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HEAVE PLATE DYNAMICS FOR A POINT ABSORBING WAVE ENERGY CONVERTER

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ABSTRACT

Wave energy converters (WECs) often include a submerged heave reaction plate located at a depth where the surrounding water is minimally disturbed by the passing surface waves. Heave plates provide a relatively stationary and artificially massive body to which WECs transfer the reactionary forces generated as their power-takeoffs (PTOs) resist the motion induced by the incoming waves. To date little research has focused on the dynamics of heave plates in the context of wave energy. This paper provides a review of relevant literature, much of which focuses on heave plates used to reduce the heave response of SPAR type oil and natural gas platforms. A numerical analysis of the dynamics of symmetric and asymmetric heave plates is also presented. The time-domain solver TD_xxx was used to simulate the WEC and heave plate dynamics in both sinusoidal and spectral seas. The phase relation of the heave plate's motion relative to that of the surface float provides incite into the importance of added mass in determining system dynamics. The effects on system loading that result from heave plate vertical asymmetry are also considered.

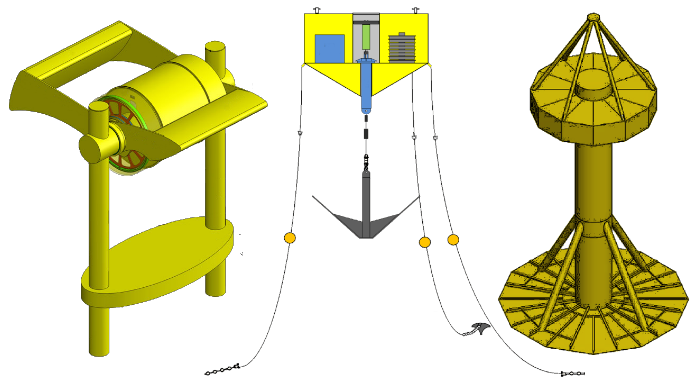

FIGURE 1. WECs often use a heave plate as an integral part of their PTO; a few examples are shown above. From left to right they are Columbia Power Technologies' StingRay, Oscilla Power's TDU2, and the Department of Energy's Reference Model 3.

1 Introduction

Heave plates are a common component of many current and former WEC designs (Figure 1). Heave plates are used to transfer the reaction forces of the PTO to the relatively stationary water deep below a WEC. Despite the key role that heave plates play in determining WEC dynamics and PTO efficiency, they have received little published research in the context of wave energy. Although the coupled dynamics of heave plate and WEC have been considered by some authors [1–4], a literature review revealed only two patents and no technical papers that focused specifically on the heave plate [5, 6]. It is also common to completely remove heave plate dynamics from numerical models by assuming either a rigid attachment to the bottom, or a tension mooring [7].

Section 2 presents the basic theory needed to understand the hydrodynamic behavior of heave plates, and Section 3 provides a brief review of some of the papers focused on heave plate dynamics. In Section 4, the initial results of a time-domain analysis of WEC motions and loading are presented. A modified version of the open source Matlab toolbox TD_xxx, written by Andrew Hamilton of the Monterey Bay Aquarium Research Institute (MBARI), was used to perform the analysis [8]. Specifically, the effects of heave plate hydrodynamic asymmetry on system motion and PTO loading are presented.

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2 Basic Theory

A body accelerating through water will accelerate some of the water that surrounds it. This added system inertia is called added mass. Heave plates accelerate a large volume of water, and thus the inertia of the added mass can dominate the dynamics of the heave plate. The force applied to a heave plate which is required to accelerate the added mass is in phase with the acceleration of the plate. The coefficient of added mass (C_a) can be defined as the mass of the water being accelerated (m_w) divided by the mass of the water displaced by the heave plate ($C_a = (m_w)/(\rho_w V_d)$ [9, §4.15]. However, due to the relative thinness of heave plates, the mass of the water they displace is very small in comparison to their added mass. For this reason the added mass is often non-dimensionalized in different ways, and values given as the coefficient of added mass may be inconsistent between publications. It is always important to check how the value was non-dimensionalized. Within this paper, I will use the term qualitatively to express the effect of design variables on the added mass of the system.

Water in the path of the plate will be forced around the edges of the plate forming vortices and turbulence that dissipate energy. This effect is known as form drag. The force required to overcome drag is in phase with the velocity of the heave plate. The drag coefficient (C_d) is defined as:

$$C_d = \frac{2F_d}{\rho v^2 A}, \tag{1}$$

where F_d is the drag force, v is the velocity of the heave plate, and A is the planform area of the heave plate.

