>
<td>Steel Piles</td>
</tr>
<tr>
<td>Anti-Corrosion Coating</td>
</tr>
<tr>
<td><strong>Tubing</strong></td>
</tr>
<tr>
<td>ASTM A333 Gr. 6 Mechanical Tubing</td>
</tr>
<tr>
<td>Anti-Corrosion Coating*</td>
</tr>
<tr>
<td>Welding</td>
</tr>
<tr>
<td><strong>Assembly</strong></td>
</tr>
<tr>
<td>Assembly</td>
</tr>
<tr>
<td><strong>Pendent Weight</strong></td>
</tr>
<tr>
<td>Concrete Weight</td>
</tr>
<tr>
<td>28 mm Plasma 12 Strand</td>
</tr>
<tr>
<td><strong>Moorings</strong></td>
</tr>
<tr>
<td>1000 kg Stingray Anchor</td>
</tr>
<tr>
<td>36mm Studlink Chain</td>
</tr>
<tr>
<td>28 mm Plasma 12 Strand</td>
</tr>
<tr>
<td><strong>Variable Buoyancy</strong></td>
</tr>
<tr>
<td>Steel (Pressure Vessel, Bulkheading)</td>
</tr>
<tr>
<td>ASTM A333 Gr. 6 Piping</td>
</tr>
<tr>
<td>Welding</td>
</tr>
<tr>
<td>2&quot; Ball Valves</td>
</tr>
<tr>
<td><strong>Lifting</strong></td>
</tr>
<tr>
<td>M18Winch</td>
</tr>
<tr>
<td><strong>Total</strong></td>
</tr>
<tr>
<td></td>
</tr>
</tbody>
</table>
Table 23. Cost of Submerged Buoyant Platform (continued).

<table>
<thead>
<tr>
<th>Turbine Diameter</th>
<th>13 m (43 ft)</th>
<th>17 m (57 ft)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Unit Price</td>
<td>Qty.</td>
</tr>
<tr>
<td><strong>Pontoons</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Steel Piles</td>
<td>$1.87 lb</td>
<td>44,635</td>
</tr>
<tr>
<td>Anti-Corrosion Coating</td>
<td>$4.02 ft$</td>
<td>9,197</td>
</tr>
<tr>
<td><strong>Tubing</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>ASTM A333 Gr. 6 Mechanical Tubing</td>
<td>$1.64 lb</td>
<td>29,756</td>
</tr>
<tr>
<td>Anti-Corrosion Coating*</td>
<td>$4.02 ft$</td>
<td>5,058</td>
</tr>
<tr>
<td>Welding</td>
<td>$1.38 lb</td>
<td>29,756</td>
</tr>
<tr>
<td><strong>Assembly</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Assembly</td>
<td>$0.57 lb</td>
<td>74,391</td>
</tr>
<tr>
<td><strong>Pendent Weight</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Concrete Weight</td>
<td>$96 ea.</td>
<td>4</td>
</tr>
<tr>
<td>28 mm Plasma 12 Strand</td>
<td>$14.63 ft$</td>
<td>197</td>
</tr>
<tr>
<td><strong>Mooring</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1000 kg Stingray Anchor</td>
<td>$3,537 ea.</td>
<td>4</td>
</tr>
<tr>
<td>36mm Studlink Chain</td>
<td>$3,000 shot</td>
<td>4</td>
</tr>
<tr>
<td>28 mm Plasma 12 Strand</td>
<td>$14.63 ft$</td>
<td>1,378</td>
</tr>
<tr>
<td><strong>Variable Buoyancy</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Steel (Pressure Vessel, Bulkheading)</td>
<td>$1.87 lb</td>
<td>2,790</td>
</tr>
<tr>
<td>ASTM A333 Gr. 6 Piping</td>
<td>$1.64 lb</td>
<td>1,488</td>
</tr>
<tr>
<td>Welding</td>
<td>$40,958</td>
<td>1</td>
</tr>
<tr>
<td>2&quot; Ball Valves</td>
<td>$1,370 ea.</td>
<td>20</td>
</tr>
<tr>
<td><strong>Lifting</strong></td>
<td>$23,000-$175,000 ea.</td>
<td>2</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

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A fixed-structure gravity foundation platform that would extend to mid-depth, illustrated in Figure 27, was considered for the following advantages:

- The platform would be below most surface traffic.
- The concept would be simple and robust.
- Material costs would likely be low.

Disadvantages include:

- The platform mounting structure must extend at least half the distance to the surface to place turbines in the high-velocity region.
- Maintenance and turbine installation/retrieval would likely be difficult and expensive.
- Scour would have to be considered.
The platform would include a box-shaped concrete base with sufficient weight and dimensions to resist tipping and sliding. This base would support a mounting structure designed as a truss sufficient to prevent yielding and buckling in its members. Figure 28 and Figure 29 show front and side views of this platform, respectively. Costs for constructing the structure onshore were determined from RS Means (2011) and quotes from steel producers and fabricators, and quotes for utilizing crane barges that could install the foundation were obtained.

Figure 28. Front view of the Gravity Foundation platform. Fixed dimensions are given in meters. All other dimensions vary with maximum turbine size.
Specific Design Criteria

- The foundation must place the turbine hub 15 m (49 ft.) above the seafloor.

- The foundation must prohibit tipping or sliding. The foundation must have a minimum factor of safety of 3 (three) in the worst loading scenario.

- In the event of failure, the foundation must slide rather than tip. Specifically, the tipping factor of safety must exceed the sliding factor of safety by 25%.

- The foundation must resist cracking, e.g. during installation. That is, it must have a bending safety factor of 5 (not including reinforcing steel) under worst-case bending.

- Each member of the truss structure must have a safety factor of 3 (three) against material yielding and 4 (four) against buckling.

The following assumptions were used in the analysis:

- A tipping condition is that in which the entire normal force acts at the rear...
lower corner of the level foundation.

- Friction can be sufficiently modeled by Coulomb's Law of Friction, in which the maximum friction force equals the normal force times a coefficient of friction between the two surfaces.
- The weight of the turbine is neglected for the tipping analysis. (This ensures that the foundation will be secure even if used to test a lightweight, high drag turbine.)
- A 2.5 m/s (5 knots) current is uniform over the entire depth.

**Governing Equations**

**Foundation Design**

A Free Body Diagram of the fixed gravity foundation platform is shown in Figure 30 and the variables therein are in Table 24.

Figure 30. Free Body Diagram of Gravity Platform. The normal force N is located at the down-current edge of the platform base to model the onset of tipping. Drag force on the mounting structure was assumed negligible compared to turbine drag.
Table 24. Statics variables for Fixed Gravity Foundation.

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>(W_f)</td>
<td>Foundation Weight (Dry)</td>
</tr>
<tr>
<td>(W_t)</td>
<td>Turbine Weight</td>
</tr>
<tr>
<td>(D_f)</td>
<td>Foundation Drag</td>
</tr>
<tr>
<td>(D_t)</td>
<td>Turbine Drag</td>
</tr>
<tr>
<td>(r_{lf})</td>
<td>Distance from bottom to Turbine Drag</td>
</tr>
<tr>
<td>(B_f)</td>
<td>Buoyant Force</td>
</tr>
<tr>
<td>(H_f)</td>
<td>Foundation Height</td>
</tr>
<tr>
<td>(L_f)</td>
<td>Foundation Length</td>
</tr>
</tbody>
</table>

To prevent sliding, the maximum friction force must equal or exceed the total drag. That is,

\[
F_{\text{fmax}} \geq F_f = D_c + D_f. \tag{80}
\]

Sliding was modeled using the Coulomb model of friction,

\[
F_{\text{fmax}} = N \mu_s \tag{81}
\]

where \(F_{\text{fmax}}\) is the maximum applicable friction force and \(\mu_s\) is a static coefficient of friction for sand-gravel, given by AASHTO (Taly, 2010) as 0.55 for concrete on medium sand, gravel. Neglecting the weight of the turbine, the normal force, \(N\), is the weight of the foundation minus the weight of displaced water, so that

\[
N = W_f - B_f. \tag{82}
\]

The weight of the foundation is

\[
W_f = \rho_c g (L_f w_f H_f), \tag{83}
\]

where \(\rho_c\) is the density of the concrete, and \(w_f\) is the width of the foundation. The buoyancy force is

\[
B_f = \rho g (L_f w_f H_f). \tag{84}
\]

Here drag on the foundation is given by
\[
D_f = \frac{1}{2} \rho C_D U^2 A,
\]  
(85)

in which the coefficient of drag, \( C_D \), is given by Hoerner (1965) as 1.05 for a block on a flat surface.

To prevent tipping, moments applied to the platform about point \( c \) must balance, so that

\[
\sum M_c = 0,
\]  
(86)

so that

\[
D_f \frac{H}{2} + D_t r_t \leq (W_f - B_f) \frac{L}{2}.
\]  
(87)

The equals sign pertains to the onset of tipping, shown in Figure 30; the “greater than” sign corresponds to the platform resting solidly on the sediment, with normal force \( N \) acting to the right of point \( C \). As a result, there exist two factors of safety for the foundation: A tipping safety factor and a sliding safety factor, given by the maximum resisting moment over the design moment, and the maximum friction force over the drag force, respectively. Thus, the safety factors are

\[
SF_{tip} = \frac{(w_f - b_f) \frac{L}{2}}{D_f \frac{H}{2} + D_t r_t}
\]  
(88)

and

\[
SF_{slide} = \frac{(w_f - b_f) \mu_s}{D_t + D_f}.
\]  
(89)

Additionally, the low tensile strength of concrete necessitates a consideration of bending due to an uneven seafloor. A free-body diagram of the worst possible loading case is shown in Figure 31.
The maximum bending stress in the base was approximated by the formula

\[ \sigma_{\text{bend}} = \frac{Mc}{I}. \]  

(90)

Here \( M \) is the maximum bending moment, \( c \) is the distance from the neutral axis, and \( I \) is the area moment of inertia of the beam. Neglecting the ameliorating effects of steel rebar, the associated safety factor is concrete's Ultimate Tensile Strength divided by the maximum bending stress,

\[ SF_{\text{tensile}} = \frac{\text{Ultimate Tensile Stress}}{\sigma_{\text{bend}}}. \]  

(91)

Using the above analysis, the foundation dimensions were iterated for each turbine scale-up factor of interest to minimize weight under the constraints listed in Table 25, using a Generalized Reduction Gradient (GRG) nonlinear forward difference solver in the Microsoft Excel® Solver package.

Table 25. Gravity Foundation base constraints.

<table>
<thead>
<tr>
<th>Constraint</th>
<th>Requirement</th>
</tr>
</thead>
<tbody>
<tr>
<td>Platform will not slide:</td>
<td>( SF_{\text{slide}} \geq 3 )</td>
</tr>
<tr>
<td>Platform will not crack in bending:</td>
<td>( SF_{\text{tensile}} \geq 5 )</td>
</tr>
<tr>
<td>Platform will slide before tipping:</td>
<td>( SF_{\text{tip}} \geq 1.25SF_{\text{slide}} )</td>
</tr>
<tr>
<td>Platform cannot be excessively narrow:</td>
<td>( w_{r} \geq 0.85L_{f} )</td>
</tr>
</tbody>
</table>

The results of this analysis are shown in Table 26.
Table 26. Gravity Foundation base dimensions.

<table>
<thead>
<tr>
<th>Turbine Diameter</th>
<th>4 m (14 ft)</th>
<th>9 m (29 ft)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Foundation</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Foundation Height, $H_f$</td>
<td>1.4 m</td>
<td>4.6 ft</td>
</tr>
<tr>
<td>Width of Foundation, $W_f$</td>
<td>7.3 m</td>
<td>24.0 ft</td>
</tr>
<tr>
<td>Foundation Length, $L$</td>
<td>8.6 m</td>
<td>28.3 ft</td>
</tr>
<tr>
<td>Foundation Mass, $m$</td>
<td>213,053 kg</td>
<td>469,702 lbm</td>
</tr>
</tbody>
</table>

Legend:
- Iterated values
- Calculated values

Table 26. Gravity Foundation base dimensions (continued).

<table>
<thead>
<tr>
<th>Turbine Diameter</th>
<th>13 m (43 ft)</th>
<th>17 m (57 ft)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Foundation</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Foundation Height, $H_f$</td>
<td>2.7 m</td>
<td>8.8 ft</td>
</tr>
<tr>
<td>Width of Foundation, $W_f$</td>
<td>9.9 m</td>
<td>32.3 ft</td>
</tr>
<tr>
<td>Foundation Length, $L$</td>
<td>11.6 m</td>
<td>38.0 ft</td>
</tr>
<tr>
<td>Foundation Mass, $m$</td>
<td>734,901 kg</td>
<td>1,620,180 lbm</td>
</tr>
</tbody>
</table>

Legend:
- Iterated values
- Calculated values

Scour

Anti-scour structure must be designed with care. Rocker (1985) cites an example in which hinged concrete scour protection slabs were broken off of their main structure by deep water wave-induced scour in 30 m (100 ft.) of water. However, Gerwick points to successful installations of steel skirts around gravity foundations that reduce scour while increasing the foundation's ability to resist sliding. Another method, currently being implemented for offshore wind gravity foundations at the Thornton Bank Offshore Wind Farm off the Belgian coast uses layers of coarse sediment and gravel to minimize scour (Terra et Aqua). A steel scour skirt was designed using ¼” ASTM 252 Gr. 1 steel.
Support Structure

Statics

The mounting structure for the turbine was designed using Circular Hollow Section (CHS) truss members because of their high resistance to buckling and comparatively low drag coefficient, shown in Figure 32 and Figure 33 respectively.

![Figure 32. Comparison of the masses of hollow and open sections under compression in relation to the loading (European Steel Design Education Programme, 1994).](image)

A three-dimensional support structure was designed and analyzed with SolidWorks® finite element software. The analysis was first conducted using truss members (all joints pinned). Axial forces in each truss element were extracted and Euler's buckling analysis was conducted. In this analysis the axial load under which each element will buckle is given by

\[ P_{\text{crit}} = \frac{\pi^2EI}{L^2}, \]

where \( E \) is the elastic modulus, \( I \) is the area moment of inertia, and \( L \) is the effective length of the member. Under pinned end conditions—assumed for this analysis as a
worst case—the effective length is the actual length of the member. Under fixed end condition (e.g. welding) the effective length is half the actual length. So using welded joints increases the critical load by a factor of four and thus quadruples the buckling safety factor, given by

\[
SF_{buckling} = \frac{P_{\text{crit}}}{P},
\]

where \( P \) is the axial force in the member. This analysis was used to select section properties which resulted in the each member having \( SF_{buckling} \geq 4 \). Standard structural tubing sizes meeting those requirements were incorporated into the design, which was then reanalyzed using Solidworks® FEA software for both von Mises stress failure and for buckling using rigid connections (simulating a welded structure). For simplicity in construction, the entire mounting structure was designed using only two sizes of mechanical tubing. Future detailed design would need to consider the distributed transverse drag load on each member. In each scenario the weight and drag forces of the turbine were applied to the truss structure along with vertical forces accounting for the moment arm between the top of the truss structure and the turbine’s center of drag. The results for the final iteration truss design under loads corresponding to the 13 m (43 ft.) representative turbine are shown in Figure 34 and Figure 35. Note that the deflections illustrated in Figure 35 are greatly exaggerated; the maximum deflection is on the order of millimeters.
Figure 34. Finite element stress analysis of truss structure for Fixed Gravity Foundation Platform, capable of supporting a 13 m (43 ft.) turbine in a 2.5 m/s current. Maximum normal stress is shown in Pa. The yield stress for the chosen material (ASTM A333 Gr. 6) is 240 MPa.

Figure 35. Finite element buckling analysis of truss structure for Fixed Gravity Foundation Platform, capable of supporting a 13 m (43 ft.) turbine in a 2.5 m/s current. Color graph shows the magnitude (RESULTANT) of the displacement vector U, $U_{RES}$. The load factor of 3.3 is the buckling safety factor.

The final mechanical tubing diameters for this scale are shown (in mm) in Figure 36.
Figure 36. Mechanical tubing diameters capable of supporting a 13 m (43 ft.) turbine in a 2.5 m/s current. Members of equal outer radii also share inner radius dimensions. Dimensions are in meters. Base is 11.4 m (37 ft.) wide.

Once a mounting structure of sufficient dimensions was designed for a 13 m (43 ft.) turbine, the results were scaled to find approximate dimensions for platforms for different turbine sizes. The results are shown in Table 26.

Table 26. Gravity Foundation mounting structure dimensions.

<table>
<thead>
<tr>
<th>Turbine Diameter</th>
<th>4 m (14 ft)</th>
<th>9 m (29 ft)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>OD1 70 mm</td>
<td>150 mm 5.9</td>
</tr>
<tr>
<td></td>
<td>ID1 60 mm</td>
<td>130 mm 5.1</td>
</tr>
<tr>
<td></td>
<td>OD2 50 mm</td>
<td>90 mm 3.5</td>
</tr>
<tr>
<td></td>
<td>ID2 40 mm</td>
<td>80 mm 3.1</td>
</tr>
</tbody>
</table>

Legend:
- Iterated values

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Table 26. Gravity Foundation parameters (continued).

<table>
<thead>
<tr>
<th>Turbine Diameter</th>
<th>Foundation Height, ( H_f )</th>
<th>Width of Foundation, ( \text{w}_f )</th>
<th>Foundation Length, ( L )</th>
<th>Foundation Mass, ( m )</th>
</tr>
</thead>
<tbody>
<tr>
<td>13 m (43 ft)</td>
<td>2.7 m 8.8 ft 3.6 m 11.8 ft</td>
<td>9.9 m 32.3 ft 11.3 m 37.0 ft</td>
<td></td>
<td></td>
</tr>
<tr>
<td>17 m (57 ft)</td>
<td>11.8 m 38.0 ft 13.3 m 43.5 ft</td>
<td>734,901 kg 1,620,180 lbm 1,285,038 kg 2,833,023 lbm</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Legend: iterated values

Dynamics—Vibration

Neglecting vibrational issues in marine structures can be catastrophic (Tomlinson, 2008, p. 413). Thus, after determining suitable dimensions for each member of the support structure to withstand its static loading, the vibrational response of each member was characterized using the method set forth by Tomlinson. According to standard beam theory the natural frequency of a beam is found to be

\[
f_N = \frac{K'}{L^2} \sqrt{\frac{E I}{M}}
\]

(94)

where \( E \) is Young’s Modulus, \( I \) is the area moment of inertia, \( M \) is mass/unit length of the beam (including the mass of the water contained in the beam and the mass of the displaced water), \( L \) is the length of the beam, and \( K' \) is a factor of 3.56 for the first mode of vibration in members with ends fixed against both translation and rotation.

The Strouhal number can then be used to find the fluid velocity at which the frequency of vortex shedding will match the member’s natural frequency. This critical velocity is given by

\[
V_{\text{crit}} = K f_N d_o.
\]

(95)

where \( d_o \) is the outer diameter of the member and \( K \) is given in Table 27.
According to Mittal and Kumar (1999), “in-line oscillations are significant only if the mass of the cylinder is not too large compared with the mass of the surrounding fluid it displaces.” Since the mass of the cylinder is on the order of the mass of the surrounding fluid, it was assumed that only cross-flow motion is significant. The critical velocity (that which would cause the onset of cross-flow motion) was determined for each member and the lowest was 3.8 m/s (7.4 knots)—far higher than the maximum velocities seen in the channel. Thus, the recommended designs will experience negligible vortex-induced vibration.

Because of its unique resistance to corrosion in seawater even after being welded, 316L stainless steel was originally investigated (Specialty Steel Industry of North America (SSINA), 2011). However, its cost was prohibitive. Several steels commonly found in marine applications were considered for this unique structure. Their properties are listed alongside alternatives in Table 28.

<table>
<thead>
<tr>
<th>Material</th>
<th>Yield Strength</th>
<th>Ultimate Tensile Strength</th>
<th>Modulus of Elasticity</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stainless Steel 316L</td>
<td>290 MPa (42 ksi)</td>
<td>558 MPa (81 ksi)</td>
<td>193 GPa (27,992 ksi)</td>
</tr>
<tr>
<td>Stainless Steel 410</td>
<td>410 MPa (59 ksi)</td>
<td>483 MPa (70 ksi)</td>
<td>210 GPa (30,458 ksi)</td>
</tr>
<tr>
<td>ASTM A252 Grade 1</td>
<td>205 MPa (30 ksi)</td>
<td>345 MPa (50 ksi)</td>
<td>210 GPa (30,458 ksi)</td>
</tr>
<tr>
<td>ASTM A252 Grade 2</td>
<td>240 MPa (35 ksi)</td>
<td>414 MPa (60 ksi)</td>
<td>210 GPa (30,458 ksi)</td>
</tr>
<tr>
<td>ASTM A252 Grade 3</td>
<td>310 MPa (45 ksi)</td>
<td>4550 MPa (66 ksi)</td>
<td>210 GPa (30,458 ksi)</td>
</tr>
<tr>
<td>ASTM A333 Grade 6</td>
<td>240 MPa (35 ksi)</td>
<td>415 MPa (60 ksi)</td>
<td>200 GPa (29,008 ksi)</td>
</tr>
<tr>
<td>ASTM A514 Grade F</td>
<td>590 MPa (86 ksi)</td>
<td>800 MPa (116 ksi)</td>
<td>210 GPa (30,458 ksi)</td>
</tr>
</tbody>
</table>
Table 28. Material Properties (continued).

