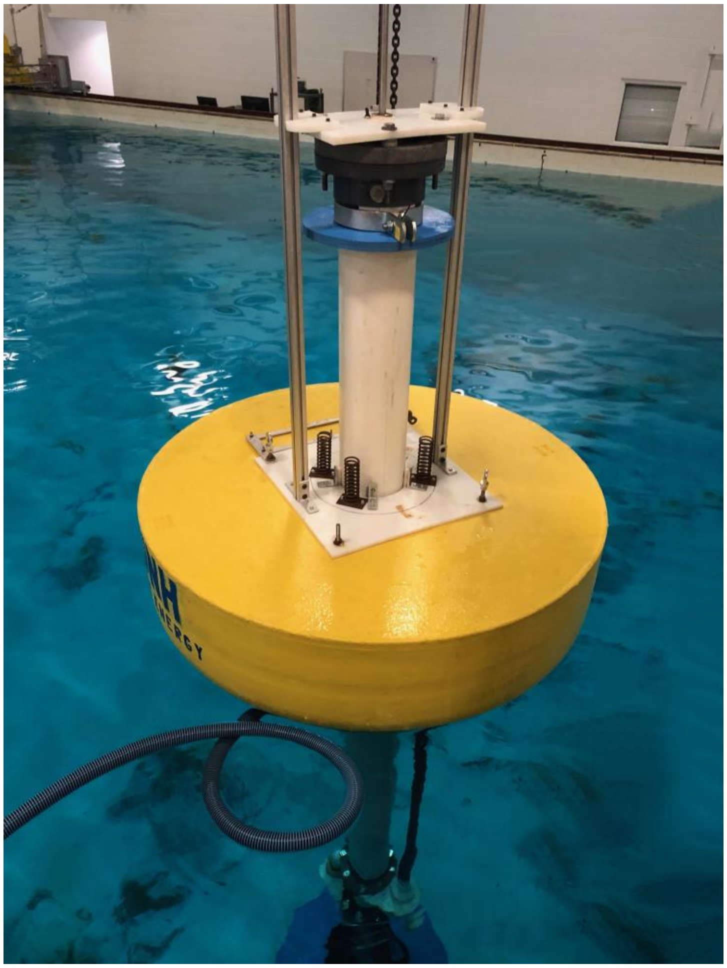
Wave Powered Water Pump



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ABSTRACT

Kelp is being explored as a potential source of renewable energy in the form of biofuel and is popularly grown for human consumption. When grown in the open ocean, the fertilization of giant kelp plants requires the sunlight that only reaches the ocean's surface as well as the nutrients found at depth. This paper details the design, partial fabrication, and preliminary testing of a wave powered water pump-buoy system created to draw up and disperse the nutrient rich, deep water at the surface where the kelp grows. In order to complete this task, the team first refurbished the student Wave Energy Conversion Buoy by removing all of the old equipment inside of the spar. This returned the buoy to its most basic state, leaving just spar and float mechanism. Utilizing a strict budget, the team was then able to purchase the required materials to build the internal pump system. The assembly of the buoy began in March 2020, but due to the implications of COVID-19, the team was unable to finish construction. Despite this, the partially fabricated buoy was tested in the UNH Engineering Tank, which proved that the system did operate. Unfortunately, the planned ocean testing, data acquisition, and subsequent analysis were cancelled as the university closed down. However, valuable information regarding the feasibility of the system operation was obtained and creates confidence that the concept can complete the design task with the investment of future time and resources.

ACKNOWLEDGMENTS

The WPWP team would like to express our sincerest thanks to all the people who have helped in the progress of the Wave Powered Water Pump project this year. A special thanks to our project advisor, Dr. Swift, and graduate advisor, Aaron Russell. The team would also like to thank technical expert John Ahern, the captain of the UNH Gulf Challenger, Bryan Soares, and Ross Hansen of the Shoals Marine Laboratory. This project is sponsored in part by the New Hampshire Sea Grant College Program**,** the UNH Parents Association, and the College of Engineering and Physical Sciences Dean's Office.

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1.0 INTRODUCTION

1.1 Background

Kelp is under investigation and trial as a potential source of renewable energy in the form of biofuel. It also can be grown and sold in large amounts for human consumption along with other products such as toothpastes, shampoos, dairy products, and some pharmaceuticals [1]. To grow kelp in massive amounts sustainably and economically, engineers and scientists utilize open-ocean farming. The kelp plants in these nurseries require both the direct sunlight that reaches the oceans' surface and the nutrients found deep on the seafloor. Existing mechanisms such as kelp elevators move kelp up and down the water column in an effective way but use large amounts of electricity [2]. This energy consumption dilutes the positive effects of kelp as biofuel and costs a significant amount of money for maintenance.

1.2 Objectives

The overall goal of this project was to convert an existing student-built wave energy converter (WEC) to a wave driven ocean circulating water pump. Original specific objectives were to

- Refurbish the damaged student WEC
- Remove the previous electricity generating system
- Design a pump replacement
- Tank test the system
- Ocean test the system

1.3 Approach

This design project aims to use naturally available wave energy to directly power a water pump system in a large, floating buoy dubbed the Wave Powered Water Pump (WPWP). This buoy constantly oscillates with passing waves and was designed to pull the nutrient-rich water from roughly 30 meters deep and release it at the surface where the kelp grows. Theoretically, an array of these buoys could surround an offshore kelp farm and provide a constant supply of cool, nutrient-rich water to the growing kelp without the use of electricity. These point absorber water pumps could also be implemented in offshore fish farms, where the combination of sunlight and the movement of penned fish warms surface temperatures and thus requires a constant cooling system to keep the fish healthy.

Beginning in September of 2019, the WPWP team was tasked with designing and manufacturing this large water pump buoy. Several years ago, from 2016-2017, another senior design group made a large-scale buoy that utilized ocean waves to harness and store electricity. This pump required more mechanical work and needed several systems to function efficiently, leading to more potential points of failure. When the team tested their Wave Energy Conversion

Buoy (WECB) in the ocean, a storm came through and knocked the buoy off its mooring. It was thrashed around in the storm and eventually clashed with the rocky shoreline, effectively damaging the WECB. The floating buoy and its spar, however, were salvaged and the buoy was returned to the Chase Ocean Engineering building, where it sat on the high bay cement floor for 3 years. The WPWP team aimed to convert this energy conversion buoy to a functioning water pump that would be more functional, efficient, and robust in operation. Over the course of their senior year, the team worked on SolidWorks models and Simulink simulations to model, design, and analyze a solution. The design work was only half the battle, as the full-scale model needed to be physically machined and manufactured before its final test and data analysis. In April 2020, the COVID-19 pandemic struck the United States, and universities across the country were forced into remote operation. Unfortunately, this meant the WPWP team never had the chance to test their final product in the ocean and gather the data they worked to prepare for.

2.0 STUDENT WEC

2.1 Wave Energy Converter (WEC)

The UNH WECB was a point absorber wave energy device [3]. It utilized the relative motion between its stationary spar and follower buoy to drive a linear generator system. The buoy rides atop the waves, moving up and down independently from the spar. It was attached to a connecting rod which pulled the coils of the generator along with the motion of the buoy. The primary spar was kept stationary through the implementation of a heave plate system, which was attached to the bottom of the spar using an extension pipe. Since the heave plate extended below the wave motion, the drag force on the heave plate counteracted the waves' buoyancy effect on the spar and kept it relatively stationary. This heave plate effect also allows for self-preservation of the system in extreme storm waves with wave heights that would likely compromise the system. The buoy will all move in unison during these large storm waves since the heave plate will not extend below the wave motion, preventing the buoy from attempting to operate in destructive conditions.

