Triton
Survival Down-Selection

DE-FOA-0001310: Next Generation Marine Energy Systems - Durability & Survivability
Award Number: DE-EE0007346
Principle Investigator: Tim Mundon. Ph.D.
Submission Date: September 28, 2018

Document contributes towards M2, D4

Following drawings provided:
<D8_Design_Drawings_Baseline+NonselectedApproaches.pdf>
<D8_Design_Drawings_Final_Design.pdf>

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2. Summary / Scope

This report summarizes the ideation and down-selection of an advanced survival configuration for the Triton WEC. Building on the baseline design, OPI identified and analyzed a number of configurations that allow for a combination of high Annual Energy Production and high survivability. Each of these concepts were sufficiently vetted through numerical analysis and a design review in order to develop sufficient confidence as to their effect on the design. As part of the design review meeting, we identified the critical failure points associated with each concept, and used this to help to guide the subsequent evaluation and study. The most promising survival strategy, which involves submerging the WEC below the water surface, was down-selected for detailed numerical analysis and physical model testing.

References

D8_Design_Drawings_Package.pdf

3. Background and objective

Overview

The ultimate goal for this project is to identify a survival strategy for the Triton WEC, that will allow for a reduction in system peak structural stresses by 50% compared to those experienced when the system is in an operational configuration, while simultaneously allowing for a reduction of capital cost by 15% due to the elimination of overdesign to account for uncertainty.

The intent is to develop methods to mitigate the loads imposed on the structure by extreme waves. However, understanding how the device will react in these conditions is a key part of this project. Nonlinearities imposed by extreme waves on WEC's makes numerical model prediction of design loads difficult and thus we will also be working to identify the most reliable and computationally efficient methods.

As it is prudent to initially identify a number of feasible strategies before subjecting them to investigation, we have identified three possible survival strategies for the Triton WEC that may be good options to allow cost-effective load reduction in extreme seas. Numerical modeling will be used to comparatively evaluate these strategies and decide upon an optimal survival strategy,

This report presents the evaluation of the practicality of these survival concepts and identify improvements or concerns that should be addressed for each, including potentially identifying other strategies, or variants to these strategies that might provide better performance. In the evaluation, we will be using tendon peak load and strategy cost as the primary metrics for comparing the performance of competing alternatives.

Design wave conditions

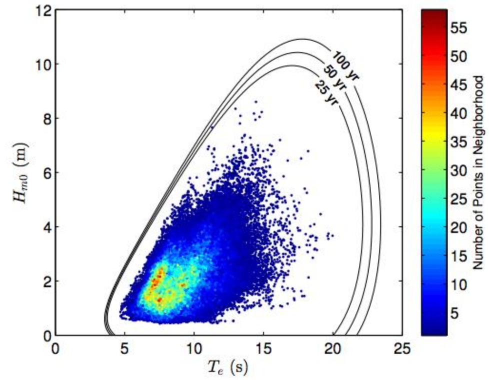

Figure 1. *Extreme wave contours for CDIP128/NDBC 46212 (2004-2012).*

This work will use Humboldt Bay in California as the reference location for which to evaluate the performance of a survival strategy. Figure 1 shows the extreme wave contours for a representative climate at this location[1]. Five data points on the 50-year contour, summarized in Table 1, were selected as design wave conditions for evaluating the baseline design. As the Bretschneider model was reported as being the best fit to the spectra[1], this representation was used in the analysis.

Table 1. *Design wave conditions sampled from the 50-yr contour in Figure 1.*

Label	T_e [s]	H_s [m]
EC1	5.5	2.9
EC2	7.6	4.9
EC3	10.1	6.8
EC4	13.0	8.7
EC5	15.1	9.6

4. Triton baseline system architecture

The Triton WEC architecture is described in the attached drawings and shown in Figure 2, with structural parameters detailed in Table 2. It is a two-body, multi-modal point absorber consisting of a surface float connected by three flexible tendons to a submerged reaction structure ('heave plate'). Mechanical energy is extracted from the environment in the form of wave-induced heave, pitch, roll, surge, and sway motion

[1] Dallman, A. R., & Neary, V. S. (2014). *Characterization of US Wave Energy Converter (WEC) Test Sites: A Catalogue of Met-Ocean Data* (No. SAND2014-18206). Sandia National Laboratories (SNL-NM), Albuquerque, NM (United States).

of the surface float through its reaction against the heave plate. The resulting tension variation in each tendon is transmitted to an independent linear powertrain, consisting of a linear hydraulic load transfer mechanism and linear generator, housed inside the surface float. This system was tested at 1:20 scale in a physical model as part of the DoE's Wave Energy Prize competition, providing data that has enabled numerical model validation in operational and some limited extreme conditions.

Table 2. *Key system physical dimensions for <u>Baseline</u> system*

Parameter	Value
System displacement	1950T
Draft	30-83m
Surface Float LOA	30m
Surface Float Beam	23m
Surface Float Mass	250T
Heave Plate Mass	1700T
Heave Plate Diameter	30m
Heave Plate Height	6m

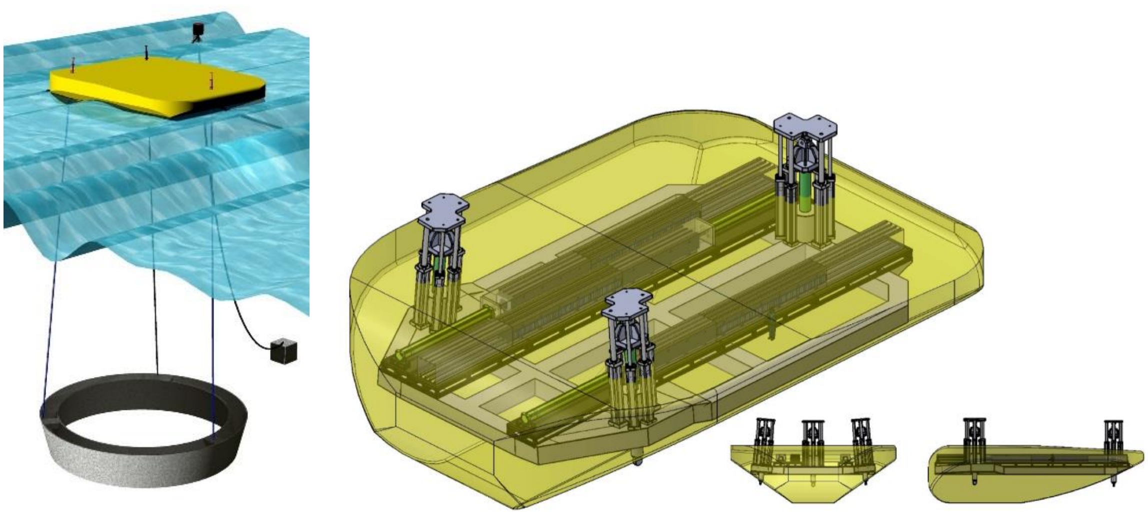

Figure 2. *Isometric conceptual view of the full system (left) and Triton hull (right) with load transfer units. More detailed drawings provided in* Appendix B*: Baseline Drawings*

Triton Hull

Note that the physical masses provided in Table 2 are <u>derived from preliminary numerical modeling work that was completed to determine a system that would be survivable in the 1:50 year contour</u> to establish a starting point for this work (and is described below). It is understood that these may not be realizable and it is therefore the purpose of this project to identify a better survival strategy that will allow a more realistic and better performing system configuration to be used.

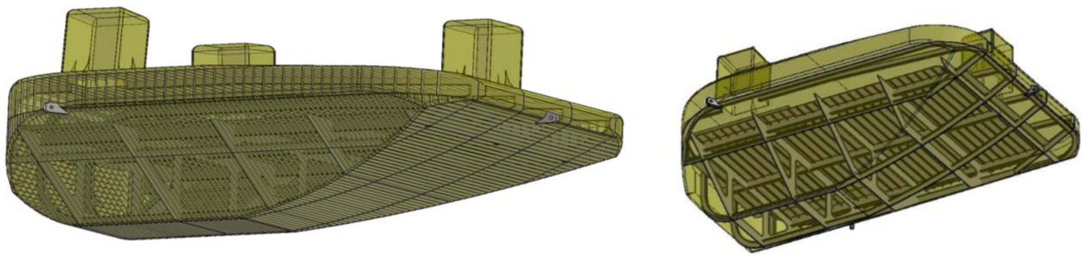

Figure 3. *Triton surface float hull structural arrangement.*

A preliminary arrangement for the hull superstructure is provided in the attached drawings and shown in Figure 3. This shows our starting point for the frames and stiffeners that might be required. Assuming a steel hull, the overall structural mass could be in the range of 260-450T depending upon the plate thicknesses used. At present for the upper estimate, we are assuming 20/18mm for the hull plate and main frames respectively, while we have considered 9/12mm thickness for as the lower mass estimate.

