OPT
OCEAN POWER TECHNOLOGIES

Final Scientific Report
COVER PAGE

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CONTENTS

1 EXECUTIVE SUMMARY

The project conducted under DOE contract DE-EE0002649 is defined as the Advanced, High Power, Next Scale, Wave Energy Converter. The overall project is split into a seven-stage, gated development program. The work conducted under the DOE contract is OPT Stage Gate III work and a portion of Stage Gate IV work of the seven stage product development process. The project effort includes Full Concept Design & Prototype Assembly Testing building on our existing PowerBuoy® technology to deliver a device with much increased power delivery. Scaling-up from 150kW to 500kW power generating capacity required changes in the PowerBuoy design that addressed cost reduction and mass manufacturing by implementing a Design for Manufacturing (DFM) approach. The design changes also focused on reducing PowerBuoy Installation, Operation and Maintenance (IO&M) costs which are essential to reducing the overall cost of energy. In this design, changes to the core PowerBuoy technology were implemented to increase capability and reduce both CAPEX and OPEX costs. OPT conceptually envisaged moving from a floating structure to a seabed structure. The design change from a floating structure to seabed structure would provide the implementation of stroke- unlimited Power Take-Off (PTO) which has a potential to provide significant power delivery improvement and transform the wave energy industry if proven feasible.

2 PROJECT OBJECTIVES

The objective of this project was to exploit all the experiences and intellectual property, which Ocean Power Technologies (OPT) has acquired over the last two decades about wave energy devices into a focused effort to explore additional ways to achieve the ultimate aim of the company and industry, namely, attainment of a full-scale, utility size wave power generating device capable of mass deployment in wide areas of the world.

The Project seeks to accomplish four key goals:

1. Investigate scale-up the PowerBuoy wave energy production from 150kW capability per installed unit to 500kW or higher.
2. Explore methods to Increase the power extraction efficiency to installed capital cost.
3. Seek to optimize the PowerBuoy design to increase robustness and reliability.
4. Investigate means to reduce the complexity of installation and maintenance techniques to reduce life cycle cost.

OPT will leverage its extensive experience in the development, deployment and testing of its PowerBuoy technology and associated lessons learned to investigate and implement additional design and performance improvements and optimization.

3 TASKS PERFORMED: OPT STAGE-GATE III-PB500 HYDRODYNAMIC ANALYSIS OF STRCTURAL CONCEPTS

3.1 Summary

This section of the report summarizes results from a set of tank tests. The primary goal of these tests was to reduce the risks, as far as possible, associated with choosing the overall design optimization leverage points for a wave energy point absorber system. The primary driver is summarized by the desire to minimize the Cost of power and Energy of installed power generation capacity. The cost of energy in either of these measures correlates directly with structural, mooring, Power Take-Off (PTO) loads and rated PTO power. This project is based on the PowerBuoy point absorber technology. Therefore, the power capacity for a given sea state is largely a function of the geometry of the float, its force and stroke limits, and the mooring mechanism for the spar component of the system. The tank test was therefore planned to evaluate candidate mooring systems and floats for these metrics.

Initial evaluations of different topologies of the system lead to three distinct mooring systems and three distinct floats (see Figure 1). A test plan was thus devised to evaluate the nine possible topologies for mooring and energy conversion performance. A key challenge for this test was the large difference between operational sea states and survival sea states required to fully evaluate the systems.

<table>
<tr><th></th><th></th><th colspan="3">Float</th></tr>
<tr><th></th><th></th><th>Symmetric</th><th>Cylinder w/Plate</th><th>Rhombus</th></tr>
<tr><td rowspan="3">Mooring</td><td>Monopile</td><td colspan="3" rowspan="3">The broad test goals were to evaluate the coupled performance metrics of 3 floats and 3 mooring configurations.

* Float power performance
* Float load/power shedding vs. draft
* Mooring and float loads
* Mooring power performance</td></tr>
<tr><td>Gimbal</td></tr>
<tr><td>TLP</td></tr>
</table>

Figure 1: PB500 SG III PowerBuoy Configurations

The survival model was developed at a scale of 35:1. And the operational scale model was developed at 26:1.

A summary of power performance is presented in Figure 2. This figure shows estimates for power using OPT in house proprietary and site specific predictions for wave climate analysis tool. Test results provided the ability to measure the power into the PTO as well as the power delivered to the float by the wave environment.

			Float		
			Symmetric	**Cylinder w/Plate**	**Rhombus**
Mooring	**Monopile**	**EMEC**	40-90	90-140	110-160
		Reedsport	70-120	90-140	130-180
		Portland	90-140	120-190	180-250

Figure 2: Estimates of Annual Average Power in Kilowatts (kW) Extrapolated from Operational Test Results (assuming 100% PTO Efficiency)

The following summarize results and observations made from these tests:

- Further study is required in order to define the behavior and limits of potential efficiency gains in the case of the gimbal mooring system over the design space.
- The monopile mooring provided a consistent power capture.
- The monopile mooring experienced larger forces and moments.
- The monopile mooring experienced larger float to spar loads.
- Float 2 experienced large operational surge loads and large forces and moments due to slam.
- The disturbance waves caused by the presence and motion of all configurations appeared to decay quickly. It was nearly indistinguishable by the time it radiated out to the location of the wave probes.

Test results are presented in two main sections: Power and Loads. All power values given in this report correspond to mechanical input power.

A study was also performed to determine the applicability of the three mooring options at various sites around the globe based upon available wave resource power and local bathymetry. This analysis is included as a section within this report.

3.2 Test Configurations and Float Geometries

The test plan evaluated 3 mooring configurations and 3 float geometries.

3.3 Instrumentation and Calibrations

As noted earlier, these tests were divided into two regimes; operational and survival. Consistency in instrumentation of the test was maintained where possible, but certain aspects of either test required unique accommodations. Following is a summary list of the instrumentation:

- Capacitive wave probes for sea surface elevation.
- Force dynamometer to measure mooring loads
- Force dynamometer to measure float to spar loads.
- Pressure sensors
- Qualisys IR motion tracking system.
- PTO Force and velocity.

3.4 Model Testing PTO

The PTO was calibrated at OPT by OPT personnel. The drive parameters were adjusted so that the motor thrust was achieved based on manufacturer specification. A review of the maximum operational condition results showed that PTO stroke limit did not conflict with the modeling in a statistically significant frequency.

3.5 Hydrodynamic Parameters

OPT proprietary, in house numerical codes referred to 'Frequency Domain Model', were used. Both codes are based on hydrodynamic characteristics derived from the geometry of the equilibrium wetted surface of the WEC. The parameters depend on the frequency of motion as well as that of the incident wave environment.

3.6 Loads

The structural loads of interest include mooring loads at interface with the sea bottom and the loads transmitted between the float and spar. Mooring loads were measured in both the operational model as well as the survival model. The primary reference for loads in these results will be from the survival model because it was in this model that float forces and pressure were measured coincident with the mooring loads.

3.7 Orcaflex Grounding

An important goal at the outset of these tests was to use the results to provide a basis to tune our Orcaflex modeling tools. It is desirable to explore a design space around the conditions tested in wave tank. With this in mind, the following process was employed in an attempt to achieve this goal:

- Create a systematic way to replicate the wave environment.
- Use a succession of increasingly complex models in order to validate the models construction and to evaluate the influence of the various model parameters.
- Adjust the parameters in order to capture the design metrics of concern.
- Evaluate the model performance over the test space
- Make predictions for cases not tested, specifically other survival drafts and water depths

The first step in this process utilizes the interface between Matlab and Orcaflex to transfer a time history of wave elevation from each test into an Orcaflex model. The process seems reliable and effective. OPT then explored the simplest survival model, the monopile exposed to the design regular wave.

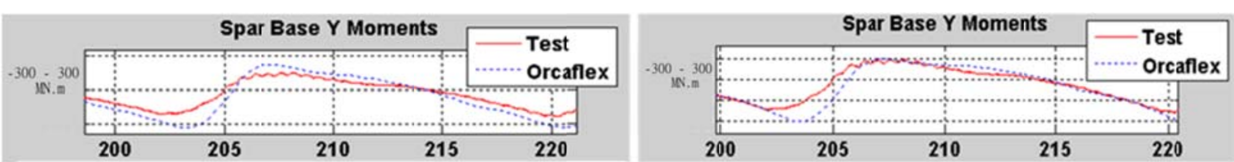

Figure 3: Comparison of conventional (left) and tuned (right) coefficients

Figure 4 shows a comparison of Orcaflex results to test results for tuned as well as conventional values for an example of a hydrodynamic coefficient. The focus for this stage was to try and get the base mooring moment as close to the test results as possible. Here we see that the moment experienced on the back side of the wave matches very well. It should be noted that it is possible to get better agreement in this test, but agreement in more complex models suffers dramatically if we force this model to match.

The next test considered was the TLP in the survival regular wave. It required very careful perturbations in the coefficients in order to match motions and certain loads. A dramatic difference in both load magnitude as well as dynamic motions was observed.

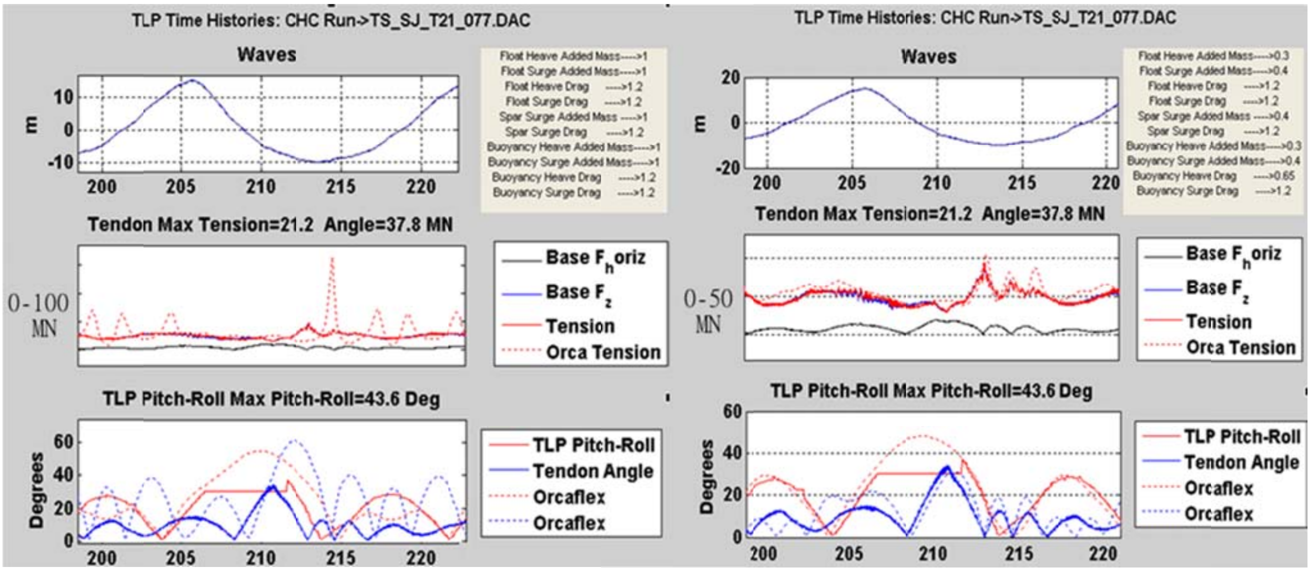

Figure 4: Comparison of Conventional (left) and Tuned Coefficients (right)

Next, performance of Orcaflex for the monopile configuration in irregular waves was assessed. Figure 5 shows a portion the 100 year storm time history. The agreement here is very good for almost all of the time series except for slam events.