<table>
<thead>
<tr>
<th>Material</th>
<th>Shear Modulus of Elasticity</th>
<th>Poisson's Ratio</th>
<th>Endurance Limit</th>
<th>Density</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>GPa</td>
<td>ksl</td>
<td>MPa</td>
<td>ksl</td>
</tr>
<tr>
<td>Stainless Steel 316L</td>
<td>77</td>
<td>11,168</td>
<td>0.5</td>
<td>279</td>
</tr>
<tr>
<td>Stainless Steel 410</td>
<td>80</td>
<td>11,603</td>
<td>0.5</td>
<td>242</td>
</tr>
<tr>
<td>ASTM A252 Grade 1</td>
<td>80</td>
<td>11,603</td>
<td>0.5</td>
<td>173</td>
</tr>
<tr>
<td>ASTM A252 Grade 2</td>
<td>80</td>
<td>11,603</td>
<td>0.5</td>
<td>207</td>
</tr>
<tr>
<td>ASTM A252 Grade 3</td>
<td>80</td>
<td>11,603</td>
<td>0.5</td>
<td>228</td>
</tr>
<tr>
<td>ASTM A333 Grade 6</td>
<td>0.3</td>
<td>207</td>
<td></td>
<td></td>
</tr>
<tr>
<td>ASTM A514 Grade F</td>
<td></td>
<td>400</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

It is important to note that high-strength steels do not necessarily offer weight savings in this application. Gerwick (2007) emphasizes that when buckling and vibration are key concerns, stiffness, rather than yield stress, governs failure. Also, the harsh Muskeget Channel environment mandates that any steel used must be specified for low-temperature service to prevent premature fatigue failure, especially in welded joints. Particularly, it must show high Charpy impact values at low temperatures. The American Petroleum Institute classifies steels in groups I—III by strength and classes C—A by toughness. For the reasons above, a Group I, Class A steel is desirable, so quotes were sought for ASTM A333 Grade 6 tubing (American Petroleum Institute, 1993).

Corrosion

According to Corus (2005), steel in the continually immersed zone “acquires a protective blanket of corrosion products and marine growth” and exhibits an average mean corrosion rate of 0.035mm/year/side. Tomlinson (2008) echoes this in saying that in the continuously immersed zone, piles should use bare steel or cathodic protection. He quotes a study by Morley and Bruce (1983) of steel piles in the UK that
reports an average loss of thickness of 0.05 mm/year in the immersion zone, with a 95% maximum probable rate of 0.14 mm/year. Furthermore, he points out that if the interior of a tubular member is sufficiently isolated from the external environment, the oxygen in the trapped seawater will quickly be “used up in the early corrosion process, leaving none to maintain the corrosion.”

A sacrificial anode system is often an economical anti-corrosion measure. Such a system could be implemented simply by using commercially available zinc shaft collars around the truss members or by mounting zinc bars between members, as is sometimes practiced in offshore structures. While it would require occasional maintenance, the anodes would simply need to be replaced when they are observed to be depleted. As to concerns of biofouling, Blackwood et al. (2010) published their findings that “anodes remain effective even after being completely coated with biofouling”.

If cathodic protection is used in conjunction with high-strength steels, Billingham et al. (2003) emphasizes that great care must be taken to mitigate hydrogen cracking. Gerwick (2007) adds that cathodic protection is prohibited in areas where the flow of water is restricted.

**Costing**

**Materials and Construction**

Material costs for the gravity foundation were based on shore forming and estimating the city-factored cost of concrete from RS Means (2011) with Overhead and Profit included, assuming that concrete would account for the entire weight of the structure (i.e. neglecting the possibility of using sediment as fill.) This included
forming materials and anti-corrosion treatment. The components of the foundation are shown for each platform size in Table 29. The cost of the anti-scour skirt was determined from the per-pound estimate of ASTM A252 Gr. 3.

<table>
<thead>
<tr>
<th>Turbine Diameter</th>
<th>4 m (14 ft)</th>
<th>9 m (29 ft)</th>
<th>13 m (43 ft)</th>
<th>17 m (57 ft)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material Type</td>
<td>Percent by Weight</td>
<td>Weight, tons</td>
<td>Percent by Weight</td>
<td>Weight, tons</td>
</tr>
<tr>
<td>Cement Portland, type I,II</td>
<td>11%</td>
<td>26</td>
<td>56</td>
<td>89</td>
</tr>
<tr>
<td>Sand+stone, Crushed</td>
<td>26%</td>
<td>62</td>
<td>126</td>
<td>210</td>
</tr>
<tr>
<td>Aggregate bank gravel, --coarse loaded at site</td>
<td>67%</td>
<td>157</td>
<td>338</td>
<td>543</td>
</tr>
<tr>
<td>Water</td>
<td>16%</td>
<td>38</td>
<td>81</td>
<td>130</td>
</tr>
<tr>
<td>Air</td>
<td>6%</td>
<td>14</td>
<td>30</td>
<td>49</td>
</tr>
<tr>
<td>Total</td>
<td>100%</td>
<td>235</td>
<td>505</td>
<td>810</td>
</tr>
</tbody>
</table>

The material cost of the mechanical tubing that constitutes the support structure was estimated from a quote from American Steel for ASTM A333 Gr. 6 mechanical tubing, also on a per-pound basis. Although it is recommended that the steel mounting structure be left bare as a cost-saving measure, the cost of corrosion protection was included in the estimate in case it is deemed necessary. This cost was based on a quote from L.B. Foster (2010) for marine-grade epoxy coating over the exterior surface area of the platform. Welding costs were estimated from a quote supplied to Jeff Byrne for his V2 design (2013). The material costs were subtracted from a quote that included deck beams and welded mechanical tubing and the remainder was assumed to be the welding cost, which was reduced to dollars per pound of tubing. (While this is clearly an over-estimate of the fraction which is welding cost, it is also worth noting that the quotes used to estimate the welding cost were for A36 steel, which may be easier to weld than ASTM A333 Gr. 6).
**Installation**

The installation cost was based on a crane barge of sufficient capacity to carry and install the foundation, in use for 7 (seven) days with 4 (four) days of mobilization/demobilization. Quotes were obtained from Manson Construction of Los Angeles (2010) and Weeks Marine of New Jersey (2012) for crane barges of various capacities. Alternative installation methods are under investigation. Gerwick (2007) describes detailed steps for constructing a gravity foundation “raft” consisting of a concrete honeycomb structure whose buoyancy is moderated by controlling the amounts of compressed air in each cell. The steel anti-scour skirt could also be utilized for buoyancy during installation. Such methods will bear further investigation in the more detailed phase of design. The estimated costs of a Fixed Gravity Foundation platform for the range of turbine sizes are shown in Table 30.

<table>
<thead>
<tr>
<th>Turbine Diameter</th>
<th>4 m (14 ft)</th>
<th>9 m (29 ft)</th>
<th>13 m (43 ft)</th>
<th>17 m (57 ft)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Base</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Cement</td>
<td>$0.12</td>
<td>111,137</td>
<td>178,220</td>
<td>311,631</td>
</tr>
<tr>
<td>Aggregate-coarse</td>
<td>$23.50</td>
<td>338</td>
<td>543</td>
<td>949</td>
</tr>
<tr>
<td>Anti-corrosion Treatment</td>
<td>$10.05</td>
<td>338</td>
<td>543</td>
<td>949</td>
</tr>
<tr>
<td>Forming</td>
<td>$10.95</td>
<td>250</td>
<td>401</td>
<td>700</td>
</tr>
<tr>
<td>Anti-scour Skirt</td>
<td>$1.87</td>
<td>1,848</td>
<td>2,467</td>
<td>3,501</td>
</tr>
<tr>
<td><strong>Support Structure</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>ASTM A333 Grade 6 Steel</td>
<td>$28-</td>
<td>2,667</td>
<td>10,667</td>
<td>24,000</td>
</tr>
<tr>
<td>Welding</td>
<td>$1.38</td>
<td>10,667</td>
<td>24,000</td>
<td>42,667</td>
</tr>
<tr>
<td>Anti-corrosion Coating</td>
<td>$4.02</td>
<td>444</td>
<td>1,000</td>
<td>1,778</td>
</tr>
<tr>
<td><strong>Installation Vessel</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mobilization/ Demobilization</td>
<td>$18,000-</td>
<td>4</td>
<td>4</td>
<td>4</td>
</tr>
<tr>
<td>Working</td>
<td>$45,000-</td>
<td>7</td>
<td>7</td>
<td>7</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td>$422,929</td>
<td>$598,074</td>
<td>$686,425</td>
<td>$828,854</td>
</tr>
</tbody>
</table>

O&P, shipping included throughout
Site work not included
A telescoping pile gravity foundation platform, illustrated in Figure 37, was considered for the following advantages:

- The turbine mounting would reside below most surface traffic.
- The platform would be accessible from the surface for turbine installation, maintenance, and retrieval.

Disadvantages include:

- Scour would have to be considered.
- Moving underwater parts are vulnerable to biofouling, etc.

The Telescoping Gravity platform would comprise a concrete base, four telescoping piles rigidly connected by a truss structure, and a turbine mounting.
structure. The uppermost section of each telescoping pile would act as a buoyancy chamber. Devices would be deployed at mid-depth (with telescoping piles collapsed to minimum length, as shown in the left half of Figure 37) for the duration of testing. For installation, service, and retrieval, the turbine mounting platform would be raised above the surface (as shown in the right half of Figure 37). This would be accomplished by forcing air into each of the uppermost pile sections. Rate of ascent and final vertical position would be controlled by mechanical control arms, shown in Figure 37. Dimensions of a gravity foundation were obtained by designing the base as a simple box-shaped concrete structure with sufficient weight and dimensions to resist tipping and sliding. The pile sections were designed to resist the axial loading, bending moment, and shearing forces. Costs were estimated for constructing the structure onshore, and quotes were obtained for crane barges that could install the platform.

**Specific Design Criteria**

- The foundation must prohibit tipping or sliding. A minimum factor of safety of 3 was specified for both failure modes.
- The foundation must resist cracking, e.g. during installation. That is, it must have a bending safety factor of 5 (not including reinforcing steel) under worst-case bending.
- Each pile section must have a safety factor of 2 against material yielding and 5 against shearing. (The high shearing safety factor is to prevent local buckling in the wall of the hollow cylinder.)
- Maximum horizontal deflection when platform is fully extended must be less
than 0.3 m (1 ft.), neglecting the stiffening cross members.

The following assumptions were used in the analysis:

- A tipping condition is that in which the entire normal force acts at the rear lower corner of the level foundation.

- Friction can be sufficiently modeled by Coulomb’s Law of Friction, in which the maximum friction force equals the normal force times a coefficient of friction between the two surfaces.

- The weight of the turbine is neglected for the tipping analysis. (This ensures that the foundation will be secure even if used to test a lightweight, high drag turbine.)

- A 2.5 m/s (5 knots) current is uniform over the entire depth.

Foundation Design

Governing Equations

A Free Body Diagram of the fixed gravity foundation platform is shown in Figure 38 and the variables used therein are given in Table 31.
Figure 38. Free Body Diagram of gravity foundation for Telescoping Pile platform. The normal force $N$ is located at the down-current edge of the platform base to model the onset of tipping. Drag force on the truss structure was assumed negligible compared to drag on the turbine, telescoping piles, and foundation.

Table 31. Statics variables for gravity foundation for Telescoping Pile platform

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$W_f$</td>
<td>Foundation Weight (Dry)</td>
</tr>
<tr>
<td>$W_t$</td>
<td>Turbine Weight</td>
</tr>
<tr>
<td>$D_f$</td>
<td>Foundation Drag</td>
</tr>
<tr>
<td>$D_t$</td>
<td>Turbine Drag</td>
</tr>
<tr>
<td>$r_f$</td>
<td>Distance from bottom to Turbine Drag</td>
</tr>
<tr>
<td>$B_f$</td>
<td>Buoyant Force</td>
</tr>
<tr>
<td>$H_f$</td>
<td>Foundation Height</td>
</tr>
<tr>
<td>$L_f$</td>
<td>Foundation Length</td>
</tr>
<tr>
<td>$b_f$</td>
<td>Foundation Breadth</td>
</tr>
</tbody>
</table>

To prevent sliding, the maximum friction force must equal or exceed the total drag. That is,

$$F_{f_{\text{max}}} \geq F_f = D_t + D_f + D_p.$$  \hspace{1cm} (96)

Sliding was modeled using the Coulomb model of friction,
\[ F_{\text{max}} = N \mu_s \]  

(97)

where \( F_{\text{max}} \) is the maximum applicable friction force and \( \mu_s \) is a static coefficient of friction for sand-gravel, given by AASHTO (Taly, 2010) as 0.55 for concrete on medium sand, gravel. The anti-sliding, anti-tipping effects of the anti-scour skirt were ignored in this analysis. Neglecting the weight of the turbine, the normal force, \( N \), is the weight of the foundation minus the weight of displaced water, so that

\[ N = W_f + W_p - B_f - B_p, \]  

(98)

where \( W_f \) is the total weight of the piles and \( B_p \) is the buoyant force on the piles.

The weight of the foundation is

\[ W_f = \rho_c g (L_f w_f H_f), \]  

(99)

where \( \rho_c \) is the density of the concrete, and \( w_f \) is the width of the foundation. The buoyancy force is

\[ B_f = \rho g (L_f w_f H_f). \]  

(100)

Here drag on the foundation is given by

\[ D_f = \frac{1}{2} \rho C_D U^2 A, \]  

(101)

in which the coefficient of drag, \( C_D \), is given by Hoerner (1965) as 1.05 for a block on a flat surface.

To prevent tipping, moments applied to the platform about point \( C \) must balance, so that

\[ \sum M_C = 0, \]  

(102)

so that
The equals sign pertains to the onset of tipping, shown in Figure 30; the “greater than”
sign corresponds to the platform resting solidly on the sediment, with normal force $N$
acting to the right of point $C$. As a result, there exist two factors of safety for the
foundation: A tipping safety factor and a sliding safety factor, given by the maximum
resisting moment over the design moment, and the maximum friction force over the
drag force, respectively. Thus, the safety factors are

$$ SF_{\text{tip}} = \frac{(W_f - B_f + W_p) L_f}{D_f + D_t + D_p r_f^2} $$

(104)

and

$$ SF_{\text{slide}} = \frac{(W_f - B_f + W_p) h_s}{D_f + D_t + D_p} $$

(105)

Additionally, the low tensile strength of concrete necessitates a consideration of
bending due to an uneven seafloor. A free-body diagram of the worst possible loading
case is shown in Figure 39.

![Free Body Diagram (in vertical) of Telescoping Gravity base supported at a single
point subject only to the larger vertical forces.](image)

The maximum bending stress in the base was approximated by the formula

$$ \sigma_{\text{bend}} = \frac{Mc}{I} = \frac{\left(\frac{L_f}{2} - L_s\right)\left(\frac{W_f}{2} - \frac{W_p}{2}\right)\left(\frac{H_f}{2}\right)}{\frac{1}{12}b_f H_f^2} $$

(106)

Here $M$ is the maximum bending moment, $c$ is the distance from the neutral axis, and

Reproduced with permission of the copyright owner. Further reproduction prohibited without permission.
$l$ is the area moment of inertia of the beam. Neglecting the ameliorating effects of steel rebar, the associated safety factor is concrete's Ultimate Tensile Strength divided by the maximum bending stress.

$$SF_{\text{tensile}} = \frac{\sigma_{\text{UTS}}}{\sigma_{\text{bend}}}$$

(107)

Using the above analysis, the foundation dimensions were iterated for each turbine scale-up factor of interest to minimize weight under the constraints listed in Table 32, using a Generalized Reduction Gradient (GRG) Nonlinear forward difference solver in the Microsoft Excel® Solver package.

<table>
<thead>
<tr>
<th>Table 32. Gravity Foundation base constraints.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Platform will not slide: $SF_{\text{slide}} \geq 3$</td>
</tr>
<tr>
<td>Platform will not crack in bending: $SF_{\text{tensile}} \geq 5$</td>
</tr>
<tr>
<td>Platform will not tip: $SF_{\text{tip}} \geq 3$</td>
</tr>
<tr>
<td>Platform cannot be excessively narrow: $w_f \geq 0.85L_f$</td>
</tr>
</tbody>
</table>

**Scour**

Anti-scour structure must be designed with care. Rocker (1985) cites an example in which hinged concrete scour protection slabs were broken off of their main structure by deep water wave-induced scour in 30 m (100 ft.) of water. However, Gerwick points to successful installations of steel skirts around gravity foundations that reduce scour while increasing the foundation's ability to resist sliding. Another method, currently being implemented for offshore wind gravity foundations at the Thornton Bank Offshore Wind Farm off the Belgian coast uses layers of coarse sediment and gravel to minimize scour (Terra et Aqua). A steel scour skirt was designed using \( \frac{1}{4}'' \) ASTM 252 Gr. 1 steel.
Support Structure

Statics

Each of the four telescoping members was modeled as a series of concentric beams, as shown in Figure 40. The variables used in the Free Body Diagrams are listed in Table 33.

Table 33. Statics variables for pile section analysis.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wpn</td>
<td>Weight (dry) of pile section</td>
</tr>
<tr>
<td>Wt</td>
<td>Turbine Weight</td>
</tr>
<tr>
<td>Dpn</td>
<td>Drag on pile section n</td>
</tr>
<tr>
<td>Dn</td>
<td>Turbine Drag</td>
</tr>
<tr>
<td>V1n, V2n</td>
<td>Horizontal force from pile section n+1</td>
</tr>
<tr>
<td>R1n, R2n</td>
<td>Horizontal reaction force from pile section n-1</td>
</tr>
<tr>
<td>R1, R2</td>
<td>Vertical reaction force on pile</td>
</tr>
<tr>
<td>r1</td>
<td>Distance from bottom to Turbine Drag</td>
</tr>
<tr>
<td>d1</td>
<td>Pile diameter</td>
</tr>
<tr>
<td>l_{ovp}</td>
<td>Overlap between sections</td>
</tr>
</tbody>
</table>
This system was analyzed using singularity equations, in the method described by, for example, Beer et al. (2012). In this analysis the shear forces are integrated along the axis of the beam to find the bending moment distribution, which is integrated to find the slope of the beam along the axis, which is integrated to find the total deflection. The constants of integration arising in the process are determined by the boundary conditions. These boundary conditions are that the slope and deflection at the end of each pile section must match that in the adjacent section at the same vertical location. Additionally, the displacement and slope at the base of pile section \( n=1 \) (the bottommost section) are zero.

This analysis was implemented in MATLAB®. Due to geometric conditions and the given water depth, it was decided that each telescoping pile would consist of two pile sections. A pile wall thickness of 2 inches was specified and the outer diameter of the smallest pile section was iterated until the maximum bending and shear stresses in the pile sections were acceptable. (The inner diameter of each subsequent pile section was set to the outer diameter of the pile section above, with the same wall thickness.) Bending stress was calculated as

\[
\sigma_B = \frac{Mc}{I_y} = \frac{M(d_{an})}{I_y}.
\]  

Here \( M \) is the local bending moment, \( c \) is the distance from the neutral axis, and \( I_y \) is the area moment of inertia about the neutral axis. Shear stress was calculated as

\[
\tau = \frac{VQ}{tl} = 2\frac{V}{A}
\]  

for a thin walled circular cylinder. Here \( V \) is the shear force; \( Q \) is the first moment of the cross-sectional area above the neutral axis; \( t \) is twice the wall thickness; \( I \) is the...
moment of inertia of the entire cross-section; and $A$ is the area of the cross-section. Safety factors for bending and shear were defined respectively as

$$SF_B = \frac{\sigma_y}{\sigma_{B_{\text{max}}}},$$  \hspace{1cm} (110)$$

and

$$SF_Y = \frac{\sigma_y}{\tau_{\text{max}}},$$  \hspace{1cm} (111)$$

Pile section diameters were iterated until $SF_B \geq 2$ and $SF_Y \geq 5$. For each design, maximum deflection was checked to ensure that it did not exceed the specified 0.3 m (1 ft.) Figure 41 shows the shear, and bending distributions, the slope, and the deflection along the telescoping pile—for a turbine size of 13 m (43 ft.)—of a system in which each telescoping pile consists of two sections. The final design results are shown in Table 34.
Figure 41. Shear force, bending moment, total slope and horizontal deflection along a telescoping pile with 2 sections. Values are calculated along each pile section. Dashed lines denote values associated with a lower pile in an overlap region. The lack of apparent slope in the shear-force diagram shows that the distributed drag on each pile section is small compared to the effect of turbine drag.

Table 34. Telescoping pile section diameters.

<table>
<thead>
<tr>
<th>Turbine Diameter (representative in-stream axis)</th>
<th>Pile Diameter, m</th>
<th>Section 1 (lower)</th>
<th>Section 2 (upper)</th>
</tr>
</thead>
<tbody>
<tr>
<td>m</td>
<td>ft</td>
<td></td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>14</td>
<td>0.60</td>
<td>0.50</td>
</tr>
<tr>
<td>9</td>
<td>29</td>
<td>1.15</td>
<td>1.05</td>
</tr>
<tr>
<td>13</td>
<td>43</td>
<td>1.70</td>
<td>1.60</td>
</tr>
<tr>
<td>17</td>
<td>57</td>
<td>2.20</td>
<td>2.10</td>
</tr>
</tbody>
</table>

Although the telescoping pile sections were designed to take the full load of the turbine, bracing members were added to the lower section to stiffen the structure. This is important because any curvature in the sections could increase friction significantly. The stiffening elements in the structure were designed using Circular
Hollow Section (CHS) truss members because of their high resistance to buckling and comparatively low drag coefficient, shown in Figure 42 and Figure 43 respectively.