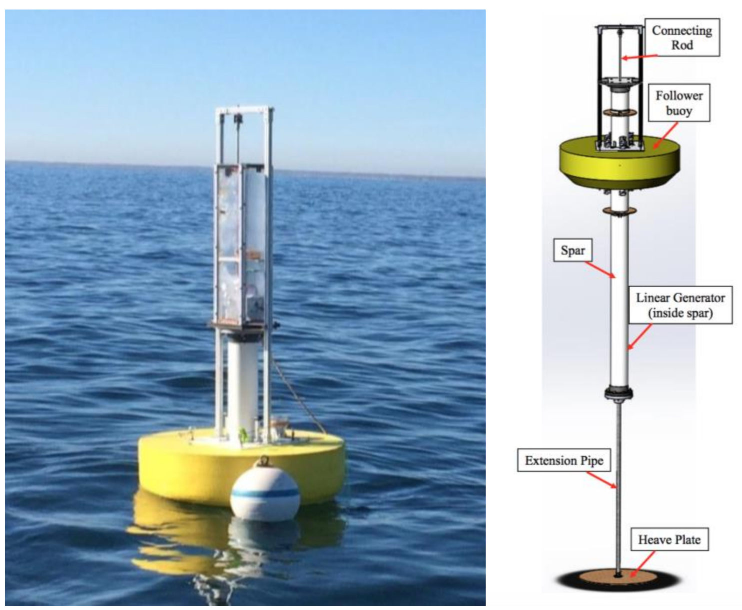

Figure 1: Student WECB in ocean tests (left). Labeled SOLIDWORKS drawing (right).

2.2 Reconditioning the WECB

The previous team to work on the buoy, Aaron Russell and Chelsea Kimball, designed, fabricated, and installed a linear generator inside the PVC spar. To recondition the buoy for use as a water pump, the entire spar was subsequently gutted to provide a location for the internal pump components to reside. To remove the linear generator, the metal riggings which the buoy slides along were adjusted. The generator was then carefully removed from the top of the spar. The previous team had also installed a stopper, which was cemented towards the bottom of the spar to prevent water leakage into the system. To build the pump within this space, the bottom cap was removed. At this point, the buoy was in a condition for the installation of the new pump mechanism to be driven by the follower float.

3.0 WATER PUMP DESIGN

3.1 Design Concepts

In the early stages of design, the team first investigated a piston style pump design. This featured an internal piston pump which utilized the entire volume of the spar. The intake valve was to be located at the bottom of the spar, while the output was located above the buoy. An additional check valve was located on the piston head to move the water from the bottom of the spar, to the top of the buoy.

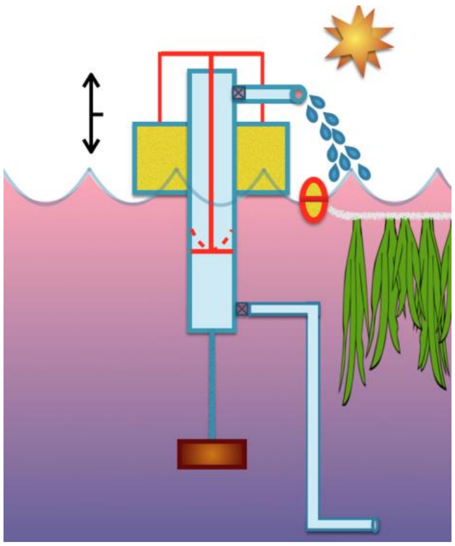

Figure 2: Initial Pump Design Concept

The major drawback of this design was that the check valves lie within the PVC spar of the buoy. If the valves were to break, or if any maintenance were required on them, they would be relatively inaccessible at sea. In addition, the additional mass of water required to fill the spar meant that the current buoy would sink during operation since the buoyancy force of the primary spar would not be sufficient to counteract the additional weight. Thus, this design was no longer taken into consideration and, the team then took a high-level conceptual approach to pump design considerations. After initial research, three main categories of pump design were identified: a bellows style pump, piston-cylinder design, and a modified bilge pump design. To utilize a bilge pump for the purpose of this project, the team would have had to retrofit a prefabricated pump to be driven by the float. As the team wanted to fabricate their own design, this style of pump was eliminated for further evaluation. The team analyzed each of the remaining pump styles, a bellows style pump and two piston-cylinder designs, in depth for manufacturability, cost, potential flow rate, and potential maintenance requirements.

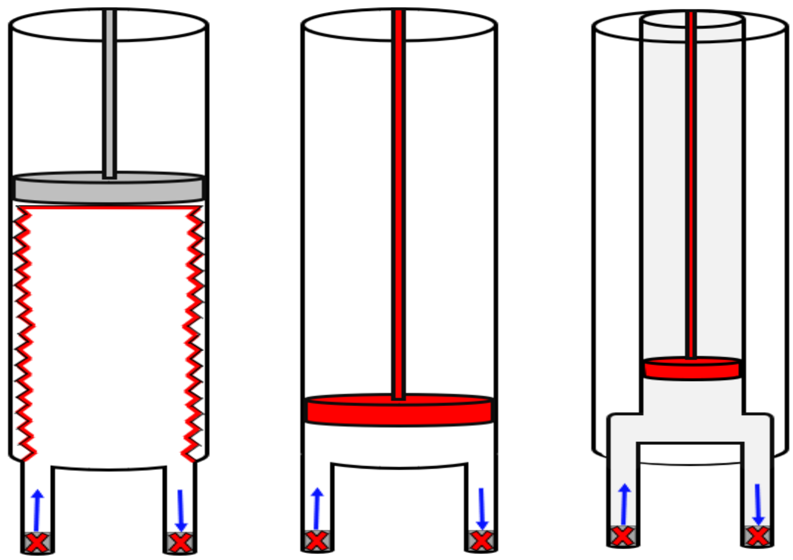

Figure 3: Considered Pump Design Options

The image above shows the three considered designs: a bellows style pump (left), a classic piston-cylinder pump (center), and an internal piston-cylinder pump (right). The bellows pump is a positive displacement pump that utilizes the expansion and contraction of its accordion style walls to draw in and release water. This wall design also offers low leakage, as the water is fully contained within the chamber. The major drawback of the bellows pump is its difficulty to manufacture. In order to accommodate the stroke length and spar diameter, a custom rubber bellows would be required, which would add significant cost to the design. There was also concern about the lifetime of an internal rubber diaphragm as a rubber bellows under constant operation may develop weak points that could lead to premature failure. Since the bellows would be an expensive component of the pump, early replacement of the component would be extremely detrimental to the feasibility of the design.

The classic piston pump design would utilize the entire diameter of the spar, allowing for the maximum displaceable amount of water. With this design, the system's flow rate was dictated by the size of the spar; therefore, if a large spar was required to provide the buoyancy force necessary to keep the system afloat, the flow rate of the system would already be defined. This could be a significant issue for deployment locations where wave patterns would be inadequate to provide the necessary force to operate the pump.

The final design considered was an internal piston-cylinder pump. This design would allow for better maintenance and leakage control than the classic piston pump, as the valves are located on the outside of the PVC spar. The location of the piston shaft within the spar also provides the pump system with a layer of protection during adverse weather. Since the pump system would be installed within the spar, the piston cylinder diameter and desired flow rates could be optimized for a specified operational point in various deployment locations.

3.2 DESIGN SELECTION

The motion of the buoy was modeled based on the amplitude and frequency of the waves. For both the upstroke and downstroke motions, the minimum and maximum spar forces were estimated to determine the capability of the buoy. Three internal pump designs were considered, and the potential volume flow rate of each pump type was calculated. All three designs could sustain satisfactory flow rates with a minimum average flow rate of 40 GPM with a 1 meter stroke and 3 second wave period (Appendix 01). This was estimated assuming the entire pump volume was moved forward during one cycle. The input wave dimensions, stroke length and period, are based off the fair weather conditions at the offshore site, when the WPWP would operate most efficiently. Considering each alternative, the following decision matrix was utilized.