Ultimately is hoped that a finished and fully equipped WEC float mass in the region of 450T can be achieved. It is estimated that the PTO mass maybe in the order of 150T, leaving around 300T for the structural mass of the hull and other subsystems.

Triton linear drivetrain

The system drivetrain is a key factor managing the system response in extreme waves. In particular the drivetrain provides a reactive force to the WEC, which can in the case of operational waves be used to tune the device to develop maximum power, or in the case of large waves, detune the device so the response is minimized. However in the case of very large waves, the ability of the PTO to manage extreme loads will be limited.

The Triton consists of three independent drivetrains, one per tendon, each with its own generator and all are electrically aggregated for export. Force and velocity from the movement of the WEC relative to the heave plate is applied as a tensile force to input hydraulic input cylinders that are mounted in-line with each tendon. This is then transferred through a hydrostatic hydraulic connection to output hydraulic cylinders, which directly drive a linear generator. This hybrid approach combines the controllability advantages of direct drive systems with the flexibility and power conditioning of hydraulic systems, while providing efficiency in advance of any other conventional WEC solution.

The drivetrain can aid in load management through its ability to modify its apparent spring and damping and most importantly to dynamically add additional (hydraulic) energy dissipation to provide greatly increased damping values. This energy is dissipated through heating of the hydraulic fluid, via a valve network, which can be dissipated to the ocean through seawater cooling.

Based on drivetrain development work through a Wave Energy Scotland project, the range of achievable spring rates is 300-1500 kN/m. The maximum damping coefficient is 1200 kN/m/s (peak generator damping) to 3000 kN/m/s (with supplemental hydraulic dissipation).

Numerical Representation

For all calculations presented in this report, the device was numerically modeled at full-scale using a time-domain simulation (OrcaFlex). The surface float was represented using linearized hydrodynamic properties produced from a panel method solver (NEMOH). The PTO was represented by a linear spring-damper characteristic, which based on our previous work, has been found to be a good approximation of the Triton drivetrain. The heave plate hydrodynamic coefficients in all modes of motion were informed from forced oscillation experiments performed in OPI's laboratory, and these coefficients have been validated with CFD by NREL. Numerical output has been validated and tuned by comparison with 1:20 scale physical model test results completed as part of the DoE Wave Energy Prize.

The mooring was represented using linear springs that act on the surface float, where the nominal pretension stiffness are equivalent to the realistic three-point mooring pictured in Figure 2. This was used in order to evaluate the effect of changing the mooring stiffness, which is one of the survival strategies described in Section 7.

It should be emphasized that these initial simulations employ linearized hydrodynamic assumptions, which tend to weaken for large waves, although it is expected that general performance trends should be accurate. In light of this, over the course of the project, models of increasing complexity (i.e. weakly nonlinear mid-fidelity models and full CFD) will be developed to hone in on more precise engineering design values.

Survivability considerations

The maximum stroke for each tendon's drivetrain is 3m, although it may be possible to extend this to 4m. Work so far has indicated that increasing the stroke will directly result in an ability to capture more energy in higher H_S as well as provide a reduced risk of slack tendon and end-stop events. However, for the purposes of this project, we will assume a 3m stroke with a mechanical end-stop condition imposed in the last 0.25m on either end of travel. This mechanical limitation is provided by a combination of soft hydraulic "cushion stops" in the primary hydraulic cylinder as well as elastomeric 'hard' mechanical stops, as shown in Figure 4. This offers increased resistance to motion when nearing the extremes of travel and therefore enhanced mechanical protection. It is intended that the soft end stops may be engaged infrequently during operation, while the hard end stops would be reserved as a safety event only.

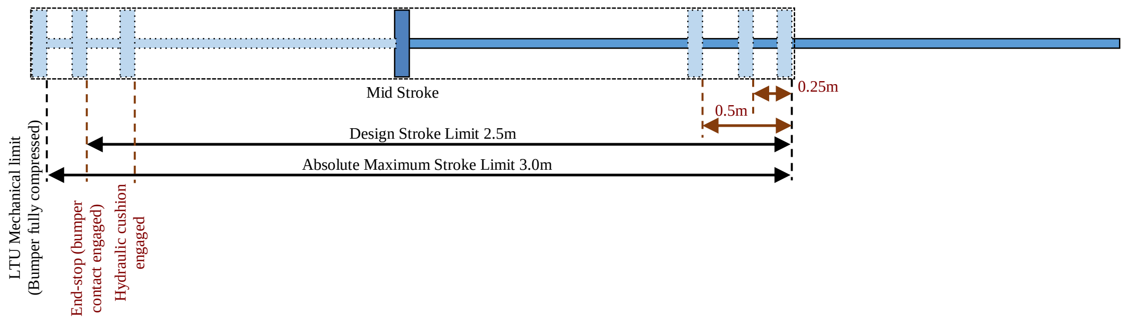

Figure 4. *Input cylinder stroke diagram*

For numerical modeling efforts, these end-stops were implemented as a mean-load offset spring as shown in Figure 5. Once the end-stop is engaged, the stiffness gradually ramps up and then asymptotes to infinity at the hard limits of travel.

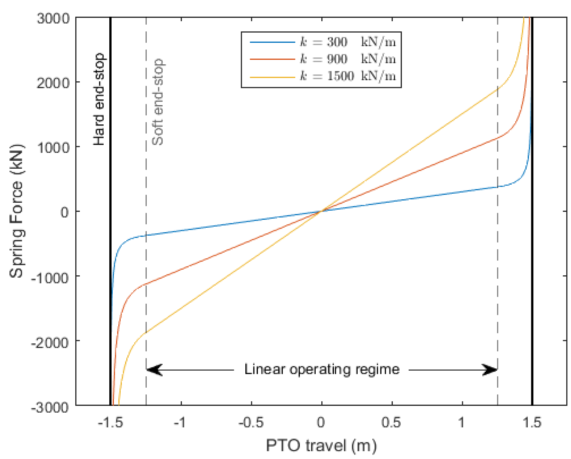

Figure 5. *Return spring profile for different spring rates*

System Analysis: Evaluation Criteria

The two criteria that must met for a survivable Triton design are minimal (ideally zero) *end-stop events* and minimal (ideally zero) *slack events*, both of which can potentially damage key components. An example simulation of an incorrectly configured system in extreme wave condition *EC4,* is shown in Figure 6 . As shown in the right tendon tension plot, slack events can lead to snap loads many times greater than typical fluctuations.

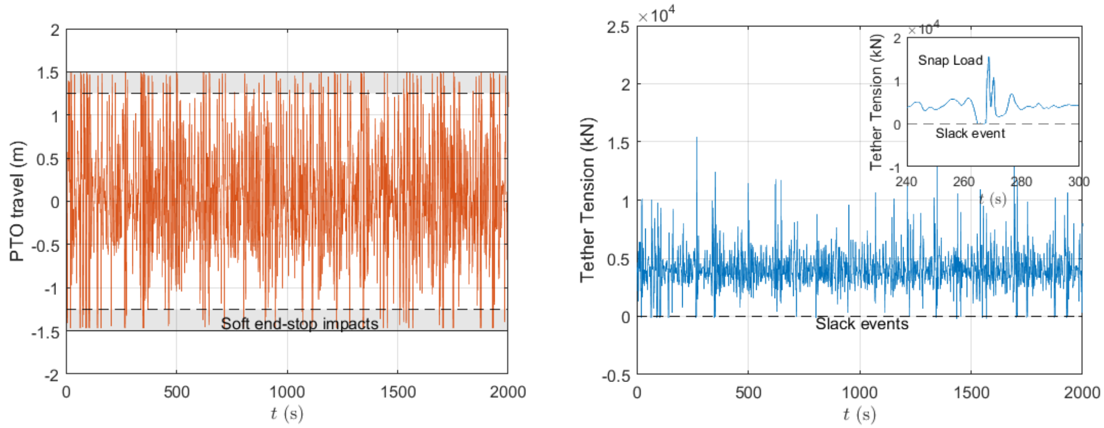

Figure 6. *Sample time series of tendon travel (left). Sample time series of tendon tension (right), where the inset represents a slack-snap event.*

Baseline system description

The baseline concept defined here is completed solely to enable quantification of the impact of design changes made over the course of this project. Our original intent was to define the baseline design as one which used ONLY a combination of an increased heave plate mass (i.e. to eliminate slack events) and an appropriately tuned PTO damping/stiffness to achieve survivability. Despite significant modeling effort, we have concluded that a practical solution is not achievable using only these two variables (with the stroke permitted in our current drivetrain design). However, we subsequently found that increasing tendon (material) elasticity in addition to these two variables can be used to further de-tune the system and enable a survivable design. Appendix A shows further details of this strategy. Unfortunately, while this resulted in a survivable design, it had an impractically low AEP.

