DESIGN OF A MOORING SYSTEM FOR AN INERTIA TUBE WAVE ENERGY CONVERTER

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INTRODUCTION

The Healy [1] buoy-inertia tube wave energy converter (WEC) is under development using a scale-up approach incorporating mathematical modeling. The feasibility of the concept was evaluated using a vertical (heave) motion mathematical model in which empirical coefficients were determined and the mathematical model validated using wave tank testing. The predicted WEC performance results were sufficiently positive that a system with a 10 kW power capacity is now in the design phase for testing at sea. The test location is the University of New Hampshire (UNH) Center for Ocean Renewable Energy (CORE) site south of the Isles of Shoals, NH. This paper describes the design of the WEC's mooring system.

A mooring system, consisting of anchors, chain, rope and associated hardware, was developed for the buoy-inertia tube device designed by Healy [1]. Initial design concepts were based on previous moorings used at the CORE site. The dynamics of the WEC mooring system were investigated using the UNH developed finite element computer program Aqua-FE [2] and wave tank testing. The Aqua-FE model was then used to design a slack mooring system sufficient for holding the WEC on station while minimizing interference with its energy absorption function. The Aqua-FE model was created and validated by comparison to wave tank measurements made using a 1/9.4 scale physical model in experiments conducted in the UNH 36.6 m long by 3.66 m wide by 2.44 m deep wave tank. The Aqua-FE model was then applied to predict full scale response to seas representing extreme storms expected at the UNH CORE offshore test site. Predicted mooring loads were used to specify mooring system hardware.

The WEC design considered here makes use of a buoy rigidly connected to a long, vertical inertia tube which is open at the top and bottom (see Figure 1). A piston-rod assembly is enclosed and connected to the power take-off (PTO) mechanism. Due to inertia of water within the tube, relative motion between the piston and buoy-inertia tube structure occurs, and this drives the PTO. In this design, the PTO consisted of an air compressor and turbine connected to a generator. The overall mass was 30,000 kg, the buoy diameter was 4.98 m, and the inertia tube length was 9.10 m.

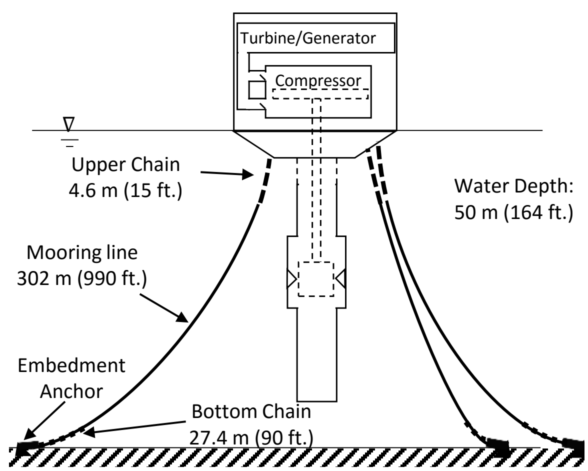

FIGURE 1. BUOY-INERTIA TUBE WEC. THE PISTON ROD DRIVES COMPRESSED AIR THROUGH A UNIDIRECTIONAL TURBINE, WHICH IS CONNECTED TO A GENERATOR. THREE-LEGGED MOORING SCHEME HOLDS BUOY ON STATION. NOT TO SCALE.

MOORING DESIGN CONCEPT

To minimize interference with the buoy's wave response, the mooring should be a slack, single or three-line system using embedment anchors. The single-line system might be used for a temporary, test deployment in which generated power is measured and dissipated by an on-board resistor bank. The three-line system, shown in Figure 1, would be used to hold the system in place when a power cable to the sea floor is used to transmit power to a shore location. The three-line system also provides redundancy for long-term deployment applications.

The design process was based on previous experience with mooring systems used to deploy large fish cages and feed buoys, as well as other WECs at the Isles of Shoals site. In each case, hardware and anchor decisions were made based on mooring line loads predicted using the finite element computer program Aqua-FE.

AQUA-FE WAVE RESPONSE COMPUTER PROGRAM

Aqua-FE was used because it can simulate the effects of wave, current and storm events on large complex systems for which motion and mooring loads are of interest [3, 4]. Whereas boundary element methods are valid only for small-amplitude displacements, the finite element approach accounts for variation in buoyancy, drag, and added mass due to the large heave and pitch motions which the device exhibits in extreme seas.