![Figure 42. Comparison of the masses of hollow and open sections under compression in relation to the loading (European Steel Design Education Programme, 1994).](image)

![Figure 43. Approximate drag coefficient curves for single section (smooth surface) members with various corner radii, r, depending on the Reynolds number, Re (European Steel Design Education Programme, 1994).](image)

**Dynamics—Vibration**

Neglecting vibrational issues in marine structures can be catastrophic (Tomlinson, 2008, p. 413). According to standard beam theory the natural frequency of a beam is found to be

\[
 f_N = \frac{K'}{L^2} \sqrt{\frac{EI}{M}},
\]

where \(E\) is Young's Modulus, \(I\) is the area moment of inertia, \(M\) is mass/unit length of the beam (including the mass of the water contained in the beam and the mass of the displaced water), \(L\) is the length of the beam, and \(K'\) is a factor of 3.56 for the first mode of vibration in members with both ends fixed. The Strouhal number can then be used to find the fluid velocity at which the frequency of vortex shedding will match the member's natural frequency. This critical velocity is given by

101
\[ V_{\text{crit}} = K f_N d_o. \] (113)

where \( d_o \) is the outer diameter of the member and \( K \) is given in Table 35.

<table>
<thead>
<tr>
<th>1.2</th>
<th>Onset of in-line motion</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.0</td>
<td>Maximum in-line motion</td>
</tr>
<tr>
<td>3.5</td>
<td>Onset of cross-flow motion</td>
</tr>
<tr>
<td>5.5</td>
<td>Maximum cross-flow motion</td>
</tr>
</tbody>
</table>

Table 35. Coefficients for modes of Vortex-Induced Vibration (Tomlinson, 2008).

According to Mittal and Kumar (1999), “in-line oscillations are significant only if the mass of the cylinder is not too large compared with the mass of the surrounding fluid it displaces.” Since the mass of the cylinder is on the order of the mass of the surrounding fluid, it was assumed that only cross-flow motion is significant. The above equations can be solved to find the required combined relative stiffness, \( CRS \) of any member of a given length subjected to a given fluid velocity, defined by

\[ CRS = d_o \sqrt{EI/M} = \frac{V_{\text{crit}} L^2}{K K'}. \] (114)

To prevent cross-flow vibration, the required combined relative stiffness of each structural member (based on its length) was computed and a cross-section with sufficient CRS (including a safety factor of two) was chosen.

**Material**

Because of its unique resistance to corrosion in seawater even after being welded, 316L stainless steel was originally investigated (Specialty Steel Industry of North America (SSINA), 2011). However, its cost was prohibitive. Several steels commonly found in marine applications were considered for this unique structure. Their properties are listed alongside alternatives in Table 36.

<table>
<thead>
<tr>
<th>Material</th>
<th>Yield Strength</th>
<th>Ultimate Tensile Strength</th>
<th>Modulus of Elasticity</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>MPa</td>
<td>ksi</td>
<td>MPa</td>
</tr>
<tr>
<td>Stainless Steel 316L</td>
<td>290</td>
<td>42</td>
<td>558</td>
</tr>
<tr>
<td>Stainless Steel 410</td>
<td>410</td>
<td>59</td>
<td>483</td>
</tr>
<tr>
<td>ASTM A252 Grade 1</td>
<td>205</td>
<td>30</td>
<td>345</td>
</tr>
<tr>
<td>ASTM A252 Grade 2</td>
<td>240</td>
<td>35</td>
<td>414</td>
</tr>
<tr>
<td>ASTM A252 Grade 3</td>
<td>310</td>
<td>45</td>
<td>4550</td>
</tr>
<tr>
<td>ASTM A333 Grade 6</td>
<td>240</td>
<td>35</td>
<td>415</td>
</tr>
<tr>
<td>ASTM A514 Grade F</td>
<td>590</td>
<td>86</td>
<td>800</td>
</tr>
</tbody>
</table>

Table 36. Material Properties (continued).

<table>
<thead>
<tr>
<th>Material</th>
<th>Shear Modulus of Elasticity</th>
<th>Poisson's Ratio</th>
<th>Endurance Limit</th>
<th>Density kg/m³</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>GPa</td>
<td>ksi</td>
<td>MPa</td>
<td>ksi</td>
</tr>
<tr>
<td>Stainless Steel 316L</td>
<td>77</td>
<td>11,168</td>
<td>0.5</td>
<td>279</td>
</tr>
<tr>
<td>Stainless Steel 410</td>
<td>80</td>
<td>11,603</td>
<td>0.5</td>
<td>242</td>
</tr>
<tr>
<td>ASTM A252 Grade 1</td>
<td>80</td>
<td>11,603</td>
<td>0.5</td>
<td>173</td>
</tr>
<tr>
<td>ASTM A252 Grade 2</td>
<td>80</td>
<td>11,603</td>
<td>0.5</td>
<td>207</td>
</tr>
<tr>
<td>ASTM A252 Grade 3</td>
<td>80</td>
<td>11,603</td>
<td>0.5</td>
<td>228</td>
</tr>
<tr>
<td>ASTM A333 Grade 6</td>
<td>0.3</td>
<td></td>
<td></td>
<td>207</td>
</tr>
<tr>
<td>ASTM A514 Grade F</td>
<td></td>
<td></td>
<td></td>
<td>400</td>
</tr>
</tbody>
</table>

It is important to note that high-strength steels do not necessarily offer weight savings in this application. Gerwick (2007) emphasizes that when buckling and vibration are key concerns, stiffness, rather than yield stress, governs failure. Also, the harsh Muskeget Channel environment mandates that any steel used must be specified for low-temperature service to prevent premature fatigue failure, especially in welded joints. Particularly, it must show high Charpy impact values at low temperatures. The American Petroleum Institute classifies steels in groups I—III by strength and classes C—A by toughness. For the reasons above, a Group I, Class A steel is desirable, so quotes were sought for ASTM A333 Grade 6 tubing (American Petroleum Institute, 1993).
According to Corus (2005), steel in the continually immersed zone “acquires a protective blanket of corrosion products and marine growth” and exhibits an average mean corrosion rate of 0.035 mm/year/side. Tomlinson (2008) echoes this in saying that in the continuously immersed zone, piles should use bare steel or cathodic protection. He quotes a study by Morley and Bruce (1983) of steel piles in the UK that reports an average loss of thickness of 0.05 mm/year in the immersion zone, with a 95% maximum probable rate of 0.14 mm/year. Furthermore, he points out that if the interior of a tubular member is sufficiently isolated from the external environment, the oxygen in the trapped seawater will quickly be “used up in the early corrosion process, leaving none to maintain the corrosion.”

A sacrificial anode system is often an economical anti-corrosion measure. Such a system could be implemented simply by using commercially available zinc shaft collars around the truss members or by mounting zinc bars between members, as is sometimes practiced in offshore structures. While it would require occasional maintenance, the anodes would simply need to be replaced when they are observed to be depleted. As to concerns of biofouling, Blackwood et al. (2010) published their findings that “anodes remain effective even after being completely coated with biofouling”.

If cathodic protection is used in conjunction with high-strength steels, Billingham et al. (2003) emphasizes that great care must be taken to mitigate hydrogen cracking. Gerwick (2007) adds that cathodic protection is prohibited in areas where the flow of water is restricted.
Lifting

The telescoping piles will be raised by buoyant forces. Air will be pumped into the upper pile section (or released from a compressed air tank). The position and rate of ascent will be moderated by scissor arms (as shown in Figure 37).

As an alternative, water was also considered as the pumping fluid. A seawater pump could be mounted on either the foundation or the rising platform and used to pump pressurized seawater into the pile sections to effectively form a seawater hydraulic system. The U.S. Navy and other researchers have been investigating comparable systems in recent years (Krutz & Chua, 2004; Jokela & Kunsemiller, 1996), but the Muskeget Channel system would require much less pressure than most other systems because of the large cross-sectional areas of the piles. However, a seawater system would have to overcome major difficulties. For instance, the interface between pile sections would have to remain sealed while subjected to large lateral forces in a corrosive environment.

Costing

Materials/Construction

Material costs for the gravity foundation were based on shore forming and estimating the city-factored cost of concrete from RS Means (2011) with Overhead and Profit included, assuming that concrete would account for the entire weight of the base structure (i.e. neglecting the possibility of using sediment as fill.) This included forming materials and anti-corrosion treatment. The components of the foundation are shown for each platform size in Table 37. The cost of the anti-scour skirt was determined from the per-pound estimate of ASTM A252 Gr. 3.
Table 37. Gravity base weight breakdown.

<table>
<thead>
<tr>
<th>Turbine Diameter</th>
<th>4 m (14 ft)</th>
<th>9 m (29 ft)</th>
<th>13 m (43 ft)</th>
<th>17 m (57 ft)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Material</strong></td>
<td><strong>Type</strong></td>
<td><strong>Percent by Weight</strong></td>
<td><strong>Weight, tons</strong></td>
<td></td>
</tr>
<tr>
<td>Cement</td>
<td>Portland, type I, II Sand+stone, crushed bank gravel, coarse loaded at site</td>
<td>11%</td>
<td>25</td>
<td>60</td>
</tr>
<tr>
<td>Aggregate</td>
<td></td>
<td>67%</td>
<td>155</td>
<td>364</td>
</tr>
<tr>
<td>Water</td>
<td></td>
<td>16%</td>
<td>37</td>
<td>87</td>
</tr>
<tr>
<td>Air</td>
<td></td>
<td>6%</td>
<td>14</td>
<td>33</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td></td>
<td>100%</td>
<td>231</td>
<td>544</td>
</tr>
</tbody>
</table>

The material cost of the piles was estimated from a quote from L.B. Foster for ASTM A252 Gr. 3 steel piles on a per-pound basis. Although it is recommended that the steel mounting structure be left bare as a cost-saving measure, the cost of corrosion protection was included in the estimate in case it is deemed necessary. This cost was based on a quote from L.B. Foster (2010) for marine-grade epoxy coating over the exterior surface area of the platform. The cost of the cross bracing and turbine mounting structure was scaled from an estimate by J.F. White (2011) of $50,000 for the corresponding elements of the pile foundations for a turbine size of 13 m (43 ft.). This cost was assumed to vary linearly with the turbine size.

The cost of the variable-buoyancy system was estimated by using the per-pound cost of ASTM A252 Gr. 3 as the cost of the integrated ballast tanks, the cost of ASTM A333 Gr. 6 for the necessary piping, and the per-pound welding cost as above. The price of twenty (20) stainless steel 2” ball valves with remote activation was obtained from Swagelok. The most expensive type was used in order to compensate for other valves, etc. not included in the cost analysis. (Corrosion in these components will need to be given careful consideration during the detailed design phase because...
stainless steel acts as the sacrificial anode to most structural steels.)

**Installation**

The installation cost was based on a crane barge of sufficient capacity to carry and install the foundation, in use for 7 (seven) days with 4 (four) days of mobilization/demobilization. Quotes were obtained from Manson Construction of California (2010) and Weeks Marine of New Jersey (2012) for crane barges of various capacities. The estimated costs of a Telescoping Pile platform for various turbine capacities are shown in Table 38.

**Table 38. Cost of Telescoping Gravity Foundation Platform.**

<table>
<thead>
<tr>
<th>Turbine Diameter</th>
<th>Unit cost</th>
<th>Unit</th>
<th>4 m (14 ft) Quantity</th>
<th>Cost</th>
<th>9 m (29 ft) Quantity</th>
<th>Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Base</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Cement</td>
<td>$0.12</td>
<td>lb</td>
<td>50,828</td>
<td>$6,245</td>
<td>119,585</td>
<td>$14,694</td>
</tr>
<tr>
<td>Aggregate-coarse</td>
<td>$23.50</td>
<td>ton</td>
<td>155</td>
<td>$3,638</td>
<td>364</td>
<td>$8,558</td>
</tr>
<tr>
<td>Anti-corrosion Treatment</td>
<td>$10.05</td>
<td>CY.</td>
<td>114</td>
<td>$1,148</td>
<td>269</td>
<td>$2,701</td>
</tr>
<tr>
<td>Forming</td>
<td>$10.95</td>
<td>SFCA</td>
<td>1,091</td>
<td>$11,943</td>
<td>1,827</td>
<td>$20,005</td>
</tr>
<tr>
<td>Anti-scarb Skirt</td>
<td>$1.87</td>
<td>lb</td>
<td>31,465</td>
<td>$58,913</td>
<td>48,263</td>
<td>$90,364</td>
</tr>
<tr>
<td>Support Structure</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Piles</td>
<td>$1.87</td>
<td>lb</td>
<td>166,248</td>
<td>$311,270</td>
<td>348,537</td>
<td>$652,575</td>
</tr>
<tr>
<td>Anti-corrosion Coating</td>
<td>$4.02</td>
<td>ft^2</td>
<td>202</td>
<td>$812</td>
<td>423</td>
<td>$1,702</td>
</tr>
<tr>
<td>Platform</td>
<td>$1,852</td>
<td>EA</td>
<td>1</td>
<td>$1,852</td>
<td>1</td>
<td>$14,815</td>
</tr>
<tr>
<td>Variable Buoyancy</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Steel (Pressure Vessel, Bulkheading)</td>
<td>$1.87</td>
<td>lb</td>
<td>10,390</td>
<td>$19,454</td>
<td>21,784</td>
<td>$40,786</td>
</tr>
<tr>
<td>ASTM A333 Gr. 6 Piping</td>
<td>$1.64</td>
<td>lb</td>
<td>8312</td>
<td>$13,646</td>
<td>17427</td>
<td>$28,609</td>
</tr>
<tr>
<td>Welding</td>
<td>1</td>
<td></td>
<td>$1,852</td>
<td>1</td>
<td>$14,815</td>
<td></td>
</tr>
<tr>
<td>2&quot; Ball Valves</td>
<td>$1,370</td>
<td>EA</td>
<td>20</td>
<td>$27,400</td>
<td>20</td>
<td>$27,400</td>
</tr>
<tr>
<td>Installation</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mobilization/Demobilization</td>
<td>$24,000</td>
<td>days</td>
<td>4</td>
<td>$96,000</td>
<td>4</td>
<td>$96,000</td>
</tr>
<tr>
<td>Working</td>
<td>$52,500</td>
<td>days</td>
<td>7</td>
<td>$367,500</td>
<td>7</td>
<td>$367,500</td>
</tr>
<tr>
<td>Total</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>$921,673</td>
<td></td>
<td>$1,380,524</td>
<td></td>
</tr>
</tbody>
</table>

O&P, shipping included throughout
Site work not included
Table 38. Cost of Gravity Foundation Platform (continued).

<table>
<thead>
<tr>
<th>Turbine Diameter</th>
<th>Unit cost</th>
<th>13 m (43 ft)</th>
<th>17 m (57 ft)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Unit</td>
<td>Quantity</td>
<td>Cost</td>
</tr>
<tr>
<td><strong>Base</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Cement</td>
<td>$0.12 lb</td>
<td>202,484</td>
<td>$24,880</td>
</tr>
<tr>
<td>Aggregate-coarse</td>
<td>$23.50 ton</td>
<td>617</td>
<td>$14,491</td>
</tr>
<tr>
<td>Anti-corrosion Treatment</td>
<td>$10.05 C.Y.</td>
<td>455</td>
<td>$4,573</td>
</tr>
<tr>
<td>Forming</td>
<td>$10.95 SFCA</td>
<td>2,551</td>
<td>$27,929</td>
</tr>
<tr>
<td>Anti-scour Skirt</td>
<td>$1.87 lb</td>
<td>62,802</td>
<td>$117,585</td>
</tr>
<tr>
<td><strong>Support Structure</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Piles</td>
<td>$1.87 lb</td>
<td>530,827</td>
<td>$993,881</td>
</tr>
<tr>
<td>Anti-corrosion Coating</td>
<td>$4.02 ft^2</td>
<td>644</td>
<td>$2,592</td>
</tr>
<tr>
<td>Platform</td>
<td>$1,852 EA</td>
<td>1</td>
<td>$50,000</td>
</tr>
<tr>
<td><strong>Variable Buoyancy</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Steel (Pressure Vessel, Bulkheading)</td>
<td>$1.87 lb</td>
<td>33,177</td>
<td>$62,118</td>
</tr>
<tr>
<td>ASTM A333 Gr. 6 Piping</td>
<td>$1.64 lb</td>
<td>26541</td>
<td>$43,573</td>
</tr>
<tr>
<td>Welding</td>
<td>1</td>
<td>$50,000</td>
<td>1</td>
</tr>
<tr>
<td>2&quot; Ball Valves</td>
<td>$1,370 EA</td>
<td>20</td>
<td>$27,400</td>
</tr>
<tr>
<td><strong>Installation</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mobilization/Demobilization</td>
<td>$24,000 days</td>
<td>4</td>
<td>$96,000</td>
</tr>
<tr>
<td>Working</td>
<td>$52,500 days</td>
<td>7</td>
<td>$367,500</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td></td>
<td>$1,882,521</td>
<td></td>
</tr>
</tbody>
</table>

O&P, shipping included throughout
Site work not included

It is important to note that the largest crane barge for which a quote was obtained is not sufficient for lifting the combined weight of the platform for turbine sizes 13 m (43 ft.) or 17 m (57 ft.) Alternative installation methods could include temporary buoyancy and towing to the site. Gerwick (2010) describes detailed steps for constructing a gravity foundation “raft” consisting of a concrete honeycomb structure whose buoyancy is moderated by controlling the amounts of compressed air in each cell. The telescoping piles or the steel anti-scour skirt could also be utilized for buoyancy during installation.
A subsurface, four-pile group with mounting structure at mid depth, illustrated in Figure 44, was considered for the following advantages:

- The platform would be below most surface traffic.
- A pile group offers greater resistance to lateral loading.
- A pile group would reduce the required depth of penetration into the seafloor.

Disadvantages include:

- Platform installation, maintenance, and turbine installation/retrieval would likely be expensive.
- Scour would need to be considered.

This platform concept would comprise four fixed piles connected by stiffening.
(cross-bracing) members. This platform would remain in a fixed position and developers would be responsible for installing their devices on the platform at mid-depth.

**Specific Design Criteria**

- The foundation must place the turbine hub 15 m (49 ft.) above the seafloor.
- Each pile must be able to act as an independent cantilevered beam. (This overpredicts the diameter of the piles required.)
- The structure must be able to sustain a 2.5 m/s (5 knots) current uniform over the entire depth. (This over-estimates the total drag.)
- Steel pipe piles must be used (based on offshore oil and other industry practice).
- 1" (0.025 m) wall thickness must be used (to accommodate in-situ welding, as per Tomlinson (2008)).
- Each pile must have a safety factor against yielding of at least two.
- The pile group must have a safety factor against uplift of at least five.

The following assumptions were made for the analysis:

- The maximum bending moment exists at the seafloor (i.e. the top of the sediment layer).
- Due to the lack of information on the sediment composition below the seafloor, two possible cases were assumed:
  1. The depth of the sand-gravel mixture is sufficient to secure the piles.
  2. Bedrock exists just below the seafloor.
Governing Equations—Statics

Pile analysis began by designing a single pile as a beam cantilevered from the seafloor, of sufficient diameter to withstand the forces applied by the current and one quarter of those on the mounted turbine. A simple Free-Body Diagram is shown in Figure 45 and the associated variables are listed in Table 39.

![Free-Body Diagram of a single pile in the four-pile group.](image)

Horizontal equilibrium requires that

\[ R_x = D_p + \frac{D_t}{4} \]  

And vertical equilibrium requires that

\[ R_z = W_p + \frac{W_t}{4} \]  

Balancing moments about the base of the pile yields

\[ M_{xy} = D_p \frac{h}{2} + \frac{D_t}{4} r_t, \]  

where

Table 39. Pile Statics variables.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>( r_t )</td>
<td>Turbine Hub from Seafloor</td>
</tr>
<tr>
<td>( h )</td>
<td>Length of Pile</td>
</tr>
<tr>
<td>( W_p )</td>
<td>Pile Weight</td>
</tr>
<tr>
<td>( W_t )</td>
<td>Turbine Weight</td>
</tr>
<tr>
<td>( D_p )</td>
<td>Pile Drag</td>
</tr>
<tr>
<td>( D_t )</td>
<td>Turbine Drag</td>
</tr>
<tr>
<td>( R_z )</td>
<td>Vertical Reaction Force</td>
</tr>
<tr>
<td>( R_x )</td>
<td>Horizontal Reaction Force</td>
</tr>
<tr>
<td>( M_{xy} )</td>
<td>Reaction Moment</td>
</tr>
<tr>
<td>( D_o )</td>
<td>Pile Diameter</td>
</tr>
</tbody>
</table>
\[ D_p = \frac{1}{2} \rho C_D A U_a^2, \quad (118) \]

in which

\[ A = d_o h. \quad (119) \]

Using these forces and moments, the stresses at the base of the pile were calculated.

Axial stress is given by

\[ \sigma_A = -\frac{F_{Rx}}{A_c}, \quad (120) \]

where the cross-sectional area is

\[ A_c = d_o \pi t, \quad (121) \]

where \( t \) is the thickness of the pile wall. Maximum bending stress, acting at the outer edge of the beam is

\[ \sigma_B = \frac{M_{xc}}{I_y} = \frac{M_{Ry}(\frac{dx}{2})}{I_y}. \quad (122) \]

where the area moment of inertia of a pipe is

\[ I_y = \frac{\pi d^4 t}{8}. \quad (123) \]

Maximum shear stress in a thin-walled hollow cylinder, acting at the neutral axis of the beam is given by

\[ \tau_{max} = 2\frac{V}{A}. \quad (124) \]

where \( V \) is the shearing force.

Assuming a long pile, shear stress (being a minimum at the outer edge of the beam) is disregarded, so that the maximum normal stress in the pile is the sum of bending and stresses at the downstream outer edge of the pile,
\[
\sigma_{\text{max}} = \sigma_B + \sigma_A.
\]  
(125)

A factor of safety for the pile then, based on compressive failure of the pile material, is

\[
SF = \frac{\sigma_{\text{yldcomp}}}{\sigma_{\text{max}}}. 
\]  
(126)

Because of the interdependence between pile diameter and drag on the pile, the diameter was iterated until the safety factor equaled 2 for the chosen material. Results are combined with limits due to soil mechanics and are summarized in Table 39. The material chosen was ASTM A252 Grade 3 steel, which is a common material for marine piling. Its properties are listed in Table 40, alongside alternatives.