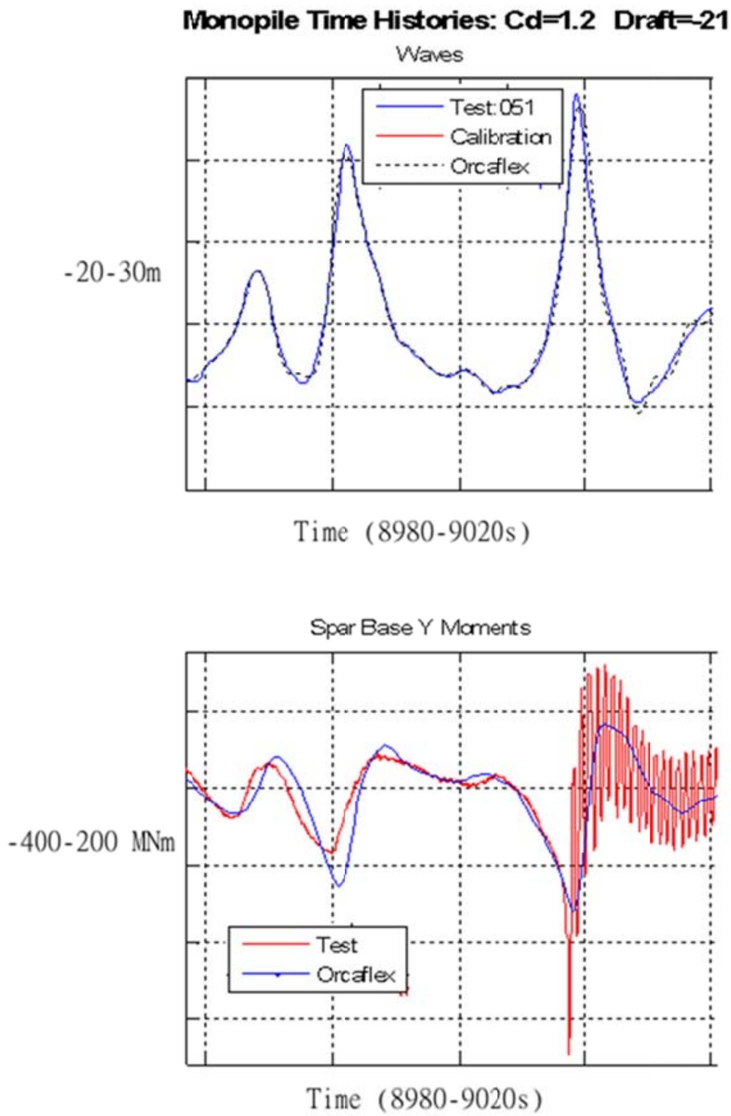

Figure 5: Orcaflex vs. Test-Monopile in irregular waves

3.8 Load Summary Comparison

The coefficients determined in the above analyses were transferred into the Orcaflex models. The following is noteworthy: First, the heave added mass is considerably lower for the float in the TLP configuration. Second, the drag on the buoyancy chamber seems low.

An Orcaflex model of each test performed for the symmetric float in the survival model test was generated using these coefficients along with the measured wave elevations at one of the wave probes. The results of these simulations are summarized and compared to test results in Figure 6.

- It was observed that Orcaflex consistently under-predicts loads in the higher sea states for the monopile and gimbal.

- Conversely, it was observed that the TLP is over-predicted. Another important finding is that perturbations in the hydrodynamic coefficients can have a strong impact on dynamics and loads. It can therefore be assumed that similar perturbations in inertia and buoyancy can lead to similar effects.
- The intermediate sea states appear well modeled by Orcaflex with marginal over-predictions. These results could be brought closer in line with test results by adjusting coefficients based on Keulegan Carpenter number or similar treatment.
- The lower sea states are greatly over-predicted by Orcaflex. Again, coefficients could be adjusted in order to fit the results to test measurements.

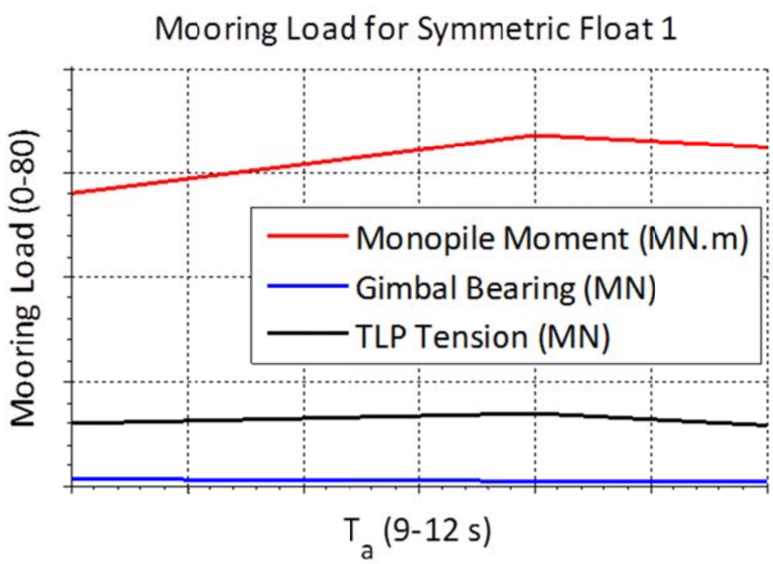

Figure 6: Predicted OrcaFlex loads from survival tests

3.9 Orcaflex Power Estimates

An initial set of Orcaflex operational models were constructed in a similar way as those used in the survival load study. The primary difference within this study was that the float was modeled in a different manner. The float component of each model was defined using an Orcaflex 'Vessel' type body. This type of modeling allows for frequency dependent added mass and damping coefficients to be assigned to the body. Figure 7 shows a summary of the Orcaflex power predictions.

The impulse response function (IRF) allows Orcaflex to capture the memory effects associated with motions of the float. These effects diminish with time but theoretically persist to infinite time. The length of time that Orcaflex considers the memory effect has a default of 20 seconds.

		Symmetric					Cylinder w/Plate					Rhombus				
H_s (m)	5.5				500-600	420-500				750-1000	510-600				550-750	550-750
	3.5			180-270		120-210			380-470		200-210			280-370		180-210
	2			50-100					150-160					80-140		
	1	5-30	5-30	5-30			15-55	15-55	30-40			5-30	5-30	5-30		
		5	6	7	9	12	5	6	7	9	12	5	6	7	9	12
		T_a (s)					T_a (s)					T_a (s)				

Figure 7: Orcaflex Power Predictions (kW)

3.10 Loads from Operational Orcaflex Model

The operational loads from the operational Orcaflex model were compared to values from the tests. Figure 8 shows a summary of the loads from the operational Orcaflex models and compares these values to test results. The results show that Orcaflex does generally predicts within 10% of the 3 hour return period for these key loads.

		Float					
		Symmetric		Cylinder w/Plate		Rhombus	
Mooring	Monopile Moment (MN.m)	90-100%	60-70%	90-100%	90-100%	60-70%	70-80%
	Gimbal Bearing (MN)	90-100%	160-170%				
	TLP Tension (MN)	120-130%	80-90%	100-110%	100-110%	90-100%	60-70%
		7	12	7	12	7	12
		T_a (s)		T_a (s)		T_a (s)	

Figure 8: For Hs=5.5m, comparison of operational OrcaFlex Mooring loads to test results

3.11 Orcaflex Grounding Summary

- Survival loads are well modeled. Slam requires careful treatment.
- Power was modeled well for the three floats.
- Operational models appear to give reasonable results for use in estimating fatigue loads.

3.12 Site Analysis

3.12.1 OBJECTIVE

The suitability of regions such as US East Coast, US West Coast, Alaska, Hawaii, United Kingdom, Spain, Portugal, for a conceptual PB500 deployment was considered based on (1) mooring requirements and (2) incident wave power. Population centers are indicated on the regional maps, to help identify sites near high power demand.

3.12.2 METHOD

- Water depth and seabed slope. For each of the target regions (US East Coast, US West Coast, Alaska, Hawaii, United Kingdom, , Spain, Portugal,), a gridded dataset of water depth (bathymetry) was downloaded from NOAA. The dataset, known as ETOPO1, contains water depth and land elevation at 1 minute spatial resolution, or equivalently 1-2 km.
- Locations off coast. For each targeted region, the coastline was extracted as a contour of zero water depth defined by a series of longitude-latitude points. At each coastline point, the local coastline orientation was determined. A coast-perpendicular line was then drawn to extend beyond the maximum offshore extent permitted for a conceptual PB500 deployment site (5 terrestrial miles).
- Suitability of site. If there were points along the coast-perpendicular line where the water depth, seabed slope, and distance from shore were suitable to a concept PB500 mooring, that coastline location was flagged as acceptable.
- Proximity to population centers. Place names and their populations were downloaded from the GEONet Names Server. While sites were not screened due to distance from population centers, cities with population over 1-5 million are labeled.
- Incident wave power. The incident wave power presented in this report is from a data product: a global wave analysis called WorldWaves performed by FUGRO. WorldWaves is the output of a global wave model that incorporates wave buoy and satellite measurements for accuracy. In deep water it has a relatively coarse spatial resolution, but in shallow coastal waters the spatial resolution is increased to better account for shoaling and refraction. Using WorldWaves, FUGRO distributes an image showing annual average incident wave power (Figure 9). For each of OPT's targeted regions, close-ups from the map were extracted, magnified, and overlaid with the local coastline, resulting in a regional map of incident wave power.

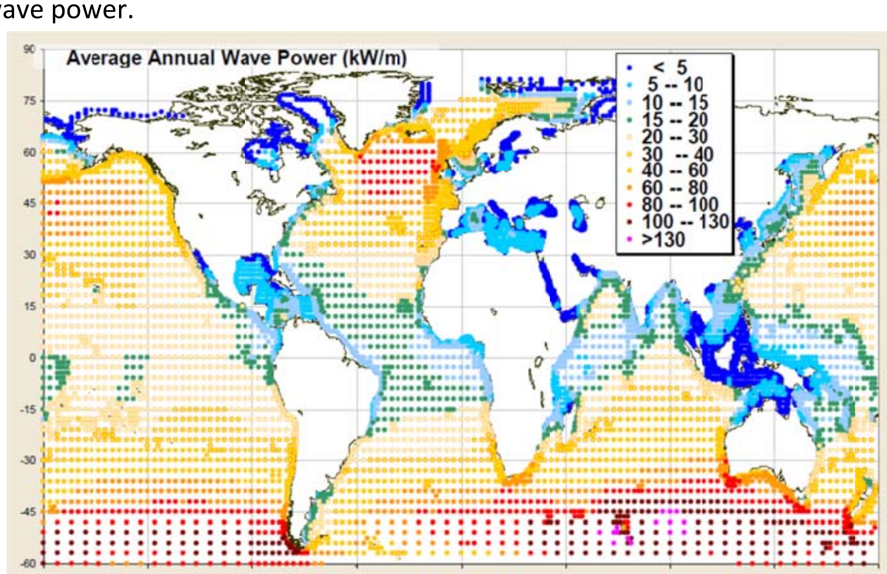

Figure 9: Annual average incident wave power around the globe. WorldWaves analysis and image from FUGRO

3.12.3 SITE ANALYSIS CONCLUSIONS

For each of the target regions, the percentage of total coastline points suited to each mooring approach is shown in Figure 10. Only the geographic criteria (water depth, seabed slope, and distance to shore) were considered when assessing site suitability.

With the exception of the Western United States, the monopile offers the greater percentage of suitability. For the Western United States' coast, on the other hand, either choice (monopole or TLP) is possible in the vast majority of this geographic area.