Table 1: Decision Matrix of Different Pump Designs

	Weight	Piston Pump	Bellows Pump	Internal Piston Pump
Manufacturability	.7	2	3	5
Average Flow Rate	.8	4	4	4
Lifespan	.6	2	3	4
Cost	.8	5	2	3
Maintenance	.5	1	4	4
Totals	X	10.3	10.6	13.5

The internal piston pump design was selected to move forward in the final design. After extensive research involving contact with bellows pump manufacturers, it was decided that the budget could not support a custom bellows pump. The piston pump design, utilizing the full internal volume of the spar could lead to many issues because of the uncertainty of the inner diameter of the eight-inch spar. Designing an internal pump would be easier to eliminate these uncertainties and control the pump system so that in practice there would be less risk of failure.

3.3 HYDROSTATICS

For the pump system to function in the ocean environment it needs to stay buoyant and upright on its own in a varying wave field and have a draft at the desired spar section which will promote efficiency. These hydrostatics values were calculated by understanding how much the entire system will weigh and how much volume of the system will be submerged in the environment. This was done by initially weighing the 8-inch diameter spar and the float with the rolling crane system in the highbay. With these initial weights calculated we now have to see how much volume we have to work with because this volume determines how much mass we need to add to the system to reach our desired draft location. Since the system has very little external structure other than the spar the volume used in the calculations was easily found. To acquire the mass of the entire structure the team needed to measure and record the mass and volume, if necessary, of each new piece for every idea pursued. With the cylindrical shape of the system and the fact that you could cut it in half vertically at any angle and it would be symmetrical. This allowed the team to disregard all horizontal displacements and treat the system as a one-dimensional hydrostatics problem. This allowed for the assumption that all weights will act through the center of the spar. The one outlier for this assumption is the elbow brackets that attach to the T mount in the center. There are two of these pieces on opposite sides of the spar and with equal weights the assumption was made that these would balance each other in the horizontal and we only need to account for the vertical contributions. The float was already put together and the team was able to calculate it's weight in its entirety and since it is uniform at any angle it can also be assumed that the weights will only act in the vertical. Another problem that the team tackled in this analysis was the change in mass due to the intake and outtake of seawater. To account for this the team assumed that there should be a constant half stroke length full of 1025 kg/m^3 seawater in the center of the spar which is to be added to the hydrostatics.

Table 2: Basic Hydrostatic Elements

	Spar (empty)	Float	Pump System (internal)
Total Weight (kg)	43.59	82.37	80.1664
Submerged Volume (m^3)	0.120728	0.084	0

This piecing together of buoyancy and weight serves to maximize the efficiency of the pump by placing the buoy in the center of the stroke length and gives it equal allowable motion in the positive and negative y-directions. To do this the waterline of the buoy needed to be determined since the buoy and the spar react independently from each other. The water line for the buoy turned out to be 6.614 inches from the bottom of the float which then required the waterline for the spar to be 10.822 feet from the bottom of the spar, not including the heave plate system. The draft line was achieved by adding 19.36 kilograms to the bottom of the spar. The heave plate is a large circular flat wooden piece that is attached below the spar and acts like a parachute holding the base of the spar stable by interacting with the lower sections of the water column that are unaffected by surface orbital motion. This helps to keep that spar stationary at its desired waterline while waves pass and move the float and not together in the waves.

When the initial tests were done in the wave tank, the buoy sat very closely within the desired waterline with only an offset of 2-5 inches. This meant that the hydrostatics were very close and only minor details needed to be added to correct the slight error. In addition, there exists untapped efficiency that allows the pump to move more water with a given wave environment. Since the hydrostatics were conducted quickly and changed often to account for the fast build schedule the group worked through in the last active weeks of school, there were a few things that were added to the system that were not implemented into the hydrostatics at the point of initial testing, which are now inaccessible. In the final days of testing there was an emphasis on speed of fabrication and a belief that the team would have several weeks for final calculations upon return. Therefore, measurements on additional components such as the second top hubcap were not a top priority. The weight of these components was estimated to the best of our ability; however, true weights and volumes of these elements should be implemented into the hydrostatics for an exact calculation. However, estimates were made based of of known material densities and properties. Another aspect that the team was unable to complete was calculation of the metacentric height, which would prove the initial stability of the structure and determine whether the buoy would overturn in certain wave environments. When building the system, it was apparent that much of the weight came from the bottom T bracket system and the probability of the buoy flipping appeared negligible, so the calculation was not conducted while on site. However, if the team were able to return, it would have been calculated and proved what critical wave environment the system could handle without capsizing.

3.4 FLOW ESTIMATIONS

From here, a proof of concept model was designed. This design, named the “alpha model,” simplifies the full model by only incorporating the internal piston, tee, and check valves with corresponding tubing. A schematic of this is shown below in Figure 4.

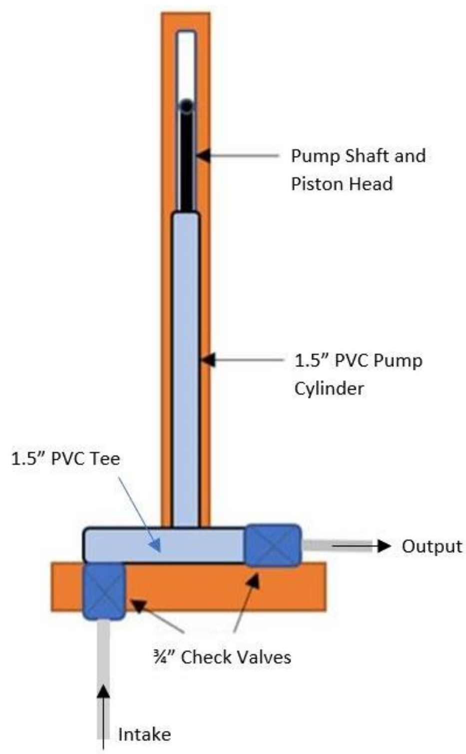

Figure 4: Alpha Model Schematic

Using available materials found in the scrap boxes at the Chase Engineering Lab, and remaining necessary materials purchased from Home Depot, the team began assembly of the prototype. The assembly of the Alpha Model was finished in late November and testing was completed by mid-December. In addition to a physical model, a theoretical model was created so that it could be calibrated against this model with intentions to scale the theoretical model and predict flow values of any wave powered water pump using similar parameters.

A particularly difficult component of modeling the wave powered water pump is the fluid response to the mechanical input and the resulting flow rate from varying input waveforms. System nonlinearities and the incorporation of check valves produce significant challenges for conventional numerical modeling methods and require complex analysis for sufficiently accurate results. However, with some simplifying assumptions including the assumption of laminar flow and ignorance of minor losses due to pipe geometry, a basic fluid model can be approximated. SIMULINK tools allow for quick compilation of flow rate data through computer simulation.

To define the SIMULINK model of the system, all fluid components are approximated as electrical element equivalents.

$$\Delta P = qR_f \tag{1}$$

$$R_f = \frac{128\mu l}{\pi D^4} \tag{2}$$

Intake and outtake pipe flow is assumed to have pipe resistance, calculated using Equations (1) and (2) above, where μ is fluid viscosity, l is tubing length, and D is the tubing inner diameter.

$$\Delta P = I_f \frac{dq}{dt} \tag{3}$$

$$I_f = \frac{\rho L}{A} \tag{4}$$

The flow is also assumed to have a fluid inertance, defined using the relationships shown in Equations (3) and (4), where ρ is fluid density, L is tubing length, and A is tubing cross section area.

$$\Delta P = \frac{1}{C} \int q dt \tag{5}$$

$$C = \frac{Al}{B} \tag{6}$$

The intake and outtake check valves are modeled as ideal diodes and the pump cylinder is modeled as a capacitor with a pressure drop described in Equation (6). Where A is the cylinder cross section area, l is cylinder stroke length, and B is the bulk modulus. With this electrical representation of the system, a waveform input can easily be introduced as an AC voltage source. The full SIMULINK model is pictured in the figure below.

