Deltawave: A Wave Energy Converter

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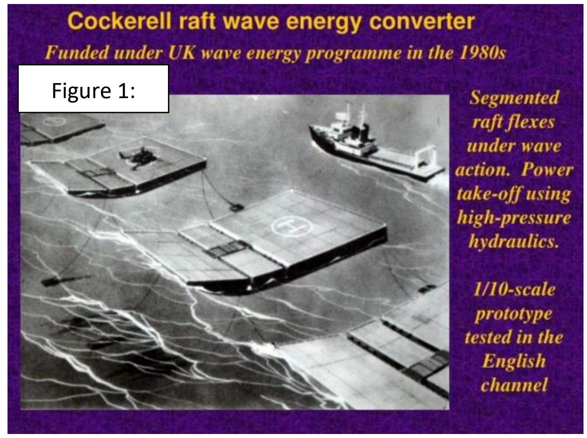


Figure 1:

Deltawave is a wave energy converter (WEC) that is a variant of the well-known 'Cockerell raft' design. In a Cockerell raft, the passage of underlying waves causes two or more rafts joined to one another at their ends to rotate relative to each other (see Figure 1). Resistance to this pitching captures the wave energy as useful power. A rectangular Cockerell raft is steered normal to incoming unidirectional waves. Because it resists the wave at the interface of its two media (water and air) and allows no corner in the interface for energy leakage, a Cockerell raft has a theoretical wave energy conversion higher than 80% (Ref 1). The 'Pelamis' WEC is an example of a Cockerell raft (Figure 2, Ref 2).

Figure 2: Pelamis Wave Power System

Deltawave modifies the Cockerell raft design from a rectangle to a triangle and connects six neighboring rafts in a 'hexraft' configuration, at the center of which is the connection, called a 'hexaxle'. Thus, the raft assembly covers the ocean in a hexagonal pattern as shown in Figure 3. Since each triangular raft can pitch in three directions, a Deltawave assembly can absorb wave power from multiple directions at once. As with the original Cockerell design, Deltawave covers the ocean surface (>80% areal coverage), giving the wave little opportunity to evade the WEC and deform into the overlying air. Hence, Deltawave is expected to retain the high energy conversion ratio of the classic Cockerell raft, but in the open ocean, where waves come from multiple directions simultaneously. Also, unlike a standard Cockerell raft, Deltawave doesn't need to be realigned when the wave direction shifts. This makes Deltawave like a buoy-type of WEC, but with a higher energy conversion ratio.

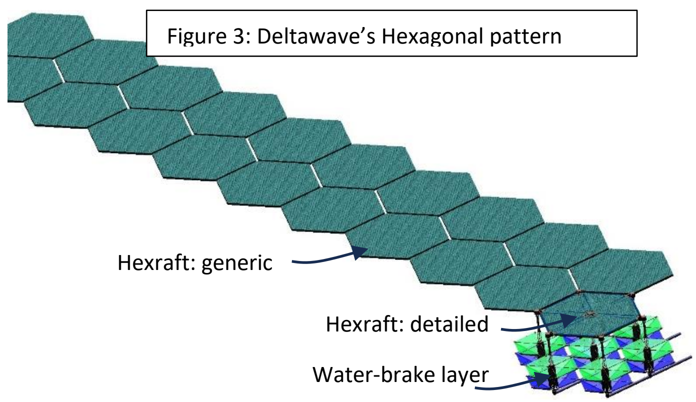
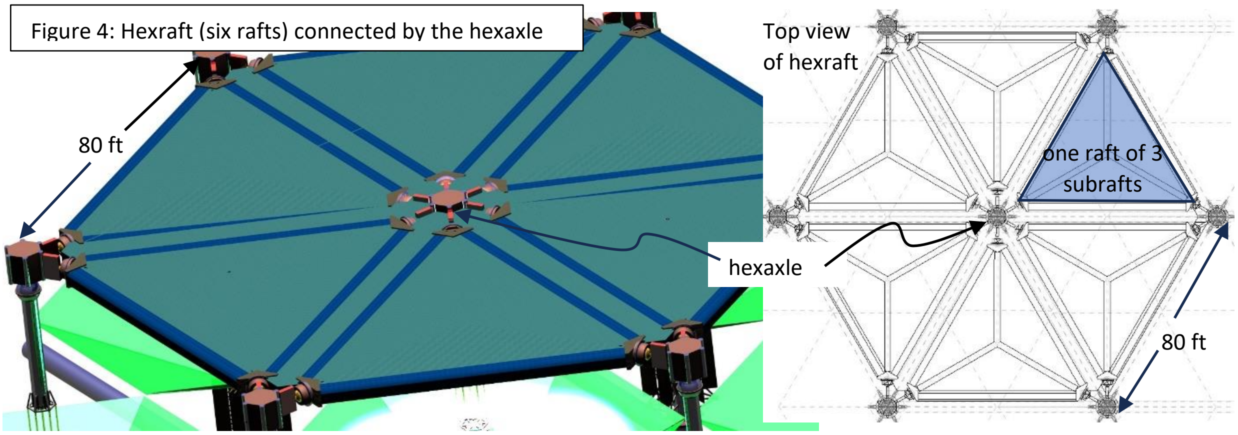