Initially developed to model aquaculture cages, buoys and mooring systems, Aqua-FE incorporates truss, buoy, and stiffener elements. It uses a nonlinear Lagrangian formulation to accommodate large displacements of the finite elements. The Newmark integration scheme is utilized to solve the nonlinear equations of motion. Wave and current loading on truss elements was incorporated into the model using a Morison equation formulation [5] modified to include relative motion between the structural element and the surrounding fluid. The program calculates both the normal and tangential drag coefficients, at each time step, as a function of Reynolds number as described by [6] who revised the approach taken by [7]. This research computer program can, however, be modified to employ user-specified drag coefficients and incident fluid flow. Aqua-FE has been extensively used to study a variety of aquaculture and buoy systems and has compared well with physical model testing and *in-situ* experiments for different buoys, cage types and mooring configurations [8-12].

Based on drawings and weight information provided by Healy [1], an Aqua-FE finite element model, shown in Figure 2, was generated for the buoy-inertia tube configuration. Since the objective was to predict maximum mooring loads during extreme storms, the piston in the numerical model was locked to the buoy to simulate the device's survival mode.

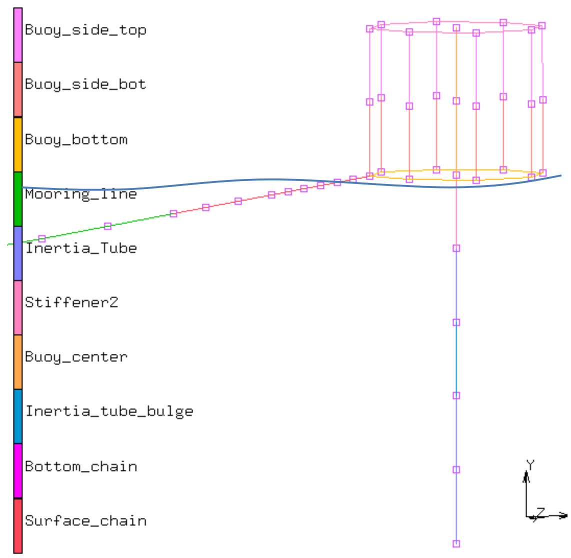

FIGURE 2. AQUA-FE MODEL OF WEC DEVICE. LINES REPRESENT ELEMENT AXES. STIFFENER ELEMENTS ARE NOT SHOWN. APPROXIMATE STILL WATERLINE (BLUE) IS SHOWN FOR REFERENCE.

In Aqua-FE, solid bodies like the buoy and inertia tube are represented by an array of truss and stiffener elements, and different combinations can be used to best represent the damping, mass and buoyancy characteristics of the system's components. In this application, these decisions were made by comparing Aqua-FE predictions with previously obtained wave tank data using a 1/9.4 scale physical model [13]. Element properties were optimized for best agreement between Aqua-FE predictions for the full-scale system and tank measurements Froude-scaled to full-size as described in the next section.

WAVE TANK TESTING

The Aqua-FE model for this system was evaluated using scale model results for free-release tests in heave (vertical displacement) and pitch (angular motion), as well as heave response to a series of monochromatic waves. Wave periods, Froude-scaled to full size, spanned the range of periods observed at the UNH site. A scaled slack mooring was used to hold the buoy-inertia tube physical model in position, while motion was recorded using an optical system. In these tank tests, the PTO was represented by a linear spring and damper as shown in Figure 3.

The 1/9.4 scale physical model buoy was fabricated from closed cell foam, and the inertia tube from PVC pipe. The piston-rod assembly was also made from plastic. The spring was selected to have a linear spring constant within the range of values (scaled) provided by the developer of the compressor/turbine PTO. The air damper was procured using a similar process. Both model scale PTO components were calibrated using a "dry" free-release test of the piston. The physical model hull was intentionally built underweight so that mass changes could easily be incorporated through ballasting. For the tests described here, a suitable pipe flange was used as ballast to achieve the design weight.