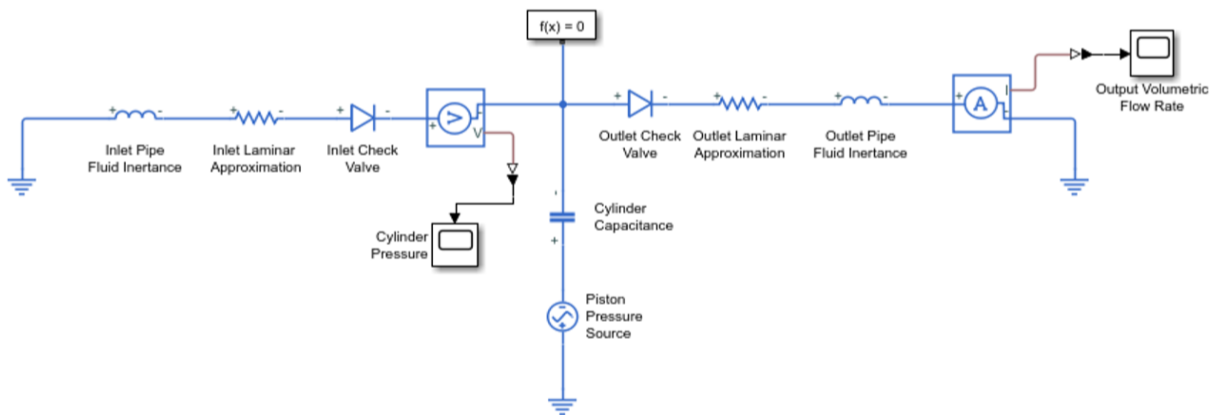

Figure 5: Simulink Model of System

3.5 PRELIMINARY TESTING

To accurately test the effectiveness of the Alpha Model pump, an oscillating force input was required. In doing so, the piston shaft is then hand driven at various approximated constant frequencies using two, three, and six second periods. This force input would simulate the force generated by the buoy-piston on the piston cylinder spar. A four-bar linkage was designed in order to have more control while hand-driving the pump. For more accuracy, a motor could be implemented to drive the system, but none were available that could meet the torque requirements and the budget intended for this model could not cover purchasing a new one. Figure 3 shows a diagram of the experimental set up:

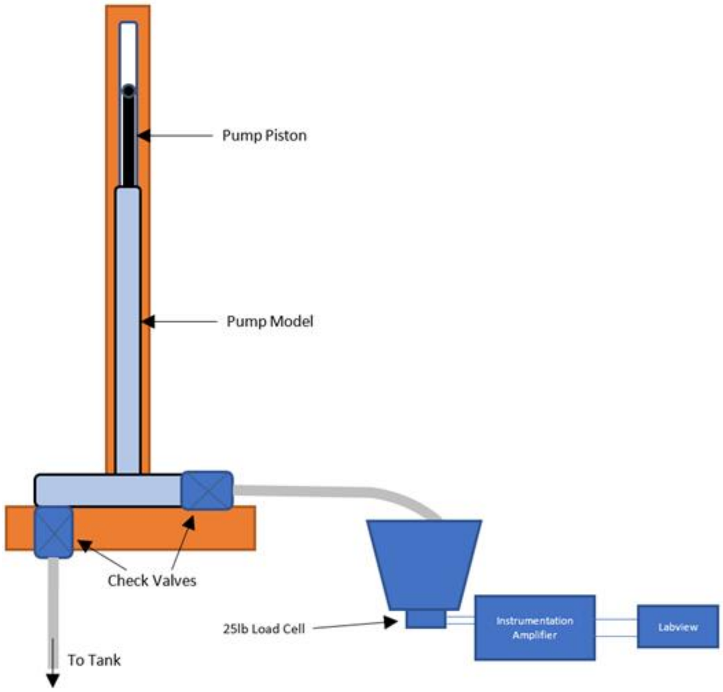

Figure 6: Alpha Model Experimental Design

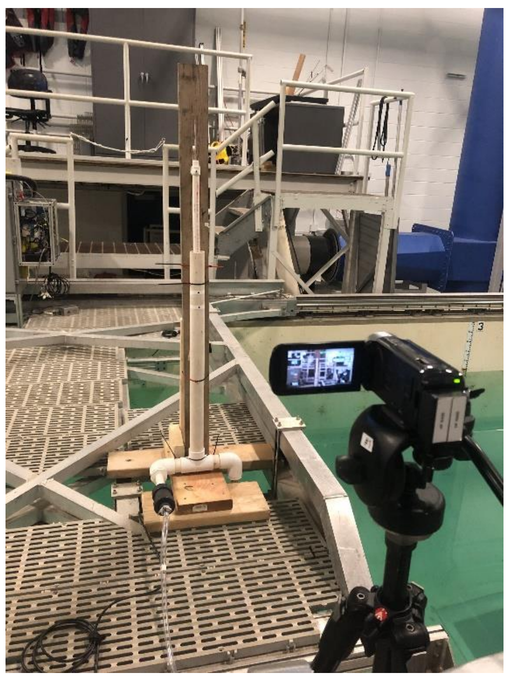

Figure 4: Alpha Model Kinovea Setup

To track force input for a simulated wave, a camera was used to record the position of the piston, figure 4. This data was eventually processed using Kinovea, a position analysis software, which produced x and y components of the "wave." To collect the water output of the pump, a 25-pound strain gage load cell was placed directly under a 5-gallon bucket with a sheet of acrylic used to evenly distribute the weight. Figure 5 shows the experimental setup in practice in the Chase Ocean

Engineering Wave Tank. This sensor gathered the force data, which was used to calculate the system's experimental output mass flow rate. To begin data collection, the load cell is calibrated using weights ranging from 2 pounds-force to 11 pounds-force. The voltage output of the load cell is amplified using an SGA/A Strain Gage Transducer Amplifier and read using the oscilloscope.

Figure 7: Alpha Model in Wave Tank

All voltage readings from the force sensor were converted to mass measurements using the sensitivity of the load cell, which is found using the slope of the voltage versus mass calibration curve. The tracking software Kinovea was used to collect position vectors with respect to time from the video data and was derived twice to find acceleration of the piston shaft. Newton's Second Law can be used to convert the acceleration data into force data; however, to account for friction losses in the spar driving piston, a constant force was applied and the acceleration calculated to determine an effective mass for the spar.

$$F = m_{eff} * a \tag{7}$$

$$P = F * A_{piston} \tag{8}$$

Using the relations in Equations (7) and (8) the force and pressure on the water were calculated. The pressure vector is input into the SIMULINK model to drive the simulated pump.

$$\frac{dm}{dt} = \frac{1}{\rho} q \tag{9}$$

$$m = \frac{1}{\rho} \int q dt \tag{10}$$

SIMULINK outputs the pump volumetric flow rate and can be converted into a mass flow rate and time dependent mass with equations (9) and (10). The efficiency and mass flow rate of the alpha model were calculated based upon a 1.7671 in^2 circular cross-sectional area and an 18 inch stroke length with a three second period, position input and output force. These dimensions correspond to a maximum theoretical volumetric flow rate of 2.75 gallons per minute with the 3 second period. The experimental result for a wave input with this these settings was 2.4 gallons per minute corresponding to a pump efficiency of 86% based on the theoretical model outputs compared to the experimental model outputs. The theoretical full-scale system mass flow rate is calculated as 68 gallons per minute, incorporating the efficiency deficit. Further testing would be beneficial to determine and improve the model accuracy between different stroke length, periods, and effective areas.

4.0 FULL SCALE MODEL

A Simulink Model estimated the efficiency of the Alpha Model and gave subsequent pump rates. This provided the results used to design the full-scale specifications that would be used in the final WPWP model for deployment.

4.1 PISTON HEAD AND SHAFT

The internal reciprocating pump system was selected as the final choice for the pump design. This design utilizes a four-inch inner diameter PVC pipe for the piston cylinder and a PVC tee head that reduces to three inches to connect to three-inch check valves. Calculations were performed to determine the maximum and average volumetric flow rate the system would produce 82.8 gpm and 41.4 gpm, respectively. Three-inch inner diameter, single inline spring, brass check valves were chosen and purchased for both the inlet and outlet of the system. To internally connect these pieces, a 3”x3”x4” tee was purchased. A variety of 3 in fittings, including male to female adapters, 90-degree elbows, and couplers were purchased. In addition, a 1.5 in diameter hose was purchased, along with all the necessary fittings such as hose clamps, barbed transitions, and slip to female reducer connections to transition from the 3 in piping to the 1.5 in hosing.