<table>
<thead>
<tr>
<th>Material</th>
<th>Yield Strength</th>
<th>Ultimate Tensile Strength</th>
<th>Modulus of Elasticity</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>MPa</td>
<td>ksi</td>
<td>MPa</td>
</tr>
<tr>
<td>Stainless Steel 316L</td>
<td>290</td>
<td>42</td>
<td>558</td>
</tr>
<tr>
<td>Stainless Steel 410</td>
<td>410</td>
<td>59</td>
<td>483</td>
</tr>
<tr>
<td>ASTM A252 Grade 1</td>
<td>205</td>
<td>30</td>
<td>345</td>
</tr>
<tr>
<td>ASTM A252 Grade 2</td>
<td>240</td>
<td>35</td>
<td>414</td>
</tr>
<tr>
<td><strong>ASTM A252 Grade 3</strong></td>
<td><strong>310</strong></td>
<td><strong>45</strong></td>
<td><strong>4550</strong></td>
</tr>
</tbody>
</table>

Table 40. Material properties (ASTM International, 2010; 2011; 2009).

<table>
<thead>
<tr>
<th>Material</th>
<th>Shear Modulus of Elasticity</th>
<th>Poisson's Ratio</th>
<th>Endurance Limit</th>
<th>Density</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>GPa</td>
<td>ksi</td>
<td></td>
<td>MPa</td>
</tr>
<tr>
<td>Stainless Steel 316L</td>
<td>77</td>
<td>11,168</td>
<td>0.5</td>
<td>279</td>
</tr>
<tr>
<td>Stainless Steel 410</td>
<td>80</td>
<td>11,603</td>
<td>0.5</td>
<td>242</td>
</tr>
<tr>
<td>ASTM A252 Grade 1</td>
<td>80</td>
<td>11,603</td>
<td>0.5</td>
<td>173</td>
</tr>
<tr>
<td>ASTM A252 Grade 2</td>
<td>80</td>
<td>11,603</td>
<td>0.5</td>
<td>207</td>
</tr>
<tr>
<td><strong>ASTM A252 Grade 3</strong></td>
<td><strong>80</strong></td>
<td><strong>11,603</strong></td>
<td><strong>0.5</strong></td>
<td><strong>228</strong></td>
</tr>
</tbody>
</table>

Table 40. Material properties (continued).
A pile’s bearing ability is broken down into its resistance to vertical and lateral loading. Both analyses are described below.

**Vertical Capacity**

For Case 2, in which bedrock exists just below the seafloor, it is assumed that the pile material will fail before the bedrock (Das, 2000) if the pile is embedded a depth of 3 diameters. For Case 1, in which there is a sufficient depth of sand-gravel to secure the pile, several methods are available for calculating a pile’s vertical bearing capacity. Meyerhof’s method (as described by Das) was used. This method calculates the point bearing capacity of the pile tip and the friction bearing capacity of the pile.

Given the pile diameter and vertical reaction force found in the mechanics analysis, the pile depth into the sediment was iterated until the required safety factor was met. A safety factor of seven was imposed because of the high uncertainty involved with soil analysis. The soil parameters used in this analysis, along with intermediate values and the results of the Meyerhof calculations for sample pile dimensions are given in Table 41.
Table 41. Pile vertical capacity sample calculations, using Meyerhof’s method.

<table>
<thead>
<tr>
<th>Sediment Type</th>
<th>Sand</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pile Diameter</td>
<td>Do</td>
</tr>
<tr>
<td>Vertical Reaction Force</td>
<td>298,330</td>
</tr>
<tr>
<td>Pile Depth in Sediment</td>
<td>d</td>
</tr>
<tr>
<td>Soil Density</td>
<td></td>
</tr>
<tr>
<td>Point Bearing</td>
<td></td>
</tr>
<tr>
<td>End Condition</td>
<td>end</td>
</tr>
<tr>
<td>Point Area</td>
<td>Ap</td>
</tr>
<tr>
<td>Unit Weight</td>
<td>gamma</td>
</tr>
<tr>
<td>Soil Friction Angle</td>
<td>phi</td>
</tr>
<tr>
<td>Bearing Capacity Factor</td>
<td>N * q</td>
</tr>
<tr>
<td>Effective Vertical Stress</td>
<td>q'</td>
</tr>
<tr>
<td>Limiting Point Resistance</td>
<td>Q_l</td>
</tr>
<tr>
<td>Skin Friction</td>
<td></td>
</tr>
<tr>
<td>Effective Earth Pressure Coefficient</td>
<td>K</td>
</tr>
<tr>
<td>Average Effective Overburden Pressure</td>
<td>σm bar'O</td>
</tr>
<tr>
<td>Soil-pile friction angle</td>
<td>δ'</td>
</tr>
<tr>
<td>Critical Depth</td>
<td>L'</td>
</tr>
<tr>
<td>Embedment Ratio</td>
<td>L-D</td>
</tr>
<tr>
<td>Ultimate Skin Resistance</td>
<td>Q_s</td>
</tr>
<tr>
<td>Vertical Load Safety Factor:</td>
<td>SF_vert</td>
</tr>
</tbody>
</table>

Legend:

Environmental parameters
Iterated values
Design inputs
Calculated values
*Pile diameter and vertical reaction force are determined from the mechanics analysis.

**Lateral Capacity**

The ultimate lateral bearing capacity of a pile is significantly more complicated than the vertical capacity. As with the vertical capacity, a pile in bedrock is assumed to fail in material before the supporting rock gives way. But for Case 1, which entirely assumes a sand-gravel mix, Brom’s method was used. This method is
described for soils below the water table in the DOT Federal Highway Administration publication, *Design and Construction of Driven Pile Foundation* (1998). Given the pile diameter and lateral reaction force found in the mechanics analysis, the pile depth into the sediment was iterated until Brom's analysis showed that the pile was "long." This means that the pile material will yield before the soil. The inputs and results of each step of that analysis are shown in Table 42 for sample pile dimensions.
### Table 42. Pile lateral capacity using Brom's method, sample calculation.

<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Pile Diameter</td>
<td>Do 0.88 m</td>
</tr>
<tr>
<td>Pile Depth In Sediment</td>
<td>d 6.5 m</td>
</tr>
<tr>
<td>Soil Type</td>
<td>Cohesionless</td>
</tr>
<tr>
<td>Coefficient of Horizontal Subgrade Reaction</td>
<td>Kh 10857000 N/m²³</td>
</tr>
<tr>
<td>Kh adjusted for loading, soil conditions</td>
<td>Kh_cyc 5428500 N/m²³</td>
</tr>
<tr>
<td>Eccentricity of Applied Load</td>
<td>ec 15 m</td>
</tr>
<tr>
<td>Shape Factor</td>
<td>Cs 1.3</td>
</tr>
<tr>
<td>Resisting Moment of Pile</td>
<td>My 6391111 N-m</td>
</tr>
<tr>
<td>Length Factor</td>
<td>eta 0.374 /m</td>
</tr>
<tr>
<td>Dimensionless Length Factor</td>
<td>etaD 2.4</td>
</tr>
<tr>
<td>Pile Length Type</td>
<td>Intermediate</td>
</tr>
<tr>
<td>Rankine passive pressure coefficient for cohesionless soil</td>
<td>Kp 3.7</td>
</tr>
<tr>
<td>Average Effective Unit Weight</td>
<td>gamma' 6955 N</td>
</tr>
<tr>
<td>Cohesion</td>
<td>cu 0</td>
</tr>
<tr>
<td>Long</td>
<td></td>
</tr>
<tr>
<td>Dimensionless Factor</td>
<td>My/(b²<em>gamma</em>Kp) 394.1</td>
</tr>
<tr>
<td>Dimensionless Factor</td>
<td>ec/b</td>
</tr>
<tr>
<td>Dimensionless Load Factor</td>
<td>Qu/(Kp<em>b³</em>gamma) 16.8</td>
</tr>
<tr>
<td>Short</td>
<td></td>
</tr>
<tr>
<td>Dimensionless Factor, D/b</td>
<td>7.3</td>
</tr>
<tr>
<td>Dimensionless Factor, ec/D</td>
<td>2.3</td>
</tr>
<tr>
<td>Dimensionless Load Factor</td>
<td>Qu/(Kp<em>b³</em>gamma) 2.3</td>
</tr>
<tr>
<td>Ultimate Lateral Load</td>
<td>Qu 1,364,266 N</td>
</tr>
<tr>
<td>Lateral Safety Factor</td>
<td>SF_lat 7.9</td>
</tr>
<tr>
<td>Recommended Safety Factor</td>
<td>NY_SF 2.5</td>
</tr>
<tr>
<td>Max allowable load</td>
<td>Qm 545707</td>
</tr>
<tr>
<td>Deflection at point of loading</td>
<td>y 0.001 m</td>
</tr>
</tbody>
</table>

**Legend:**
- **Environmental parameters**
- **Iterated values**
- **Design inputs**
- **Values calculated from formulae**

*Pile diameter and the reaction bending moment are determined from the mechanics analysis.*
**Uplift**

In addition to lateral and vertical bearing capacity, uplift must be considered to prevent pull-out in a pile group. A basic, worst-case view of this scenario—treating the pile group as rigid and neglecting the weight of a turbine and the reaction moment on each pile—is shown in Figure 46.

![Free Body Diagram of pile group.](image)

Figure 46. Free Body Diagram of pile group.

Summing the moments about point B shows that equilibrium is maintained if

\[
R_{ZA}l + W_pl = \frac{h}{2} 2D_p + \frac{h}{2} D_t. \tag{127}
\]

So the Ultimate Skin Resistance \(Q_s\) in Table 41 must exceed \(R_{ZA}\) in this analysis to prevent pull-out. This requirement was quantified by defining an uplift safety factor,

\[
SF_{uplift} = \frac{Q_s}{R_{ZA}}. \tag{128}
\]

which was found to be greater than five for each design.
**Scour**

Anti-scour structure must be designed with care. Rocker (1985) indicates that a steel skirt extending one diameter beyond each pile can adequately protect against scour and such features were included in the design of the pile foundation platform. Planned biofouling and scour experiments in the Muskeget Channel this summer will inform an investigation of the economics and effectiveness of various methods.

**Results**

The design processes above were integrated in the following procedure for each maximum turbine size specified:

- A pile wall thickness (1 inch for all designs) was selected (to accommodate in-situ welding).
- The diameter of each pile was iterated until the mechanics analysis showed that the required safety against yielding was met.
- The depth to which the pile would be driven into the soil was iterated until requirements for both vertical and lateral capacity were satisfied.
- Uplift was analyzed to ensure that the specified safety factor was satisfied.
- Vortex-Induced Vibration (VIV) was analyzed and it was found that the piles as designed would not experience significant vibration unless subjected to velocities at least twice those expected in the channel. Also, cross-bracing members were designed with sufficient Combined Relative Stiffness such that VIV would not occur unless subjected to the same velocities.

The results of the design of the four-pile platform are shown in Table 43.
Table 43. Four-pile platform parameters.

<table>
<thead>
<tr>
<th>Turbine Diameter</th>
<th>4 m (14 ft)</th>
<th>9 m (29 ft)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pile Depth in Sediment</td>
<td>5 m 18 ft</td>
<td>8 m 26 ft</td>
</tr>
<tr>
<td>Turbine Distance from Bottom</td>
<td>15 m 49 ft</td>
<td>15 m 49 ft</td>
</tr>
<tr>
<td>Pile Height from Bottom</td>
<td>15 m 49 ft</td>
<td>15 m 49 ft</td>
</tr>
</tbody>
</table>

**Pile**

<table>
<thead>
<tr>
<th>Material</th>
<th>ASTM A252 Grade 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of Piles</td>
<td>4</td>
</tr>
<tr>
<td>Diameter of Pile</td>
<td>0.34 m 1.12 ft</td>
</tr>
<tr>
<td>Thickness of Pile</td>
<td>0.0254 m 1.00 in</td>
</tr>
<tr>
<td>Mass of Pile</td>
<td>4,366 kg 9,625 lbm</td>
</tr>
<tr>
<td>Compression Safety Factor</td>
<td>2.0</td>
</tr>
<tr>
<td>Max Stress / Endurance Limit</td>
<td>0.7</td>
</tr>
<tr>
<td>Pile Type in Soil</td>
<td>Long</td>
</tr>
<tr>
<td>Velocity for Transverse Vibration</td>
<td>1.4 m/s</td>
</tr>
</tbody>
</table>

Table 43. Four-pile platform parameters (continued).

<table>
<thead>
<tr>
<th>Turbine Diameter</th>
<th>13 m (43 ft)</th>
<th>17 m (57 ft)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pile Depth in Sediment</td>
<td>10 m 32 ft</td>
<td>12 m 38 ft</td>
</tr>
<tr>
<td>Turbine Distance from Bottom</td>
<td>15 m 49 ft</td>
<td>15 m 49 ft</td>
</tr>
<tr>
<td>Pile Height from Bottom</td>
<td>15 m 49 ft</td>
<td>15 m 49 ft</td>
</tr>
</tbody>
</table>

**Pile**

<table>
<thead>
<tr>
<th>Material</th>
<th>ASTM A252 Grade 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of Piles</td>
<td>4</td>
</tr>
<tr>
<td>Diameter of Pile</td>
<td>0.95 m 3.12 ft</td>
</tr>
<tr>
<td>Thickness of Pile</td>
<td>0.0254 m 1.00 in</td>
</tr>
<tr>
<td>Mass of Pile</td>
<td>14,786 kg 32,598 lbm</td>
</tr>
<tr>
<td>Compression Safety Factor</td>
<td>2.0</td>
</tr>
<tr>
<td>Max Stress / Endurance Limit</td>
<td>0.7</td>
</tr>
<tr>
<td>Pile Type in Soil</td>
<td>Long</td>
</tr>
<tr>
<td>Velocity for Transverse Vibration</td>
<td>7.8 m/s</td>
</tr>
</tbody>
</table>

**Legend:**
- Iterated values
- Design inputs
- Calculated values

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Costing

Several Marine Contractors, including J.F. White, Pihl U.S., and Sea and Shore, were contacted for rough estimates of the cost for installation of two 56" (1.42 m) piles or four 24" (0.61 m) piles (corresponding to early designs for the 13 m (43 ft) turbine platform). J.F. White proposed the following installation procedure:

A marine piling operation would be mobilized and consist of a 54' x 180' barge with a 200 TN lattice boom crane set on top. All construction materials, templates and equipment would also be placed on the barge. The barge would be mobilized from a main land marine facility and towed to the location of work. The sequence of work would be to construct templates, install piles and set platforms. In the event that bedrock is encountered above the proposed pile tip elevation, JFW would use a "down the hole hammer" to remove the bedrock and create a rock socket. Concrete and reinforcing steel would then be placed in the toe of the pile to provide the required embedment and stability of the pile system.

The contractor provided estimates for installing piles of the aforementioned size for both the case in which bedrock exists just beneath the seafloor and that in which there is sufficient sediment overburden to hold the piles, while strongly recommending that soil testing be conducted before installation. These estimates were scaled under the assumption that the entire installation cost was proportional to the volume of sediment removed by drilling or enclosed by the pile. The estimate carried $50,000 to construct the platform to which the turbine would mount. This was assumed to vary linearly with the size of the turbine. The results from this analysis are shown in Table 44.
### Table 44. Cost of Fixed Four-pile Platform.

<table>
<thead>
<tr>
<th>Turbine Diameter</th>
<th>Unit Cost Unit</th>
<th>4 m (14 ft) Qty.</th>
<th>Cost</th>
<th>9 m (29 ft) Qty.</th>
<th>Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Foundation</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Piles, Installed</td>
<td>$68,553 EA</td>
<td>4</td>
<td>$274,215</td>
<td>4</td>
<td>$956,860</td>
</tr>
<tr>
<td>Anti-corrosion Coating</td>
<td>$4.02 ft²</td>
<td>740</td>
<td>$2,980</td>
<td>1562</td>
<td>$6,286</td>
</tr>
<tr>
<td>Anti-scour Mat</td>
<td>$1.87 lb</td>
<td>1854</td>
<td>$3,471</td>
<td>2271</td>
<td>$4,253</td>
</tr>
<tr>
<td><strong>Support Structure</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Platform</td>
<td>$1,852-$118,519 EA</td>
<td>1</td>
<td>$1,852</td>
<td>1</td>
<td>$14,815</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td></td>
<td>$282,518</td>
<td></td>
<td>$982,215</td>
<td></td>
</tr>
</tbody>
</table>

O&P, shipping included throughout
Site work not included

### Table 44. Cost of Fixed Four-pile Platform (continued).

<table>
<thead>
<tr>
<th>Turbine Diameter</th>
<th>Unit Cost Unit</th>
<th>13 m (43 ft) Qty.</th>
<th>Cost</th>
<th>17 m (57 ft) Qty.</th>
<th>Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Foundation</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Piles, Installed</td>
<td>$68,553 EA</td>
<td>4</td>
<td>$1,991,978</td>
<td>4</td>
<td>$3,204,822</td>
</tr>
<tr>
<td>Anti-corrosion Coating</td>
<td>$4.02 ft²</td>
<td>2508</td>
<td>$10,092</td>
<td>3565</td>
<td>$14,345</td>
</tr>
<tr>
<td>Anti-scour Mat</td>
<td>$1.87 lb</td>
<td>2693</td>
<td>$5,042</td>
<td>3119</td>
<td>$5,841</td>
</tr>
<tr>
<td><strong>Support Structure</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Platform</td>
<td>$1,852-$118,519 EA</td>
<td>1</td>
<td>$50,000</td>
<td>1</td>
<td>$118,519</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td></td>
<td>$2,057,112</td>
<td></td>
<td>$3,343,526</td>
<td></td>
</tr>
</tbody>
</table>

O&P, shipping included throughout
Site work not included

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A surface-piercing, self-raising, two-pile foundation, as shown in Figure 47, was considered for the following advantages:

- The platform would greatly reduce the cost and ease of turbine installation/retrieval and maintenance by bringing the device to the sea surface for service.
- A permanent, visible infrastructure presence could be useful for navigation and for public relations (Barrett, 2012).
Disadvantages include the following:

- Marine traffic would need to avoid the surface presence.
- A surface presence could raise objections over alterations to the existing viewscape.
- The surface structure could become a target for vandalism.
- Scour would need to be considered.

This platform concept, shown in Figure 47, would include a mounting structure raised and lowered along two upright piles which provide the integrity of the overall structure. A working platform would rigidly connect the two piles. A winch, wire-rope, and chain system (described later) would provide the lifting capability.

**Specific Criteria**

- The foundation must place the turbine hub 15 m (49 ft.) above the seafloor.
- In addition to current loading, the platform must survive a 15 m (49 ft.) storm wave.
- Each pile must be able to act as an independent cantilevered beam. (This over-predicts the diameter of the piles required.)
- The structure must be able to sustain a 2.5 m/s (5 knots) current uniform over the entire depth. (This over-estimates the total drag.)
- Steel pipe piles must be used (based on offshore oil and other industry practice).
- 2” (0.05 m) wall thickness must be used (in order to allow for in-situ welding).
- Each pile must have a safety factor against yielding of at least two.

The following assumptions were made for the analysis:

- The maximum bending moment exists at the seafloor.
Due to the lack of information on the sediment composition below the seafloor, two possible cases were assumed:

1. The depth of the sand-gravel mixture is sufficient to secure the piles.
2. Bedrock exists just below the seafloor.

**Governing Equations—Statics**

Piles were analyzed as beams cantilevered from the seafloor, of sufficient diameter to withstand the forces applied by the current and the mounted turbine. A simple Free-Body Diagram is shown in Figure 48 and the associated nomenclature is given in Table 45.

![Free-Body Diagram of a single pile in a 2-pile group](image)

**Table 45. Pile Statics Variables**

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>( r_t )</td>
<td>Turbine Hub from Seafloor</td>
</tr>
<tr>
<td>( h )</td>
<td>Length of Pile</td>
</tr>
<tr>
<td>( W_p )</td>
<td>Pile Weight</td>
</tr>
<tr>
<td>( W_t )</td>
<td>Turbine Weight</td>
</tr>
<tr>
<td>( D_p )</td>
<td>Pile Drag</td>
</tr>
<tr>
<td>( D_t )</td>
<td>Turbine Drag</td>
</tr>
<tr>
<td>( D_w )</td>
<td>Wind Drag on Pile</td>
</tr>
<tr>
<td>( R_x )</td>
<td>Vertical Reaction Force</td>
</tr>
<tr>
<td>( R_h )</td>
<td>Horizontal Reaction Force</td>
</tr>
<tr>
<td>( M_b )</td>
<td>Reaction Moment</td>
</tr>
<tr>
<td>( d_p )</td>
<td>Pile Diameter</td>
</tr>
</tbody>
</table>

**Figure 48. Free-Body Diagram of a single pile in a 2-pile group**

Horizontal equilibrium requires that

\[
R_x = D_p + D_w + \frac{D_t}{2}.
\]  

(129)

And vertical equilibrium requires that
Balancing moments about the base of the pile yields

\[ M_b = D_p \frac{h}{2} + D_w h + \frac{D_k}{2} t, \]  

(131)

where

\[ D_p = \frac{1}{2} \rho C_D A_s U^2, \]  

(132)

and

\[ D_w = \frac{1}{2} \rho_{air} C_D A_w U_w^2. \]  

(133)

Here the submerged area is given by

\[ A_s = h d_o, \]  

(134)

and the area exposed to the wind is given by

\[ A_E = h_2 d_o, \]  

(135)

where \( h_2 \) is the height of the pile above the waterline. A value of \( C_D = 0.7 \) was used to calculate both \( D_w \) and \( D_p \). Furthermore, a design wind speed of 15 m/s (30 knots) was used to calculate the wind drag on the pile. Using these forces and moments, the stresses at the base of the pile were calculated. Axial stress is given by

\[ \sigma_A = -\frac{R_z}{A_c}, \]  

(136)

where the cross-sectional area is

\[ A_c = d_o \pi t. \]  

(137)

Here \( t \) is the thickness of the pile wall. Maximum bending stress, acting at the outer edge of the beam is

\[ \sigma_B = \frac{M_{yc}}{I_y} = \frac{M_{RY}(\frac{d_o}{2})}{I_y}, \]  

(138)

where the area moment of inertia of a pipe is
Maximum shear stress in a hollow cylinder, acting at the neutral axis of the beam is given by

$$I_y = \frac{\pi d^4 t}{8}.$$ (139)

where \( V \) is the shearing force. Assuming a long pile, shear stress (being a minimum at the outer edge of the beam) is disregarded, so that the maximum normal stress in the pile is the sum of bending and stresses at the downstream outer edge of the pile,

$$\sigma_{\text{max}} = \sigma_B + \sigma_A.$$ (141)

A factor of safety for the pile then, based on compressive failure of the pile material, is

$$SF = \frac{\sigma_{\text{yield comp}}}{\sigma_{\text{max}}}.$$ (142)

Because of the interdependence between pile diameter and drag on the pile, the diameter was iterated until the safety factor equaled 2 for the chosen material, ASTM A252 Grade 3 steel, which is a common material for marine piling. Results were combined with soil mechanics and wave loading analysis, and are given in Table 52. Steel properties are listed in Table 46, alongside alternatives.