Figure 3: Deltawave's Hexagonal pattern

Deltawave's rafts are equilateral triangles 80 ft on a side (Figure 4). Triangles are structurally optimized shapes regarding strength-to-weight, which is why they are found frequently in construction. As the individual triangular rafts pitch, the connection of their apices to the hexaxle is rotatable and extensible except in the vertical direction. This allows each of the three raft apices to pitch, roll, and yaw relative to its hexaxle as needed for swells up to 30 feet, without tugging on or torquing the hexaxle in any horizontal direction. The torque of each raft apex on the hexaxle occurs exclusively in the vertical direction, and it is the resistance to this vertical motion that captures the

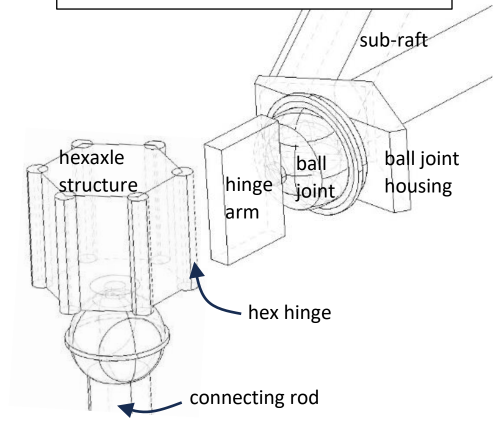
Figure 4: Hexraft (six rafts) connected by the hexaxle



wave's energy. Although the hexaxle is designed to accommodate 30-ft swells, for conservatism Deltawave's power conversion chain only allows 20 feet of extension.

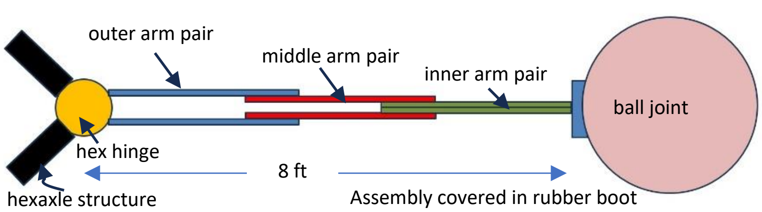
Figure 5 shows an exploded view of the hexaxle linkage. There are six hex-hinges on each hexaxle. The hinges swing horizontally only, and only through a limited arc so neighboring rafts avoid colliding. Attached to each hex hinge is a hinge arm, which is connected at its other end to a ball joint located just inside the apex of the raft. Furthermore, the hinge arm is capable of extension by up to 5 feet, giving the connection sufficient leeway to allow the raft to pitch in swells up to 30 feet without pushing or pulling the hexaxle in the horizontal plane (Figure 6).