The plot lists the target regions in order of overall incident wave power as estimated visually from Figure 9. Few locations are suited to the gimbal and TLP (10%). In the United Kingdom, over 15% of the coastline is suited to a conceptual PB500 deployment, led marginally by the monopile. In Alaska, the TLP could be deployed along 25% of the coastline, twice the locations suited to the monopile or gimbal. In Portugal and Spain, over 20% of the coastline is suitable for monopile deployment, about 5% higher than the gimbal. On the US West Coast, over 30% of the coastline is suitable to the TLP, 10% higher than the monopile. Hawaii is dominated by the TLP (20%) with few sites suitable to the monopile (<5%), but the results are limited because the region is small relative to the spatial resolution of the datasets used in the analysis. Few sites along the US East Coast are suited to a concept PB500 deployment (5%). Over all the regions considered, the three mooring approaches give similar results (~15%), as is true in all regions except for the North Pacific (Alaska, US West Coast, and Hawaii).

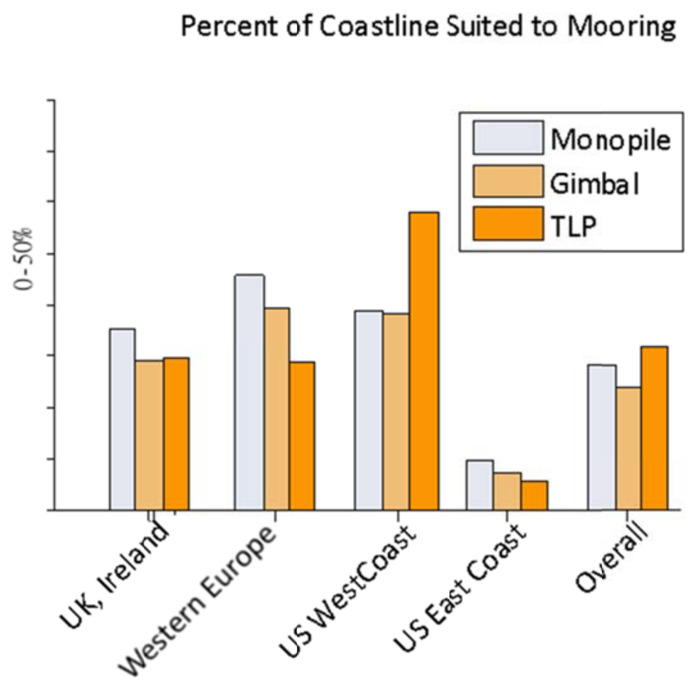

Figure 10: For each of the target region, percentage of coastline suited to deployment of monopile (blue), gimbal (light orange), and TLP (dark orange). Regions are listed in order of overall incident wave power.

3.13 Conclusion/Recommendation

A summary of the pros and cons of the various float/mooring options is shown in Table 1. Considering the power results summarized in Figure 2, the two highest power options at Reedsport are the float 3/monopile (approximately 150 kW) and float 3/TLP (approximately 120 kW) configurations.

Table 1: Summary of Pros and Cons of each combination of tested geometry

	Rhombus
Monopile	Pro Highest power configuration of all cases studied; 600-610 kW mechanical power Best agreement between predicted and measured Con Large float size Estimated base moments (-5.5m survival) 550-750 MN.m @ 40m depth 750-1000 MN.m @ 50m depth Float moment: 40-100 MN.m
TLP	Pro Second highest power studied Avoid base moment load Con Large float size High tether loads (15-60 MN) @ maximum operating sea state

Given float 3 at the best float condition, the choice then becomes the mooring system: monopile vs. TLP. Although a direct comparison of the mooring loads is not possible between the two systems, the evidence suggests that the peak TLP loads are relatively easier to handle in comparison with the survival moments associated with the monopile. Thus, we are left to choose between the reduced loadings of the TLP against the increased power output of the monopile.

It is recommended that a few alternative geometries (perhaps three or four options) based upon float 3 be tested so as to minimize the survival loads.

3.14 Future Testing

The following are some of the lessons learned in this set of wave tank tests.

- Involve as many people representing various engineering disciplines and operations as possible from the outset. Valuable advice helps avoid a number of mistakes in this kind of test.

- Video. Time must be allotted to carefully plan the location of the camera and the quality of the pictures, as well as the synchronizing of the camera with the test events. Camera angle picture quality is extremely critical.
- File naming conventions during the testing program must be specified carefully. A fixed set of rules helps immensely as data is reviewed and post-processing routines are written.
- Some system dynamics modeling must be performed ahead of time.
- The external losses such as mechanical interface motion friction and their relationship with system efficiency and float geometry should be characterized and included in this type of modeling if possible.

4 TASKS PERFORMED: OPT STAGE GATE III-PB500 POWER TAKE-OFF SYSTEM

This section details the initial conceptual design of the PB500 project Power Take-Off (PTO) system including the component testing and final recommendations.

4.1 Loading Requirements

PTO design loading requirements were developed through computer simulation, performed by OPT's systems engineering group utilizing OPT proprietary engineering tools. Such requirements as PTO force and PTO operational duty cycle, were then used to carry out a trade study involving a variety of linear to rotary mechanical systems. Careful considerations were given to low percentage occurrence Peak operating conditions versus more frequent and lower severity operating conditions.

4.2 Rack and Pinion based linear to rotary conversion system

A rack and pinion PTO concept was investigated. For this concept, several key technologies are required for its implementation; thus, some initial design, analysis, and testing was necessary to fully vet each component of the buoy. These technologies include:

- Rack and pinion
- Input rod
- Input rod cabling
- Speed increaser
- Brake
- Linear seal
- Flexible cable

The details for the design, analysis, and testing are described below.

4.2.1 WIRE ROPE SYSTEM

As part of the PTO encapsulation within the float, one concept is to use a rack, fixed to the spar through external means but within the float. Therefore, all corrosion and fouling issues are eliminated from the PTO system. To meet other requirements related to tide, maintenance and survivability, the concept of a wire rope pulley system was envisioned that would have the capability of adjusting the position of the rack within the float relative to the spar

The wire rope system acts as a pulley. Assuming a survivability float submergence depth of 15m to 25m from the nominal waterline, the system would consist of 2 ropes, a haul-down rope and a hoist rope. The preload in the rope is critical to prevent any snagging/snap loads that may results in a sudden, catastrophic failure of the wire rope system. The end of the wire ropes at the input rods would terminate using a standard open socket configuration while the ends of the ropes at the winch drum will terminate on the drum using a standard swaged button design.

Upon developing the general concept of the wire rope system, several rope companies were approached to provide solutions. Off the shelf rope used in oil and gas applications was considered and configurations of the rope based system were assessed accordingly. An example of such a rope is shown in Figure 11.

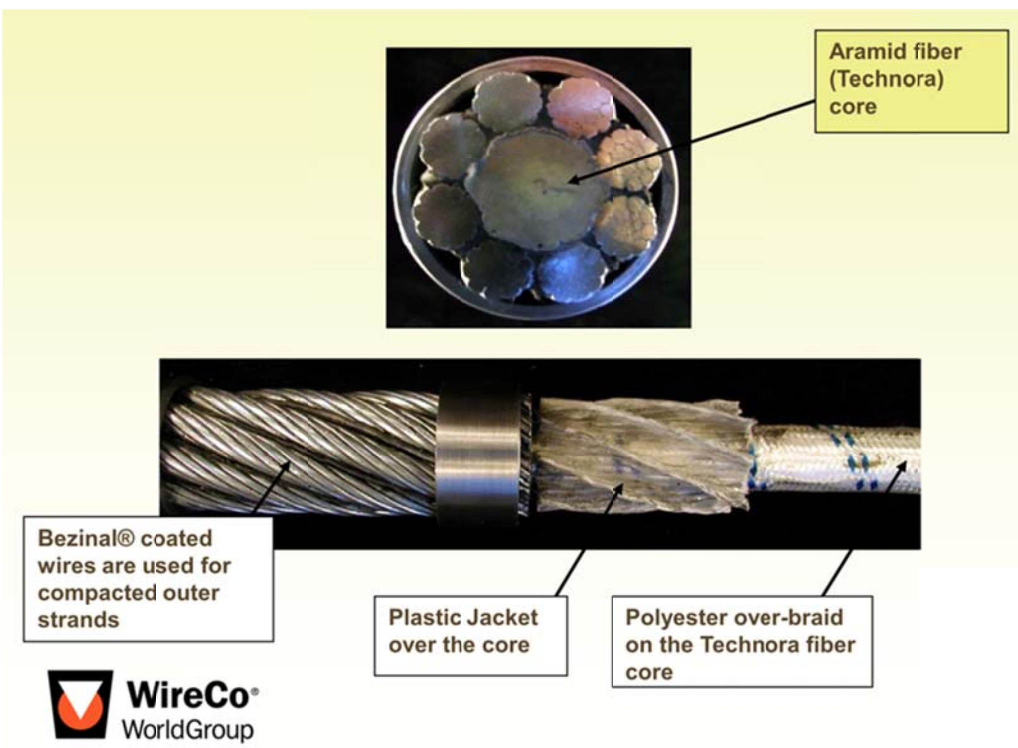

Figure 11: WireCo's Bezinal coated hybrid rope

4.2.2 PINION GEAR

The pinion gear is the primary power transmission element in the Rack & Pinion motion conversion system. Therefore it is necessary to understand the pinion size required to transmit the design torque at the number of design cycles with an acceptable design life.

Commercial companies were approached for pinion gear sizing, and each company made recommendations based on the anticipated MetOcean conditions and associated load requirements.

Each company made very different recommendations for pinion size and module for each design configuration. Careful consideration must be given to pinion size as it has a direct impact on generator efficiency and size.

4.2.3 SPEED INCREASER

Using a speed increaser to increase the rotational motion could allow the use of smaller more efficient generators. The following sections detail research into three types of speed increasers.

4.2.3.1 GEARBOX

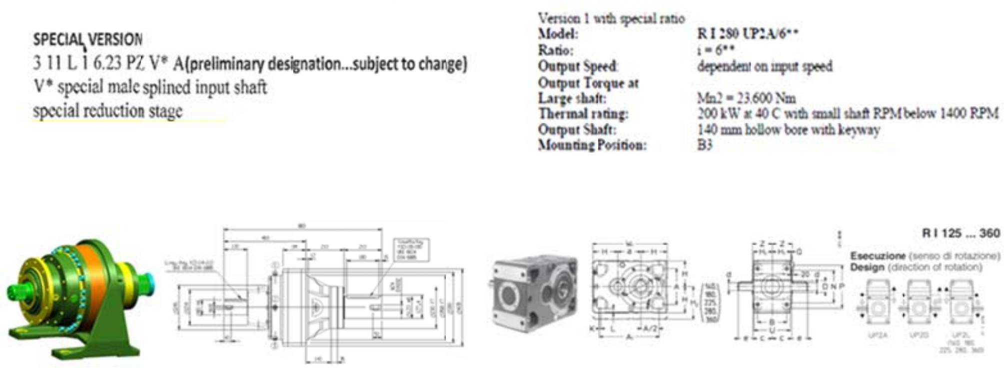

Figure 12: Potential Gearbox Model Selections

4.2.3.2 BELT REDUCTION (BELT BOX)

OPT worked with the appropriate vendors to identify a belt and pulley configuration which could serve as a speed increaser for the PB500 concept. Two concept configurations were considered. A conceptual drawing of the belt box configuration in Figure 13, note that the conceptual design is 1.2 meters in length and height and 0.285 meters in width.