As a result of the above analysis, we have therefore chosen to incorporate a rudimentary survival strategy into the baseline design. This strategy distributes loads to parallel, high elasticity lines in extreme wave conditions. These high elasticity lines are only engaged during extreme conditions. The additional lines are connected between the surface float and the heave plate and remain slack under normal operation. Under extreme waves, the tension on the additional lines is increased using a winch mechanism to support the full load of the heave plate and thus the normal tendons will become slack. These 'survival lines' will have a high elasticity, and will therefore absorb energy through stretching (see Figure 20) to help eliminate slack events. This simple strategy thus allows the system to produce a practical AEP and can be used for comparative evaluation. This basic survival strategy would still require a substantial amount of overdesign (i.e., increased capital cost) in the system and the mass properties (presented above in table Table 2) are highly aggressive. In addition, this strategy requires permanently installed winches that would be expensive and a line elasticity that may be difficult to achieve (~30%). The cost of these system modifications has been included in the revised LCOE and CAPEX estimates reported in Deliverable 3.

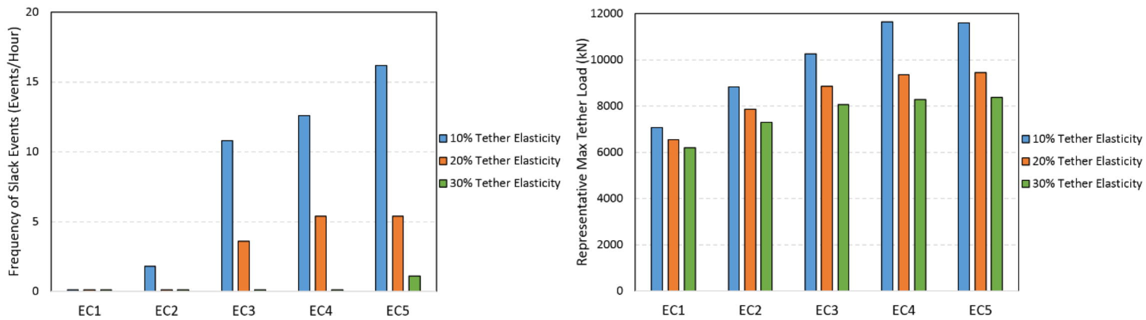

Figure 7. *Baseline survival configuration (i.e. heave plate winched up on elastic lines) parameters as a function of tendon elasticity for the front tendon. Slack event frequency (left). Baseline representative max tendon load (right).*

Figure 7 shows numerical results for this baseline system in survival mode, i.e. the heave plate is winched up on highly elastic lines, thus decoupling the tendons from the PTO, with the winch loads directly transferred to the float structure. As shown, for the 250T/1700T system a ~30% survival line elasticity or perhaps a little larger (possibly combined with a further de-tuning method summarized in Section 10), is needed to eliminate slack events. The representative maximum tendon load on the front tendon, defined here as the average of the 20 largest peaks (in a 2000s simulation), is approximately 8200kN.

5. Preliminary survival considerations

As stated, a goal of this project is to allow reduction of heave plate weight through the introduction of an improved survival strategy (relative to the Baseline). An aspirational system comprises a 450T float and 1500T heave plate. This configuration has a lighter and more manageable heave plate weight compared to the Baseline, and the aim is that it will be enabled through a suitable survival strategy. The analysis below however follows from a 750T/1200T configuration (i.e. the 1:20 scale Wave Energy Prize system) since the numerical model has been validated by the WEP tank tests. The overarching feasibility of different survival strategies should be insensitive to this choice.

Work in Task 1.1 identified the need to incorporate a rudimentary survival strategy in the baseline system in order to make it viable in large waves, i.e. to avoid slack events and minimize end-stop events. In this baseline strategy, extreme loads are distributed into parallel, high-elasticity lines during extreme wave conditions. These high elasticity lines are only engaged (using a winch mechanism) during extreme conditions.

During initial project conception, we identified three additional survival strategies:

1. Hard-mating the heave plate and surface float (with winches):
2. Distributing tension across lifting lines (by winch or by ratcheting lines)
3. Lifting heave plate close to surface float (with winches), without mating the two

Intial discussion with naval architects Glosten on the feasibility of these strategies concluded that strategies 1 and 3 would be impractical due to a number of reasons, In particular, it would be preferable to avoid the use of permanently installed winches given their probable size and cost. While winches will be used for installation and recovery, relocating them from to a (dedicated) deployment vessel for installation, was more likely to be cost effective. Secondly, while the idea of mating the heave plate and float together as a survival mechanism was considered at length, it was ultimately concluded that it would involve too much cost and risk for an automated mating operation in anything other than mild waves. Given that such a strategy would need to be employed once wave height passed a certain (large) threshold, it would become a difficult and risky operation.

With these concerns in mind, alternative survival strategies were developed to replace 1 & 3 above. These are described below.

1. Multiple Auxiliary Lines *(Variant of #2 above)*
2. Increased Mooring Tension
3. Submerged Surface Float

6. Multiple auxiliary lines

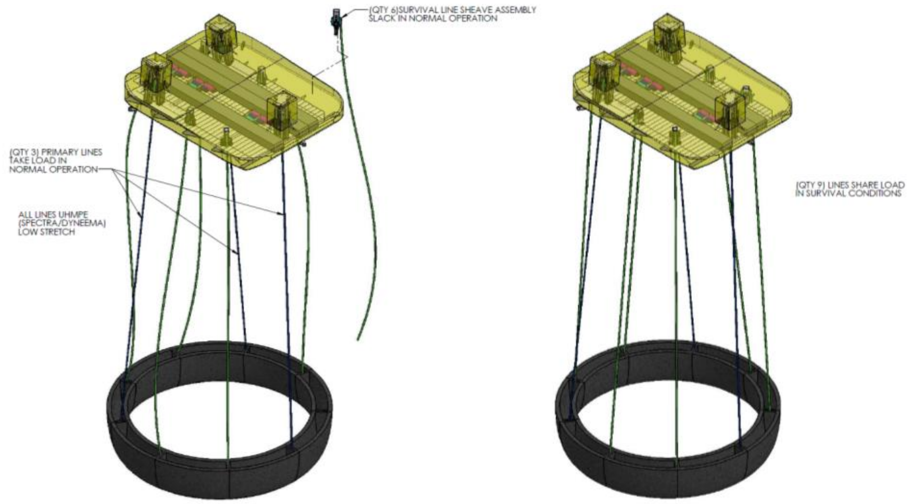

Figure 8. *(Left) System shown with six auxiliary tendons that remain slack in normal operation. (Right) System shown in survival configuration with load shared between 9 total tendons.*

Strategy

In addition to the three operational tendons that connect to the PTO, six auxiliary lines of moderate elasticity connect the heave plate to the surface float structure. These lines are slack in normal operation, but tensioned in survival mode such that the heave plate load is spread equally between both the 6 'survival' lines and the three normal load lines.

Importantly, in this strategy, we allow slack events to happen, but with the knowledge that any snap load is distributed more evenly between the tendons and transferred more predictably to the float. Lower line loads would thus protect the main load lines and the survival lines could be sized appropriately. The survival line elasticity would be moderately high, such as nylon or polyester (~10% extension in working regime). As the main load lines would be much stiffer than the survival lines, the drivetrain return spring would be set to provide a compatible elasticity – which would actually allow power generation during this time.