<table>
<thead>
<tr>
<th>Material</th>
<th>Yield Strength</th>
<th>Ultimate Tensile Strength</th>
<th>Modulus of Elasticity</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>MPa</td>
<td>ksi</td>
<td>MPa</td>
</tr>
<tr>
<td>Stainless Steel 316L</td>
<td>290</td>
<td>42</td>
<td>558</td>
</tr>
<tr>
<td>Stainless Steel 410</td>
<td>410</td>
<td>59</td>
<td>483</td>
</tr>
<tr>
<td>ASTM A252 Grade 1</td>
<td>205</td>
<td>30</td>
<td>345</td>
</tr>
<tr>
<td>ASTM A252 Grade 2</td>
<td>240</td>
<td>35</td>
<td>414</td>
</tr>
<tr>
<td>ASTM A252 Grade 3</td>
<td>310</td>
<td>45</td>
<td>455</td>
</tr>
</tbody>
</table>
Table 46. Material properties (continued).

<table>
<thead>
<tr>
<th>Material</th>
<th>Shear Modulus of Elasticity</th>
<th>Poisson’s Ratio</th>
<th>Endurance Limit</th>
<th>Density</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>GPa</td>
<td>ksl</td>
<td>MPa</td>
<td>ksl</td>
</tr>
<tr>
<td>Stainless Steel 316L</td>
<td>77</td>
<td>11,168</td>
<td>0.5</td>
<td>279</td>
</tr>
<tr>
<td>Stainless Steel 410</td>
<td>80</td>
<td>11,603</td>
<td>0.5</td>
<td>242</td>
</tr>
<tr>
<td>ASTM A252 Grade 1</td>
<td>80</td>
<td>11,603</td>
<td>0.5</td>
<td>173</td>
</tr>
<tr>
<td>ASTM A252 Grade 2</td>
<td>80</td>
<td>11,603</td>
<td>0.5</td>
<td>207</td>
</tr>
<tr>
<td>ASTM A252 Grade 3</td>
<td>80</td>
<td>11,603</td>
<td>0.5</td>
<td>228</td>
</tr>
</tbody>
</table>

In addition to static loading, low-cycle fatigue due to tidal cycles was also considered. The endurance limit listed in Table 46 is the uncorrected limit, calculated as 50% of the ultimate tensile strength. To properly consider the effect of fatigue, the corrected endurance limit must be used, defined by the equation

\[ S' = C_{load} C_{size} C_{surf} C_{temp} C_{reliab} S', \]

whose terms are listed in Table 47 using the method for fully-reversed bending described by Norton (2006).

Table 47. Endurance limit correction factors.

<table>
<thead>
<tr>
<th>( S' )</th>
<th>( C_{load} )</th>
<th>( C_{size} )</th>
<th>( C_{surf} )</th>
<th>( C_{temp} )</th>
<th>( C_{reliab} )</th>
<th>( S' )</th>
</tr>
</thead>
<tbody>
<tr>
<td>228 MPa</td>
<td>1</td>
<td>0.6</td>
<td>0.41</td>
<td>1</td>
<td>0.81</td>
<td>46 MPa</td>
</tr>
</tbody>
</table>

Using the corrected endurance limit, a S-N diagram (showing material strength, S, as a function of loading cycles, N) was created to show the effect of fatigue. An example (for the case of a platform capable of deploying a 9 m diameter turbine) is shown in Figure 49.
In this figure, $S_m$ is the strength at $10^3$ cycles, given as $S_m = 0.9S_{ut}$, where $S_{ut}$ is the ultimate stress. The strength of the material is taken to decrease logarithmically from $S_m$ to $S_e$ between $10^3$ and $10^6$ cycles. A design life of 20 years corresponds to 14,600 tidal cycles over which the turbine and structure drag loading will be fully reversed.

In the example shown, the calculated allowable stress at this point in the life cycle will be 326 MPa. Since the fully-reversed bending stress is 155 MPa, this results in a fatigue safety factor of about 2.2, which is higher than the safety factor of 2 required against yielding in the static analysis.

**Governing Equations—Wave Loading**

Since it would be permanently fixed to the seafloor, the Two-pile Foundation Platform must be capable of surviving a storm wave event. The design wave used was equal to the largest single wave observed at nearby Block Island, RI, during the 2012 Super-storm Sandy, with a height of 15 m and a period of 14 s (Seymour et al., 2012). The forces and moments that this wave would exert on the platform structure were determined as follows.

The problem of wave forces on a vertical cylinder is well known (see, for
example, Techet (2004)). Morrison’s Equation states that the total force in the
direction of wave propagation, $F_w$ is

$$F_w(t) = \rho C_m V \dot{U} + \frac{1}{2} \rho C_d A |U|,$$  \hfill (144)

where $\rho$ is the fluid density, $C_m$ is the cylinder’s mass coefficient, $V$ is the volume, $U$
is the fluid velocity, and $C_d$ is the coefficient of drag. The mass coefficient for a
cylinder in oscillating fluid flow is found from Table 48.

**Table 48. Coefficients of mass and drag (Clauss et al., 1992).**

<table>
<thead>
<tr>
<th>KC</th>
<th>Re&lt;10^5</th>
<th>Re&gt;10^5</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$C_d$</td>
<td>$C_m$</td>
</tr>
<tr>
<td>&lt; 10</td>
<td>1.2</td>
<td>2.0</td>
</tr>
<tr>
<td>\geq 10</td>
<td>1.2</td>
<td>1.5</td>
</tr>
</tbody>
</table>

Here the Reynolds number is defined as

$$Re = \frac{d_0 U_a}{v},$$  \hfill (145)

where $U_a$ is the amplitude of the wave velocity and $v$ is the dynamic viscosity of the
fluid. In this application, $Re$ was generally on the order of $10^7$. Also, $KC$ is the
Keulegan-Carpenter number, given as

$$KC = \frac{U_a T}{d_o},$$  \hfill (146)

where $T$ is the wave period. $KC$ was generally on the order of 20 or higher. Thus, a
mass coefficient of 1.5 was used.

The fluid velocity, $U$, is given by

$$U = U_{wave} + U_{current},$$  \hfill (147)

where $U_{current}$ is assumed constant and, from linear wave theory,

$$U_{wave} = \frac{H}{2} \sigma \left( \frac{\cosh(h+z)}{\sinh k h} \right) \sin kx \sin \sigma t.$$  \hfill (148)

Here $H$ is the wave height; $\sigma$ is the wave radian frequency; $k$ is the wave number; $h$
is the water depth; $z$ is the vertical coordinate with $z=0$ corresponding to the mean water
level; and \( t \) is time.

The largest stress in each pile (modeled as a cantilevered beam) will be the bending stress at the base. Thus, the overturning moment from each of four contributions must be considered:

- Viscous loading on the pile.
- Viscous loading on the turbine.
- Inertial loading on the pile.
- Inertial loading on the turbine.

The maximum bending moment at the base of the pile due to the viscous drag on the pile, \( M_D \), is found by integrating the product of the maximum drag force on the pile and the moment arm from the seafloor to the surface. That is,

\[
M_{D_{max}} = \int_{-h}^{0} \frac{1}{2} \rho C_d d \phi \left( \frac{H}{2} \left( \frac{\cosh(k(h+z))}{\sinh kh} + U_{current} \right) \right)^2 (h+z) dz \quad (149)
\]

\[
= \frac{1}{16(-1+e^{2hk})^2} (e^{2hk} \rho C_d d \phi [H^2 \sigma^2 (1 + 2h^2 k^2 - \cosh 2hk + 2hk \sinh 2hk) + 16u_{cur} (H \sigma [-hk + hk \cosh 2hk + 2 \sinh hk - \sinh 2hk] + h^2 k^2 u_{cur} \sinh^2 2hk)])
\]

The maximum bending moment at the base of the pile due to inertial wave forcing, \( M_I \), is found by integrating the maximum inertial force on the pile from the seafloor to the mean water level (because the surface elevation is zero when horizontal fluid acceleration is at a maximum). Using linear wave theory and integrating the first term of Equation (144) gives

\[
M_I = \int_{-h}^{0} \rho C_m V \left( \frac{H}{2} \sigma^2 \frac{\cosh(k(h+z))}{\sinh kh} \right) z dz \quad (150)
\]

\[
= \rho C_m \pi d \phi \frac{H}{2} \sigma^2 \frac{kh \sinh(kh) - \cosh(kh)}{k^2 \sinh(kh)}
\]

The maximum bending moment at the base of the pile due to viscous loading on the turbine is

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The inertial force on the turbine, $F_{it}$, was assumed to be that of a flat disc multiplied by the solidity of the turbine, $S$, which is the actual projected area of the device divided by its outline area. (A value of $S = 0.3$ was used.) Thus,

$$ F_{it} = M' \frac{d\gamma}{dt} S. $$

(152)

This is assumed to be valid if the turbine under test is be braked. Since extreme wave events can generally be forecast days in advance, this should always be the case during such events. In the above equation, $M'$, is given by Lamb (1932) as

$$ M' = \frac{8}{3} \rho \pi a^3, $$

(153)

where $a$ is the radius of the disc (or the radius of the in-stream axis turbine with the same area as the device mounted).

From Equation (148) it is evident that the maximum values of $U_{\text{ave}}$ and $U_{\text{wave}}$ occur 90 degrees out of phase. Therefore, maximum viscous and inertial loads cannot be simply summed to find the maximum total load. Rather, the maximum total bending moment takes the form

$$ M = M_y \sin(\sigma t) + M_i \cos(\sigma t), $$

(154)

where $t$ represents time and, again, $M_y$ and $M_i$ are magnitudes. Setting the time derivative of this equation to zero shows that the maximum combined moment occurs at time $t = \sigma^{-1} \tan(M_y/M_i)$. Using this value of $t$ in Equation (154) and simplifying yields

$$ M_{\text{max}} = \sqrt{M_y^2 + M_i^2}, $$

(155)

(This result can also be obtained by observing that sine and cosine are orthogonal functions). The bending stress induced by this total moment was calculated using the
method described in the statics analysis. Pile dimensions were iterated until a safety factor of 2 was achieved.

**Governing Equations—Soil Mechanics**

A pile’s bearing ability is broken down into its resistance to vertical and lateral loading. Both analyses are described below.

**Vertical Capacity**

For Case 2, in which bedrock exists just below the seafloor, it is assumed that the pile material will fail before the bedrock (Das, 2000) if the pile is embedded a depth of 3 diameters. For Case 1, in which there is a sufficient depth of sand-gravel to secure the pile, several methods are available for calculating a pile’s vertical bearing capacity. Meyerhof’s method (as described by Das) was used. This method calculates the point bearing capacity of the pile tip and the friction bearing capacity of the pile. Given the pile diameter and vertical reaction force found in the mechanics analysis, the pile depth into the sediment was iterated until the required safety factor was met. A safety factor of seven was imposed because of the high uncertainty involved with soil analysis. The soil parameters used in this analysis, along with the intermediate values and the results of the Meyerhof calculations for sample pile dimensions are given in Table 49.
Table 49. Two-Pile vertical capacity sample calculation using Meyerhof's method.

<table>
<thead>
<tr>
<th>Soil</th>
<th>Pile Diameter 1, m</th>
<th>5.64 ft</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vertical Reaction Load</td>
<td>1,356,003</td>
<td></td>
</tr>
<tr>
<td>Sediment Type</td>
<td>Sand</td>
<td></td>
</tr>
<tr>
<td>Pile Depth in Sediment</td>
<td>15 m</td>
<td>49 ft</td>
</tr>
<tr>
<td>Soil Density</td>
<td>Dense</td>
<td></td>
</tr>
</tbody>
</table>

**Point Bearing**

| End Condition         | open               |         |
| Point Area            | 0.14 m^2           | 1.55 ft^2 |
| Unit Weight           | 17,000 N/m^3       | 108 lbf/ft^3 |
| Soil Friction Angle   | 0.61 rad           | 35 deg  |
| Bearing Capacity Factor| 143                |         |

"Atmospheric" Pressure 100,000 Pa 14.5 psi

**Unit Point Resistance**

| Effective Vertical Stress | 239,091 Pa | 35 psi |
| Point Resistance          | 4,934,803 N | 1,109,388 lbf |
| Limiting Point Resistance | 722,608 N  | 162,449 lbf |

**Skin Friction**

| Effective Earth Pressure Coefficient | 0.60 |
| Average Effective Overburden Pressure   | 270,000 Pa | 39 psi |
| Soil-pile friction angle               | 0.61 rad | 35 deg |
| Critical Depth                         | 27 m | 89 ft |
| Embedment Ratio                        | 11     |         |
| Ultimate Skin Resistance               | 9,740,335 N | 2,189,714 lbf |
| Vertical Load Safety Factor:           | 7      |         |

Legend:

<table>
<thead>
<tr>
<th>Environmental parameters</th>
<th>Iterated values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Design inputs*</td>
<td>Values calculated from formulae</td>
</tr>
</tbody>
</table>

*Pile diameter and vertical reaction force are determined from the mechanics analysis.

**Lateral Capacity**

The ultimate lateral bearing capacity of a pile is significantly more complicated than the vertical capacity. As with the vertical capacity, a pile in bedrock is assumed to fail in material before the supporting rock gives way. But for Case 1, which entirely assumes a sand-gravel mix, Brom's method, as described in the DOT Federal Highway Administration publication, *Design and Construction of Driven Pile Foundation* (1998) was used. Given the pile diameter and lateral reaction force found
in the mechanics analysis, the pile depth into the sediment was iterated until Brom's analysis showed that the pile was "long." This means that the pile material will yield before the soil. The inputs and results of each step of that analysis are shown in Table 50 for sample pile dimensions.

<table>
<thead>
<tr>
<th>Table 50. Two-Pile lateral capacity sample calculation using Brom's method.</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Brom's Method</strong></td>
</tr>
<tr>
<td>Pile Diameter</td>
</tr>
<tr>
<td>Pile Depth in Sediment</td>
</tr>
<tr>
<td>Soil Type</td>
</tr>
<tr>
<td>Coefficient of Horizontal Subgrade Reaction</td>
</tr>
<tr>
<td>Kh adjusted for loading, soil conditions</td>
</tr>
<tr>
<td>Eccentricity of Applied Load</td>
</tr>
<tr>
<td>Shape Factor</td>
</tr>
<tr>
<td>Resisting Moment of Pile</td>
</tr>
<tr>
<td>Length Factor</td>
</tr>
<tr>
<td>Dimensionless Length Factor</td>
</tr>
<tr>
<td>Pile Length Type</td>
</tr>
<tr>
<td>The result at this stage is sufficient; Brom's analysis shows that the pile will fail before the soil.</td>
</tr>
<tr>
<td>Rankine passive pressure coefficient for cohesionless soil</td>
</tr>
<tr>
<td>Average Effective Unit Weight</td>
</tr>
<tr>
<td>Cohesion</td>
</tr>
<tr>
<td>Long</td>
</tr>
<tr>
<td>Dimensionless Factor</td>
</tr>
<tr>
<td>Dimensionless Load Factor</td>
</tr>
<tr>
<td>Ultimate Lateral Load</td>
</tr>
<tr>
<td>Lateral Safety Factor</td>
</tr>
<tr>
<td>Recommended Safety Factor</td>
</tr>
<tr>
<td>Max allowable load</td>
</tr>
<tr>
<td>Factor</td>
</tr>
<tr>
<td>Deflection at point of loading</td>
</tr>
</tbody>
</table>

Legend: Environmental parameters
Iterated values
Design Inputs*
Values calculated from formulae

*Pile diameter and the reaction bending moment are determined from the mechanics analysis.
Neglecting vibrational issues in marine structures can be catastrophic (Tomlinson, 2008, p. 413). After a suitable pile wall thickness and diameter were selected, vortex-induced vibration was investigated.

According to standard beam theory the natural frequency of a beam is found to be

$$f_N = \frac{K'}{L^2} \sqrt{\frac{EI}{M}},$$

where $E$ is Young's Modulus, $I$ is the area moment of inertia, $M$ is mass/unit length of the beam (including the mass of the water contained in the beam and the mass of the displaced water), $L$ is the length of the beam, and $K'$ is a factor of 3.56 for the first mode of vibration in members with both ends fixed. The Strouhal number can then be used to find the fluid velocity at which the frequency of vortex shedding will match the member’s natural frequency. This critical velocity is given by

$$V_{crit} = K f_N d_o,$$

where $d_o$ is the outer diameter of the member and $K$ is given in Table 35.

| Table 51. Coefficients for modes of Vortex-Induced Vibration (Tomlinson, 2008). |
|---|---|
| 1.2 | Onset of in-line motion |
| 2.0 | Maximum in-line motion |
| 3.5 | Onset of cross-flow motion |
| 5.5 | Maximum cross-flow motion |

According to Mittal and Kumar (1999), “in-line oscillations are significant only if the mass of the cylinder is not too large compared with the mass of the surrounding fluid it displaces.” Since the mass of the cylinder is on the order of the mass of the surrounding fluid, it was assumed that only cross-flow motion is significant. So a value of $K=3.5$ was chosen. Equation (157) was evaluated for each
design and the minimum velocity required for the onset of vortex-induced vibration was found in all cases to exceed twice that seen in Muskeget Channel.

**Scour**

Anti-scour structure must be designed with care. Rocker (1985) indicates that a steel skirt extending one diameter beyond each pile can adequately protect against scour. Such structures, made of 1 inch steel, were incorporated into the design and are shown in Figure 47.

**Lifting**

Two main concepts were considered for raising/lowering the platform: A rack-and-pinion system and a hydraulic winch system. Each must be capable of lifting the weight of the turbine plus the friction force between the turbine mounting structure and the piles. The coefficient of friction was taken to be 0.5 for wet steel on steel (a worst-case approximation). Thus, the friction force was half the drag force on the turbine at max current.

Significant mechanical advantage can be achieved in the winch system by using block and tackle configurations. However, this should be avoided in the splash zone and underwater because of the harsh environmental factors (including biofouling, corrosion, and ice blockage). Thus, the platform was designed to house this system in the protection of the above-surface platform. This would reduce the total cost of the required marine grade winches from $300,000 to $90,000 (as per Lantech). A resulting design is shown in Figure 50.
In this design, each winch coils a wire rope, which is connected to a length of chain, which is attached to the turbine mounting structure. This allows the winch to coil only the wire rope, while only the chain is submerged or exposed in the splash zone. Issues of wire rope set were addressed by including clevises below the fairleads, which would be capable of bearing the full tension in the chains when the mounting platform is not being raised or lowered. It should be noted that a hydraulic drive system would need to be incorporated to power the winches.

A rack-and-pinion system would provide a robust operating system with excellent positional control during the raising/lowering process and during operation. However, such systems are costly. LeTourneau Technologies quoted a system at $406,000 (including the electric induction drive system) using the smallest unit they offer (which could handle all turbines of the scales investigated). Thus, this concept would only be applied to platforms capable of deploying in-stream axis turbines of 13 m (43 ft.) diameter or greater.

Figure 50. Close-up of work-platform and lifting structure. 42 ft. support vessel and workers are shown for scale. Each winch is connected to opposite side wire rope/chain. Design shown is for mounting a 6 by 10 m (20 by 33 ft.) cross-flow axis turbine. Larger systems would use a rack-and-pinion lifting system.
Results

The design processes describe above were integrated in the following procedure for each maximum turbine size specified:

- A pile wall thickness (2 inches for all designs) was selected.
- The diameter of each pile was iterated until the mechanics analysis showed that the required safety against yielding was met.
- The depth to which the pile would be driven into the soil was iterated until requirements for both vertical and lateral capacity were satisfied.
- Vortex-Induced Vibration (VIV) was analyzed and it was found that the piles as designed would not experience significant vibration unless subjected to velocities at least twice those expected in the channel.
Table 52. Two-pile Surface-Piercing Platform parameters.