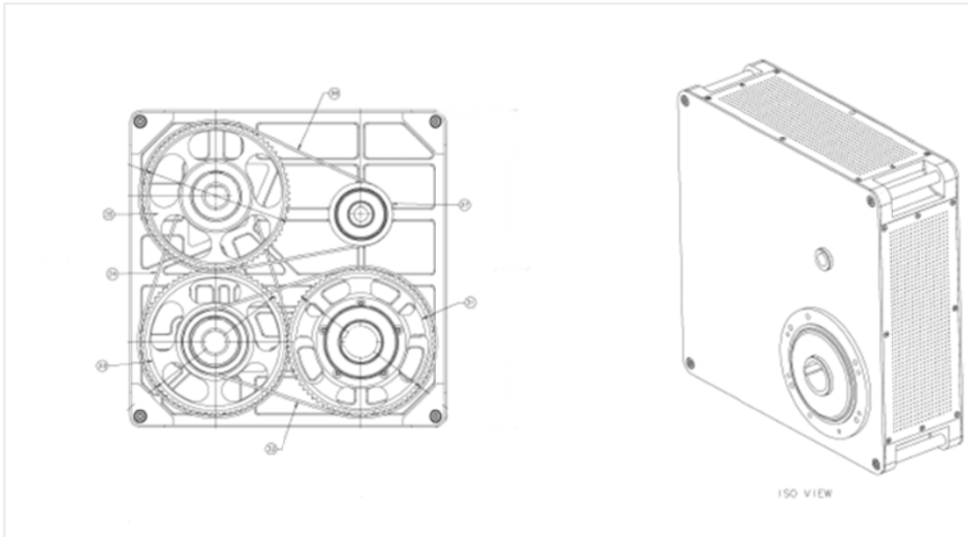

Figure 13: Box Conceptual Drawing

4.2.3.3 CHAIN REDUCTION (CHAIN BOX)

OPT worked with a supplier to select a chain drive configuration which could serve as a speed increaser for the PB500 concept. A concept design configuration shown in Figure 14, consisting of 24 chain boxes was assessed. Note that the conceptual design is 2.9 meters in length, 1.9 meters in height, and 0.635 meters in width.

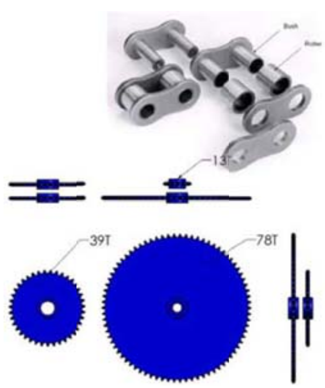

Figure 14: Chain Drive

Due to the size of the chain box further research was not performed.

4.2.3.4 SPEED INCREASER ECONOMIC COMARISON

Table 2 compares the three systems proposed for use as a speed increaser in the concept PB500 system.

Table 2: Speed Increaser Economic Comparison

Gearbox vs. Belt Box Comparison

	Gear Box				Belt Box		Chain Box
	Gearbox 1		Gearbox 2				
Number of Power Take Off Systems	16	24	16	24	24		24
Peak Power (kW)	320 - 430	215 - 285	320 - 430	215 - 285	215 - 285		215 - 285
No. Stages	1	1	1	1	3		3
Anticipated Efficiency	98%	98%	98%	98%	94%		94%
Size (m^3)	0.3	0.3	0.22	0.22	0.43		3.5
Cost per Unit	\$8k - \$12k	\$8k - \$12k	\$5k - \$7k	\$5k - \$7k	\$16k - \$24k		
Maintenance Item	Seals & Oil	Seals & Oil	Seals & Oil	Seals & Oil	Belt	Sprocket	
25 Year Total Cost per Buoy	\$150k - \$450k	\$225k - \$675k	\$125k - \$375k	\$175k - \$525k	\$500k - \$1500k		

4.2.3.5 RECOMMENDATIONS

It is clear from the preceding analysis that the gearbox speed increaser is the best value approach balancing complexity and cost.

4.2.4 FLOAT BRAKE

The mechanical float brake is classified as the device which allows transition from one functional state to another such as from Normal Operation to Survival Mode when exceedingly destructive sea states are present (severe storms), or from Normal Operation to Maintenance Mode, etc.

The mechanical requirements for the mechanical float brake are:

- The Mechanical Locking System shall be designed to handle the operational loads and wear in Sea State 6.
- The wear components of the Mechanical Locking System shall be designed to be easily replaceable at sea within the normal maintenance requirements.
- The Mechanical Locking System shall be designed to activate with full force within less than 2 seconds.

4.2.4.1 FLOAT BRAKE CONFIGURATIONS

Four float brake configurations were considered early on in the design process each of which could be employed in the various conceptual systems, they are as follows.

- External Brake (exposed to sea)
    - Linear: Float to Spar
    - Rotary: Spar to Sheave
- Internal Brake (enclosed in float)
    - Linear: Float to Rack
    - Float to Pinion

4.2.4.2 CALIPER BRAKES

The following caliper brake models were recommended by various suppliers to meet the braking force requirements.

(A) HYDRAULIC APPLIED

External Linear –

- Tangential Brake Force Required
    - Ft = 300kN to 900kN
- Supplier 1/
    - Quantity / Buoy: 3
    - Operating Pressure: 200 bar

Internal Linear –

- Tangential Brake Force Required
    - Ft = 300 kN to 900kN
- Supplier 1/
    - Quantity / Buoy: 2

Figure 15: Hydraulically Actuated Brake Example

External Rotary –

- Tangential Brake Force Required
    - Disc Size = 0.75 meter to 3m
    - Reduced Linear PTO Force = 200 kN 700kn
- Supplier 1/
    - Quantity / Buoy: 1
    - Operating Pressure: 200 bar

(B) SPRING APPLIED

External Linear –

- Tangential Brake Force Required
    - Ft = 300 kN to 900kN
- Supplier 1/
    - Quantity / Buoy: 4

Internal Linear –

- Tangential Brake Force Required
    - Ft = 300 kN to 900kN
- Supplier 1/
    - Quantity / Buoy: 2
- Supplier 2/
    - Quantity / Buoy: 2

Figure 16: Example of Spring Actuated Brake

External Rotary –

- Tangential Brake Force Required
    - Disc Size = 0.75meter to 3m
    - Reduced Linear PTO Force = 200 kN to 700kN
- Supplier 1/
    - Quantity / Buoy: 2

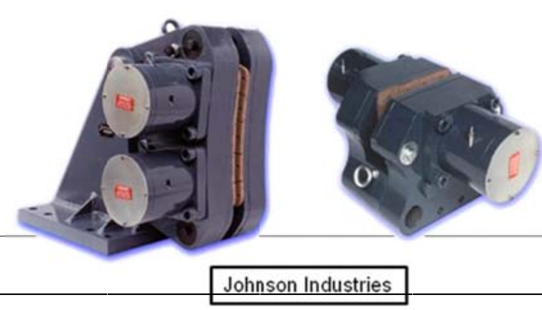

Figure 17: Example of Spring Actuated Brake

Internal Rotary –

- Tangential Brake Force Required
    - Disc Size = 0.25m to 1m
    - Torque = 20kN – m to 50kN
- Supplier 1/
        - Quantity / Buoy: 1
- Supplier 2/
        - Quantity / Buoy: 1

(C) COMPARISON OF HYDRAULICALLLY APPLIED VS. SPRING APPLIED BRAKES

The following list of pros and cons that compares the advantages and disadvantages of both hydraulically applied and spring applied brakes.

Hydraulic Applied –

- Pros –
    - Pad wear is automatically compensated for increased piston stroke
    - Fewer moving parts
    - No stress on components unless activated
- Cons -
    - Hydraulic pressure required to maintain clamping

Spring Applied –

- Pros –
    - Brake is fail safe
- Cons –
    - Manual adjustment required to compensate for pad wear
    - Components always under stress
    - Seal life reduced due to constant release pressure required

Selection of brake application style should take the above advantages and disadvantages into consideration.

Figure 18: Supplier 1/ Hydraulic Applied

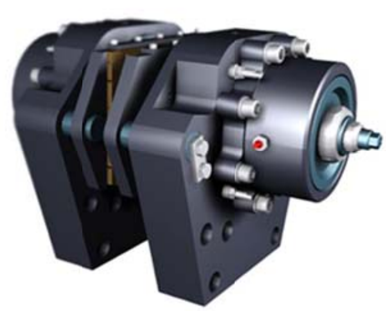

Figure 19: Supplier 1/ Spring Applied

4.2.4.3 HYDRAULIC POWER UNIT

Each brake system will require a HPU (Hydraulic Power Unit) to apply or release the brake. The size and capacity of the HPU will be determined after the final brake configuration is selected. The HPU shown below is an example of an off the shelf product which could be used to control any of the brake systems.

Specifications (see Dellner HPU Spec) –

- Pressure Capacity : 200 bar
- Power Requirements
  - Pump
    - 380-420 V, 50 Hz; 0.75 kW
    - 440-480 V, 60 Hz; 0.86 kW
  - Valves
    - 24 V DC, 110 V AC, 230 V AC
    - Amps
- Flow Rate; 2.2 (dm^3/min)
- Price: $3,640.00 each (see Dellner HPU Quote Stage III)

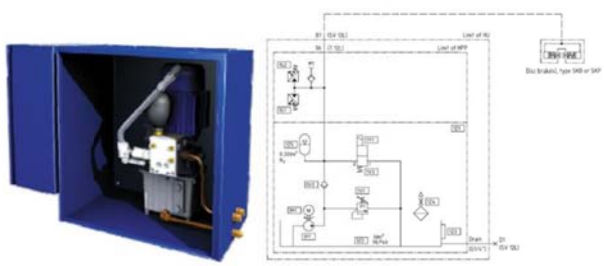

Figure 20: Supplier 1/ Hydraulic Power Unit

4.2.4.4 ECONOMIC COMPARISON

Table 3 is an economic comparison of each brake model recommended for use on the concept PB500. It is important to note that each brake is capable of maximum different tangential forces; each brake is compared on an equal level by comparing the cost per kN of braking force and the size.

Table 3: Float Brake Economic Comparison

	Caliper Brake				Rod Lock
	Spring Applied	Hydraulic Applied	Spring Applied	Hydraulic Applied	Spring Applied
Number of Brakes Required Per Buoy (Assuming Linear Internal)	2	2	2	2	8
Salt Water Compatible	Yes	Yes	No	No	No
Max Tangential Force / Brake (kN)	250 - 350	340 - 460	530 - 720	340 - 460	340 - 460
Size (m^3)	0.2	0.05	0.22	0.3	0.1
Capital Cost per Buoy	$16k - $24k	$10k - $14k	N/A	N/A	$150k - $250k
Maintenance Item	Seals & Oil		Seals & Oil		Seals & Oil
25 Year Total Cost per Buoy	$15k - $45k	$10k - $30k	$40k - $110k	$10k - $25k	$175k - $525k

From the table above it is seen that hydraulic applied brakes are much less expensive than spring applied brakes. Final brake recommendations are made at the end of this report.

4.2.5 SHAFT COUPLING

Flexible shaft couplings are required to reduce the precision alignment required during PTO assembly. Three coupling types, shown in Figures 21 and 22, are presented as possible solutions for varying input torque requirements.