Figure 5: Exploded view of hexaxle



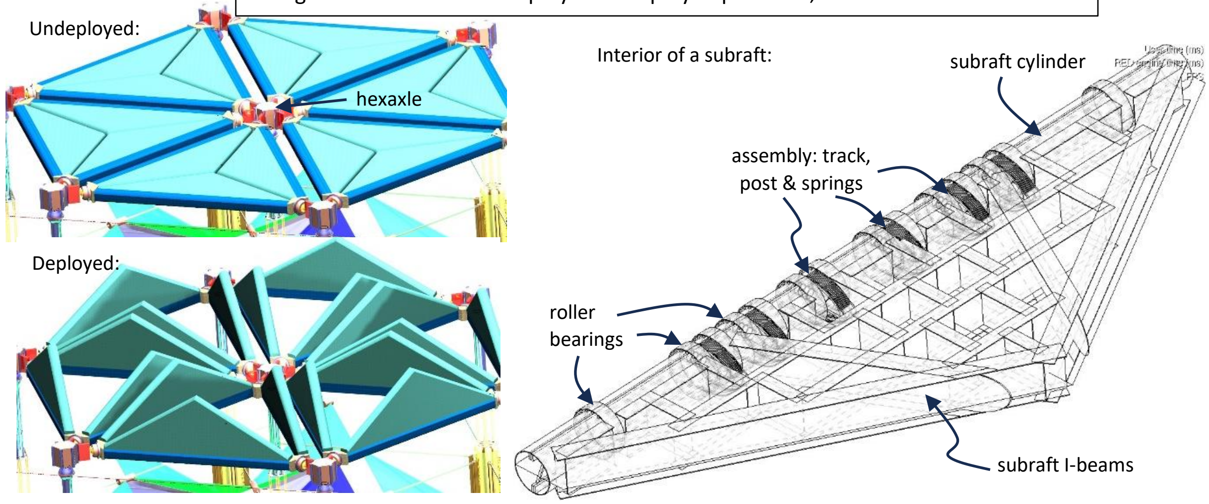
Deltawave is also designed to withstand storm waves of 60 feet or more, although not with the same high energy conversion efficiency. Each Deltawave raft is composed of three 'subrafts', which are rigged to rotate +/- 90 degrees when confronted with enough pressure, as shown in Figure 7. Once rotated, water can surge past the opened raft, which buries it and partially insulates it from the violence at the surface. Deltawave's rafts are designed to withstand being buried in water to a depth of 50 feet. Once the violence has past, the subrafts are spring loaded to preferentially return to their working position and lock in place.

Figure 6: Top view of a hinge arm (fully extended)



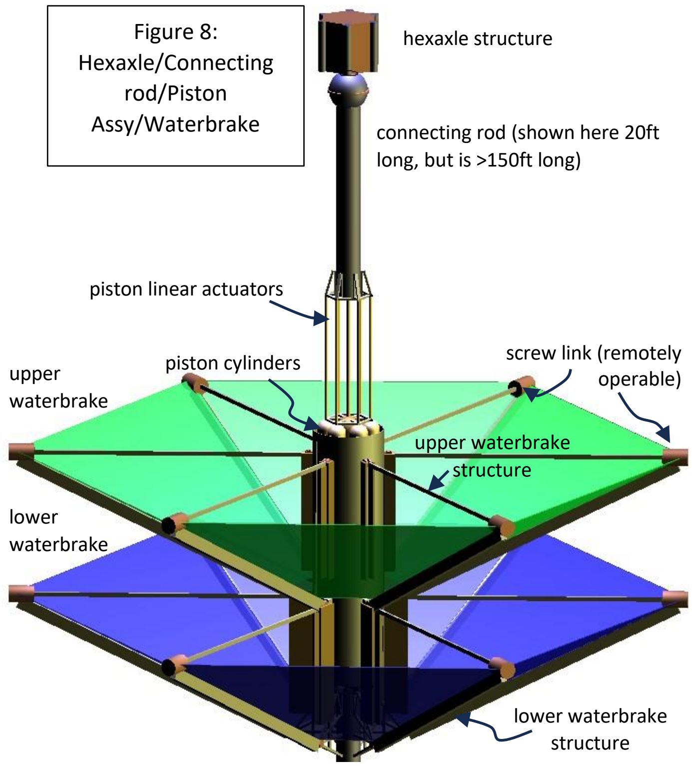
The hexaxle is connected via a connecting rod to an underwater power conversion chain (PCC) system as shown in Figure 8. This could simply be an array of piston pumps whose product is pressurized seawater. The pistons would have a maximum extensibility of 20 feet, above which the subrafts would be forced to deploy to protect the structure.