Initially, the full-scale piston was machined out of a 3-inch inner diameter PVC cap with two O-ring grooves distanced one inch apart and offset one inch from the base of the cap shown in figure 6. The O-rings were circular and 4 inches in diameter with a ⅛ inch thickness. These materials were found in stock in the Chase Ocean Engineering high bay.

Figure 8: Initial Piston Head

The original idea for the piston-shaft involved cementing the stainless-steel shaft that transfers the driving force from the buoy into the cup of the 3-inch PVC cap, also possibly including a metal plate to distribute the force. After returning from break, this design was reexamined, and the team decided a stronger piston head should be placed instead of the cemented PVC because the force driving into this head may cause it to fail. Ideas of aluminum and steel were the most fitting material to use moving forward due to their strength and low corrosive properties. Due to manufacturing feasibility and weight, the aluminum material was chosen for the final design. A SolidWorks part was created for the piston head which was designed to have a diameter of 3.97 inches and a height of 3.00 inches. A small filet was cut from the top and bottom of the piston head to reduce the risk of sharp corners interacting with the inside of the pump cylinder. X-profile oil-resistant O-rings were chosen for the final design rather than the standard circular profile O-rings. Using an X-profile or quattro seal helps increase the seal between the piston and the bore because there exist four points of contact. These O-rings are also more beneficial in dynamic applications and typically serve a longer life. The Buna-N material is also tear resistant, which was very important considering the purpose of these O-rings. Choosing the correct size O-ring was difficult because the dimensions required sat directly between two O-ring sizes. The O-ring grooves of the aluminum piston shaft were sized to fit the smaller of the two, but both sizes were purchased in case the seal with the smaller O-ring was too loose. Two O-ring grooves seemed sufficient, but a third groove was added for redundancy and to assist with stability if necessary. In the final design, each O-ring groove had a depth of 0.17 inches and a width of 0.22 inches. The O-rings, which had a thickness of 0.21 inches, would sit in the 0.17-inch groove providing an additional 0.04 inch of rubber outside of the original diameter of the piston head. Refer to Appendix 02 for the SolidWorks drawing that includes these dimensions. This additional rubber would form the seal between the piston head and the pump cylinder. The thread chosen was 3/16-10 and a series of three scrap metal steel pipes were machined to fit the same thread, with a ⅝ inch diameter. The piston head was threaded in the center to a depth of 1.5 inches, which was chosen to meet the requirements that the depth be twice the major diameter of the rod [4]. Drawings were created in SolidWorks of the final design for the piston head prior to machining. Figure 7 shows the three-dimensional SolidWorks model of the full-size piston head design.

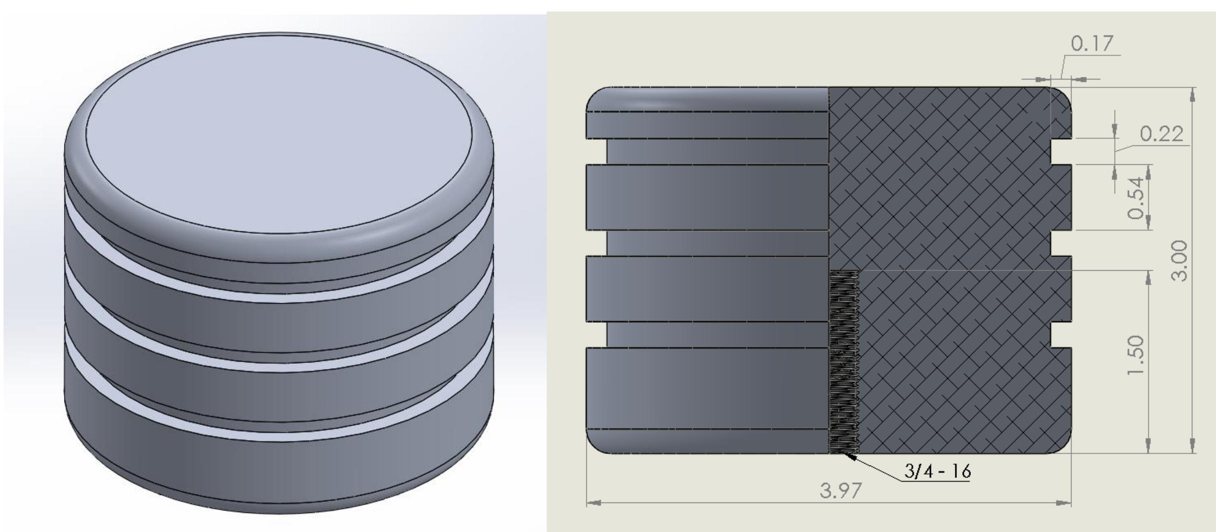

Figure 9: SolidWorks Model (left) and Drawing (right) of Aluminum Piston Head

To ensure that the piston-shaft design would not fail under maximum compression or tension some mechanic calculations were performed. These include buckling, deflection, axial tension, pin shear and bearing gathered from the *Machine Design: An Integral Approach 4th ed.* [4]. There were no failure modes which suggested inoperability with these dimensions.

4.2 TEE HOUSING

Two primary methods were considered for the pump tee housing design. The first was a direct fabrication method with a hole to be machined through the primary spar to provide inlet and outlet flow connections to the check valves. This was the most efficient method and used the least amount of materials; however, there were a few concerns regarding manufacturability and lifetime maintenance. The second design, which was selected for fabrication, was a modular design that was attached to the main spar via a flange connection and connected to the heave plate through an identical fitting. A modular design was preferred because a direct machining method would require a large milling machine with ample clearance to accommodate the 15-foot spar. Additionally, if the tee assembly were to be damaged during operation, an entirely new spar would have to be manufactured, which would be time consuming and costly. Alternatively, the modular design only requires a short segment of pipe to be machined (16-18 inches) and if the tee housing were damaged, a new segment could be manufactured quickly for a lower cost. In addition to the central tee housing pipe, support bracing to maintain check value orientation and stability was designed and implemented as seen in the final tee housing assembly below, Figure 8.

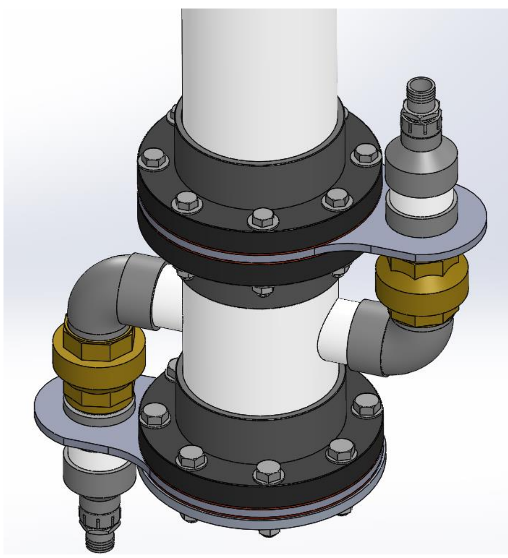

Figure 10: Tee Housing Assembly

4.3 FINAL DESIGN

With each component of the buoy analyzed and designed, a final model of the wave powered water pump in SOLIDWORKS was compiled, Figure 9, and the full-scale buoy fabrication was completed.