To repeat: In this arrangement, we are not eliminating slack events, but rather making them far more predictable and of a much lower and more manageable magnitude. The major reason slack loads are to be avoided is that they typically result in an <u>unpredictable</u> snap loads. This unpredictability means the structural design must contain substantial safety factors, driving the cost and risk up significantly. However, the predictability of the loads can be seen to be roughly proportional to the number of lines supporting the mass. By using 9 lines, we can therefore significantly reduce the unpredictability of snap load magnitude. Furthermore, loads are distributed much more evenly across lines and across the hull and heave plate structure, thus becoming far more manageable. An additional benefit is that this survival mode can be engaged and disengaged very quickly.

Implementation

Given the drivetrain's 3m stroke before the engagement of end-stops (+/- 1.5m about the mean position), these additional lines will have approximately 2m of slack in operating conditions. Once it is determined that the system needs to enter a survival configuration, the slack will be removed from the six additional lines to enable the system to have nine lines sharing loads between the surface float and heave plate. In the event of extreme climate, survival lines would be tensioned by way of a hydraulically or electrically actuated cam mechanism. A cam would be advantageous as it allows a variable mechanical advantage to be provided; with it increasing as the line is pulled in, allowing a smaller actuator. Figure 9, shows one possible method using two cylinders, although equally a rotary hydraulic or electric actuator could be used. It may be possible to find a mechanical arrangement where a single cylinder can be used.

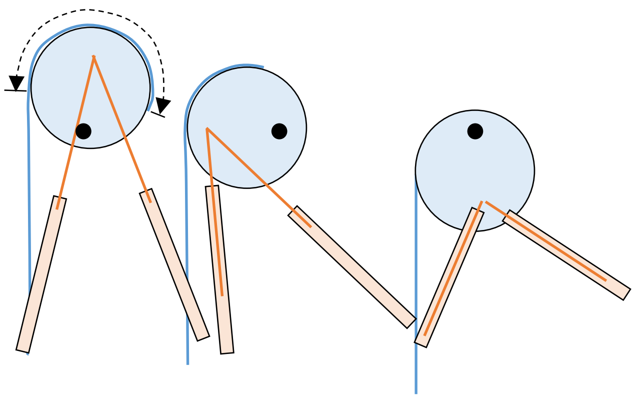

Figure 9. *Possible cam implementation with two hydraulic cylinders. Cam diameter and piston stroke would be >1m for a ~1.5m length of line taken up*

Another method would be to run the line over a sheave and terminate it on the deck. The sheave would be mounted on a shaft that can be raised or lowered by hydraulic cylinders to remove the slack. The hydraulic cylinders provide a reduction in line length of two times the cylinder stroke. Using two cylinders of approximately 1m in stroke, as shown in Figure 2, a bore size of 12 inches would be suitable given a standard pressure rating of ~3000 psi (assuming an extreme load of twice the heave plate weight).

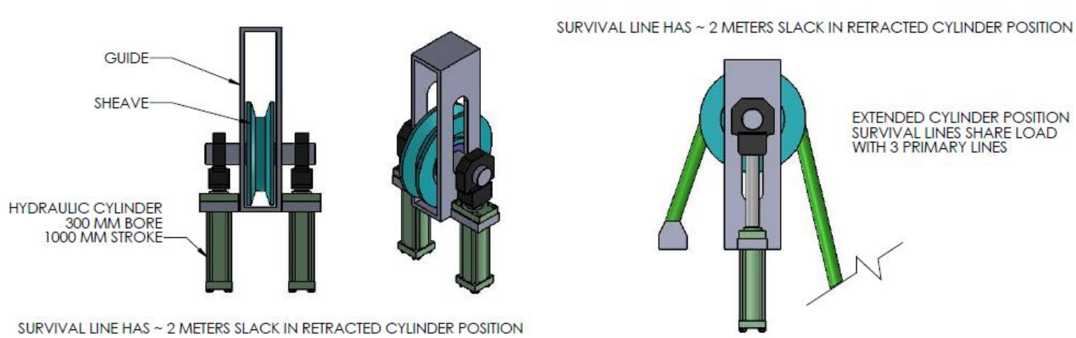

Figure 10. *Potential sheave/cylinder arrangement.*

Preliminary feasibility

A detailed performance comparison with the other strategies is shown in Section 9.

A set of basic numerical simulations were run in Orcaflex at the five selected 50-year wave conditions with different drivetrain settings. Six auxiliary lines were added to the numerical model connecting the heave plate to the hull structure. The auxiliary line elasticity was set to be equal to the drivetrain stiffness. Specifically, simulations were conducted with drivetrain stiffness = 300, 900, or 1500 kN/m, corresponding to a matching auxiliary tendon elasticity of 20%, 8%, or 5%.

It was found that using auxiliary tendons encourages significantly more slack events (due to the reduced mean static load on each line). However, the additional lines result in lower peak loads and also allow management of the drivetrain stroke, thus preventing end-stop impacts. Overall, this strategy appears to be effective and furthermore allows power capture even when the survival lines are engaged. Compared to this baseline, which demonstrated representative peak loads of 8200kN, this strategy yielded peak loads of 6000kN, which is roughly commensurate with the heave plate weight reduction, enabled by the load sharing. Optimizing the auxiliary line stiffness will likely enable further reduction in peak loads. A slightly stiffer survival line may be more effective because the 3 primary tendons are connected to dampers and therefore carry more load; this may be examined in the future.

7. Mooring line pre-tensioning

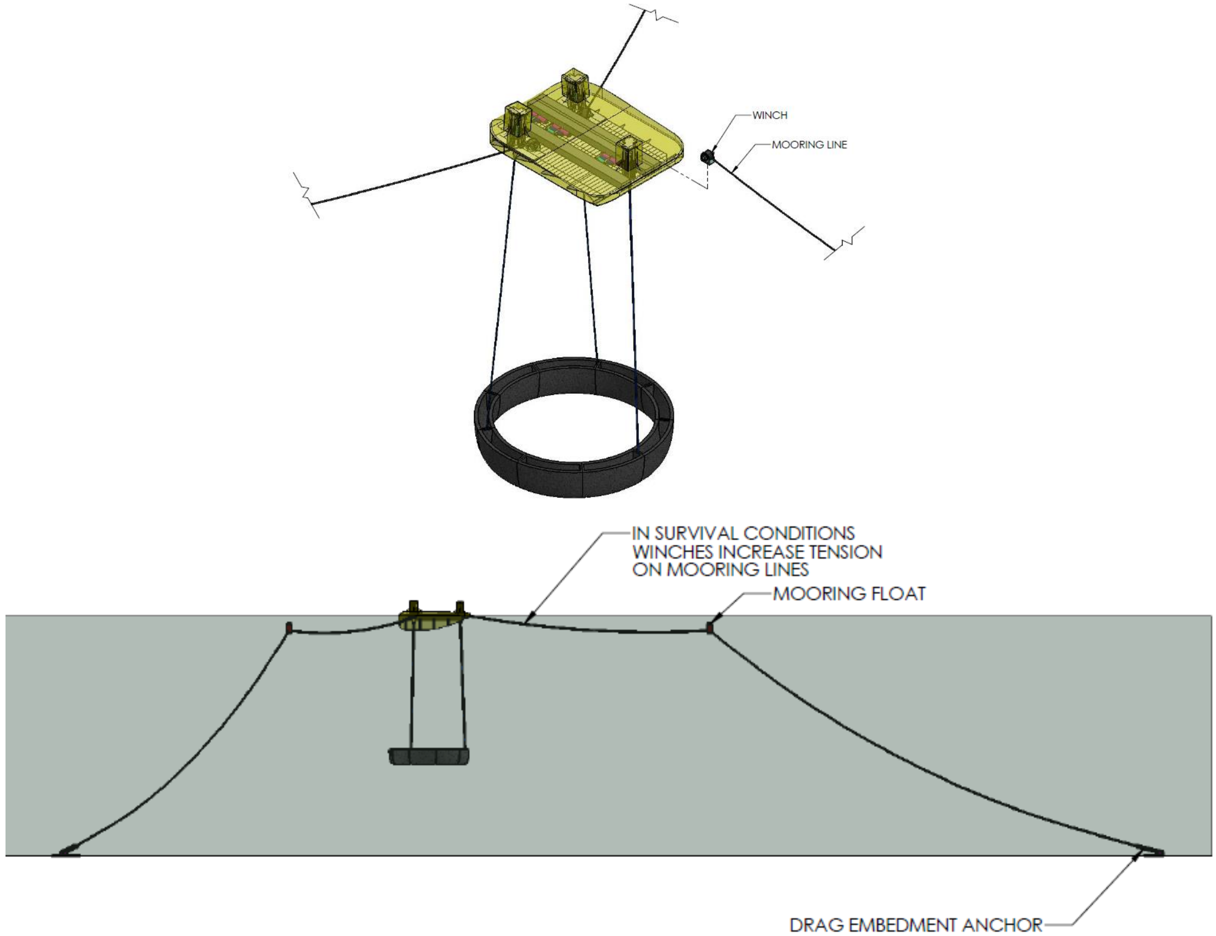

Figure 11. *(Top) System shown with mooring lines running to winches aboard surface float. (Bottom) Mooring arrangement demonstrating increased tension on mooring lines.*