Figure 21: Bellows Coupling

Design Loading –

- Maximum Torque: approximately 4500N-m

Supplier 1/ Bellows Coupling –

- o Rated Torque = 4000 N-m
- o Peak Torque = 6000 N-m
- o Misalignments
    - ▪ Angular : 1.5 deg
    - ▪ Parallel: 0.4 mm
    - ▪ Axial: 3.5 mm

Supplier 2/

- o Rated Torque = 2825 N-m
- o Peak Torque = 5650 N-m
- o Misalignments
    - ▪ Angular : 3 deg
    - ▪ Parallel: 2.6 mm
    - ▪ Axial: 9.1 mm

Figure 22: Disc Coupling

4.2.5.1 SHAFT COUPLING FOR PB500 CONCEPT

Design Loading –

- Maximum Torque: approximately 40,000 N-m
    - o 16 Pinion PTO
    - o No Gearbox

Supplier 1/ in Compression Coupling –Rated Torque = 14,751 N-m

- Peak Torque = 44,254 N-m
- Misalignments
    - o Angular : 0.5 deg
    - o Parallel: 1.5 mm
    - o Axial: 2.5 mm
- Key Features
    - o Severe shock load protection / dampening
    - o Intrinsically fail safe
    - o Couplings proven in marine industry

Figure 23: SUPPLIER 1/Rubber in Compression Coupling

4.2.6 LINEAR BEARINGS

Linear bearings are required for the rack and pinion PTO option as a guide for the rack within the float. Therefore, initial life calculations were performed to size a bearing and mitigate any future design risk.
Using the criteria provided by the manufacturer, the estimated life is ~7 years using the specific supplier bearing product. However, adding an additional bearing block to the linear bearing results in a life increase to ~18 years. Also, two linear bearing rails may be used in parallel to further increase the bearing life to 25 years. Therefore, the risk is relatively low at this point.

4.2.7 ROTARY BEARINGS

Rotary bearings are required for the rack and pinion option as a support for the pinion within the float. Therefore, initial life calculations were performed to size a bearing and mitigate any future design risk. The assumptions for these calculations were:

- 1 pinion per PTO
- 2 bearings per pinion

Therefore, using the standard rotary bearing life equation:

$$Life = \left(\frac{C}{P}\right)^{\frac{10}{3}} \times \left(\frac{16667}{vel}\right)$$

Where P is the equivalent load, C is the dynamic load rating of the bearing, v is the velocity in rpm and the life is calculated in hours.

Using the criteria provided by the manufacturer, the estimated life is ~30 years using a specific supplier pillow block bearing product. Therefore, the risk to the project from a rotary bearing standpoint is relatively low at this point. Additionally, standard off-the-shelf roller bearing could also be used with similar and higher load ratings.

4.2.8 LINEAR SEAL

A linear seal is required in the rack and pinion PTO design in order for the input rod to penetrate the float as the input rod penetrates the spar. However, as the input rod passes through the top and bottom of the float, the linear seal will be subjected to a constantly submerged condition at pressures of 2.5 bar to 10 bar during a survivability state. Therefore, an investigation into different linear seals was conducted.

A specification for the linear seal was developed and presented to 3 seal manufacturers some of the key specifications were:

- Target life: 5 years or more is desired
- 2 million to 5 million cycles per year

Each vendor prepared a seal design for OPT which were purchased for testing. The following section describes each design.

Some additional notes from each of the seal manufacturers were:

- All seals will leak "somewhat" over time
- Testing is the only method to predict leakage rate and life
- Bar sealing capability is not problematic, especially in a "quasi-static" state

4.2.9 FLEXIBLE CABLE & MANAGEMENT SYSTEM

Flexible power and communications cable will traverse between the float and the spar of the PB500 to transmit power from the PTO system to the subsea cable attached to the spar. The electrical and mechanical requirements for the cable management systems are detailed in OPT's specifications. The following sections focus on the products and components of each cable management system proposed.

Flexible cable was sized based on voltage and current requirements that were defined and approved by OPT.

Table 4 lists companies which were sent an RFP to meet OPT's requirements:

Table 4: Cable Companies Sourced

Company	Cable Management System	Response
Philatron	Cable Coil	Proposed Solution
Cable Science	Cable Coil	Proposed Solution
IGUS	Cable Carrier	Proposed Solution
Draka	Cable Carrier	Proposed Solution
Prysmian	Cable Carrier	Proposed Solution
Conductix	Cable Reel	Proposed Solution

4.2.10 SHOCK ABSORBERS

The following shock absorbers are proposed as product solutions to meet the needs of multiple PTO and Tidal Compensation Systems. The use of these off the shelf shock absorbers is described in more detail in the Tidal Compensation System section of this report.

8 Shocks / Buoy

- Configuration:
  - 8 shocks per impact direction
  - Size: (L x W x H)
  - 737 mm x 333 mm x 333 mm

16 Shocks / Buoy

- Configuration:
  - 8 shocks per impact direction

Figure 24: Example of an off the shelf Shock Absorber

- Size: (L x W x H)
- 737 mm x 333 mm x 333 mm

4.2.11 LOCKING MECHANISM

A locking mechanism for the float is required during maintenance and survivability states where the float is raised on the spar or lowered underwater. Therefore, the float can be locked to the spar without PTO assistance. Two possible configurations were investigated:

- Shear Pin Mechanism
- Latch Mechanism

An initial design and analysis were completed for each below.

4.2.11.1 SHEAR PIN MECHANISM

The shear pin mechanism is intended to lock the float to the spar using an electrically or hydraulically actuated pin, the two mechanism types are detailed below.

(A) ELECTRICALLY ACTUATED

The shear pin mechanism design used the following assumptions:

- 4 Shear pins per float
- Float weight = 300 to 600 tons

Using these assumptions, a shear pin mechanism was designed as shown in Figure 25. The design consists of:

- 6” diameter shear pin
  - 17-4 PH stainless steel material
  - Tapered pin
- Linear actuator – 2 options
  - up to 15,000 lbf capability
  - Hydraulic piston (alternative)

Each shear pin mechanism mounts to the top of the float at equally spaced positions with a matching receiver on the spar. The locking/unlocking process would proceed as follow:

- The float would use the internal or external method if necessary to adjust to locking position
- A feature (i.e., guide pin or equivalent) would align the shear pin mechanism on the float with the receiver.
- Once the float achieved the locking position, the shear pin mechanism will activate and the shear pins will engage the receiver.
- The positioning device, would disengage; thus, leaving the shear pins to constrain the float

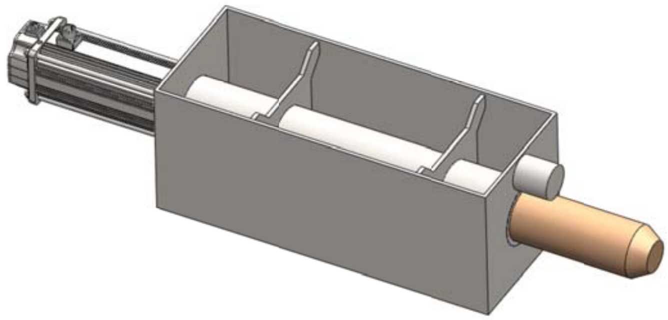

Figure 25: Shear pin actuator

During the preliminary design, an initial FEA on the shear pin mechanism was conducted to evaluate the design feasibility. A symmetric model of the shear pin mechanism engagement with the receiver was developed as shown in Figure 26. The results showed stresses in the receiver of ~500 MPa (see Figure 27). However, this is a conservative analysis showing high hot spot stresses; therefore, further refined FEA model would likely reduce the maximum stress. Additionally, modification of the design will also be used to further reduce the stress in the receiver. Next, the shear pin showed a maximum stress of 462 MPa (see Figure 28). The shear pin analysis also was conservative and the design can be modified by slightly increasing the pin diameter to decrease the maximum stress. Therefore, the overall technical risk to the shear pin concept is low and can be mitigated through an optimized design.

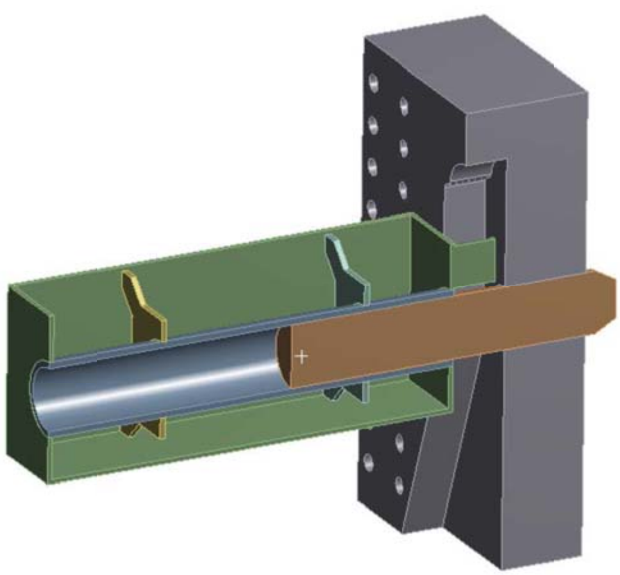

Figure 26: Symmetric model of the shear pin mechanism

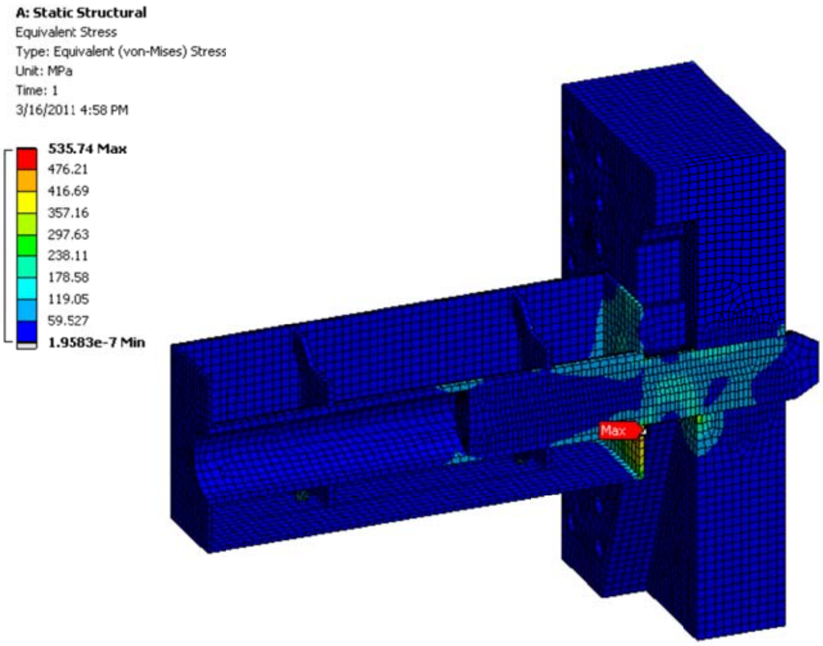

Figure 27: Overall FEA results

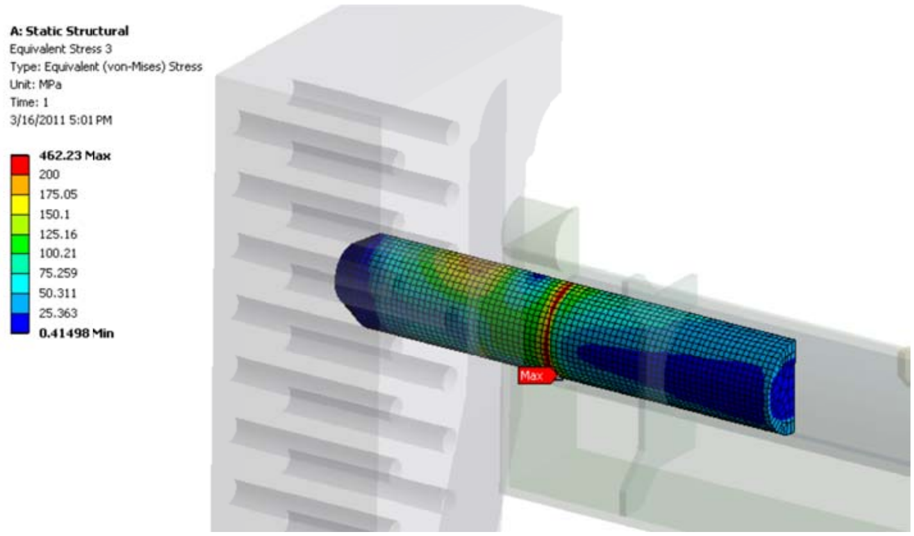

Figure 28: FEA results for shear pin

(B) HYDRAULICALLY ACTUATED

The shear pin mechanism design used the following assumptions:

- 4 Shear pins per float
- Float weight = 300 to 600tons

Using these assumptions, an off the shelf shear pin mechanism was specified as shown in Figure 29; the design and locking capacity are shown below.