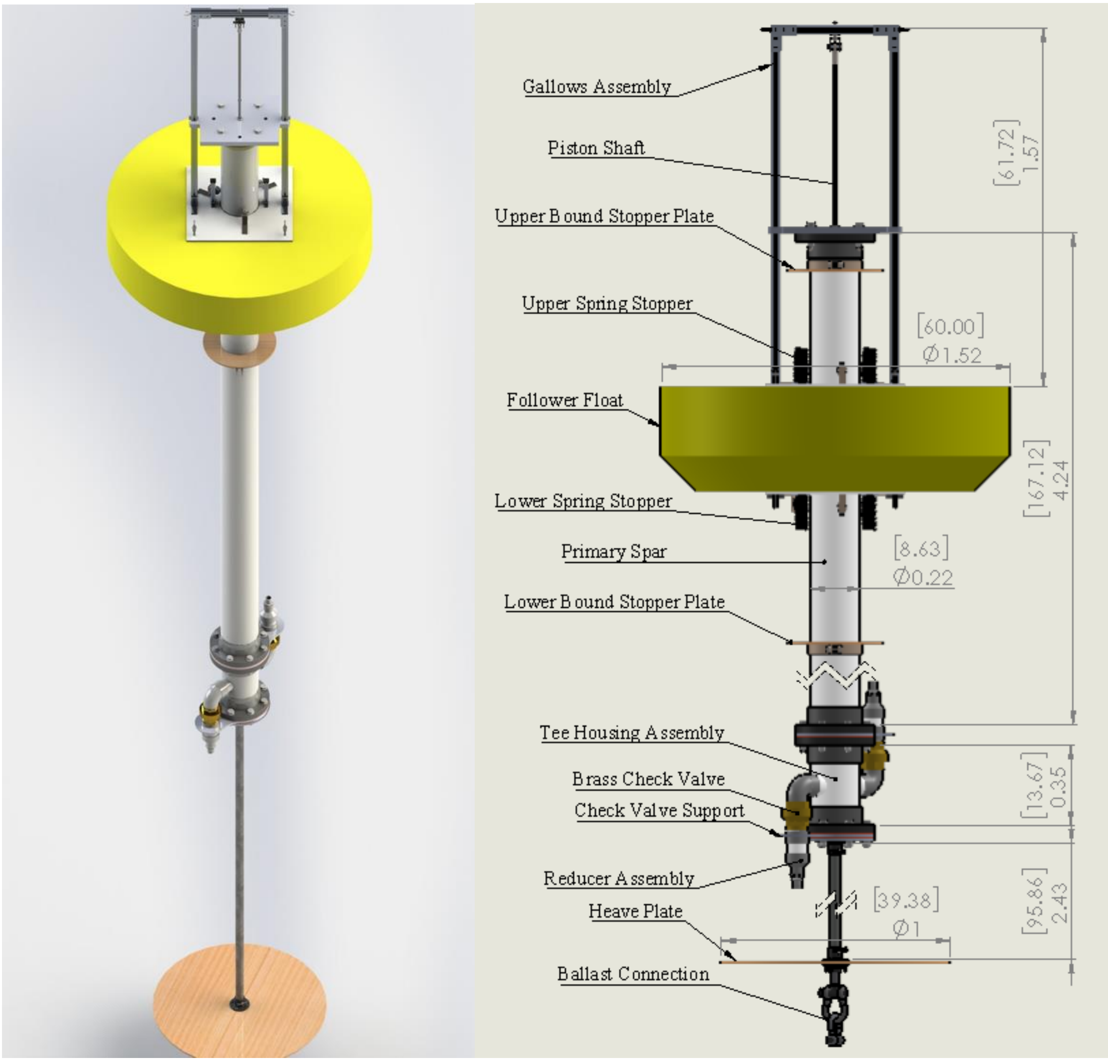

Figure 11: Final Solidworks model rendering (left) and dimensioned model drawing (right). Dimensions provided in meters [inches].

The pre-existing float, gallows, heave plate, and primary spar were reused from the wave energy conversion buoy. For the center piston cylinder, two segments of 4-inch diameter schedule 40 PVC piping were connected with a PVC union. The piston shaft segments, and the central pipe for the tee housing were cut by Aaron Russell and the shaft segments were also turned to the final thread dimensions. The piston head was manufactured by Scott Campbell in the College of Engineering and Physical Sciences student machine shop from 6061 Aluminum stock to exactly fit the 4-inch PVC pipe dimensions from the piston cylinder. All the PVC fittings, check valves, reducers, supports and flanges for the tee housing were assembled with heavy duty PVC cement and Teflon tape as needed before being bolted to the primary spar and connected to the piston cylinder. The heave plate connection was finally bolted to the bottom flange allowing the full heave plate and ballast to be connected directly prior to deployment in the open ocean testing. In a full production scale buoy, there would also be an addition of floatation foam to the central spar between the piston cylinder and the 8-inch spar. This would be added before the piston cylinder support bracing so that the opening could be accessed from the top of the buoy where the foam would be poured (Figure 12). Before deployment, the 1.5-inch diameter inlet and outlet hose would be connected to the reducer/check valve assembly by hose clamps with an additional floatation buoy attached to the outlet hose and a weight attached to the inlet hose.

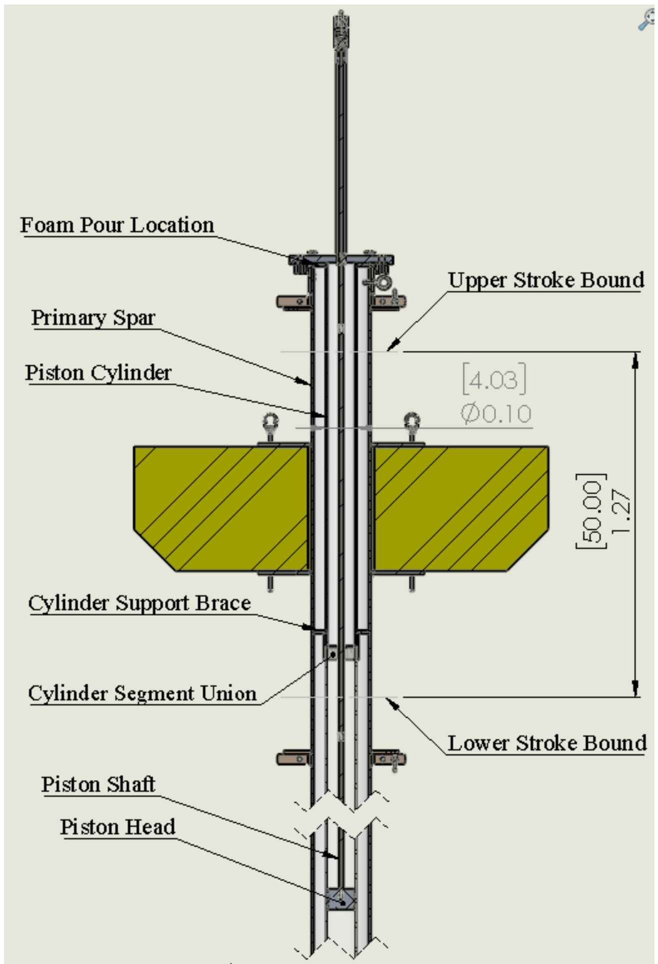

Figure 12: Labeled internal cylinder diagram with maximum stroke area and cylinder diameter in meters [inches].

The final buoy design would measure between 24.5 feet and 28.5 feet in total length with the heave plate attached, depending on the position of the float within the stroke length and buoy diameter would measure 5 feet based upon the widest buoy component, the float. A usable stroke length of approximately 50 inches (1.27 meters) as seen in Figure 12 in conjunction with the 4-inch diameter piston cylinder would combine for a maximum volume transfer of 2.72 gallons per stroke dependent upon empirical efficiency and actual wave motion. Given the preliminary testing results that resulted in an approximately 74% efficiency, if the buoy was deployed in the current state and maximum stroke length realized, then a 2 gallons per stroke flow rate would be achieved.

This corresponds to a volumetric flow rate for different wave periods as described in the table below.

Table 3: Flow Rate for Different Wave Heights and Periods. Wave heights were selected to be uniform for comparison purposes only.