Strategy

For operational waves, Triton's mooring system is designed to have large horizontal compliance, thus allowing for power capture from the relative surging motion between the surface float and the (less mobile) heave plate. This soft mooring is beneficial in that it allows a lighter, less expensive anchoring solution. Work so far has indicated that tendon slack events can be closely correlated with excessive surge motion of the surface float in large waves. This is somewhat intuitive, as we know that as the system surges, it couples to pitch motion (resulting in energy capture). This can be understood by considering the sequence of events in a single large wave; initially, the surface float can be seen to surge backwards, resulting in the heave plate pitching up. When the wave then causes the float to surge forward, the heave plate, due to its larger inertia and drag, restores more slowly than the float, resulting in a slack tendon events (Figure 12). This effect has been seen in both numerical and physical models when the waves are sufficiently large and typically affects the front tendon the most, although slack events have been observed in rear tendons as well.

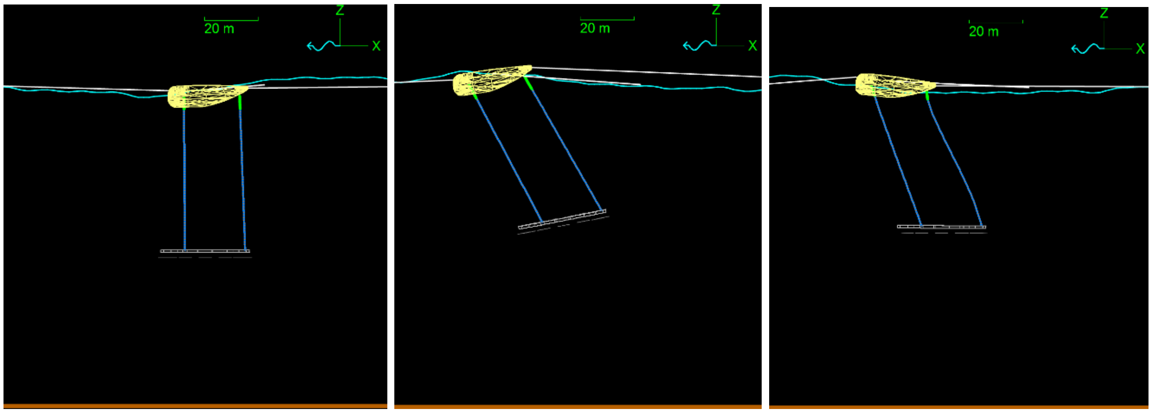

Figure 12. *Time-lapse images of the system experiencing a large surge excursion during a large wave event. As the surface float surges backward and then forward, a slack event initiates in the front tendon (shown in the third frame).*

When severe conditions are expected, the proposed strategy is to significantly increase the tension on the mooring lines with winches. The intent is to reduce the surge and hence pitch motion of the system, which are the primary contributors to the slack events. The 'cost' of this method is that the anchors will need to be significantly oversized relative to the baseline, and designed for some uplift. Numerical modeling will be conducted to determine the extent of the required anchor solution and gauge the associated cost. This modeling will also determine the effect of mooring line tensions on the system behavior and whether this would be an effective survival strategy within the entire 1:50 contour.

Implementation

This strategy would require appropriately sized winches (one for each of the three mooring lines) as well as high holding power anchors that could provide a suitable holding force.

Preliminary feasibility

A detailed performance comparison with the other strategies is shown in Section 9.

Preliminary numerical simulations were performed for three different mooring configurations - 2.5x, 5x, and 10x the baseline mooring pretension and stiffness, which are denoted as "L", "M" and "H".

It was found that while increasing the mooring stiffness does reduce slack event frequency, as expected, it appears that an extremely stiff mooring (~20x baseline) may be required to completely eliminate them. According to trends in the preliminary numerical model simulations, this would require mooring forces in the order of 700T (nearly an order of magnitude higher than for the other strategies, ~80T).

Though slack events can be mitigated using this strategy by restricting surge motion of the float, end-stop impacts appear more difficult to manage (as the mooring does not restrict heave motion of the float), unless the drivetrain is modified to allow additional spring and damping.

8. Sinking the surface float using ballast

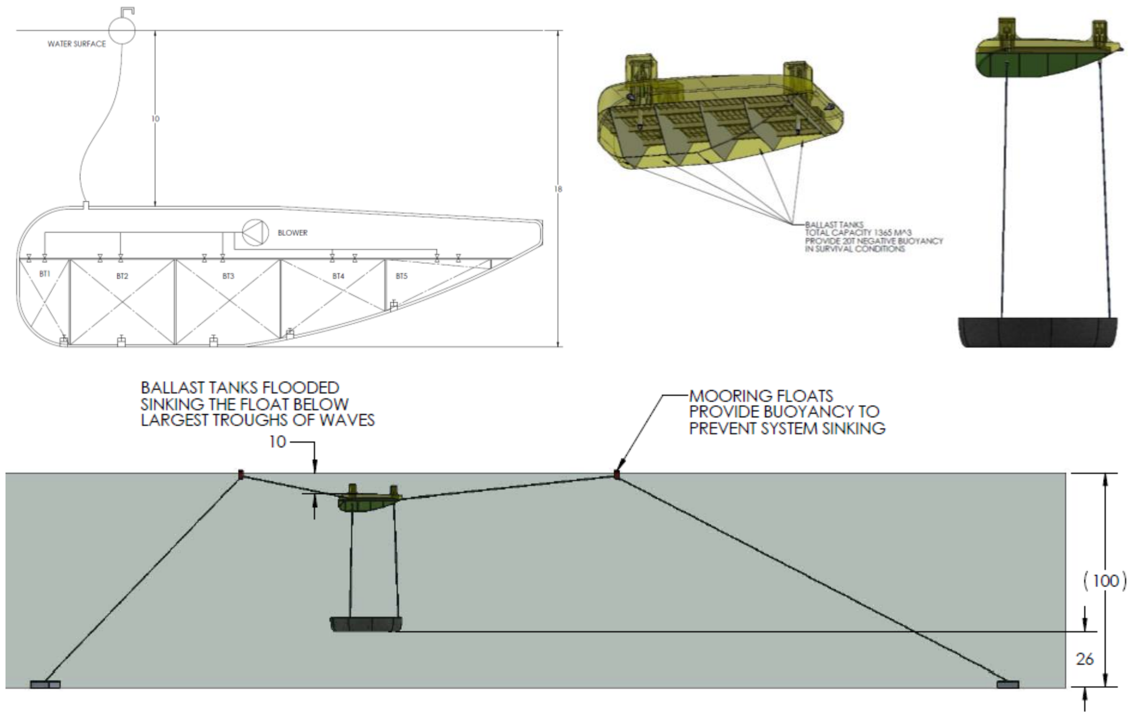

Figure 13. *(Top left) Preliminary schematic of ballast tanks and basic piping and instrument arrangement. (Top right) Location of ballast tanks in surface float. (Bottom) System shown with surface float sunk below the surface.*

Strategy

The third survival concept is to decrease the wave excitation forces on the surface float by sinking it below the surface. Moving below the surface will significantly reduce the dynamic loads on the float and hence the tendons and drivetrain. This strategy can be implemented by the addition of water ballast, giving the entire system a few 10's of tonnes of negative buoyancy. A rough position of 10m below the water surface was initially proposed; however some more in depth analysis is needed to determine an effective submergence depth. Oversized mooring floats, or a (small) separately deployed float, can be used to maintain the submerged depth. Surprisingly, this solution requires significantly less adaptive hardware than the other two strategies and has the potential to be the simplest and lowest cost option. Sealing the float from water ingress would be somewhat straightforward using bulkheads and common marine engineering techniques. The risks associated with this approach, will, of course, need to be thoroughly evaluated.