Specifications –

- Locking Force: 1000 kN
- Working Pressure: 200 bar
- Price: $16,900.00 each

Figure 29: Example of off the shelf Hydraulically Actuated Shear Pin Lock

Each shear pin mechanism mounts to the top of the float at equally spaced positions with a matching receiver on the spar. The locking/unlocking process would proceed as follow:

- The float would use internal or external method if necessary to adjust to locking position
- A feature (i.e. guide pin or equivalent) would align the shear pin mechanism on the float with the receiver
- Once the float achieved the locking position, the shear pin mechanism will activate and the shear pins will engage the receiver.
- The shear pin safety lock would engage the shear pin locking it in place
- The positioning device, would disengage; thus, leaving the shear pins to constrain the float

4.2.11.2 LATCH MECHANISM

An alternative to the shear pin concept would be a latch mechanism. The latch mechanism design also used the following assumptions:

- 4 Latch mechanisms per float
- Float weight = 300 to 600 tons

Using these assumptions, a latch mechanism was designed as shown in Figure 30. The design consists of:

- 8” diameter latch
  - 17-4 PH stainless steel material
- Pillow block bearings – 2 required
  - Off the shelf example – dynamic capacity = 1.6MN each
- Linear actuator – 2 options
  - Of the shelf example: up to 15,000 lbf capability
  - Hydraulic piston (alternative)

Each latch mechanism mounts to the top of the float at equally spaced positions with a matching receiver on the spar, similarly to the shear pin mechanism concept. The locking/unlocking process would proceed as follows:

- The float would use internal or external method if necessary to adjust to locking position
- Once the float achieved the locking position, the latch mechanism will activate, rotate, and engage the receiver.
- The positioning device, would disengage; thus, leaving the latch to constrain the float

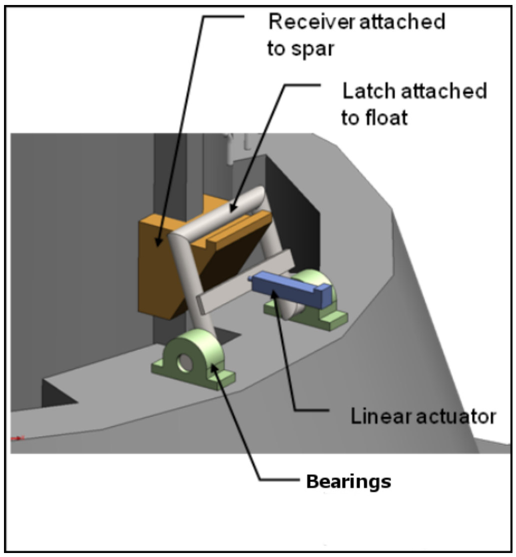

Figure 30: Latch mechanism design

During the preliminary design, an initial FEA on the latch mechanism was conducted to evaluate the design feasibility similarly to the shear pin concept. A model of the latch mechanism engagement with the receiver was developed as shown in Figure 31. The results showed stresses in the receiver of ~318 MPa (see Figure 32). However, this is a conservative analysis with a simplified model; therefore, a further refined design and FEA model would likely reduce the maximum stress. Next, the latch (8" diameter rod weldment) showed a maximum stress of 255 MPa Therefore, the overall technical risk to the latch mechanism concept is low and can be mitigated through an improved design similar to the shear pin concept.

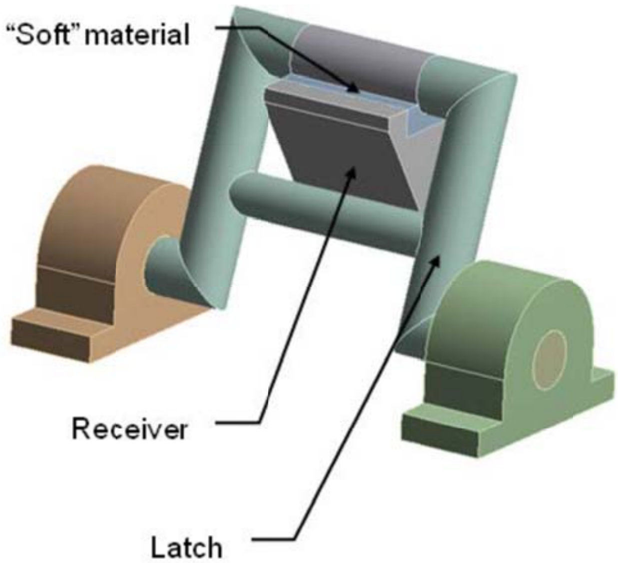

Figure 31: Latch mechanism FEA model

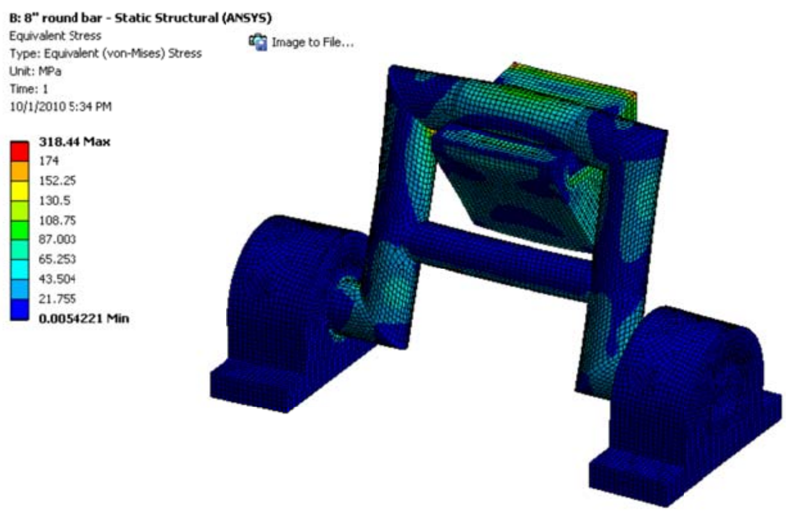

Figure 32: FEA results for latch mechanism

4.2.12 RACK & PINION PTO SYSTEM CONFIGURATIONS

4.2.12.1 RACK & PINION PTO SYSTEM CONFIGURATION #1

This Rack & Pinion PTO System Configuration, employs a fixed input rod input system and a rack and pinion PTO mounted inside the float acting against a rack mounted inside the float. The rack assembly is connected to the input rod via rod locks mounted to the rack assembly and clamping against the input rod. The generator and pinion are stationed inside the float, the pinion engages the rack. Waves drive the relative linear motion between the spar and the float, and thus the relative linear motion created between the rack and the pinion will convert the linear wave motion to rotational motion driving the generator.

4.2.12.2 RACK & PINION PTO SYSTEM CONFIGURATION #2

This Rack & Pinion PTO System employs a wire rope input and adjustable sheave input system and a rack and pinion PTO mounted inside the float acting against a rack mounted inside the float. The wire rope is connected to a rack inside the float through two input rods. The generator and pinion are stationed inside the float, the pinion engages the rack. Waves drive the relative linear motion between the spar and the float, and thus the relative linear motion created between the rack and the pinion will convert the linear wave motion to rotational motion driving the generator.

- PTO are completely internal to the float
    - Generator, gearbox (if necessary) and pinions are fixed to the float structure
    - Each PTO module can be repaired/replaced with relative ease

4.2.13 TIDAL COMPENSATION SYSTEMS

Tidal compensation is required for a PowerBuoy operating at sea subject to variations in the mean waterline due to raising and lowering tides. The tidal compensation system allows a PowerBuoy to adjust the free stroke of the input system as the mean waterline changes throughout the day due to tidal variations. Figure 33 shows the tidal variations throughout the world.

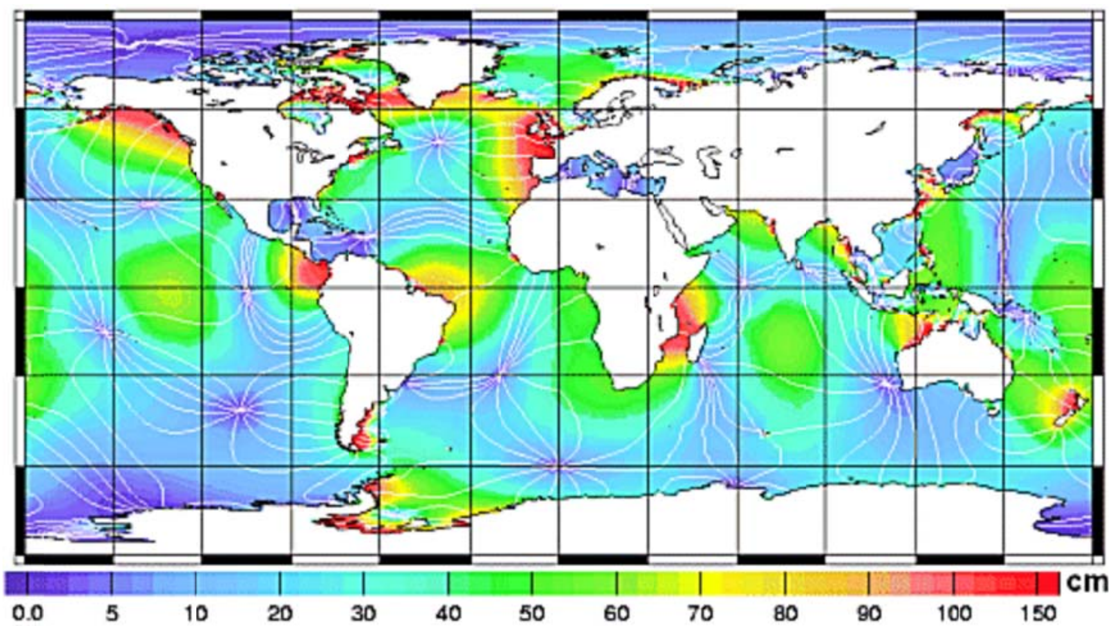

Figure 33: Map showing tidal variations across the globe. Red areas represent large variations in water level, purple areas represent zero or very low tidal variation. Image Credit: Legos/CNRS.

The sections below describe tidal compensation system concepts intended for use on PB500 concept PowerBuoy.

4.2.13.1 FIXED INPUT ROD INPUT RACK & PINION PTO SYSTEM

A fixed input rod PTO system previously discussed employs an input rod passing through the float and attaching to the spar at both ends. The rack assembly is connected to the input rod via rod locks mounted to the rack assembly and clamping against the input rod. The generator and pinion are stationed inside the float, the pinion engages the rack. Waves drive the relative linear motion between the spar and the float, and thus the relative linear motion created between the rack and the pinion will convert the linear wave motion to rotational motion driving a generator. As the tide raises and lowers so does the mean position of the float; it is therefore necessary to adjust the mean position of the rack to maximize the stroke. The internal sliding rack tidal adjustment devices adjust the rack position to maximize the stroke.

System Pros –

- Entire adjustment system internal
- High load capacity
- Shorter section of rack required
- Capable of automatic adjustment

System Cons –

- Flexible hydraulic hoses required
- 2 Piece input rod required

4.2.13.2 WIRE ROPE INPUT SYSTEM

The basic components employed in a sheave and wire rope system.