In practice it can be difficult to separate the component forces associated with added mass and drag, and understanding the phase relation of the two component forces is critical to accurately separating their contribution to the net force acting on the heave plate.

3 Literature Review

In 1999, the oil and natural gas industry began incorporating heave plates into the design of SPAR-type deep-water platforms as a means of reducing their heave response to surface waves [10]. Wang et al. provide an overview of the design practices for SPAR platforms [11]. Oil and natural gas extraction continues to move into deeper, more energetic water as shallow-water resources are consumed. To aid this move, a small body of literature has developed that considers the hydrodynamics of heave plates. Heave plates are also being used for stabilization on floating offshore wind platforms [12].

Lake et al. (1999) found that C_a and C_d varied linearly with the Keulegan-Carpenter number (KC) [10]. The Keulegan-Carpenter number is now commonly used to provide a non-dimensional measure of the flow around a heave plate

$$KC = \frac{\pi a}{r} = \frac{2\pi a}{D} \tag{2}$$

In the preceding equation, a is the amplitude of oscillation, r is the radius of the heave plate, and D is the diameter. When experiments are performed using square plates instead of discs, D is replaced with the edge length (L) in KC.

Several studies have focused on the design parameters related to the hydrodynamic performance of heave plates [13]. Perhaps the most intuitive effect on vessel dynamics associated with the use of heave plates is an increase in damping. Damping is the effect of drag on an oscillating system; an increase in drag will increase system damping. As mentioned in Section 2, drag is related to the quantity and strength of the vortices generated as water flows around the edges of a heave plate. Therefore, drag is related to the ratio of edge length to surface area, the thickness of the plate, and the shape of the edges.

The published studies on heave plate hydrodynamics have focused on the effects produced by varying several design parameters: plate porosity, thickness, and edge shape. In addition, multiple studies have considered the efficacy of stacking heave plates and the ideal distance between stacked plates. Varying these design parameters also effects their ability to trap water, altering the added mass of the system.

In 2013, Li et al. published the results of a comprehensive study that considered many of the variables effecting drag and added mass [14]. The relevant design parameters were systematically varied to quantify their effect on both heave and added mass at KC numbers ranging from 0.2 to 1.2. They found that increasing KC reduced the coefficient of drag to a value of 6.0 at a KC of 0.6, above which C_d is approximately invariant. Porosity drastically increases C_d for heave plates oscillating at low KC, but the effect diminishes as KC increases to a value of 1.0. Added mass is reduced by porosity at all KC, as more water is allowed to pass through the plate instead of being accelerated with it. The size of the holes used to make the porous plates had little effect on C_d or C_a. For thin heave plates, it is more difficult for the water to make the turn around the corners, which increases the strength of the resulting vortices, increasing C_d. For the same reason, plates with sharp edges produced the greatest C_d. However the effect of edge shape was minor, and the ocean will rapidly blunt sharp edges.

For some WECs, it may be desirable for the hydrodynamic coefficients of a heave plate to vary with the direction of motion. For example, a heave plate may have a C_d or C_a that is greater when moving up than it is when moving down. Hydrodynamic asymmetry is discussed in Mundon et al. (2014); this paper was published after the deployment and testing of Oscilla Power's TDU2 WEC (Figure 1) [15]. Asymmetric heave plates may be

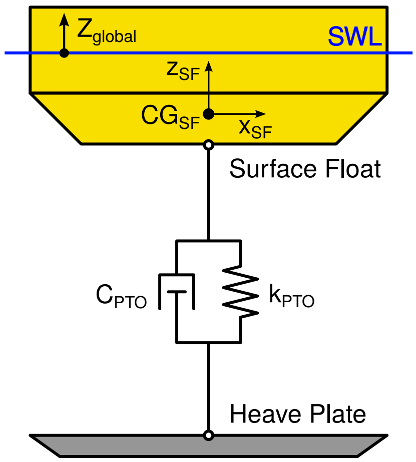

FIGURE 2. The modeled WEC consists of a surface float, heave plate, and PTO. Both the surface float and the heave plate have local coordinate systems. The global coordinate system is oriented with the origin at the SWL and positive Z up.

capable of limiting the number of slack load events in WECs that use a line under tension to connect their PTO to the heave plate. Some examples of devices and developers that have used this design include the L10 WEC built by Oregon State University and Columbia Power Technologies in 2008, the PowerBuoy built and deployed by MBARI, and all of Oscilla Power's designs [15–17].