Realized Wave Height [in]	Wave Period [s]	Average Flow Rate [Gal/s]
50	1	2
50	2	1
50	4	0.5
50	10	0.2

5.0 ENGINEERING TANK TEST

Once the system was all together it was time to test. This test was conducted in the Chase Ocean Engineering Building's engineering tank on Friday, March 13th from 12 to 4pm. The system had to be wheeled over to the tank via the rolling carts that support the structure. Once there, it was attached by rope at three locations to the spar and then attached to the inhouse crane. This crane picked up and transferred the system to the tank. The rope tripod was kept on the system for the duration of the tests to allow for easy retrieval upon completion.

Figure 13: Preparation for the Tank Test

Since the tank only has a depth of 6 meters, the original 25 plus foot length of the entire system was not able to be put into the tank. Our way around this was to shorten the heave plate bar, not shown in image, which provided the length reduction needed to not collide with the tank bottom but also still perform the task that it would in the large scale wave environment. Since the heave plate grabs and holds underwater stability and the water in the engineering tank has relatively no motion at its bottom, the heave plate would still do its job at this depth. To begin testing, the system needed to be primed by manually moving the float until water began to flow. This allowed for the system to start off lubricated by water and get rid of unforeseen pressure variations. To generate waves, two individuals stood on a raft at the opposing side of the tank and jumped on it in sync to propagate a wave ray towards the system. This worked well although the waves were not large enough to obtain a maximum pump stroke, but it was able to pump a consistent amount of water with each stroke. Knowing that it can pump consistently its maximum pumping capability needed to be tested. To achieve this, we had people stand on the float and manually pull the spar through the top of its stroke length then push the spar up to the bottom of the stroke length and then back to the top again simulating one full wave period starting at a crest working through a trough and back to a crest. This method produced a significant amount of water per stroke length and allowed us to produce approximately five gallons with 2.5 ideal full wave periods. This verified that the model worked and provided confidence that with more time working on the system it can become an efficient product.

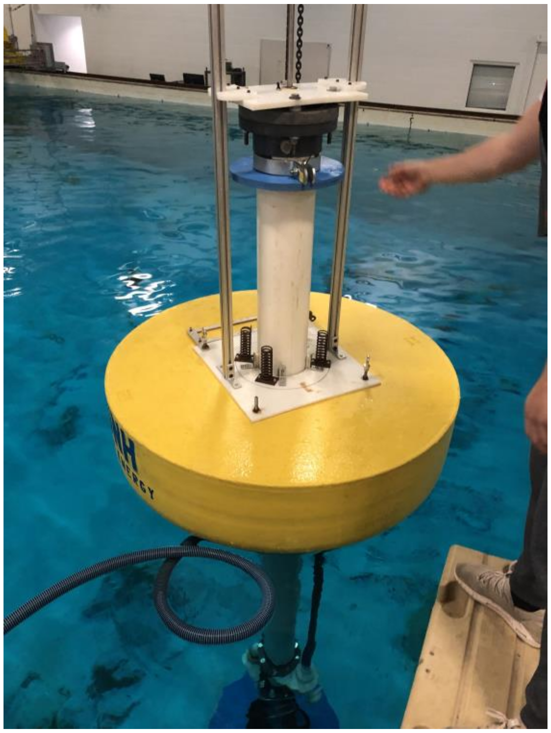

Figure 14: WPWP in the Chase Engineering Tank

6.0 REMAINING FABRICATION TASKS

Due to the COVID-19 pandemic in mid-March of 2020, the University of New Hampshire unexpectedly switched to remote learning. This project was in its final steps before testing and the team would have finished the fabrication of the buoy. The team ordered two-part closed cell marine flotation foam. This would be poured into the main spar of the buoy to secure the tee housing and maintain buoyancy within the system in the event of a leak into the main spar. The closed cell foam was crucial to the completion of the hydrostatic analysis as it added additional weight to the system without providing additional buoyancy force. The team also planned to add a heave plate directly below the spar that would be connected to the flange under the pump system. The purpose of the heave plate would be to increase the stability of the spar. This plate would be placed as low as possible to ensure it felt minimal effects from the orbital velocities of the waves. Spar stability is critical to ensure that the spar and buoy do not move together, as it is this relative motion that determines the efficiency of the pump. Minor fabrication tasks including the permanent cementing of the piston cylinder to the tee fitting and manufacturing of a spacer component to seal the top spar cap also needed to be finalized before open ocean testing of the system.

6.1 OCEAN TEST

After months of hard work designing and manufacturing the final WPWP model, the goal was to test it in the ocean. Appledore Island found amongst the Isle of Shoals off the coast of New Hampshire was chosen as the test site. The western part of the island has a small cove where the moorings are anchored. This is a location where waves can still pass through but has protection from stormy seas by the island itself. The Shoals Marine Lab is also situated on Appledore; Ross Hansen serves as the director of facilities for the laboratory and oversees all moorings. Ross was contacted and pitched the idea of mooring the WPWP in this cove for several days as it gathered data from a variety of sea states. Aware of the past mooring failure for the WECB, Ross suggested that he would come along with the team and aid in the mooring process. He planned to remove one of the weaker, temporary winter "stick" moorings and put in place a more durable and permanent boat mooring for the WPWP.

To get to the test site, the WPWP team would have required the aid of John Ahern and his trailer, which would bring the large buoy to Fort Point in New Castle, New Hampshire. Here, there lies a large dock, home of several boats along with a US Coast Guard Station. One of UNH's research vessels, the R/V Gulf Challenger, is docked there. The Gulf Challenger is a 50-foot aluminum hull boat, and with a cruise speed of 18 knots, she is the fastest and most capable research vessel in the area. She has a large deck area aft along with a below-deck laboratory that can provide a live feed of data acquisition via the deck box. Captain Bryan Soares was contacted and would have taken the WPWP team and the buoy to the testing site on his vessel. The team allocated $2800 of the budget to schedule the Gulf Challenger for two full days, one for deployment and one for retrieval. Both Ross Hanson and John Ahern were ready to help with the deployment of the buoy and ensuring safety throughout the process.

Figure 15: The R/V Gulf Challenger

As aforementioned, Ross Hansen would have been onboard to help with mooring the WPWP. The team would have replaced a winter stick with a permanent boat mooring upon arrival at the test site. Upon utilizing the winch on the stern of the ship to raise the buoy into the ocean, the mooring process would have begun. This was an extremely important aspect of the project, as improper mooring could be a recipe for disaster. The WPWP has two eyelets that would attach to the mooring pennants, as suggested by Hansen. These pennants then would have been secured to the thimble and shackles found on top of the mooring ball. The mooring ball has a large chain that reaches the seafloor and is attached to the mooring block. This mooring configuration would have been strong enough to hold during a storm and nimble enough to allow the WPWP to oscillate with the incoming waves and serve its purpose as an efficient water pump. The mooring configuration provided by Ross Hansen can be seen in the image below.