Implementation
The displaced volume of the fully submerged float is $3260m^3$. A ballast mass of 1390 tonnes is required to achieve a negative buoyancy of 20 tonnes. A ballast water volume of $1365m^3$ will achieve the target buoyancy. This can be split up into multiple ballast tanks that fit the scheme of the structural hull arrangement and the dry volume required for the drivetrain and generator equipment. The three mooring floats will provide enough positive buoyancy to keep the system at a prescribed depth and can be designed to impart limited wave forces. A separately deployed float could also be utilized to provide the required positive buoyancy.

To sink the float, flood ports will be opened at the bottom of the ballast tanks while vents at the top of the ballast tanks will be vented to atmosphere to allow the air volume to escape. Once the ballast tanks are full of water the air vents will be closed while the flood ports can be left open. Once the waves return to the normal operating conditions the float will be brought back to the surface by providing air, either from a compressed on-board source or an air-line attached to a float at the surface. This pressure must be greater than the hydrostatic pressure of 180 kPa at the flood ports at 18 meter depth. An onboard compressed air tank may need to be large, but can be maintained by a small air compressor as the cycle time for entering and leaving survival mode is not expected to be less than 24h, however this will need to be factored into the feasibility of this strategy.

Preliminary feasibility
A detailed performance comparison with the other strategies is shown in Section 9.

Preliminary numerical simulations were performed for a 10m float submergence depth with 20T of net negative buoyancy supported by the mooring. Further analysis will be conducted for different depths, though initial results suggest that 10m is promising. This configuration results in zero hard end-stop impacts and nearly zero slack events (limited events during *EC5* only). Furthermore, the characteristic peak tendon load does not exceed 6000 kN, i.e. ~1.5x the mean load.

Practical considerations of this survival strategy include:

1) A need to avoid heave plate collisions with sea floor (challenging for shallower locations). Due to the large system inertia, the system oscillated vertically with fairly large amplitude (up to ~15m) and thus there is a need to make sure the mooring (or surface float volume) has sufficient vertical stiffness to avoid disturbing the sea floor.
2) Another potential downside is the amount of time needed to perform the sinking/floating operations. This is in contrast to the auxiliary tendon strategy, which can potentially adapt in real time. Further analysis is needed to look at how frequently the survival mode needs to be engaged.

9. Numerical comparison of survival strategies

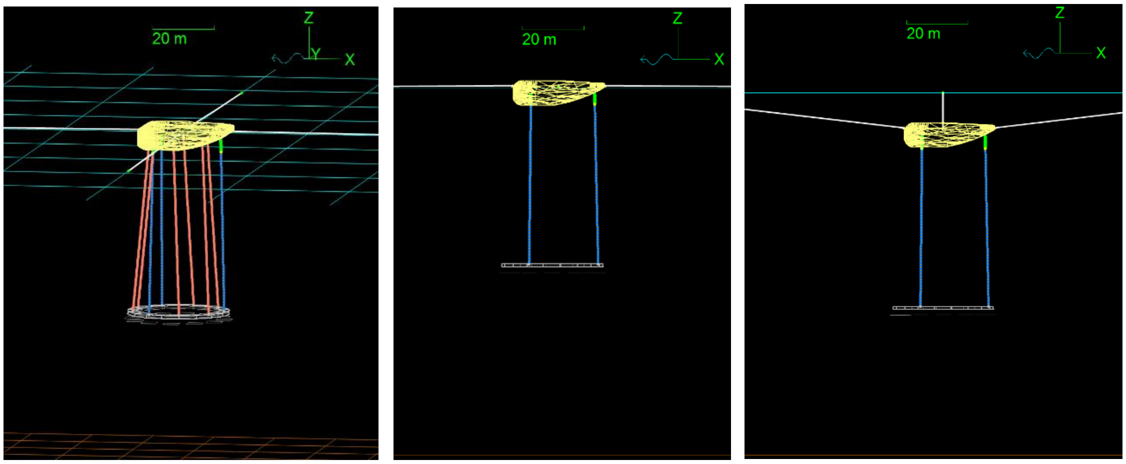

Figure 14. *Orcaflex models of the three identified survival concepts*

We identified four relevant metrics that allow a generalized comparison between the survival strategies with specific focus on system reliability and structural loads (note that for the moment, the simulations are currently performed only at 0 degrees with long crested waves, so the front tendon will be representative of the worst case):

1. ***Frequency of slack events*****:** i.e. slack events/hour, using the front tendon as a proxy (which is typically most severely affected).
2. ***Representative maximum tendon load*****:** the average of the 20 largest tension peaks in the front tendon.
3. ***Maximum drivetrain travel*****:** the peak-to-peak stroke of the front tendon.
4. ***Representative maximum mooring load*****:** the average of the 20 largest tension peaks in the front mooring line.

For each survival mode and wave condition, 9 different drivetrain configurations were selected encompassing a realistically achievable range spring/damper settings, summarized in Section 1.3). The data points summarized in Figure 15 represent the most favorable drivetrain setting, which results from a balance between minimizing slack event frequency, peak stroke, and peak tendon loads. These loads do not consider any fault cases, which would need to be examined in more detail at a later stage.

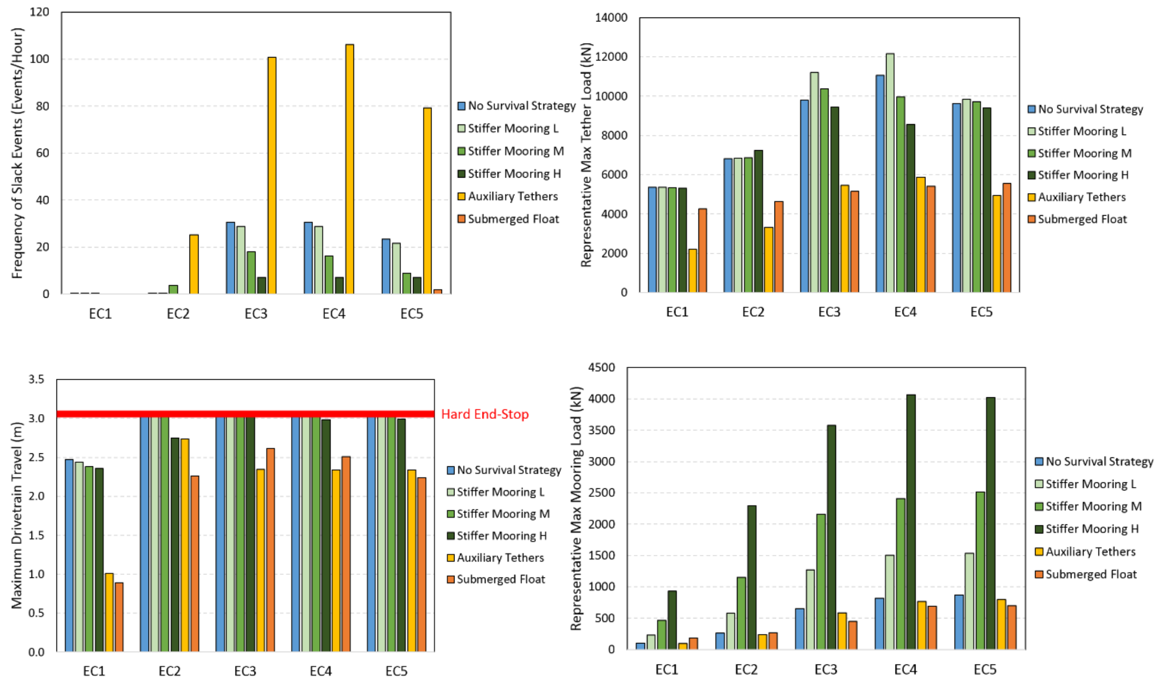

Figure 15. *Metrics of interest for the different survival concepts.*

10. Additional de-tuning strategies

In addition to the three survival strategies identified, the following methods can potentially be used in conjunction with all strategies in order to further de-tune the system. We believe that they are not feasible on their own, or combined, to provide enough de-tuning for a fully survivable system. Nevertheless, these methods may potentially be used to extend the operability limits of the device, thus enabling a higher threshold sea state before engaging survival mode and thus it may be worthwhile to consider these as an augmentation to the other survival strategies presented herein.