<table>
<thead>
<tr>
<th>Turbine Diameter</th>
<th>4 m (14 ft)</th>
<th>9 m (29 ft)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turbine Distance from Bottom</td>
<td>25 m</td>
<td>25 m</td>
</tr>
<tr>
<td>Pile Material</td>
<td>ASTM A252 Grade 3</td>
<td>ASTM A252 Grade 3</td>
</tr>
<tr>
<td>Number of Piles</td>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td>Diameter of Pile</td>
<td>1.04 m 3.42 ft</td>
<td>1.89 m 6.20 ft</td>
</tr>
<tr>
<td>Thickness of Pile</td>
<td>0.0508 m 2.00 in</td>
<td>0.0508 m 2.00 in</td>
</tr>
<tr>
<td>Pile Depth in Sediment</td>
<td>12 m 39 ft</td>
<td>17 m 56 ft</td>
</tr>
<tr>
<td>Pile Height Above Surface</td>
<td>10.5 m 34 ft</td>
<td>15.0 m 49 ft</td>
</tr>
<tr>
<td>Total Length of Pile</td>
<td>47.4 m 156 ft</td>
<td>57.1 m 187 ft</td>
</tr>
<tr>
<td>Width of structure</td>
<td>9 m 28 ft</td>
<td>13 m 43 ft</td>
</tr>
<tr>
<td>Mass of Pile</td>
<td>61,926 Kg 136,524 lbf</td>
<td>135,246 Kg 298,166 lbf</td>
</tr>
<tr>
<td>Compression SF</td>
<td>2.0</td>
<td>2.0</td>
</tr>
<tr>
<td>Max Stress/Endurance Limit</td>
<td>0.7</td>
<td>0.7</td>
</tr>
<tr>
<td>Lifting Number of Winches</td>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td>Maximum Winch Line Pull Required</td>
<td>26,594 N 5,979 lbf</td>
<td>138,414 N 31,117 lbf</td>
</tr>
<tr>
<td>Winch Selected</td>
<td>M18</td>
<td>LWS 570</td>
</tr>
<tr>
<td>Pile Type in Soil</td>
<td>Long</td>
<td>Long</td>
</tr>
<tr>
<td>Velocity for Transverse Vibration</td>
<td>4.1 m/s</td>
<td>11.1 m/s</td>
</tr>
</tbody>
</table>
### Table 52. Two-pile Surface-Piercing Platform parameters (continued).

<table>
<thead>
<tr>
<th>Turbine Diameter</th>
<th>13 m (43 ft)</th>
<th>17 m (57 ft)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Turbine Distance from Bottom</strong></td>
<td>25 m 82 ft</td>
<td>25 m 82 ft</td>
</tr>
<tr>
<td><strong>Pile Material</strong></td>
<td>ASTM A252 Grade 3</td>
<td>ASTM A252 Grade 3</td>
</tr>
<tr>
<td><strong>Number of Piles</strong></td>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td><strong>Diameter of Pile</strong></td>
<td>2.77 m 9.10 ft</td>
<td>3.69 m 12.11 ft</td>
</tr>
<tr>
<td><strong>Thickness of Pile</strong></td>
<td>0.0508 m 2.00 in</td>
<td>0.0508 m 2.00 in</td>
</tr>
<tr>
<td><strong>Pile Depth in Sediment</strong></td>
<td>21 m 71 ft</td>
<td>26 m 84 ft</td>
</tr>
<tr>
<td><strong>Pile Height Above Surface</strong></td>
<td>19.5 m 64 ft</td>
<td>24.0 m 79 ft</td>
</tr>
<tr>
<td><strong>Total Length of Pile</strong></td>
<td>66.0 m 217 ft</td>
<td>74.5 m 244 ft</td>
</tr>
<tr>
<td><strong>Width of structure</strong></td>
<td>18.0 m 57 ft</td>
<td>22.0 m 72 ft</td>
</tr>
<tr>
<td><strong>Mass of Pile</strong></td>
<td>229,387 Kg 505,711 Ib</td>
<td>344,645 Kg 759,813 Ib</td>
</tr>
<tr>
<td><strong>Compression SF</strong></td>
<td>2.0</td>
<td>2.0</td>
</tr>
<tr>
<td><strong>Max Stress/Endurance Limit</strong></td>
<td>0.7</td>
<td>0.7</td>
</tr>
<tr>
<td><strong>Lifting</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Number of Winches</strong></td>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td><strong>Maximum Winch</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Line Pull Required</strong></td>
<td>383,517 N 86,218 lbf</td>
<td>809,960 N 182,086 lbf</td>
</tr>
<tr>
<td><strong>Winch Selected</strong></td>
<td>LWD3500</td>
<td>LWD3500</td>
</tr>
<tr>
<td><strong>Pile Type in Soil</strong></td>
<td>Long 0.01</td>
<td>Long</td>
</tr>
<tr>
<td><strong>Velocity for Transverse Vibration</strong></td>
<td>20.7 m/s</td>
<td>32.6 m/s</td>
</tr>
</tbody>
</table>

### Costing

Several Marine Contractors, including J.F. White, Pihl U.S., and Sea and Shore, were contacted for rough estimates of the cost for installation of two 56" (1.42 m) piles or four 24" (0.61 m) piles. J.F. White (2011) proposed the following installation procedure:

A marine piling operation would be mobilized and consist of a 54' x 180' barge with a 200 TN lattice boom crane set on top. All construction materials, templates and equipment would also be placed on the barge. The barge would be mobilized from a main land marine facility and towed to the location of work.
The sequence of work would be to construct templates, install piles and set platforms. In the event that bedrock is encountered above the proposed pile tip elevation, JFW would use a "down the hole hammer" to remove the bedrock and create a rock socket. Concrete and reinforcing steel would then be placed in the toe of the pile to provide the required embedment and stability of the pile system.

The contractor provided estimates for installing piles of the aforementioned size for both the case in which bedrock exists just beneath the seafloor and that in which there is sufficient sediment overburden to hold the piles, while strongly recommending that soil testing be conducted before installation. These estimates were scaled under the assumption that the entire installation cost was proportional to the volume of sediment removed by drilling or enclosed by the pile. The estimate carried $50,000 to construct the platform to which the turbine would mount. This was assumed to vary linearly with the size of the turbine.

For the turbine lifting mechanism, quotes were obtained from TWG Lantech (2011) for winches of various sizes. Also, a rack-and-pinion system which could handle the required loads for any of the turbine sizes investigated was quoted by Letourneau Technologies (2011).

The results of this cost analysis are shown in Table 53.
### Table 53. Cost of Two-pile Surface-piercing Platform.

<table>
<thead>
<tr>
<th>Turbine Diameter</th>
<th>Unit</th>
<th>Quantity</th>
<th>Cost</th>
<th>Unit</th>
<th>Quantity</th>
<th>Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Foundation</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Piles, Installed</td>
<td>EA</td>
<td>2</td>
<td>$618,006</td>
<td>2</td>
<td>$1,793,351</td>
<td></td>
</tr>
<tr>
<td>Anti-corrosion Coating</td>
<td>ft^2</td>
<td>5251</td>
<td>$21,133</td>
<td>11469</td>
<td>$46,154</td>
<td></td>
</tr>
<tr>
<td>Anti-scour Matt</td>
<td>lb</td>
<td>2820</td>
<td>$5,279</td>
<td>3993</td>
<td>$7,476</td>
<td></td>
</tr>
<tr>
<td>Support Structure</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Platform</td>
<td>EA</td>
<td>1</td>
<td>$1,852</td>
<td>1</td>
<td>$14,815</td>
<td></td>
</tr>
<tr>
<td>Hydraulic Winch</td>
<td>EA</td>
<td>2</td>
<td>$46,000</td>
<td>2</td>
<td>$90,000</td>
<td></td>
</tr>
<tr>
<td>Total</td>
<td></td>
<td></td>
<td>$692,270</td>
<td></td>
<td>$1,951,795</td>
<td></td>
</tr>
</tbody>
</table>

O&P, shipping included throughout
Site work not included

### Table 53. Cost of Two-pile Surface-piercing Platform (continued).

<table>
<thead>
<tr>
<th>Turbine Diameter</th>
<th>Quantity</th>
<th>Cost</th>
<th>Quantity</th>
<th>Cost</th>
<th>Quantity</th>
</tr>
</thead>
<tbody>
<tr>
<td>Foundation</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Piles, Installed</td>
<td>2</td>
<td>$3,614,986</td>
<td>2</td>
<td>$5,865,824</td>
<td>2</td>
</tr>
<tr>
<td>Anti-corrosion Coating</td>
<td>19452</td>
<td>$78,280</td>
<td>29225</td>
<td>$117,613</td>
<td>19452</td>
</tr>
<tr>
<td>Anti-scour Matt</td>
<td>5212</td>
<td>$9,759</td>
<td>6479</td>
<td>$12,131</td>
<td>5212</td>
</tr>
<tr>
<td>Support Structure</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Platform</td>
<td>1</td>
<td>$50,000</td>
<td>1</td>
<td>$118,519</td>
<td>1</td>
</tr>
<tr>
<td>Skidder Gear Unit RH</td>
<td>2</td>
<td>$279,475</td>
<td>2</td>
<td>$279,475</td>
<td>2</td>
</tr>
<tr>
<td>Rack Skidder</td>
<td>140</td>
<td>$126,598</td>
<td>140</td>
<td>$126,598</td>
<td>140</td>
</tr>
<tr>
<td>Total</td>
<td>$4,159,097</td>
<td></td>
<td>$6,520,160</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

O&P, shipping included throughout
Site work not included
CHAPTER 8

NATURAL BERTH

A Natural Berth would comprise only an unmodified section of seafloor with monitoring equipment and electrical power connection provided and the necessary permits in place for testing hydrokinetic devices. Thus, developers would be responsible for device installation and could test integrated systems that include a turbine, a generator, and a foundation system. This would allow developers to test concepts at the highest Technology Readiness Level defined by the Department of Energy (2009) before commercial deployment.

A Natural Berth option was considered for the following advantages:

- The complete system would be tested.
- The developer would have maximum freedom
- The berth could accommodate systems up to TRL 9 (DECC stage 5) which allows for commercial demonstration.

Disadvantages include:

- The developer is faced with substantial installation, maintenance, and removal costs.
- The type of foundation is undefined, potentially raising permitting obstacles.
- The berth would not be conducive to devices in the early stages of development.

The Natural Berth option would provide a permitted, instrumented (including electrical power connection and measurement and flow measurement) section of seafloor whose baseline has been thoroughly investigated. Developers would be responsible for installing their devices, including any necessary foundation or
mooring system, and would remove devices after testing is completed. Thus, developers would be able to see how well their foundation concepts are suited to the high sediment transport environment on the Muskeget seafloor. The European Marine Energy Centre (EMEC) reports that it has successfully employed this model for several years (2011).

The Natural Berth option could exist in place of or in parallel with a platform option, alternatives for which are shown in the following section. A schematic is shown in Figure 51. Note that this figure indicates hardwired ADCP connections and also incorporates the Edgartown Tidal Power Pilot Project.

![Figure 51. Schematic of Natural Berth layout.](image-url)
CHAPTER 9

ELECTRICAL POWER CONNECTION AND INSTRUMENTATION

The test site will include, at a minimum, means of accepting the electrical power generated by the test device and instrumentation to measure the generated power and the flow conditions.

**Electrical Power Connection**

The site will be equipped with a submersible three-phase electrical power connection to transmission lines running to shore. This type of connection, shown in Figure 52, has been implemented at the European Marine Energy Center. This will connect the device to either the grid or local users via armored 3-phase XLPE undersea cabling, shown in Figure 53. If the device is connected to the grid it will be via a 4 kV line to an on-land substation along one of the routes shown in Figure 54, whose distances are given in Table 54. Determining the cost of installing these cables was outside of the scope of this thesis. However, it was noted that similar cable-laying projects on the northeastern coast of the U.S. have cost about $1 million/mile of cable.
Figure 52. Subsea electrical power connection (EMEC).

Figure 53. Typical XLPE 3-phase undersea cable with fiber-optic core (EMEC).

Figure 54. Potential cable routes.

Table 54. Grid connection distances.

<table>
<thead>
<tr>
<th>Option</th>
<th>Distance km</th>
<th>Distance Miles</th>
</tr>
</thead>
<tbody>
<tr>
<td>Option 1: Via Chappaquiddick</td>
<td>5.6</td>
<td>3.5</td>
</tr>
<tr>
<td>Option 2: Via Katama</td>
<td>8</td>
<td>5.0</td>
</tr>
</tbody>
</table>

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Instrumentation

Instrumentation will be in accordance with the guidelines set forth in *Assessment of Performance of Tidal Energy Conversion Systems*, which were developed in consultation with The European Marine Energy Centre Ltd and with other interested parties in the UK tidal energy community.

The guidelines for power measurement found therein can be satisfied by including a “3- or 2-phase power measurement device, such as a transducer...[which] shall conform to [accuracy] Class 0.5, or better, as defined in IEC 60044-1” (International Electrotechnical Commission, 2002) as close to the device as practicable. The data from this device could be transmitted to shore via the fiber-optic core of the power cable, if such a cable is laid and the necessary connector is installed.

The requirements for flow measurement can be satisfied by placing vertical-looking ADCPs up- and downstream of the test area. The specifications for the ADCPs can be easily met by, for example, the RDI Workhorse Sentinel V at 600 kHz (Teledyne RD Instruments, 2012). These would be mounted in bottom-mounted trawl-resistant housings and equipped with acoustic release mechanisms or, for a floating platform, on the platform itself. Three options exist for acquiring ADCP data from bottom-mounted systems:

- Hardwiring to the power/data cable for transmission to shore
- Equipping with acoustic modems
- Manually retrieving self-recorded data

Hardwiring is attractive for its real-time transmission, reliability, and its ability to supply power to the ADCP, allowing indefinite deployment. However, it involves expensive equipment, and its longevity is a concern. A hardwire connection on an
ADCP installed at the European Marine Energy Centre failed within one year (Devine, 2011). The designer of that system cautions against such a transmission system and questions the need for real-time ADCP data (Wood, 2011).

Acoustic modems can also provide real-time data if coupled with a gateway buoy. However, these can be plagued with reliability issues (Codiga et al., 2004), making them undesirable for this application.

Relying on the self-recording mechanism requires divers to manually retrieve data from the ADCPs. However, this method is extremely reliable and requires the least capital cost. Additionally, the cost of retrieving data manually may not greatly exceed the maintenance cost of other data-acquisition options, as divers may periodically be required to visit the devices regardless of the method used.

The baseline capital costs for flow measurement are shown in Table 55. It is important to note that data retrieval, power measurement, and connection costs are not included.

<table>
<thead>
<tr>
<th>ADCPs</th>
<th>$/unit</th>
<th>Quantity</th>
<th>Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>RDI Workhorse Sentinel</td>
<td>$30,000</td>
<td>2</td>
<td>$60,000</td>
</tr>
<tr>
<td>Trawl-resistant Bottom Mounts</td>
<td>$20,000</td>
<td>2</td>
<td>$40,000</td>
</tr>
<tr>
<td>Acoustic Release</td>
<td>$7,000</td>
<td>2</td>
<td>$14,000</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td></td>
<td></td>
<td><strong>$114,000</strong></td>
</tr>
</tbody>
</table>
Six design alternatives were identified, and basic engineering calculations were performed for each. Costs for each were estimated primarily from manufacturer and contractor quotes and estimates. The results are compared in Figure 55 for each scale investigated. These estimates do not include instrumentation or cabling cost. Note that the natural berth is not included because its structural cost is zero.

Each test facility concept provides a unique set of potential benefits. These advantages are compared in Table 56.
Figure 55. Estimated Platform Cost Comparison.
Table 56. Advantages of each design alternative.

<table>
<thead>
<tr>
<th>Developer</th>
<th>Floating</th>
<th>Submerged</th>
<th>Fixed Gravity</th>
<th>Telescoping</th>
<th>4-pile</th>
<th>2-pile</th>
<th>Natural Berth</th>
</tr>
</thead>
<tbody>
<tr>
<td>Variable depth</td>
<td>✓</td>
<td>✓</td>
<td></td>
<td>✓</td>
<td>✓</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Monitoring</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
</tr>
<tr>
<td>365 day testing</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
</tr>
<tr>
<td>Below traffic</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
</tr>
<tr>
<td>Low risk of vandalism</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
</tr>
<tr>
<td>Easy removal</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td></td>
</tr>
<tr>
<td>Simple/Robust</td>
<td>✓</td>
<td>✓</td>
<td></td>
<td>✓</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

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CHAPTER 11

CONCLUSION

Recommendation

Six design alternatives for a test platform were considered for technical and economic feasibility, in addition to a natural berth test bed concept. Neither the four-pile platform nor the fixed gravity foundation platform provided convenient access for developers. The submerged buoyant platform and the telescoping gravity foundation platform both provided access for developers but would require extensive proof-of-concept work and further development before being implemented. Since developer-friendliness and reliability were crucial factors in comparing design alternatives, all four of these concepts were rated lower after the preliminary engineering calculations and costing were completed.

Both the floating platform and the two-pile, surface piercing platform were analyzed in detail. This analysis focused on platforms capable of testing a maximum turbine diameter of 9 m (29 ft.) because the core of the maximum tidal current extends vertically over this range in the upper portion of the water column.

The floating platform would incur lower construction and installation costs (approximately $1 million) than the two-pile platform. It could be easily removed from the site when necessary, which could be very useful as the test site and testing procedures are being developed. As for a floating platform’s performance in Muskeget’s wave environment, it was found that a floating platform could typically operate for more than 90% of the year.

The two-pile, surface piercing platform would require more capital for
construction and installation (approximately $2 million). However, testing from a fixed platform can be very beneficial to the developer and a permanent presence in the Muskeget Channel could be advantageous.

Furthermore, a natural berth would be necessary for developers wishing to evaluate complete systems (including mounting structure). Thus, it is recommended that natural berth be incorporated in addition to a testing platform.

The floating platform, the two-pile platform, and the natural berth were presented to the U.S. Coast Guard Waterways Management Division for comment. It was indicated that either option could be implemented in the Muskeget Channel (E.G. LeBlanc, personal communication). It was noted that the two-pile platform could even be used as an aid to navigation.

Given the lower cost of the floating platform and the present experience with such platforms, it is recommended that the Muskeget Channel tidal energy facility implement a floating platform as the near-term testing solution. Then, as the tidal energy industry grows, demand for the facility increases, experience with the testing site is gained, and funding becomes available, a two-pile, surface piercing platform could be implemented.

Future Work

It is important to note that cost estimates for both the floating platform and the two-pile platform are based on certain assumptions (which are detailed in this document). Although this analysis was conducted carefully, exact quotes for a completed design should be sought before making final decisions. For the floating platform, this will require detailed structural design of platform. For the two-pile
platform, this will require more detailed structural analysis of the “bridge” section connecting the two piles, sub-bottom profiling and, possibly, exploratory boring to ensure that installation quotes will be accurate.
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APPENDIX A: COMPUTER CODE

The mathematical model of the dynamics of the Floating Platform system was implemented in MATLAB™. The code of the main program and significant subroutines is displayed below.