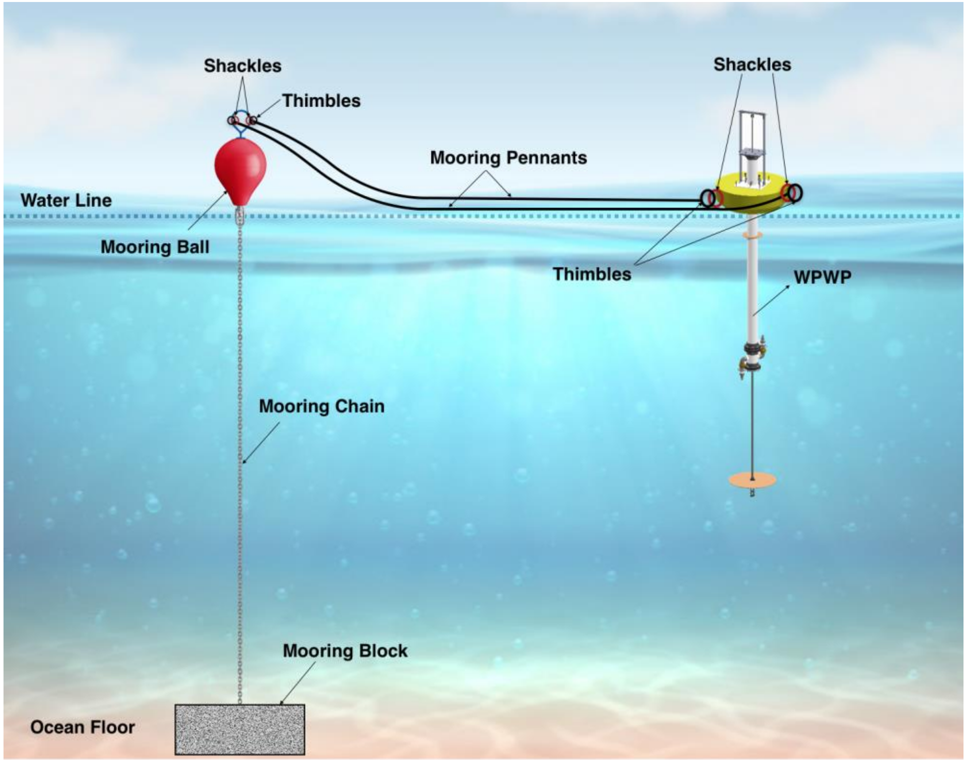

Figure 16: WPWP Mooring Configuration - Suggested by Ross Hansen

Once the buoy was in the water and securely attached to the mooring ball, it would have stayed there for 2-6 days, depending on weather, gathering a wide range of data for further analysis. The exact type of data the team would have attained was still under discussion prior to the cancellation of the project. A flow rate sensor placed at the input and output would have been necessary to determine the efficiency of the pump and how much nutrient-rich water would have been dispersed to the growing kelp at the surface. Specifically, a velocity flow meter would be the most practical instrumentation to utilize. These sensors measure the speed at a given point and integrate it over area. In addition, a CTD could have been placed at the input and outputs to see the change in water qualities with the increase of depth. A fluorometer could have been put at the seafloor, which would have provided the amount of chlorophyll in the water and could reveal the amount of nutrients in the deep water. To measure water pressure, the team would have borrowed an RBR Duet T.D. wave logger from Dr. Lippmann. It is RBR's newest data logger; a dual-channel temperature and depth, idea and wave logger. The tide recorder averages the pressure readings to provide accurate tide levels and tidal slope readings. The wave recorder measures these tides and continuously bursts to best measure high frequency waves (wave heights and periods each burst) over a several day deployment period (5). Upon recovering the WPWP, the team would have used the Ruskin data processing software from the ocean instrumentation lab. This which could be compared to the flow rate sensor data, allowing for an understanding of the flow rate seen with specific size waves and periods. Additionally, Dr. Lippmann could have also provided the team with a Spotter wave follower buoy. This device measures waves, winds, and sea surface temperature. The wave data includes wave spectrum, directional moments, bulk statistics, and motion time series. The wind sensor uses a proxy measurement of wind speed and direction estimated from the equilibrium range of wave spectrum (6). This would have provided the team with the wind data during deployment and how it correlated to wave height and period, and ultimately, the flow rate. This array of sensors would have provided the WPWP with a variety of data that would have been used to determine how well the pump worked, and how beneficial it would be if used in a real-life scenario.

6.2 FUTURE IMPROVEMENTS

Although great strides were made throughout this project team's year in design and fabrication of a wave powered pump system to transfer seafloor water to surface-level kelp farms, many improvement opportunities exist to further expand the project's scope and increase the design efficiency. The most obvious task that the team wished to complete this year was the ocean testing of the full scale buoy system as this empirical data would allow for a true validation of conceptual calculations and would provide valuable data to calibrate system models for operational estimation. Data including wave motion, buoy motion, float motion, piston shaft forces, and/or volumetric flow rates could allow for sections of the buoy to be modelled to estimate the operation in different conditions.

Another significant area for improvement is the SIMULINK model that was formulated by the team. In its current state, the model is a preliminary estimation of the alpha model pump fluid dynamics; however, the principles could be extrapolated to a model for the full scale system and used in conjunction with a two-body relative motion system to create a simulation capable of estimating the system operation for any given waveform. This model could allow for NOAA buoy data near candidate application sites to be evaluated for buoy operational feasibility without requiring the construction and deployment of the physical system.

To evaluate the buoy performance for fish farm cooling applications, it would also be beneficial to investigate the heat transfer ramifications of different diffusion systems for the outlet flow. Multi-buoy deployment may be required to achieve the necessary temperature, but evaluation of that aspect was outside of the scope for this group.

A final topic for investigation would be the possible addition of a power take-off unit to store excess energy from the wave motion that is unnecessary to achieve the desired output flow rate. Control methods should be explored for the system to efficiently determine the amount of power to syphon from the system and storage methods would require further consideration as well since efficient electrical storage is one of the current major barriers to the implementation of wave energy based renewable energy. Overall, there are a variety of avenues for this project to progress and the current buoy design work provides the necessary foundational structure required to approach any of these topics.

7.0 DISCUSSION

Over the past academic year, this project successfully transformed the Wave Energy Conversion Buoy to the Wave Powered Water Pump. The goal of this project aims to create a durable, cost-efficient water pump that meets flow requirements needed to replenish cold, nutrient rich surface water in an aquaculture setting without the use of electricity. Through the application of hydrostatics, fluid flow and mechanics calculations, the interdisciplinary engineering team designed and built a full-scale model which proved to be functional during a practical tank test in mid-March. The tank test showed the efficiency of the model and highlighted a few discrepancies that could be improved upon, specifically, managing fluid leaks between the piston head and cylinder. Unfortunately, due to the COVID-19 pandemic and the subsequent transition to remote learning, final installments nor improvements could be executed. Overall, the Wave Powered Water Pump project served as a great learning opportunity for ocean and mechanical engineering students at the University of New Hampshire. The advantages of utilizing wave power will only increase as more sustainable, clean resources become necessary and this project provides a foundational starting point for future undergraduate students to expand and improve upon.

REFERENCES

[1] *"National Oceanic and Atmospheric Administration." National Oceanic and Atmospheric Administration, www.noaa.gov.*

[2] "Biofuel Production from Kelp." *ARPA*, 6 June 2016, arpa-e.energy.gov/?q=slick-sheet-project/biofuel-production-kelp.

[3] *Kimball, Chelsea, and Aaron Russell. "Wave Energy Conversion Buoy." 27 Apr. 2017.*

[4] *Norton, Robert L. Machine Design: An Integrated Approach. Prentice Hall, 2014.*

[5] Rouse, Scott. "How Does a Flow Meter Work?" *Sierra Instruments*, 9 Oct. 2017, www.sierrainstruments.com/blog/?flow-meter-work.

[6] "Spotter Buoy." *Sofar*, 2020, www.sofarocean.com/products/spotter.

APPENDICES

Appendix 01: Average Flow Calculation Estimates for Design Matrix

$$V_{avg} = \frac{\pi h r^2}{T}$$

*Where V_{avg} is average flow rate, h is the stroke length, r is the cylinder radius, and T is the wave period

$$V_{avg,bellows} = \frac{\pi * 1m * (0.0762m)^2}{3\ seconds} \cong 0.006\frac{m^3}{s} \rightarrow 96.14\ gpm$$
$$V_{avg,piston} = \frac{\pi * 1m * (0.1016m)^2}{3\ seconds} \cong 0.0108\frac{m^3}{s} \rightarrow 171.34\ gpm$$
$$V_{avg,internal} = \frac{\pi * 1m * (0.0508m)^2}{3\ seconds} \cong 0.0027\frac{m^3}{s} \rightarrow 40\ gpm$$

$$V_{Max} = 2 * V_{avg}$$
$$V_{Max,bellows} \cong 192.28\ gpm$$
$$V_{Max,piston} \cong 342.68\ gpm$$
$$V_{Max,internal} \cong 80\ gpm$$

Appendix 02: Solidworks Drawing of Aluminum Piston Head