- ***Ballasting***: Dynamically changing the surface float mass on a sea-state to sea-state basis in order to move the system natural periods away from resonance.
- ***Additional Stroke***: Increasing the stroke from 3m to 4m (or higher) would allow more compliance between float and heave plate and hence fewer slack events.
- ***Extra high damping***: Increasing the drivetrain damping through hydraulic power dissipation.
- ***Tendon layout***: Based on the preliminary numerical results, the slack event frequency tends to be significantly greater for the front tendon than for the rear tendons, even in off-axis short crested seas. Therefore, different arrangements of the 3-tendon architecture (such as two tendons at the front, or perhaps 4-tendons) may also help mitigate slack events further. Additional numerical work would be needed here to evaluate different tendon layouts, with particular care taken to balance the device AEP and survivability.

11. Survival down-selection

Design Review 1 was held on April 5th 2017 with attendees from OPI, Glosten, and DNV-GL. Here, OPI presented preliminary numerical model results, for each of the three survival strategies in extreme waves.

Those involved in the design review agreed that the OPI team had considered all valid strategies (that they were aware/could think of) and had not dismissed or ignored any other valid alternative methodology. Key points taken from discussions during Design Review 1 are as follows:

Strategy A: Multiple lines.

- Concern was raised from that the hydraulic cylinder mechanism for tensioning the survival lines would be too heavy, expensive and unreliable. An alternative method was suggested that would use the PTO to tension the lines. It was felt that the PTO could be raised to the very top of its travel, then a cam mechanism actuated on each line in order to remove the slack (not take load) from the survival lines. The PTO could then be extended to the extreme of its stroke to transfer the load from the tendons to the survival lines.
- Multiple lines would still require the same MBL of the tendons, resulting in very expensive lines.
- Moving to groups of lines, rather than evenly spread-out lines may help to distribute peak loads between lines better. However there would be the risk of lines rubbing and abrading against each other.
- Matching line elasticity would be very difficult, and combined with this synthetic lines would creep, unpredictably causing problematic loads.
- Required survival line elasticity would be difficult to achieve as the average to peak load ratio is very high, thus lines would be very large.
- DNV-GL commented that the multiple lines provide an extra degree of redundancy, which would be beneficial. However, more end connections would mean more points of failure.
- Higher elasticity lines also tend to be of lower lifetime and thus would require more frequent replacement. Also, multiple lines may chafe against each other and further drive a reduced lifetime.

Strategy B: Increased mooring stiffness

- Uplift forces on the anchors would mean that drag embedment anchors may not be practical.
- There would be additional expense (and mass) due to the considerable extra structure on the float needed to manage the extremely high mooring loads **as well as** the tendon loads.
- Some survivability (fault) mooring cases (line break etc.) could lead to catastrophic system failure.

- Initial simulations suggest that this strategy is not effective on its own as it will not fully prevent hard end-stop impacts.
- The group agreed that this was by far the least attractive strategy. It was suggested that increasing mooring stiffness may be a way to de tune an alternative strategy to allow entry into survival at a higher Hs, i.e. use this method as an intermediate strategy, bridging a lower max operating condition with a higher survival entry criteria. However, such a strategy would still need winches on the mooring lines, so the additional cost would need to be balanced against the additional power gained.

Strategy C: Submersible system

- The submerged strategy was regarded as the superior strategy by all present and it was relayed that 4 separate reviewers at DNV-GL also agreed that this strategy is attractive and the most promising of the three. As a result the majority of the time was spent discussing this strategy.
- The concept involves lowering the surface float by ~10m by ballasting the hull to achieve a small negative buoyancy. The submerged position would be supported by the mooring floats. Glosten also presented the idea of lowering the system such that the heave plate sits upon the seafloor, similar to a tension leg platform. This provides the advantage of not requiring the large mooring floats to support the system in the submerged configuration and eliminating the chance of clashes between the heave plate and seafloor. However it was generally agreed that there may be more challenges and risks associated with this variant, such as controlling the rate of descent once fully flooded, environmental disturbance of the seafloor, potential damage to the heave plate during the set-down, and possible filling of the ring with sediment. The structure would also have to withstand a higher hydrostatic pressure if the heave plate is lowered all the way to the sea floor.
  A significant advantage was noted that the risk profile of this system is no longer focused on damage during waves, but is reduced to just the risk of entering and exiting the strategy itself. As a result the loads are far more controllable and predictable, meaning that it becomes much easier to mitigate and reduce risks by ensuring that the procedures and operations are comprehensive. This was agreed to be a significant advantage.
- There is added risk due to leakage around hull penetrations, but there are well known techniques to ensure these are maintained. Also positive pressure can be maintained in critical compartments to ensure no water entry.
- Ballast chambers would need multiple subdivisions to ensure an even lowering of system. This would also require redundancy in case of pump or valve port failure as it is critical that the surface float can be sunk when needed, however it is slightly less critical that the system is able to resurface. Additionally, it was noted that the float CoB will shift aft as the float sinks, so part of the control problem will be to

ensure that the added ballast water shifts the float CoG appropriately in order to avoid drastic pitching of the hull as it sinks.

- Biofouling and corrosion buildup may be an issue internally, but can be monitored and effective maintenance performed in-place.
- While automatic entry into the survival strategy is possible, it is expected that processes would be initiated manually (remotely) at some monitoring station
- Mooring design will be very critical for this concept. Some concern was raised that the connections between the mooring floats and the submerged hull might be a risk, undergoing large ranges of motion and have high loads, further investigation should be completed, including these in the numerical models. Also need to consider mobility and range-of-motion for the mooring connection to the float (e.g. shackles).
- There is a risk that the time to enter the strategy will be too long, and the climate may increases dangerously before being implemented. It is therefore important to ensure that the design accounts for this.
- DNV-GL pointed out that it is important to also consider the potential benefits (i.e. increased AEP) that can be enabled by using the ballast system during operational conditions.
- As noted previously, for this survival mode, the design loads are likely to be driven by large operational seas rather than survival mode seas. Work will need to be conducted to determine threshold for sinking the float and how frequently will this mode need to be engaged.
- Glosten noted that they expected this to be the lowest overall cost of the three strategies. The cost drivers for this strategy were discussed as
    - Minor additional structure to resist hydrostatic pressure from airtight compartments
    - Maintaining watertight integrity
    - Additional compartmentalization to ensure even ballasting when sinking.

Given the level of support for the submerged strategy, it was decided and agreed by all that it made sense to progress this strategy. While there is a small risk that unforeseen issues could prevent a workable system, this is true of any strategy, and the review group agreed that this likelihood is very small.

12. Appendix A: Additional information on the development of the baseline system configuration

Evaluation metrics

The goal to identify a system configuration or strategy that will allow the system avoid slack events in extreme 1:50 year waves is binary so to provide a more progressive measurement we developed metrics to quantify performance related to PTO travel and minimum tendon tension.

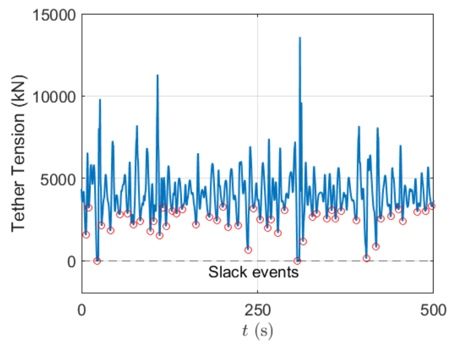

Figure 16. *Detection of tendon tension minima*

1) For quantifying and comparing the PTO travel, we evaluated the displacement significant, x_{sig}, which is defined as 4 standard deviations of travel.
2) For calculating a representative minimum tendon tension, we average the lowest 100 minima in a 2000s simulation period $\bar{T}_{min}$. The choice of 100 samples is arbitrary, but the general trends associated with this metric were insensitive to the number of minima averaged.

Influence of drivetrain spring and damping on system survivability

In normal operating conditions, there is an optimal drivetrain spring/damper setting that corresponds to maximum power absorption by the WEC. As the conditions become more extreme, optimum values of spring rate and damping are chosen to de-tune the device in order keep stroke within available limits, while ensuring zero tension (slack) events do not occur in the tendons. Given the large forces encountered during extreme waves, the damping forces required are in excess of what can be achieved from the generator alone. As such, additional dissipative damping is implemented in the drivetrain, through the use of pressure reducing valves and seawater cooling, which will approximately double the achievable drivetrain damping from 1200kN/m/s (generator alone) to 2000kN/m/s (generator + dissipative damping).