4 Time-domain Analysis

A WEC similar in design to the MBARI PowerBuoy was chosen for analysis. A schematic of the design is provided in Figure 2. A simple float with a diameter of 2.5-m and draft of 0.9-m is attached to its heave plate by a line under tension. The PTO is part of the line, and can be modeled as a spring and damper. The surface float has a mass of 2050-kg with its center of gravity located 0.38-m below the still water line (SWL), and its center of buoyancy 0.22-m below the SWL. The heave plate is suspended below the buoy at a depth of 20-m. The diameter of the heave plate is 3.57-m, which provides a planform area of 10-m^2. The mass of the heave plate is 916-kg with a thickness of 0.023-m, which leads to a wet weight of 6,688-N (approximately 1500-lbs). The PTO spring constant (k_{pto}) is specified as the wet-weight of the heave plate divided by half the maximum PTO stroke. A 4-m stroke results in a $k_{PTO} = 3,344$-N/m. PTO damping is modeled with a linear damping coefficient of $c_{PTO} = 10,000$-$N/(m/s)$.

Hydrodynamic coefficients for the buoy were determined using WAMIT, and impulse response functions were developed from those coefficients. The symmetric added mass of the heave plate was analytically determined to be approximately 25,000-kg

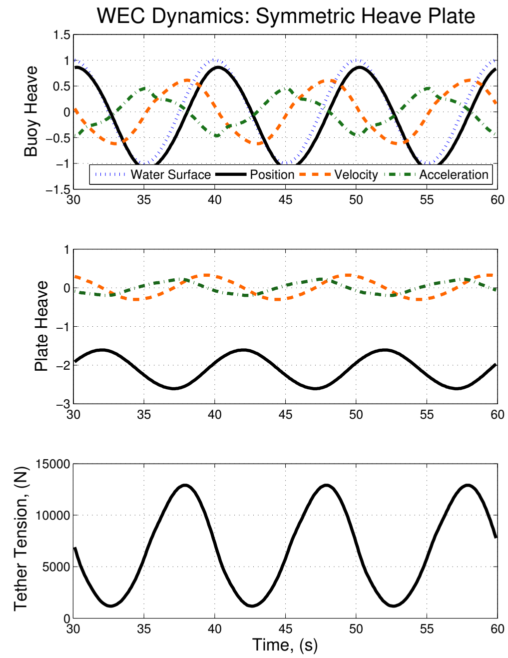

FIGURE 3. The motion of a surface float and a hydrodynamically symmetric heave plate connected by a PTO tether were modeled in 2-m 10-s sinusoidal waves. Units: Position (m), Velocity (m/s), and Acceleration (m/s^2)

based on the form of the heave plate, as described by Newman (1977) [9, §4.15]. In our study, the WEC will be operating at a $KC = 1.7$. The symmetric C_d for the heave plate was set to 6.0; the value to which the coefficient of drag converges for $KC > 1$ in the studies performed by Li et al. (2013) [14].

For the asymmetric analysis an added mass coefficient 1.3 times greater than that of a flat plat was used for upwards motion, and a value of 0.7 times the added mass of the flat plate was used for downwards motion. Asymmetric drag was specified with a C_d for upwards motion equal to 8.0, and equal to 4.0 for downwards motion. We believe these values are reasonable, and plan to perform physical tests within the year to provide hard numbers for future analyses.

Simulations were conducted using TD_xxx [8]. TD_xxx is similar to, although not as refined as, programs like OrcaFlex and ProteusDS. It uses a Runge-Kutta method (RK4) to solve the differential equations of motion for any number of interconnected 6 degree-of-freedom bodies. The program is modular in nature and can be easily expanded as needs arise. For the simple analyses presented here, the simulations completed substantially faster than real-time. TD_xxx is still in beta form, and several modifications were made to improve the code's stability and accuracy.

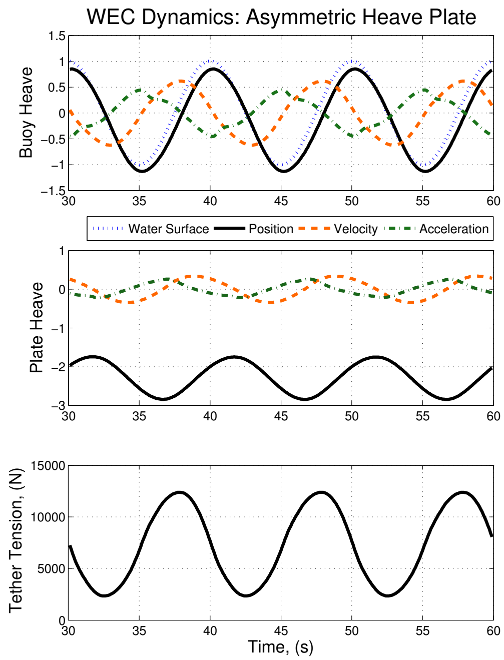

FIGURE 4. The motion of a surface float and a hydrodynamically asymmetric heave plate connected by a PTO tether were modeled in 2-m 10-s sinusoidal waves. Units: Position (m), Velocity (m/s), and Acceleration (m/s^2)