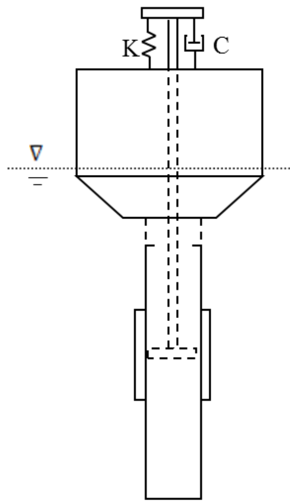

FIGURE 3. SCHEMATIC OF THE BUOY-INERTIAL TUBE DESIGN CONCEPT (NOT TO SCALE). THE PTO UNIT WAS MODELED AS A SPRING (WITH STIFFNESS K) AND DAMPER (WITH DAMPING COEFFICIENT C) IN THE 1/9.4 SCALE MODEL.

Experimental results were scaled up to full scale using Froude scaling. Froude number in the tank tests is taken to be the same as that full scale. As a consequence, the ratio of velocities and time scales is proportional to the scale ratio to the one-half power. Forces are proportional to scale ratio cubed, and power to scale ratio raised to the 3.5 power.

Before wave tests, the physical model was subject to "free release experiments". The model was lifted slightly from its equilibrium position and released. The induced decaying oscillations were recorded optically. Besides directly providing the heave damped natural frequency, a buoy virtual mass-effective damping-buoyancy restoring force mathematical model, without wave forcing terms, was fit to the heave time series yielding added mass and damping constant. The best fit yielded a full scale added mass of 23,270 kg, and a damping coefficient of 23,500 N/(m/s). The damped natural period in heave was 3.2 s. For additional information, the test was applied to pitch yielding a pitch damped natural period of 12.3 s.

The complete physical model for the wave energy device was tested in monochromatic waves spanning the normal range of device operation. The buoy was held in place by a high scope, lightweight mooring line connected to a chain terminating at a sinker-weight on the tank floor. Wave surface position, buoy heave (vertical displacement), pitch (angular displacement) and piston motion relative to the buoy were recorded optically. Time series for heave were processed to yield oscillation amplitudes normalized by wave amplitudes. The non-dimensional result, as a function of frequency, is commonly used in seakeeping analysis and is referred to as a response amplitude operator (RAO).

AQUA-FE MODEL CALIBRATION

The properties of the Aqua-FE model elements shown in Figure 3 were specified such that the overall characteristics of the model matched those of the prototype. The total cross-sectional area of the vertical members intersecting the waterplane was calculated to match the waterplane area of the buoy. The correct area moment of inertia of the waterplane (important for pitch dynamics) was achieved by properly distributing the cross-sectional area between central and outer vertical members. Representing a single vertical cylinder as nine cylinders with the same total cross-sectional area increases the total projected area, so the drag coefficient of the vertical members was decreased proportionally.

Hydrodynamic radiation effects were incorporated using results from a free-decay test of the Froude-scaled physical model of the buoy and inertia tube with the piston removed. The diameter of the submerged horizontal buoy members determines the system's added mass in heave. Thus, this diameter was adjusted so that the model's heave natural period matched that calculated from vertical free-release tests of the physical model. The drag coefficient of those members was then set so that the model's damping coefficient in heave matched that calculated from the same vertical free-release tests. Finally, the vertical distribution of mass between sections was iterated until the pitch natural period of the Aqua-FE model matched the value that was calculated from a pitch free-release test of the physical model.

Since the wave tank study was focused on normal operation instead of extreme events, the PTO was active during those tests. To account for the energy absorbed by the power take-off, the drag coefficient of the horizontal buoy members was increased until the heave displacement predicted by the Aqua-FE model reasonably matched the monochromatic wave tank results, as shown in Figure 4. The resulting effective densities and cross-sectional areas of each member are provided in Table 1.

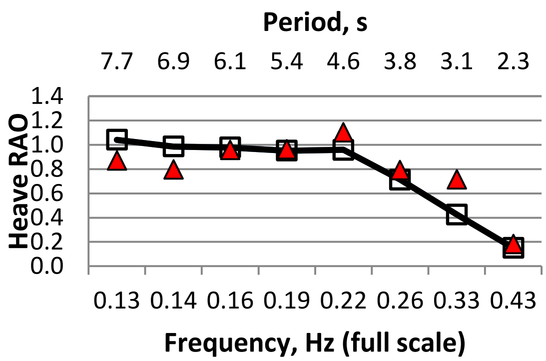

FIGURE 4. RESPONSE AMPLITUDE OPERATOR FROM AQUA-FE PREDICTIONS COMPARED TO WAVE TANK RESULTS FOR MONOCHROMATIC WAVES.