Hydrodynamic Model: Main Program

```matlab
% Model for the motion of a floating platform with a hydrokinetic turbine
% Developed by Rob Swift and Toby Dewhurst
% Coded by Toby Dewhurst
% Copyright 2013

clear;clc;close all
clearvars -global

% Global variables
global ckk mkk skk fkk h phase dir sigma0 sigmaw h u_cur g rho L bm L_wp eps s L_Db
L_Rt L_Lml K L_Rml phi_m1 a b Ab Bb CM AB AT Mt Abt Mtot Iwp C_Db A_Psd C_DT
A_PT T T0 XO s p t ind zeta odkk zeta_pass dzeta_dx_pass zetaddot_t_pass
zetaddot_t_pass q_pass t_pass qdot_pass wrange Tl_pass Tlml_pass x_r_pass
y_r_pass out_pass outO outOO x_r y_r out c_fact hcfact eta3amp L_Dmb C_Dmb A_Pmb
phi_m2 bow_pass linchk_pass

% Options:
vkk=1; % V1 barge, vkk=1; V2 barge, vkk=2
% Note: Control presence of mooring ball by setting L_Dmb in V#_parameters
%(with L_Dmb=0 for no mooring ball).
mmk=1; % WithCage==1; WithoutCage==2
ckk=2; % Exact catenary equations, ckk=1; Linearized, ckk=2; Spring approximation,
ckk=3, No mooring, ckk=4
skk=1; % Random seas, skk=1; Range of single-frequency waves, skk=2 (gets transformed
later—Not anymore); Comparing with wave tank results, skk=3. Free-release, skk=4
cfact=[2 1 2 1 2 1]; % Long wave assumption, cfact5=1. Surface level integrated over
length of barge, cfact5=2.
fkpplot=1; % Choose case for plotting results

% Get vessel parameters (platform, turbine, mooring, etc.)
load 'Free Release Tests\V1\parameters.mat'
% Get open ocean results. DISABLE if not comparing
% if skk==1
% load 'Field Data\Ocean_Results 201271922400.mat'
% staff_RAOs=RAOs; % Take care of name overlap
% clear('RAOs');
% end

eps=-pi/2;
% Turbine parameters (included in V1\V2_parameters.mat.)
if mkk==2
    At=zeros(6,6);
    Bt=zeros(6,6);
end
```

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\[ M_t = \text{zeros}(6,6); \]

\text{% Define environment;}
\text{u_cur=2; \%m/s, current speed. (Later converted to Stoke's drift velocity if u_cur<0.5)}
\text{h=26; \%water depth, m}
\text{g=9.81;}
\text{rho=1024; \%kg/m^3}

\text{% Define wave (spectrum)}
\text{SWH=2; \%significant wave height in meters}
\text{T_pk=6; \%s, Significant period}
\text{dir=1; \%waves propagating in the -x direction, dir=1; waves propagating in the x direction, dir=-1}
\text{sigma_pk=2*pi/T_pk; \%Peak frequency, rad/s}
\text{f_pk=1/T_pk; \%Peak frequency, Hz}
\text{THigh=1; \%s, shortest period}
\text{TLow=0; \%s, longest period}
\text{if skk>1}
\text{d_sigma=0.2; \%Width of frequency bands}
\text{else}
\text{d_sigma=0.5 \%Consider setting to Fourier frequencies}
\text{end}
\text{sigmaO=(2*pi/TLow:d_sigma:THigh); \%rad/s, Angular frequency without current}
\text{Get ks}
\text{Intermediate solution (if T decreases as kk increases)}
\text{deep=0; \%Start with intermediate solution.}
\text{for kk=1:length(sigmaO)}
\text{if deep=0; \%If not in deep water}
\text{[lambda(kk,1) k(kk,1)] = brute_dispf(2*pi/sigmaO(kk), h);}
\text{if lambda(kk)<2*h \%Check for deep water}
\text{deep=1;}
\text{end}
\text{else}
\text{k(kk,1)=(sigmaO(kk)).^2/g; \%Deep water dispersion relation}
\text{end}
\text{end}
\text{sigmae=sigmaO+dir*k*u_cur; \%Frequency of encounter}
\text{if length(sigmaO)>1}
\text{d_sigmae=abs(diff(sigmae));}
\text{d_sigmae=[d_sigmae(1);d_sigmae];}
\text{end}

\text{% Generate a Bretschneider spectrum}
\text{SED(:,5)=SWH^2/(16*f_pk)*(sigmaO./sigma_pk).^(-5).*exp(-5/4*(sigmaO./sigma_pk).^(-4));}
\text{if skk==1}
\text{if exist('staff_sj1side','var')==1}
\text{\%sigma=\text{max(staff_bandfreqs) - min(staff_bandfreqs)}*(pi/180)/(length(staff_bandfreqs)-1);}
\text{SED(:,5)=interpl(staff_bandfreqs*2*pi,staff_sj1side(:,5),sigma); \%since, in}
\text{the ocean experiments, it is the encounter frequency that is measured. sigmaO is}
\text{calculated later.}
\[ H = \sqrt{2 \cdot \text{SED(:,5)} \cdot \text{d_sigmae}} \] Wave Height in accordance with the energy spectrum
else
\[ H = \sqrt{2 \cdot \text{SED(:,5)} \cdot \text{d_sigm};} \] Wave Height in accordance with the energy spectrum
end

phase = rand(1, length(sigma0)) * 2 * pi; % Generate random wave phases
endif skk==2
\[ H(1: \text{length(sigma0)}) = 5 \cdot H; \]
\[ \text{H=[2 4 6 8]} \]
phase = zeros(length(sigma0));
endif skk==3 % If comparing with tank tests processed in OPIE_Reader_regular.m
load '2012-05-03 v1 Model Wave Testing\WaveTestParameters.mat'
H = H * L_scale;
sigmaO = 2 * pi / (Ts * L_scaleA0.5);
phase = zeros(length(sigma0));
\% Get k for wave tank periods
clear k
for kk=1:length(sigmaO)
if deep==0; % If not in deep water
[lambda(kk,1), k(kk,1)] = brute_disp(2 * pi / sigma0(kk), h);
if lambda(kk)<2*h % Check for deep water
deep=1;
end
else
k(kk,1)=(sigmaO(kk)).^2/g; % Deep water dispersion relation
end
end
\% k=k';
sigmae = sigmaO;
endif skk==4 % For free-release test
H=0;
sigmaO = 2 * pi / (Tnpkmean(1.2) * L_scaleA0.5);
phase=0;
end
if u_cur==0
u_cur = dir \( \cdot (H/2). \cdot k' \cdot \sigma' \); % \( \text{exp}(-L \cdot \text{Rt} \cdot (2 \cdot \pi / \text{lambda_pk}) \) \text{m/s. Stokes drift velocity for typical wave. Applied so that u_cur>0.}
fkkL=length(sigma0);
else
u_cur=u_cur*ones(1, length(sigma0));
fkkL=1;
endif skk=1
% Compute surge and pitch wave spectra from prescribed heave spectrum
SED(:,9)=kA2.*SED(:,5); % Compute surge spectrum from heave spectrum
SED(:,1)=1./tanh(k*h).A2.*SED(:,5);
end

phi_m = asin(h/L_Lml); % rad, Approximate mooring line angle from vertical
% CAUTION: Code

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does not calculate $\phi_{ml}$ as a function of time

% Solve catenary equations at equilibrium (only current forcing) and linearize catenary eqs.
$\phi_{ml}$=0; $\chi_{rad}$, Angle of mooring ball-to-barge line from horizontal
if $c_{kk}$<4
    $p_{kk}$=$p_{kk}$+1;
    figure($p_{kk}$)
    hold on
    for $f_{kk}$=1:$f_{kk}$L
        $T_{O}$=$1/2*rho*(c_{Db}A_{Ps}b+c_{Dt}A_{Pt}+c_{Db}A_{Pmb})u_{cur}(f_{kk})^2+b_{tot}(1,1)*u_{cur}(f_{kk})$;
        if $u_{cur}(f_{kk})$>0.5
            $T_{O \_mult}$=[1 1.5 1] \% Change horizontal force when finding $dT/dx$; Written to provide representative tensions for $u_{cur}$=0 and $u_{cur}$=-2
        else
            $T_{O \_mult}$=[1 (H(f_{kk})/2*sigma0(f_{kk})*tanh(k(f_{kk})*h))/u_{cur}(f_{kk})] \% Change horizontal force when finding $dT/dx$; Written to provide representative tensions for $u_{cur}$=0 and $u_{cur}$=-2
        end
        $T_{O \_mult}$=[1 10000 1]
        $d_{yin}$=[0 0 H(f_{kk})/2+0.1]; \% Change y-displacement when finding $dT/dy$
        \% Show result:
        $L_{\_Lmlk}(catx)=l_{\_Lmlk}(catx)+T_{O \_mult}*\_Lmlk(\_Lmlk(c_{kk}))$; \% Now solved for in catR.m
        for $catx$=1:2+(-1)*$c_{kk}$ \% So go through only once (at equilibrium) for exact cats, 3 times when using linearized cats
            \% No displacement, 2: x-displacement, 3: y-displacement
            $[T_{l}(catx) \_Lmlk(c_{kk}) x_{r}(catx) y_{r}(catx)]=catterR(T_{O \_mult}(catx),h+d_{yin}(catx),l_{\_Lmlk}(catx),\_Lmlk(\_Lmlk(c_{kk})))$; \% CAUTION: Inexact for elastic line
            $\text{Sup}(catx)=T_{O \_mult}(catx)/p*sinh(p*x_{r}(catx)/(T_{O \_mult}(catx)))$; \% $\text{Sup}$, Length of chain off of the seafloor
            $S_{down}(catx)=S_{down}(catx)+x_{r}(catx)+L_{\_Lmlk}(catx)*\_Lmlk(\_Lmlk(c_{kk}))$; \% Actual x-displacement (not horizontal distance from where the chain leaves the ground)
        end
        $x_{0 \text{coord}}=0:0.5:S_{down}(catx)$;
        $x_{0 \text{coord}}=0:0.5:x_{r}(catx)$;
        plot($x_{0 \text{coord}},zeros(length(x_{0 \text{coord}})), 'o')$
        plot($x_{0 \text{coord}}+S_{down}(catx),T_{O \_mult}(catx)/p*(cosh(p*x_{0 \text{coord}}/(T_{O \_mult}(catx)))-1), 'o'$)
        line([$x_{r}(catx)+S_{down}(catx)],x_{r}(catx)+cos(\_Lmlk(\_Lmlk(c_{kk}))).\_Lmlk(\_Lmlk(c_{kk}))),[y_{r}(catx)\_Lmlk(\_Lmlk(c_{kk}))\_Lmlk(\_Lmlk(c_{kk}))],[\text{Color}',[0.5 catx/3 0.5]])
        axis equal
        end
    end
end
% Set up guess for catter.m (applies when $c_{kk}$=1)
% Set up guess for catter2.m (applies when $c_{kk}$=1)
$X_{0}(f_{kk})=S_{down}(1)+x_{r}(1)+cos(\_Lmlk(1))\_Lmlk(1)$;
\[ dT_{\eta}(l, 1, \kappa^k) = (T_l(2) \cos(\phi_{m1}(2)) - T_l(1) \cos(\phi_{m1}(1))) / (x_{tot}(2) - x_{tot}(1)) \]  
\[ \text{Change in horizontal tension/change in horizontal position} \]

\[ dT_{\eta}(3, l, \kappa^k) = (T_l(2) \sin(\phi_{m1}(2)) - T_l(1) \sin(\phi_{m1}(1))) / (x_{tot}(2) - x_{tot}(1)) \]  
\[ \text{Change in vertical tension/change in horizontal position} \]

\[ dT_{\eta}(l, 3, \kappa^k) = C T_l(3) \cos(\phi_{m1}(2)) - T_l(1) \cos(\phi_{m1}(1))/d_{yin}(\text{catx}) \]  
\[ \text{Change in horizontal tension/change in vertical position} \]

\[ dT_{\eta}(3, 3, \kappa^k) = (T_l(3) \sin(\phi_{m1}(3)) - T_l(1) \sin(\phi_{m1}(1)))/d_{yin}(\text{catx}) \]  
\[ \text{Change in vertical tension/change in vertical position} \]

\[ \text{if } L_{D_{ffl}}>0 \text{ if a mooring ball is used} \]
\[ dT_{\eta}(3, 1, \kappa^k) = 0; \]
\[ dT_{\eta}(1, 3, \kappa^k) = 0; \]
\[ dT_{\eta}(3, 3, \kappa^k) = 0; \]
\[ \text{xso, the cat solution only has to be right for catx=1;} \]
\[ \text{if } \phi_{m1}(1) < 0 \text{ & & } \phi_{m1}(1) >= -\pi/2 \text{ if the cat solution wasn't even right for the first scenario...} \]
\[ \text{warn('Catx 1 failed. Horizontal spring constant invalid.'}) \]
\[ \text{pause} \]
\[ \text{otherwise, you're good to go} \]
\[ \text{end} \]
\[ \text{end} \]
\[ \text{else} \]
\[ dT_{\eta} = 0; \]
\[ \text{end} \]
\[ \text{end} \]
\[ \text{Calculate effective spring constant} \]
\[ \text{Plot initial catenary solutions} \]

\[ \text{legend('--', 'E', 'Equilibrium','--', '++ horizontal force','--', '++ vertical displacement')} \]
\[ \text{plot(xO,h,'sq')} \]
\[ \text{fill mooring values for u_{cur}>0} \]
\[ \text{if fkkL==1} \]
\[ \text{outO0=outO0(,ones(1,length(sigma0)))} \]
\[ \text{XO=xO(,ones(1,length(sigma0)))} \]
\[ \text{dT_{eta}=repmat(dT_{eta},[1 1 length(sigma0)]);} \]
\[ \text{end} \]
\[ \text{else} \]
\[ \text{dT_{eta}=zeros(length(sigma0));} \]
\[ \text{outO=0; For calculating L_{Lmlk} before ckk if statements in mrmotion.} \]
\[ \text{end} \]
\[ \text{if skk>1} \]
\[ \text{skk=length(sigma0);} \]
\[ \text{end} \]
\[ \text{if skk=2} \]
\[ \text{looplength=length(sigma0);} \]
\[ \text{elseif skk=3} \]
\[ \text{looplength=length(sigma0);} \]
\[ \text{else} \]
\[ \text{looplength=1;} \]
\[ \text{end} \]
a=zeros(6,6,looplength);
b=zeros(6,6,looplength);
Ab=zeros(6,6,looplength);
Bb=zeros(6,6,looplength);

x=-2*L:2*L; %horizontal spatial range which to display in animation
for fkk=1:looplength

%Adjust for either summing waves or calculating single wave
if skk==1
    wrange=1:length(sigmae);
else
    wrange=fkk;
end

% sigma0(kk)=fzero(@(var)(var+var^2*u_cur/g*cos(beta)-sigma(kk)),[.1 12]); %Frequency of waves without u
% disp = @(k)(sigma0(kk)^2-g*k*tanh(k*h)); % k(kk)=fzero([.01 2000]); %Wave number
%
% Xlf using strip theory (and applying forward speed/current corrections
% [Ab(fkk) Bbf(fkk)]=faltAB(sigma(fkk),u_cur,L_Db,A_PSb,g,rho)

Mtot=Mb+Mt;
% Abtot=Ab+At;
% Btot=Bb+Bt;

%Solve coupled differential equations. (Note: Pass most variables as
%global.)
qnot=zeros(1,12); %Initial conditions of all variables q, where:
%currently:
% q(1)=eta(1)
% q(2)=etadot(1)
% q(5)=eta(3)
% q(6)=etadot(3)
% ...

qnot(1)=-.1; %Trying to eliminate springing for u_cur=0
qnot(2)=0.05;

qnot(5)=0.8; %Because tstart=T-pk/4 (for solving cats) %h(fkk)/2; %Initial vertical position
% qnot(6)=
qnot(9)=0; %rad, initial angle from horizontal

%Prepare to concatenate and extract values
linch_k_pass=[];
qdot_k_pass=[];
t_k_pass=[];
zeta_k_pass=[];

...
zetadot_t_pass=[];
zetaddot_t_pass=[];
q_pass=[];
bown_pass=[];
T_pass=[];
if ckk==1
  Tl_pass=[];
  phi_ml_pass=[];
x_r_pass=[];
y_r_pass=[];
  out_pass=[];
end
% Set times to get solution (does not control time step in solver)
odkk=0; % Count times mrsmotion.m is called
eta3amp=0; % Initial amplitude for hull-interaction wave
tout_step=0.1; % s, IMU_freq;
if exist('IMU_freq')==1 % e. if comparing with ocean results from IMU_reader.m
tend=staff_t(length(staff_t));
else
  tout_step=0.1;
  tend=30*4*TPlow; % s
end
twant=0:tout_step:tend; %, %T_pk/4:tout_step:tend--replaced by multiplying u_wave
by t for t<1 in mrsmotion.m,
abstol=(1e-6);
options = odeset('RelTol',1e-6,'AbsTol',abstol,'NormControl','on');
[t,q]=ode45(@(t,q) mrsmotion(t,q,dt_eta),twant,qnot,options);

% %
% for fkk=1:skk
% Match passed outputs to twant/tout time scale
for tkk=1:length(t)
  last=1;
  for tpkk=last:length(t_pass)
    if t_pass(tpkk)>t(tkk)
      linchh_out(tkk,:,fkk)=linchh_pass(tpkk,:);
      qdot_out(tkk,:,fkk)=qdot_pass(tpkk,:);
      zeta_out(tkk,:,fkk)=zeta_pass(tpkk,:);
      dzeta_dx_out(tkk,fkk)=dzeta_dx_pass(tpkk);
      zetadot_t_out(tkk,:,fkk)=zetadot_t_pass(tpkk,:);
      zetaddot_t_out(tkk,:,fkk)=zetaddot_t_pass(tpkk,:);
      q_out(tkk,:,fkk)=q_pass(tpkk,:);
      bow_out(tkk,:,fkk)=bown_pass(tpkk,:);
      T_out(tkk,:,fkk)=T_pass(tpkk,:);
      if ckk==1
        Tl_out(tkk,fkk)=Tl_pass(tpkk);
        phi_ml(tkk,fkk)=phi_ml_pass(tpkk);
        x_r_out(tkk,fkk)=x_r_pass(tpkk);
        y_r_out(tkk,fkk)=y_r_pass(tpkk);
        out_out(tkk,fkk)=out_pass(tpkk);
      end
      last=tpkk;
      break
    end
  end
end

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end
end

%Store ode45 outputs
% tout(:,fkk)=t;
qout(:,fkk)=q;

%Repeat to extract values
% [zetap(fkk,:),setadot_p(wkk,:)] zetaddot_p(wkk,:)
dzeta_dx_p(wkk)]=wave(0,t,H(wkk),sigmae(wkk),phase(wkk),k(wkk),h);

%Find total incident velocity at rotor hub (Plotted outside of the fkk
%loop)
u_hub(:,fkk)=ones(length(t),1)*u_cur(fkk)+zetadot_t_out(:,1,fkk)
qdot_out(:,1,fkk)-qdot_out(:,9,fkk)*L_Rt;

%Find total incident fluid acceleration at the rotor hub
udot_hub(:,fkk)=zetadotdot_out(:,1,fkk)-qdot_out(:,2,fkk)-qdot_out(:,10,fkk)*L_Rt;
%Find the total (horizontal) force on the turbine
Fhub_drag(:,1,fkk)=1/2*rho*C_Dt*A_Pt*(u_hub(:,fkk)).*abs(u_hub(:,fkk));
Fhub_amass(:,1,fkk)=at(1,1)*udot_hub(:,fkk);
Fhub_inert(:,1,fkk)=mass_t*qdot_out(:,10,fkk)*L_Rt;
Fhub_fluid(:,1,fkk)=Fhub_drag(:,1,fkk)+Fhub_amass(:,1,fkk);
force_t(:,1,fkk)=Fhub_drag(:,1,fkk)+Fhub_amass(:,1,fkk)+Fhub_inert(:,1,fkk);

%Calculate RAOS
if skk>1
%    if fkk=5
%        what=8
%    end
zetarange=zetadot_t_out(floor((tend-(3/4)*tend)/tout_step):length(t),3); %Find
surface elevation for corresponding section of time series
zetarange=zetarange-mean(zetarange); %Mean, primarily for finding zero
crossings below
range=t(floor((tend-2*pi/sigma0(fkk))/tout_step):length(t)); %Get the time
vector corresponding to the cut range
lagrange1=0;
lagrange2=80; %Round(2*pi/sigmae/tout_step/2); %Range of lags to compute cross
correlation at.
roel2(:,fkk)=zeros(lagrange2-lagrange1+1,12); %Pre-allot so that the roe12
will be m-by-6
q_H(fkk,:)=zeros(1,6);
phlag(fkk,:)=.zeros(1,6);
slowlags=[pi/2 0 0 0 0 0 0 0 0 0 pi/2 0 0 0];
for qkk=1 3 5 %1:6 works because doing 2*qkk-1
%Response/wave amplitude
range=q(floor((tend-3/4*tend)/tout_step:length(t)),2*qkk-1);
%2*pi/sigma0(fkk)/tout_step:length(t),2*qkk-1); %Note, when u_cur>0, dir=-1,
encounter periods can be much larger
range=range-mean(range); %Mean, primarily for finding zero crossings
below
if qkk=1 & & 2*pi/abs(sigmae(fkk))<3/4*tend %Remove a running mean for
surge

range=range-tjranmean(range,2*pi/abs(sigmae(fkk))/tout_step);
end
q_H(fkk,qkk)=(max(range)-min(range))/2/(H(fkk)/2);
%Find phase lag. If the encounter period is long (too long to
%Find zero crossings in the simulated time, the heave and surge phases are
%likely 0, the pitch phase is likely pi/2
% if 2*pi/sigmae<2*pi e. if there will be at least 3 good zero
crossings
% [dum zcrosses]=crossed(t,zetarange); %Find the (time) zero crossings of
% the wave at the platform's CG
% [dum qcrosses(2*qkk-1)]=crossed(t,range); %Find the (time) zero
crossings of the platform response
% Calculate the autocorrelation to find the phase lag. Might not
% need to cut as above.
[R12 roel2(:,2*qkk-
1)]=correlation(range,0,zetarange,0,lagrange1,lagrange2);
[pkcor pkcorind]=findpeaks(roel2(:,2*qkk-1),'SORTSTR','descend'); %Find
the maximum correlation (how many lags).
if length(pkcorind)==0 %Account for first value giving the highest
correlation
phlag(fkk,qkk)=slowlags(2*qkk-1);
else
%tlag=(pkcorind(l)-1)*tout_step;
phlag(fkk,qkk)=rem((pkcorind(l)-1)*tout_step*sigmae(fkk),2*pi); %Find
the phase by which response lags forcing
end
% phlag(2*qkk-1,fkk)=tlag*sigmae(fkk); %Phase lag
% pkk=pkk+1;
% figure(pkk)
% plot((0:lagrange)/tout_step,roel2(:,2*qkk-1),'*-')
% title(['Cross-correlation with T='
num2str(round(2*pi/sigma0(fkk)*10)/10) ', T_e ='
num2str(round(2*pi/sigmae(fkk)*10)/10) '; Mode=' num2str(qkk) ])
% xlabel('Lag, s')
% ylabel('Cross-correlation')
% end
end
for qkk=1:6
%scaled
%RAO(fkk,:)=q_H(fkk,:).*[1 1 1 1 H(fkk)/k(fkk) 1]; %For pitch, added
would*2*2 makes slope equal 1 at low frequencies
RAO(fkk,:)=q_H(fkk,:).*[1 1 1 1/k(fkk) 1]; %For pitch, added would*2*2
makes slope equal 1 at low frequencies
end
elseif skk==1 %i.e. random seas %
%Calculate platform motion spectrum from model. Copied from Time Series final
#3
platform_freq=1/tout_step;
padyes=0;
ensembles=5;
bands=1
stds=100; %Standard deviations to filter--high # means don't filter
passes=1;
mean_bands=0
mean_bands(1)=round(20/tout_step); %mean surge over 20-second intervals
mean_bands(2)=0; %mean heave
mean_bands(3)=0; %mean pitch. CHECK current variation!
%Call function to calculate the power spectral density
ppk=pkk+1;
magnifier=ones(12);
magnifier(9)=5; %Factor by which to amplify pitch spectrum for plotting
ptcs={'g' 'k' 'r'};
clrkk=1;
for mdkk=[1 5 9]
  %Calculate and plot platform response spectra
  figure(pkk)
  hold all
  [platform_bandfreqs platform_sjlside(:,mdkk) platform_con95(:,mdkk)
  platform_bandwidth(:,mdkk) platform_sjare(:,mdkk)
  platform_var(:,mdkk)]=spectra(q(:,mdkk),platform_freq,ensembles,bands,padyes,stds,passes,mean_bands(mdkk));
  % Semilogy(bandfreqs(2:length(bandfreqs)),sjlside(2:length(sjlside)),'e y')
  plot(platform_bandfreqs(2:length(platform_bandfreqs))*2*pi,platform_sjlside(2:length(platform_sjlside)),mdkk)*magnifier(mdkk),ptcs(clrkk))
  hold off

  figure(pkk+1)
  hold all
  %Calculate and plot RAOS
  % Make sure spectra are the same length:
  % if length(platform_sjlside)>length(staff_sjlside)
  %   cutter=length(platform_sjlside)-length(staff_sjlside)
  % else
  %   cutter=0;
  % end
%staff_sjlside_int=interp1(platform_bandfreqs,staff_sjlside,platform_bandfreqs); %Get wave environment from IMUreader.m
  SED_int(:,mdkk)=interp1(sigmas,SED(:,mdkk),platform_bandfreqs*2*pi);
%Interpolate the wave environment in the model onto the banded platform frequencies
%MUST shift SED by u!
  RAOs(:,mdkk)=sqrt(platform_sjlside(:,mdkk)/SED_int(:,mdkk));
%Hold on
%plot(platform_bandfreqs*2*pi,RAOs(:,mdkk),'Color',ptcs(clrkk)) %Multiply freqs by 2pi to plot over rad/s
  if exist('staff_sjlside','var')==1
    plot(IMU_bandfreqs*2*pi,staff_RAOs(:,mdkk),'Color',ptcs(clrkk),'LineStyle','--')
  end
  clrkk=clrkk+1; %Counter to assign colors to plot
  hold off
end
figure(pkk)
hold all
% Return to this when comparing with data:
% plot(staff_bandfreqs*2*pi,staff_Sj1side(:,5)) % Plot wave spectrum with
response spectra
plot(sigmae,SED(:,5),':-')
% Check that wave spectra are the same
title('Platform Response Spectra')
xlabel('Frequency, rad/s')
ylabel('Spectral Energy Density')
legend('Surge, 'Heave', 'Pitch')
legend('DynamicLegend', 'Surge, m^2/Hz', 'Heave, m^2/Hz', 'Pitch, rad^2/Hz', 'Wave Forcing, m^2/Hz')
hold off
xlim([0 3.5])

figure(pkk+1)
title('Response Amplitude Operator')
xlabel('Frequency, rad/s')
legend('Surge, m^2/Hz', 'Surge (Ocean), m^2/Hz', 'Heave, m^2/Hz', 'Heave (Ocean), rad^2/Hz', 'Pitch, rad^2/Hz', 'Pitch (Ocean), rad^2/Hz', 'Wave spectra (heave)', 'SED wave spectra (heave)')
hold off
xlim([0 3.5])

pkk=pkk+1;
end

if dir==1
for qkk=1:3:5
% Sum responses to each sigmae
phi_sigs=0:0.01:max(sigmae); % Frequencies on which to interpolate results for the
% purpose of adding responses
[sigmax sigmax_ind]=max(sigmae); % Find location of maximum sigmae (for when dir==1)