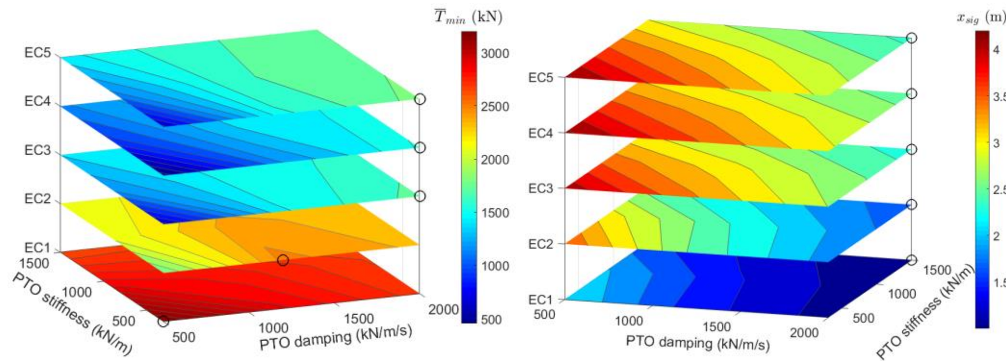

Figure 17. *Contour plots of* $\overline{T}_{min}$ *and* x_{sig} *for* [M_{float}, M_{HP}] *= [750, 1200] Te. The black circle denotes the safest spring/damping setting for each extreme wave condition.*

As shown in Figure 17, the travel (x_{sig}) is effectively limited by providing both an increasing spring stiffness and increasing damping. However, there will be a tradeoff between these two quantities, as a lower stiffness system tends to be most suitable for maximizing the minimum tendon loads. ($\overline{T}_{min}$) (i.e. reducing the number of slack events).

Influence of heave plate weight on PTO travel and slack events

In addition to tuning the drivetrain spring rate and damping, slack events can be mitigated by increasing the heave plate weight, which acts to provide a larger mean tension offset in the tendons. The effect of heave plate mass on x_{sig} and $\overline{T}_{min}$ is shown in Figure 18 for three different heave plate masses. It is important that the water line on the device is maintained, thus a heavier heave plate configuration consequently requires a lighter surface float, as such surface float/heave plate mass balances [750/1200t, 500/1450t, 250/1700t] that give the same overall system mass are used[2]. Clusters of data points for each wave condition represent different spring and damper settings, and the lines are fitted through the center of each cluster. From the trend shown in Figure 18 (left) it can be seen that a heavier heave plate significantly increases the minimum tendon tension for all wave conditions (beneficial for mitigating slack events), although Figure 18 (right) shows that the mass change has a far more limited impact on the PTO travel.

The baseline design presented here uses the 250t/1700t balance, as this provides the best survival performance. However, these masses are higher than practical and will result in substantial additional structural design to manage. In particular achieving a surface float mass of only 250t, with the strength to support a 1700t heave plate and associated power generation equipment may be extremely challenging and expensive. Thus, a further goal of this project is to allow the reduction of heave plate mass through the introduction of a suitable survival strategy.

[2] *This is primarily for ease of numerical analysis; in reality the surface float would need to maintain some minimum mass (~500Te). For the case of the 250T surface float, the mass would remain at 500Te but the water plane area would increase slightly to account for the increased overall mass.*

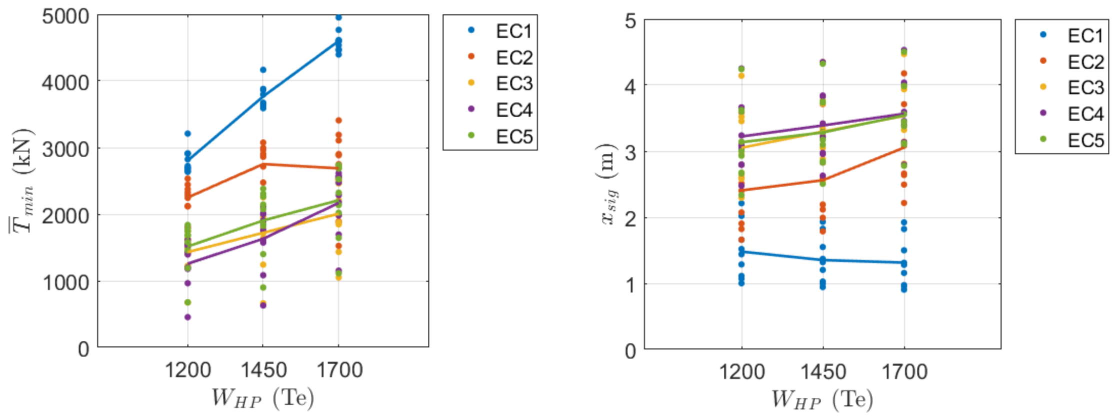

Figure 18. *Effect of heave plate weight on* $\overline{T}_{min}$ *and* x_{sig}

However, the visible trends in Figure 19 (the frequency of slack events as a function of heave plate mass) indicate that the heave plate mass would need to impractically large (i.e., the system would no longer be buoyant) in order to eliminate slack events by purely increasing the mass of the heave plate.

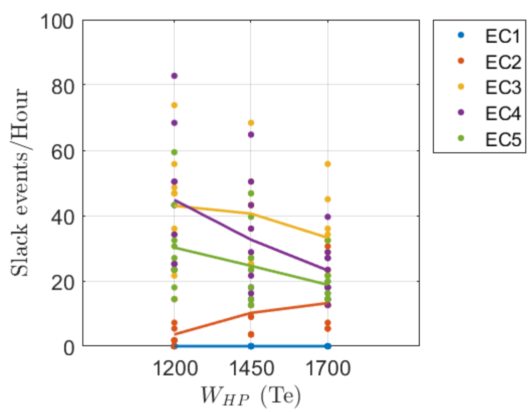

Figure 19. *Heave plate mass vs. frequency of slack events*

Influence of tendon elasticity on PTO travel and slack events

Increasing tendon elasticity is one way to detune the WEC by storing some of the captured energy as potential energy rather than transmitting it to the PTO, although this is not implemented here. The corresponding reduction in PTO travel and increase in representative minimum tendon loads is shown in Figure 20 for elasticities ranging from 0% to 20%. Note that $\overline{T}_{min}$ is the average of multiple tension minima and can still be greater than zero in the presence of slack events. We found that a tendon elasticity of ~30% is needed to de-tune the system enough to completely eliminate slack events and end-stops.

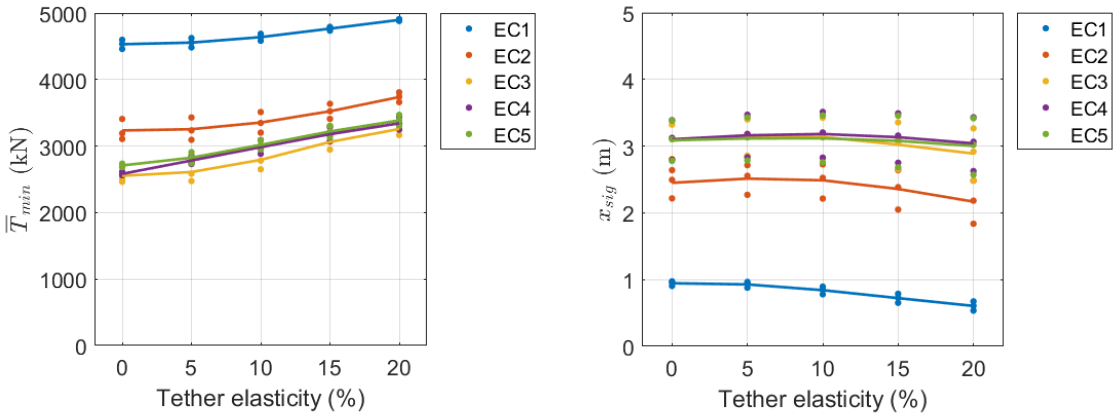

Figure 20. *Effect of tendon elasticity on* $\overline{T}_{min}$ *and* x_{sig}