TABLE 1. COMPONENTS OF WEC AQUA-FE™ MODEL.

Component	Elements per component	Effective density (kg/m^3)	Cross-sectional Area (m^2)
Buoy			
Upper vertical members	8	149.1	1.5
Lower vertical members	8	596.4	1.5
Lower horizontal members	8	253.7	0.24
Central vertical members	2	89.87	7.46
Inertia tube			
Uniform tube section	3	1193	1.76
Piston chamber	1	1821	2.63
Mooring			
Surface chain	9	1371	0.00528
Mooring line	147	1167	0.00456
Mooring chain	20	1376	0.00633
Total	206		

APPLICATION TO THE ISLES OF SHOALS TEST SITE

Once the Aqua-FE model was calibrated, it was used to compute storm wave response at the Isles of Shoals site using trial mooring system designs. The starting point was the embedment anchor, bottom chain and fiber rope design used to moor the 84 ton aquaculture feed buoy described by [14]. For the three-leg mooring system, the worst-case situation was taken to be that only one leg was holding the buoy in position. Thus, effectively, both the single and three-leg system corresponded to the same worst-case design condition.

Wave input was chosen to represent typical northeaster and superstorm conditions at the UNH Isles of Shoals wave energy test site. The initial design wave consisted of a 9 m high, 8.8 second period wave, which corresponds to a 50-year return sea-state at the CORE site [15]. To provide an assessment of the combined effects of waves and tidal current, this wave input was combined with a 10 cm/s collinear current. In addition, a wave representing the worst case conditions due to the last major hurricane, Sandy, was applied. The wave height was 11.6 m, which was the largest wave height recorded at the nearby National Data Buoy Center site at Jeffrey's Ledge [16]. The period used was 11.1 seconds corresponding to the peak storm wave energy period recorded during Sandy at Jeffrey's Ledge.

MOORING SYSTEM DESIGN

The initial mooring system was found to be adequate, except that in Sandy waves, the bottom mooring chain was lifting off the bottom slightly all the way to the embedment anchor. The bottom chain size was therefore increased so that the load on the anchor would always be horizontal.

The largest mooring line force occurred during the superstorm Sandy design wave case and was 175,800 N (40,000 lbs). This value was used to specify mooring line components which are listed in Table 2, along with pertinent capabilities and factors of safety. All safety factors meet or exceed international standards as published by [17]

TABLE 2. MOORING COMPONENTS.

Component		Reference load	Safety Factor
Anchor:	750 kg Stingray	440,000 N[a]	2.5
Shackles:	2 inch shackles	352,000 N[b]	2.0
Bottom chain:	2.5" stud-link grd3	2,129,600 N[c]	12.0
Line:	3" 12-plait Nylon	1,078,000 N[d]	6.1
Upper chain:	2" stud-link grd3	1,399,200 N[c]	8.0

[a] Holding strength
[b] Working load
[c] Proof load
[d] Breaking strength

Quotes for the recommended components were obtained from distributers. The results, in 2014 dollars, are given in Table 3. Component dimensions are shown in Figure 1.

TABLE 3. MOORING COMPONENT COSTS (2014).

Component		Qt./leg	$/leg	Cost for 3-leg	Source
Anchor	750 kg Stingray	1	$2,240	$6,721	Cortland NY
Shackles	2 inch shackles	4	$1,700	$5,100	Anchor MS Houston, TX
Bottom chain	2.5" stud-link grd3	90	$4,995	$14,985	Anchor MS Houston, TX
Line	3" 12-plait Nylon[e]	990	$14,355	$43,065	Cortland NY
Upper chain	2" stud-link grd3	15	$750	$2,250	Anchor MS Houston, TX
	Total		**$24,040**	**$72,121**	

[e] Ships with thimble terminations on each end.

The three-leg mooring system specified has built-in redundancy and adequate safety factors for the most severe storms at the CORE site actually observed over the last 15 years. Though unlikely, it is of course possible that, with an increasingly energetic environment, even worse sea states could occur in the future. Nevertheless, the recommended components are consistent with those successfully used year-round on similar sized structures at the CORE site.

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