% phi_sigs has no physical significance here; it is only convenient
% Break RAOs, lags into two interpolated curves (to remove indeterminacy)
RAOs_int1=interpl(sigmae(1:sigmax_ind),RAO(1:sigmax_ind,:),phi_sigs);
RAOs_int2=interpl(sigmae(sigmax_ind+1:length(sigmae)),RAO(sigmax_ind+1:length(sigmae)),
:,:),phi_sigs);
phlag_int1=interpl(sigmae(1:sigmax_ind),phlag(1:sigmax_ind,:),phi_sigs);
phlag_int2=interpl(sigmae(sigmax_ind+1:length(sigmae)),phlag(sigmax_ind+1:length(sigmae)),
:,:),phi_sigs);

% Decompose into components that can be added (cf. polar to
% rectangular).
Arao1=RAOs_int1.*cos(phlag_int1);
Brao1=RAOs_int1.*sin(phlag_int1);
Arao2=RAOs_int2.*cos(phlag_int2);
Brao2=RAOs_int2.*sin(phlag_int2);
% Replace NaNs with zeros (so that A1+Nan=A1).

end

end
A Rao1(isnan(A Rao1)) = 0;
A Rao2(isnan(A Rao2)) = 0;
B Rao1(isnan(B Rao1)) = 0;
B Rao2(isnan(B Rao2)) = 0;
C Rao = sqrt((A Rao1+A Rao2).^2+(B Rao1+B Rao2).^2);
eps Rao = tan((A Rao1+A Rao2)./(B Rao1+B Rao2));

figure(pkk);
plot(phi_sigs,C Rao);
title('Summed RAOs');
ylabel('RAO');
legend('Surge', 'Heave','Pitch');

end

% Find heave RAOs at bow and stern for single frequency waves
% if skk == 2 && length(sigmae)>12
% whatthe=squeeze(qout(round(end/4):end,5,:)+L/2*sin(qout(round(end/4):end,9,:)));
% RAObow=(max(whatthe)-min(whatthe))./H;
% pkk=pkk+1;
% figure(pkk);
% plot(sigmae,RAObow);
% whatthestern= squeeze(qout(round(end/4):end,5,:)-
% L/2*sin(qout(round(end/4):end,9,:)));
% RAOstern=(max(whatthestern)-min(whatthestern))./H;
% hold all
% plot(sigmae,RAOstern)
% end

catter.m

% Use fsolve to solve the catenary equations stored in cats.m
function [T1 phi_ml x_r y_r TO L_LmlK out exitflag]=catter(q,XO,h,L_Lml,S,p,K,outO,options)

% out(1)=T1
% out(2)=phi_ml
% out(3)=x_r
% out(4)=y_r
% out(5)=TO (T-horizontal)

% Use fsolve
[out,fval,exitflag] = fsolve(@(x) cats(x,p,S,q,L_Lml,h,XO,K,outO,options));

% % Or use fmincon
% A=[];
% b=[];
% Aeq=[];
% beq=[];
% lb=zeros(1,length(outO));
% ub=Inf*ones(1,length(outO));
% [out,fval,exitflag] = fmincon(@(x) catscons(x,p,S,q,L_Lml,h,XO),outA,b,Aeq,beq,lb,ub,@(x)catminconNLC(x,p,S,q,L_Lml,h,XO)


%)}

% Check that fsolve solved properly
% if exitflag==1
% NGood
% else
% exitflag
% end

% Check that some chain is on the floor
if S<out(3)
    age=input('Please enter your age:');
    age=30;
    if age>=30
        warning('Chain has lifted off of the seafloor')
    elseif age<30
        warning('This boat is off the chain!!')
    end
end

T1=out(1);
phi_m1=out(2);
x_r=out(3);
y_r=out(4);
T0=out(5);
L_Lmlk=out(6);

% %Show result:
% xcoord=0:0.1:out(3);
% plot(xcoord,out(5)/p*(cosh(p*xcoord/out(5))-1))
% axis equal

cats.m

% Supply the catenary equations
function [F]=cats(x,p,S,q,L_Lml,h,X0,K)

% x(1)=T1
x(2)=phi_m1
% x(3)=x_r
% x(4)=y_r
x(5)=T0
% x(6)=L_Lmlk

% if q(1)>0.5
% what=101
% end
% scaler=1000;
F(1)=x(5)-x(1)*cos(x(2));
F(2)=x(1)-x(5)*cosh(p*x(3)/x(5));
F(3)=q(5)+x(4)-x(6)*sin(x(2));
F(4)=x(4)-x(5)/p*(cosh(p*x(3)/x(5))-1);
\[
\text{Sup} = \frac{x(5)}{p \cdot \sinh(p \cdot x(3)/x(5))}; \quad \%m, \text{Length of chain off of the seafloor}
\]

\[
\text{Sdown} = -\text{Sup}; \quad \%m, \text{Length of chain on the seafloor}
\]

\[
F(3) = \text{Sdown} + x(3) + x(6) \cdot \cos(x(2)) - X_0 - q(1);
\]

\[
F(6) = (x(6) - L_{\text{Lm}}) \cdot x(1)/k;
\]

\begin{verbatim}
%Equations of motion reduced to 1st order for ode45

function \[qdot tester\]=mrsmotion(t,q,dT_\etaeta)
   global fkk H sigmaO sigmae k phase d dir ckk mkk skk h u_cur g rho L bm L_wp eps s L_Db
   L_RT L_Lm l L_Rml phi_ml a b Ab Bb C MB At Bt Mt Atot Btot Iwp C_Db A_PDb C_DT
   A_Pt T1 T0 X0 S p tind zeta odkk zeta_pass dzeta_dx_pass zetadot_t_pass
   zetaddot_t_pass q_pass t_pass qdot_pass wave(3,1); T_pass phi_ml_pass \[x_r_pass y_r_pass out_pass outO_pass x_r y_r_pass cfact hcfact eta3amp L_Dmb C_Dmb A_Pmb phi_ml2 bow_pass linchk_pass
   tic
   whattime=t
   if wrange=4\%pi/signae(fkk) && fkk==2
      what=16;
      end
   %Define forces
   %Note that turbine and cage are considered one item
   %Define fluid forcing on turbine
   %Calculate wave action at turbine hub (with q=zeros)
   z=L_RT;
   for wkk=wrange \%Go through once for single-freq waves; loop for skk=1 (random)
      \[zeta_p(wkk,:) zetadot_p(wkk,:) zetaddot_p(wkk,:) dzeta_dx_p(wkk)
      u_bar(wkk),]=wave(q(1),t,z,H(wkk),sigm(0(wkk),sigmae(wkk),phase(wkk),k(wkk),h,dir);
      end
   for dkk=[1 3]
      zeta_t(dkk)=sum(zeta_p(:,ddk));
      zetadot_t(dkk)=sum(zetadot_p(:,ddk));
      zetaddot_t(dkk)=sum(zetaddot_p(:,ddk));
      % dzeta_dx_t=sum(dzeta_dx_p);
   end
   if t<1 \%so that the solution at t=0+ is close to that at t=0
      zeta_t=t*zeta_t;
      zetadot_t=t*zetadot_t;
      zetaddot_t=t*zetaddot_t;
      % dzeta_dx_t=t*dzeta_dx_t;
      end
   %Note: The effect of barge motion is included in the turbine forcing
   %because it is non-linear.
   Ft(1)=1/2*rho*C_Dt*A_Pt*zetadot(1)+u_cur(fkk)-q(2)*Abs((zetadot_t(1)+u_cur(fkk)-q(2)))*At(1,1)*zetadot_t(1); Velocity term will include -q(10)*l_Rt once Bb is used instead of Bt
   Ft(3)=0; %/1/2*rho*C_Dv*At(3)*zetadot(3)*Abs(zetadot(3))+At(3,3)*zetaddot(3)
   (presently included in Fb(3))
   Ft(5)=Ft(1)*L_Rt;
   Ft(6)=0;
%end

end
\end{verbatim}
% Define fluid forcing on barge

% Calculate wave action at surface
% Sum partial contributions from each wave
z=0; L_Ob;
for wkk=wrange % Go through once for single-freq waves; loop for skk=1 (random)
    [zeta_p(wkk,:), zetadot_p(wkk,:), zetaddot_p(wkk,:), dzeta_dx_p(wkk)]
    u_bar(wkk))=wamve(q(l), t, z, H(wkk), formula, phase(wkk), k(wkk), h, dir);

% Apply heave correction factor if specified
if cfact(3)==2
    % Simplified MRS correction factor (does not include pitch-coupling);
    % Modified to include surge coupling (at least sort of).
    zeta_p(wkk,3)=sincme(k(wkk)*L/2)*zeta_p(wkk,3);
    zetadot_p(wkk,3)=sincme(k(wkk)*L/2)*zetadot_p(wkk,3);
    zetaddot_p(wkk,3)=sincme(k(wkk)*L/2)*zetaddot_p(wkk,3);

% Failed correction factor?
    zeta_p(3)=H(wkk)/L/(k(wkk)*cos(q(9)))*cos(2*k(wkk)*q(1)-
    end

% Apply surge correction factor if specified.
if cfact(1)==2
    zeta_p(wkk,1)=sincme(k(wkk)*L/2)*zeta_p(wkk,1);
    zetadot_p(wkk,1)=sincme(k(wkk)*L/2)*zetadot_p(wkk,1);
    zetaddot_p(wkk,1)=sincme(k(wkk)*L/2)*zetaddot_p(wkk,1);
end
end
zeta=sum(zeta_p,1);
zetadot=sum(zetadot_p,1);
zetaddot=sum(zetaddot_p,1);
dzeta_dx=sum(dzeta_dx_p);

% Correct for hull interaction. % Not finished! % Must stay after heave correction
% factor is applied or the heave correction factor will incorrectly be applied
% to the interaction wave.
if t>pi/phase
    if hfact==2
        if q(6)*q_pass<0
            eta3amp=abs(q(5));
        end
        Amp(fkk)=eta3amp*bm*k(fkk)*(cos(k(fkk)*L WP)+sin(k(fkk)*L WP/2))
        zeta(3)=eta3amp*Cos(k(fkk)*L WP/2-2*phase(k(fkk)))+
    end
end
end

% if skk<4
if t>1 % So that the solution at t=0+ is close to that at t=0
    zeta=t*zeta;
    zetadot=t*zetadot;
    zetaddot=t*zetaddot;
    dzeta_dx=t*dzeta_dx;
end

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end

% end

% u_tot = u_cur(fkk)(fkk) + zetadot(1); %Total fluid velocity at barge DUE TO WAVES (and currents) (Not ship motion)
Fb(1)=1/2* rho*(C_Db*A_PSb+C_Db*Pmb)*zetadot(1)+u_cur(fkk)-q(2)*abs((zetadot(1)+u_cur(fkk)-q(2)))+Btot(1,1)*(u_cur(fkk)-q(2))*ab(fk,1,1,fk)*zetadot(1); %Surge forcing on barge

% ("B"tot" because "B" gets zeroed in mrsmod)
% Long-wave assumption
Fb(3)=(C(3,3)*zetadot(3)+Mb(3,3)+Atot(3,3))*zetadot(3)*exp(-k(fkk)*z); % Long wave assumption

if cfact(S) == 1
   % Fb(3) = C(S,5)*dzeta_dx; % Pitch force on barge - Assuming center of drag (of barge)
   %CG
else
   % Allows for for superposition of waves:
   for wkk > wrange
      Fb_p(wkk,5) = rho*g*2*bm*H(wkk)/2*(-L*cos(k(wkk)*L/2)/k(wkk)+2*sin(k(wkk)*L/2))*sin(dir*sigmae(wkk)*t+dir*q(l)*k(wkk)+phase(wkk)+q(l)*L*sinme(k(wkk)*L/2)*cos(k(wkk)*q(l)-dir*(sigmae(wkk)*t+phase(wkk))); %+/q(l)*k?
% With dir changed--Wrong results
% Fb_p(wkk,5) = rho*g*2*bm*H(wkk)/2*(-L*cos(k(wkk)*L/2)/k(wkk)+2*sin(k(wkk)*L/2))*sin(q(l)*k(wkk)-dir*(sigmae(wkk)*t+phase(wkk)))+q(l)*L*sinme(k(wkk)*L/2)*cos(k(wkk)*q(l)-dir*(sigmae(wkk)*t+phase(wkk)));
   end
   Fb(:,S) = sum(Fb_p(:,5), 1);
   %CG
end
Fb(6)=0;

% Salvesen, etc. formulation of forcing
% What about C?
Ff(1) = Fb(1); % Faltinsen doesn't address surge in this formulation
Ff(3) = (H(fkk)/2*exp(-1*sigma0(fkk)*t)*2*(2*sin(k(fkk)*L/2))/2*(bm*g*rho*sigmae(fkk)*sigma0(fkk)))*a(3,3,fkk)+1i*sigma0(fkk)*b(3,3,fkk))/(exp(L_DB*k(fkk)*s)+1i*exp((-L_DB)*k(fkk)*s+1i*k(fkk)*L*cos(k(fkk)*L/2)/k(fkk))+1i*exp((L_DB)*k(fkk)*s+1i*k(fkk)*L*sin(k(fkk)*L/2)/k(fkk)));
Ff(5) = (H(fkk)/2*exp(-1*sigma0(fkk)*t)*2*(-(exp(-(-1/2)))1i*k(fkk)*L-L_DB*k(fkk)*)*(1-1i*(k(fkk)*L/2))/2)*rho*sigmae(fkk)+exp(1i*k(fkk)*L)*(bm*g*(1-i(k(fkk)*L/2))/2)*rho*sigmae(fkk)+(-sigmae(fkk)+k(fkk)*u_cur(fkk)+c(l)*l*sigmae(fkk))/2)*sigma0(fkk)+sigmae(fkk)*a(3,3,fkk));
Ff(6) = 0;

% Sum components
\[ F_{tot} = F_b + F_t; \text{ Total wave forcing on barge and turbine} \]

\textbf{if} \( ckk < 4 \)
\begin{itemize}
  \item \textbf{if} \( dkk = 0 \)
    \begin{itemize}
      \item \( out0 = out0(1:kk); \text{ out0} \)
      \item \( \text{options=optimset('Display','off','MaxFunEvals',10000,'MaxIter',10000);} \text{ Option to display output} \)
      \item \( \text{else} \)
        \begin{itemize}
          \item \( \text{options=optimset('Display','off');} \text{ , 'MaxFunEvals',10000,'MaxIter',10000);} \text{ else} \)
        \end{itemize}
    \end{itemize}
  \end{itemize}
\textbf{end}
\[
L_{Lmk} = L_{Lm} + out0(1)/k + 2; \text{ Elastic line length} \]
\textbf{end}
\[
\text{elseif} \quad L_{Lmk} = 0; \text{ Elastic line length} \]
\[
X = 0; \text{ end} \]
\[
\text{if} \quad X0 + q(1) < S \pm \text{sqrt}(L_{Lmk}) \quad \text{end} \]
\[
\text{if} \quad ckk < 4 \& \& X0 + q(1) > S \pm \text{sqrt}(L_{Lmk}) \quad \text{Check that some chain is off the ground} \quad \text{(necessary for u_cur(fkk)=0).} \]
\[
\text{if} \quad ckk = 1 \text{ Solve using exact catenary equations} \]
\textbf{elseif} \( ckk = 2 \)
\begin{itemize}
  \item \( T(1) = -(T(l) \times \cos(phi_{ml}(l)) + dT_{eta}(l,l) \times q(l) + dT_{eta}(l,3) \times (q(5) - L_{Rml} \times q(9))); \text{ Neglecting the change in horizontal position of mooring point due to pitch} \)
  \item \( T(3) = -(T(3) \times L_{Rml}); \text{ Neglecting the change in horizontal position of mooring point due to pitch} \)
  \item \( T(6) = 0; \text{ } \)
\end{itemize}
\[
\text{elseif} \quad ckk = 3 \text{ Optional (and very approximate) rear mooring line: Nada} \]
\[
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% Spring approximation

C_K=110; %N/m, Arbitrary mooring line stiffness
if q(1)>0
    T(1)=(T0-C_K*sqrt(q(1)^2+q(5)^2))/cos(phi_m); %
else
    T(1)=0;
end
T(3)=T(1)*tan(phi_m);
T(5)=-T(3)*L_ml;
T(6)=0;
elseif ckk==4
    T=zeros(1,6);
else
    T=zeros(1,6);
    T1=0;
    phi_m1=0;
    x_r=0;
    y_r=0;
end
if L_Dml>0 %If a mooring ball is used
    T(3)=0;
    T(5)=0;
end
if t<1 %So that the solution at t=0+ is close to that at t=0
    T=T;
end
qdot=zeros(length(q),1);
q(1)=etal
q(2)=etal_dot

% Solve equations of motion for each DOF (CAUTION: Only calculate filled
% rows, to avoid scaling problems with ODE45
% Atot(1,1)=At(1,1)
for kkk=1:2:5
    qdot(2*kkk-1)=q(2*kkk);
    qdot(2*kkk)=(Ftot(kk)+T(kk)-Btot(kk,kk)*q(2*kkk)-C(kk,kk)*q(2*kkk-1))/(Mtot(kk,kk)+Atot(kk,kk));
end

% Check for water-on-deck/wave slamming:
% Find surface height at bow
for wkk=wrange %Go through once for single-freq waves; loop for skk=1 (random)
    [zeta_p(wkk,:), zetadot_p(wkk,:), zetaddot_p(wkk,:), dzeta_dx_p(wkk), u_bar(wkk)] = wave(q(l)-L/2,t,0,H(wkk),sigma0(wkk),sigmae(wkk),phase(wkk),k(wkk),h,dir);
end
zeta_b(3)=sum(zeta_p(:,3));
eta_b(3)=q(5)+L/2*sin(q(9));
bower=[zeta_b(3) eta_b(3)];
tester=q(5);
linck_pass=[linck_pass; Ft(5) Fb(5) Ftot(10) zetadot_t(1)+u_cur(fkk)-q(2)];
qdot_pass=[qdot_pass; qdot']; % assigned to fkk in main fkk loop
t_pass=[t_pass t];
zeta_pass=[zeta_pass; zeta];
dzeta_dx_pass=[dzeta_dx_pass; dzeta_dx];
zetadot_t_pass=[zetadot_t_pass; zetadot_t];
zetaddot_t_pass=[zetaddot_t_pass; zetaddot_t];
q_pass=[q_pass; q'];
bow_pass=[bow_pass; bowe];
T_pass=[T_pass; T];
if ckk==1
    T1_pass=[T1_pass; T1];
    phi_m1_pass=[phi_m1_pass; phi_m1];
    x_r_pass=[x_r_pass; x_r];
    y_r_pass=[y_r_pass; y_r];
    out_pass=[out_pass; out'];
end
odkk=odkk+1;
timesolve=toc;

% draughts=[out0(1) T1(1)];
% checker=[l_L_m1 k_L_m1+1(1)/K];

wave.m

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