A sheave and wire rope input system employs a wire rope wrapped around a drum or sheave which fixes the position of the wire rope with respect to the buoy spar. The wire rope is connected to a rack inside the float through two input rods. The generator and pinion are stationed inside the float, the pinion engages the rack. Waves drive the relative linear motion between the spar and the float, and thus the relative linear motion created between the rack and the pinion will convert the linear wave motion to rotational motion driving a generator. As the tide raises and lowers so does the mean position of the float; it is therefore necessary to adjust the mean position of the rack to maximize the stroke. The sheave tidal adjustment devices enable the sheave to adjust the mean position of the rack.

The decision to adjust the rack position with respect to the spar to compensate for changing tides is made by the control algorithm. The position of the rack with respect to the spar is measured by a position sensor. The position of the float with respect to the rack is measured by another position sensor. As the tide changes, the mean position of the float with respect to the rack is recorded. Tidal compensation is required when the mean position of the float with respect to the rack moves above or below a set limit. The set limit is defined by the control algorithm.

4.2.14 FLOAT SURVIVABILITY

Float survivability position will be implemented to avoid overstressing the buoy during storm conditions. The following actions will be taken to move the float to a survival position.

- Buoy control system enters locked state
    - Brakes engage
- Float ballasting must be started
- Tidal compensation would be engaged
    - Configuration #1 – Adjustable rod lock system
        - Internal sliding rack
    - Configuration #2 – Wire rope adjustment system
        - Caliper brake sheave lock
        - Hydraulic sheave lock
- Float would gradually sink to desired position
- Locking mechanism would secure float at survivability position
- Reverse procedure to return float to the operational locked state

4.2.15 FLOAT MAINTENANCE

Maintenance to components of the buoy will be performed when the float is at the top of the spar and locked in place. Moving the float to the maintenance position must be possible without assistance. Four of the following motor driven chain fall winches mounted to the lifting shackles on the spar could perform this lift.

- For a winch system, the motor/gearbox must provide 300 t0 800kN*m of torque
    - Off the shelf solution: Gearbox = 475 kN*m rating
        - Dimensions are 1.2m diameter x 2.2m long
        - Feasible but very large
- Chain fall hoist option (Liftchain)
    - Hydraulic hoist
    - 100 ton option
    - Requires additional structure to mount
        - Design issue
    - Compact footprint

4.3 Belt Drive PTO Technology

The Belt Drive System PTO technology consists of using a double-sided belt on the float to engage a rack that is fixed relative to the spar. This technology allows the PTO to be internal or external to the system as described in the Rack & Pinion PTO System Configurations section. This belt drive also provides many advantages, such as:

- Eliminates the need for an input rod and linear seals (as part of the external configuration)

- Allows for different rack options
  - Full length rack along the spar
  - Operational range rack with tapered ends
- Alignment of PTO (float to spar)
  - Lower tolerances required
- No lubrication required
  - Belt and rack can run dry
- Compact PTO design

For this concept, several key technologies are required for its implementation. These technologies include:

- Belt drive
- Rack (options)
- Rotary bearings
- Gimbal system

The details for the design, analysis, and testing are described below. Since some of the technology from the rack and pinion system is also used on the belt drive, the following section will discuss only those components not described above in the Rack & Pinion PTO Technology section.

4.3.1 COMPONENT CONFIGURATION

The following sections detail the research performed on components necessary for the various configurations of the Belt Drive PTO system.

4.3.1.1 BELT DRIVE SYSTEM

The primary component in this concept is the belt drive technology and whether this design can meet our system requirements. Several belt manufacturers were engaged to determine initial feasibility of this concept. Only 1 manufacturer responded positively to OPT's request. Therefore, collaboration with the manufacturer was started with the concept selection process. Several belt configurations, rack configurations, and drive configurations were reviewed:

- Belt configuration
  - 14mm pitch, single-sided toothed belt
  - 14mm pitch, twin power (double-sided) toothed belt
  - 19mm pitch, single-sided toothed belt
  - 19mm pitch, twin power (double-sided) toothed belt
- Rack configuration
  - Urethane rack
  - Spliced-belt rack
- Drive configuration

- Omega drive shown in Figure 34

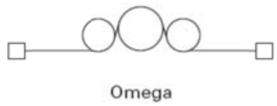

Figure 34: Omega drive configuration

- Twin Power caterpillar drive shown in Figure 35.

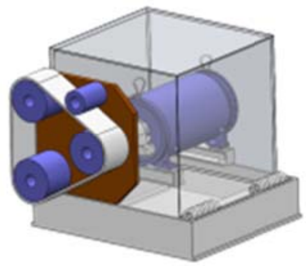

Figure 35: Twin Power caterpillar drive

- Single-sided caterpillar drive shown in Figure 36.

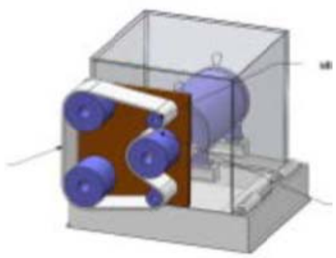

Figure 36: Single-sided caterpillar drive

Through this process, OPT and the responsive vendor developed a Pro/Con list to the development of the belt and drive configurations.

4.3.1.2 RACK

The other component required for the belt drive system is the rack which the belt drive will engage. Four options for the rack have been proposed:

- Stainless steel – Full length
- Stainless steel – Stroke length
- Stainless steel sliding rack
- Urethane rack

The following sections briefly describe the full length stainless steel and urethane racks.

4.3.1.3 STAINLESS STEEL-FULL LENGTH

The first option for the rack in our belt drive system is to use stainless steel fixed to the bearing rails of the spar with a length of approximately 20 t0 40m. The stainless steel is required for corrosion resistance in the salt water environment. The length is required for belt drive during typical operation (stroke + tidal adjustment), maintenance and survivability as required in the functional specification. In addition to the technical aspect of the concept development, OPT also included the corresponding cost information. The final full length rack design was found to be cost prohibitive.

4.3.1.4 URETHANE RACK

The final option for the PTO rack is a urethane rack. As described previously, the urethane rack development is extensive in time and cost. Therefore, further design and development must be considered if this option is selected. As for the design, the urethane rack would require either a full length (20 to 40m) rack or a shortened rack (5m to 15m) with transition mechanism for the PowerBuoy. The transition mechanism would be required to “clock” the belt drive during re-engagement with the rack when recovering from a survivability condition. The transition mechanism would consist of a short section of stainless steel rack in parallel with the urethane rack with a matching stainless steel pinion connected to the belt drive. The operation of this mechanism is as follows: As the float is raised from a survivability (submerged) position, the transition rack and pinion (stainless steel) would engage and absorb any impact effects. The pinion would then “clock” the belt drive with the rack and cause the belt to engage the urethane rack without any impact. Thus, the transition mechanism would prevent any severe impacts or mis-alignment of the urethane belt and rack which may have otherwise caused damage or failure of the PTO system.

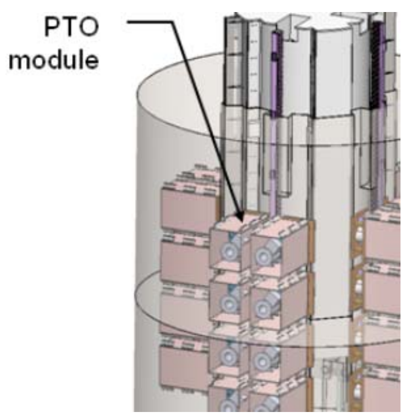

Figure 37: Urethane rack transition mechanism

4.3.2 ROTARY BEARINGS

Rotary bearings are required for the belt drive PTO option as a support for the sprockets in the drive. Therefore, initial life calculations were performed to size a bearing and mitigate any future design risk. The assumption for these calculations was:

- 2 bearings per sprocket

Therefore, using the standard rotary bearing life equation:

$$Life = \left(\frac{C}{P}\right)^{\frac{10}{3}} \times 10^6$$

Where P is the equivalent load, C is the dynamic load rating of the bearing, and the life is calculated in revolutions.

Using the criteria for a specific off the shelf bearing, the estimated life is ~12 years. Therefore, the risk to the project from a rotary bearing standpoint is relatively low at this point.

4.3.3 GIMBAL SYSTEM

Finally, a gimbal system for each PTO module is required due to the relative movement (tip/tilt) of the float with respect to the spar. Without a gimbal system, the alignment and engagement of the PTO belt drive to the rack would not be correct, leading to damage and possible failure of the system. Therefore, an initial design of the gimbal system using wire rope isolators has been completed. As shown in Figure 38, wire rope isolators mounted to the PTO module will allow the PTO to maintain correct engagement with the rack.

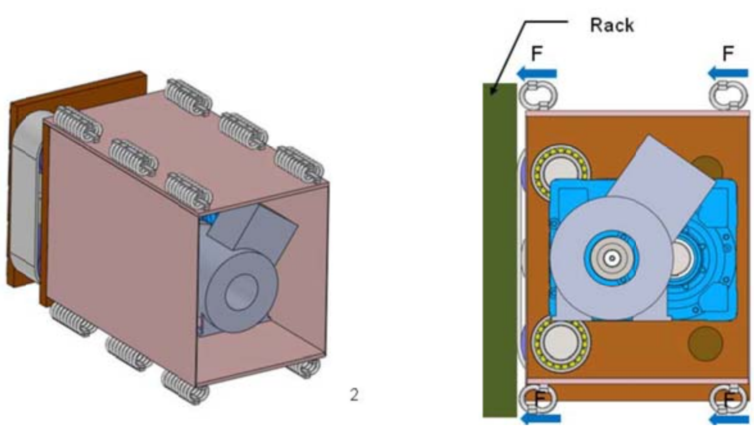

Figure 38: Wire rope isolator gimbal system

Additionally, compression springs or pneumatic isolators could also be used as an alternative to the wire rope isolators. Therefore, the gimbal system has low technical risk to the PB500 concept at this time.

4.3.4 BELT DRIVE PTO SYSTEM CONFIGURATIONS

4.3.4.1 BELT DRIVE PTO SYSTEM CONFIGURATION #1

Belt Drive PTO Configuration #1 employs a belt drive PTO mounted inside the float acting against a stainless steel rack mounted to the spar. The belt drive is mounted outside the float and connects to the generator assembly through rotary seals passing through the float skin. The belt drives stationed outside the float engaging the rack. Waves drive the relative linear motion between the spar and the float, and thus the relative linear motion created between the rack and the belt drive will convert the linear wave motion to rotational motion driving the generator.

(A) INFINITE RACK INPUT SYSTEM

The basic component configuration employed in the infinite rack input system are as follows:

Infinite Rack Input System

- 24 PTO's per buoy
  - Belt drive, gearbox, and generator
- Racks/6 PTO's each
  - Racks are mounted "back-to-back"
  - Racks are ~20 to 40 meters long

(B) BELT DRIVE PTO CONFIGURATION

- PTO systems are completely internal to the float
  - Belt drive fully immersed in sea water
  - Sealed rotary shaft into dry compartment
    - Input into parallel gearbox (and generator)

4.3.4.2 BELT DRIVE PTO SYSTEM CONFIGURATION #2

Belt Drive PTO Configuration #2, employs a wire rope input and adjustable sheave input system and a belt drive PTO mounted inside the float acting against a rack mounted inside the float. The wire rope is connected to a rack inside the float through two input rods. The belt drives are stationed inside the float engaging the rack. Waves drive the relative linear motion between the spar and the float, and thus the relative linear motion created between the rack and the belt drive will convert the linear wave motion to rotational motion driving the generator.