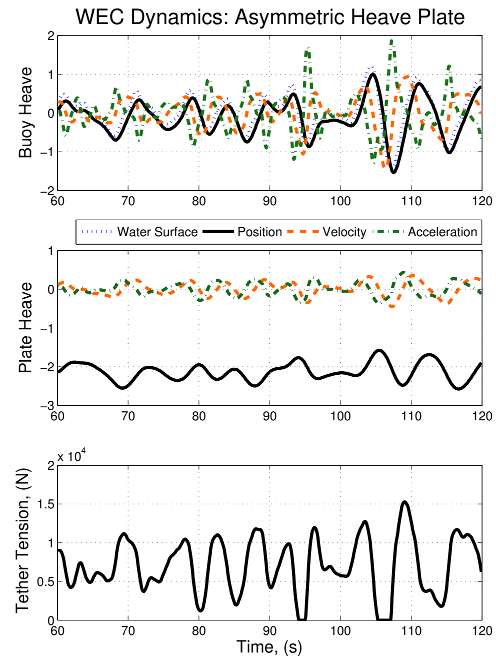

FIGURE 5. The WEC with an asymmetric heave plate was simulated in spectral waves (Pierson-Moskowitz) with a 2-m significant wave height and 10-s peak period. Two slack load events can be seen in the tether tension at $t = 93$-s and $t = 105$-s. Units: Position (m), Velocity (m/s), and Acceleration (m/s^2)

The WECs response to sinusoidal waves with a height of 2-m and a period of 10-s was simulated using symmetric hydrodynamic coefficients, and the results are shown in Figure 3. This simulation serves as a baseline against which the effects of heave plate asymmetry can be weighed. The simulation also provides a qualitative feel for the phase relations of the coupled surface float and heave plate motions. As expected, for both the heave plate and surface float, acceleration leads velocity by approximately 90°, and velocity leads position by 90°. The maximum tether tension is in phase with heave plate acceleration. This implies that the combined inertia of the heave plate and its added mass significantly affect the dynamics of the system. We also note that the heave plate moves nearly a meter vertically with each oscillation despite being heavily damped. This calls into question the assumption made by some that the dynamics of a heave plate can be modeled as a stationary body. It is also the case that due to the phase relation of the float and heave plate motion, the per cycle elongation of the PTO tether is 3.86-m which is greater than the height of the wave. This is an important point that should be considered when determining the necessary stroke of the PTO.

We now consider the effect of hydrodynamic asymmetry on the dynamics of a WEC, as shown in Figure 4. The effect of heave plate asymmetry is small; however, an increase in the skewness of the heave plate's acceleration allows the heave plate to fall more rapidly and deeper on its down stroke than was observed for the symmetric plate. The most visible effect of asymmetry is on the tether tension, the variation of which is muted by the asymmetry. One potential benefit of this behavior is a reduction in the risk of slack loads in the PTO tether. The reduction in peak tension appears to be primarily due to a loss of heave plate inertia on the down stroke as the buoy begins to heave upwards.

Slack loads have been mentioned as detrimental to system life. This is in part due to the shock load that may occur as the line comes back into tension, but also due to the internal friction between fibers that occurs when a line is re-tensioned. Two periods of slack load can be seen in Figure 5 at $t = 93$-s and $t = 105$-s. These slack loads occurred even though the height of the wave did not exceed the stroke of the WEC's PTO. The phase relation of the float and heave plate increase the necessary PTO stroke. In this simulation shock loads during re-tensioning are minimal due to the elasticity of the PTO tether. However, tethers without sufficient compliance will experience significant shock loads.

5 Conclusions

Heave plates are often used in the PTOs of WECs, yet little research has focused on their dynamics. In Section 3, we presented a brief review of heave plate research, much of which comes from the oil and natural gas communities. The review highlights the importance of certain design variables on the heave plates hydrodynamic coefficients. An analysis of WEC dynamics using hydrodynamically symmetric and asymmetric heave plates was then presented in Section 4. Added mass tends to dominate the dynamics in both cases, and asymmetry may provide a means of reducing the likelihood of slack loads in the PTO tether.

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