The basic component configuration employed in the belt drive PTO system are as follows:

- Allows for internal configuration of belt drive
  - Wire rope adjustment system
  - Adjustable rod lock system
- Trade rotary sealing requirements for linear sealing requirement
- Eliminates fouling/corrosion issues
- Reduces rack requirements
- Issues
  - Belt engagement length requires a longer rack due to overhead of PTO drives
    - Float height ~10m to 20m
    - Risk of significantly higher float weight

- Fixed input rod system may require very long input rod >30m

4.3.5 BELT DRIVE TIDAL COMPENSATION SYSTEMS

Tidal compensation is required for a PowerBuoy operating at sea subject to variations in the mean waterline due to raising and lowering tides. The tidal compensation system allows a PowerBuoy to adjust the free stroke of the input system as the mean waterline changes throughout the day due to tidal variations. The tidal variation map previously shown is provided for convenience in Figure 39.

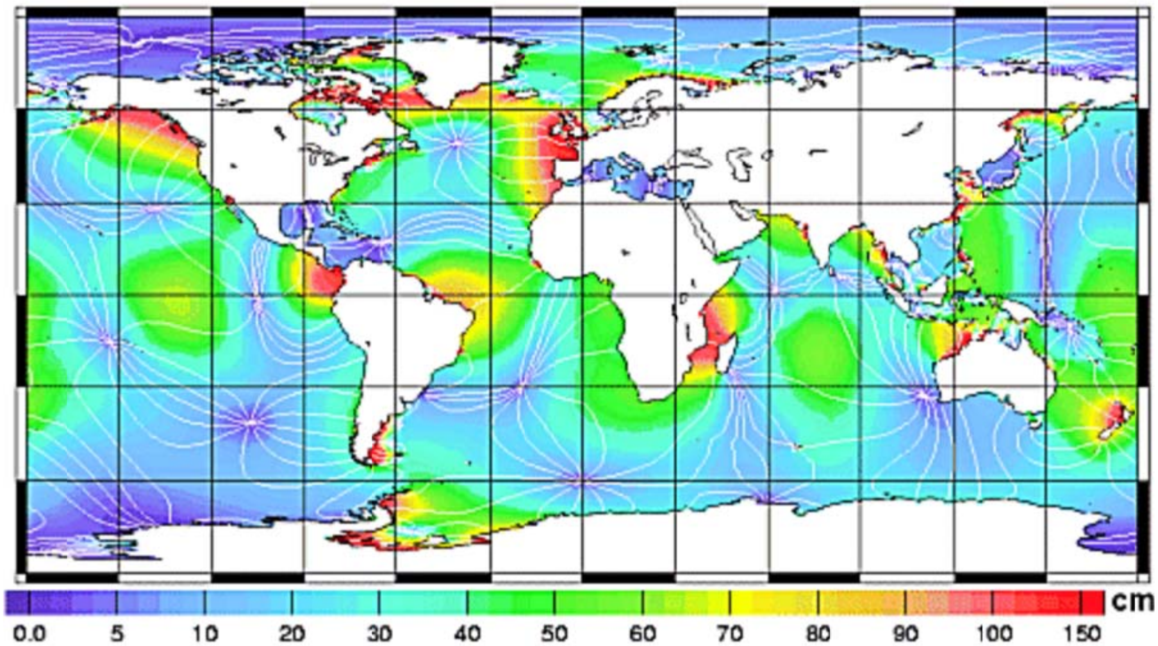

Figure 39: Map showing tidal variations across the globe. Red areas represent large variations in water level, purple areas represent zero or very low tidal variation. Image Credit: Legos/CNRS.

4.3.5.1 BELT PTO SYSTEM

The following section covers tidal compensation systems concepts intended for use on two types of Belt drive systems: Infinite rack Belt drive and External Sliding Rack.

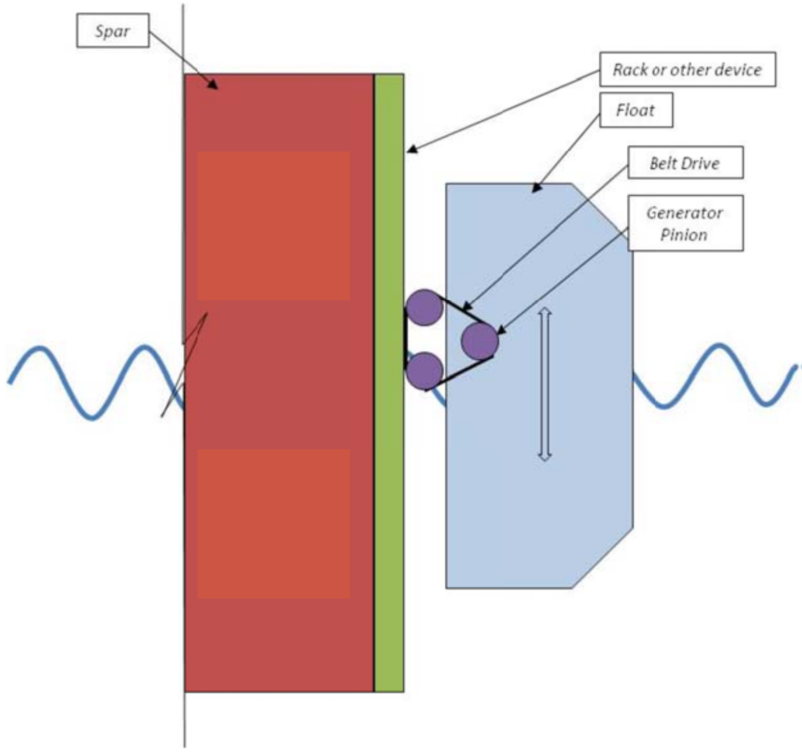

Figure 40: Belt Drive PTO System

A belt drive PTO system employs a belt drive mounted to the float acting on a rack mounted to the spar. The belt drive is located outside the spar and connected to the generator pinion inside the spar through a sealed shaft connection. Waves drive the float up and down moving the belt drive up and down the rack, generating power in the process. As the tide raises and lowers so does the mean position of the float; it is therefore necessary to employ a very long rack or adjust the mean position of the rack. The rack tidal adjustment device enables the mean position of the rack to adjust during operational loading of the PTO system.

Although the infinite rack belt drive and other options were considered, the external sliding rack belt drive PTO is discussed herein.

System Pros –

- Shorter Section of Rack Required
- Automated Tidal Adjustment (no control system required)

System Cons –

- Complex Hydraulic Circuitry
- Fatigue of Structure for Shocks
- Corrosion Resistant Requirement

During normal operation the rack is locked to the brake rail / rack guide through the caliper brake. The shock absorbers are mounted to the rack assembly and hydraulically connected to a pressure accumulator. The pressure accumulators are connected to the release port on the caliper brakes.

During an over travel condition a rogue wave forces the float beyond the normal operational travel limits. When the float travels beyond the normal operational travel limits the shock absorbers mounted on the rack impact the float hard stops and compress. The hydraulic fluid forced out of the shock absorbers pressurizes the accumulator. If the shock absorbers fully compress the caliper brakes are released. When the shock absorber pistons extend, the pressure in the accumulators is relieved and the caliper brakes re-engage locking the rack in position to the spar.

4.3.6 FLOAT SURVIVABILITY

Float survivability position will be implemented to avoid overstressing the buoy during storm conditions. The following actions will be taken to move the float to a survival position.

- Buoy control system enters locked state
    - velocity limiting brakes engage
- Float ballasting to achieve negative buoyancy must be started
- Tidal compensation would be engaged
    - Configuration #1 – Adjustable rod lock system
        - Internal sliding rack
    - Configuration #2 – Wire rope adjustment system
        - Caliper brake sheave lock
        - Hydraulic sheave lock
- Float would gradually sink to desired position
- Locking mechanism would secure float at survivability position
- Reverse procedure to return float to the operational locked state

4.3.7 FLOAT MAINTENANCE

Float maintenance would be similar to that for the Rack and Pinion PTO.

4.3.8 BELT DRIVE SYSTEM CONCLUSIONS

The following table summarizes the conclusions drawn from work on the Belt Drive System. Final system configuration recommendations are made at the end of this report.

Table 5: Rack & Pinion PTO System Conclusions

<table>
<tr><th>Component or System</th><th colspan="2">Conclusions</th></tr>
<tr><td rowspan="3">Belt Drive</td><td colspan="2">Drive design currently sized for1 MN to 4MN peak force for 24 PTO arrangement</td></tr>
<tr><td rowspan="2">Concern from manufacturer regarding alternating direction of belt drive</td><td><u>Recommend a clutch system for single direction drive</u></td></tr>
<tr><td>Require system testing to validate calculations</td></tr>
<tr><td>Rack</td><td colspan="2">multiple options that can be integrated into buoy</td></tr>
<tr><td rowspan="4">Belt</td><td colspan="2">Salt water has no significant effects on the Polychain belt</td></tr>
<tr><td colspan="2">single direction operation has shown no issues</td></tr>
<tr><td colspan="2">Reciprocating operation of the belt is implicitly included in the belt drive calculations but no supporting data is available</td></tr>
<tr><td colspan="2">Wear of belt/rack shows typical wear characteristics (qualitative) for short term testing</td></tr>
<tr><td>Tidal Adjustment</td><td colspan="2">Multiple concepts available</td></tr>
<tr><td>Survivability</td><td colspan="2">Multiple concepts available</td></tr>
<tr><td>Maintenance</td><td colspan="2">Multiple concepts available</td></tr>
<tr><td>Gimbal</td><td colspan="2">System using COTS wire rope isolators is feasible</td></tr>
<tr><td>Risk</td><td colspan="2">Some risk remains on the belt drive design that would require system level testing to evaluate</td></tr>
</table>

4.4 PTO Technology Configurations

For the four PTO configurations studied as described in the previous sections, an evaluation and a rating are given to each one of them, and final recommendations are made.

4.5 Comparison Matrix Summary

Table 6 lists the components investigated vs. each PTO system. A component receives a check mark if it is compatible with the PTO system in the column and an x if it is not compatible.

Table 6: Component Comparison Matrix (see Comparison Matrix Spreadsheet)

		RP#1	**RP#2**	**Belt #1**	**Belt#2**
		Adjustable input rod	Fixed input rod	External rack on spar	Internal rack in float
		Option?	Option?	Option?	Option?
Input Rod	Fixed Input Rod	✗	✓	✗	✓
	Wire Rope Adjustable Input Rod	✓	✗	✗	✓
Speed Increaser	Gearbox	✓	✓	✓	✓
	Beltbox	✓	✓	✓	✓
	Chain drive	✓	✓	✓	✓
Brakes	External linear brake	✓	✓	✓	✓
	Rotary spar to sheave	✓	✗	✗	✓
	Internal linear rod lock	✓	✓	✗	✓
	Internal rotary pinion caliper brake	✓	✓	✓	✓
Locking Mechanism	Latch	✓	✓	✓	✓
	Shear pin	✓	✓	✓	✓
Pinion	Vendor 1	✓	✓	✓	✓
	Vendor 2	✓	✓	✓	✓
Wire Rope	Vendor 1	✓	✗	✗	✓
	Vendor 2	✓	✗	✗	✓
Linear Seal	Vendor 1	✓	✓	✗	✓
	Vendor 2	✓	✓	✗	✓
	Vendor 3	✓	✓	✗	✓
Gimbal system	Wire Rope Isolators	✗	✓	✓	✗
	Compression Springs	✗	✓	✓	✗

4.6 Recommended PTO Component Options

Based on the above trade studies, design assessments, concept development and analyses, our recommendation for a power take off configuration would be based on configuration 2.