Figure 7: Subrafts in undeployed vs deployed positions, and Interior of a subraft



The piston pump array is connected at its bottom to a waterbrake assembly. This is composed of two upside-down umbrella-like plastic tarps with slanted spokes. The tarps may be glued to fish-netting for greater strength. The waterbrake is positioned at least 150 ft below the surface. This helps it form a proper anchor against the motion of the surface rafts. The arrow-like shape of the waterbrake resists upward motion to a greater degree than downward motion. This is designed to help the surface rafts align with the troughs of incoming waves rather than with the wave mean or with the wave crests. The PCC must likewise be designed to resist extension more than compression, as during compression the force is limited to the weight of the raft assembly.

Figure 8: Hexaxle/Connecting rod/Piston Assy/Waterbrake



Most WEC's translate wave-power into electricity locally, absorbing it into a closed loop hydraulic system with a piston pump, accumulator, turbine, and generator. Electricity is then cabled to shore. The economic analysis in Appendix B assumes such a PCC is present with Deltawave. This allows direct use of PCC cost estimates used in comparing various Marine Energy Conversion (MEC) systems, as recommended by Sandia National Labs and the National Renewable Energy Lab (Refs 3 & 4). In performing this economic analysis, Deltawave was assumed to be operating in the Pacific Northwest where the wave power resource is largest. In Appendix B, a Deltawave assembly composed of 20 hex-rafts is estimated to produce power at 36 cents/kW-hr for on-shore communities.

The waterbrakes may be independent, or they may be linked structurally to their neighbors to improve anchorage. These linkages would be remotely operable. Although not currently designed, it is expected they would resemble motorized screw assemblies. Once linked, the waterbrake assembly can rigidly resist the upward force of each connecting rod, since they won't all be driving upward at the same time. Additionally, each waterbrake would be designed to fold in a manner like an umbrella. Thus, with the linkages disconnected, waterbrakes folded, and rafts disconnected at the ball joint/hinge arm connection, a Deltawave assembly can be 'unzipped' from its neighboring rafts for maintenance. Ideally, individual rafts can be easily isolated and towed to shore for repair together with their associated waterbrakes. This kind of flexibility helps lower maintenance costs.

A Deltawave assembly is seen from the side in Figure 9. The low obstruction profile indicates that the assembly can push itself through the ocean using local wave energy from the site of its assembly to the site of its deployment. The main resistance to movement would come from the waterbrake assembly. It is estimated that with a typical NE Pacific trade wind blowing at 20 knots (34 ft/sec), enough wave power is available to move a Deltawave assembly the 2500-mile distance from Long Beach, CA to Hawaii in 0.8 to 1.4 months. If rigged for self-propulsion, and outfitted with satellite GPS positioning, Deltawave can potentially self-deploy, drastically reducing the cost of deployment.

Figure 9: Side view of Deltawave

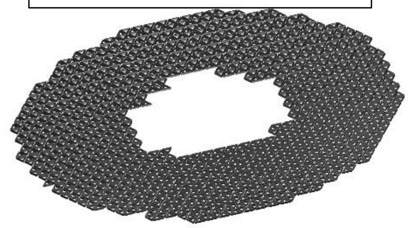
Once deployed, the Deltawave assembly would remain unanchored to the ocean floor and would use its self-propulsion and steering capability to maintain its location. The low wind profile, the cyclic nature of surface waves, and the drag imposed by the waterbrake assembly buried deep in unmoving water should make this a small parasitic load. This would reduce the cost of anchorage and allow for easy relocation if needed, since only the power takeoff would need to be manually adjusted. Each Deltawave assembly would transmit its location to satellites overhead, allowing for its location to be constantly monitored remotely at a centralized office.

Figure 10: Outer layer of rafts

The outer layer of a Deltawave assembly may have drastically reduced subraft surfaces, so that each raft resembles a hollow triangle, as shown in Figure 10. The purpose of this outer layer is to help the Deltawave assembly survive monster waves. The outer layer, because it is hollow but still floats, can more easily slide between the swell and the foam above it that forms when open ocean swells are 'topping'. Having located this boundary, the outer layer pulls the rest of the assembly behind it in the proper position over the swell. In this way monster waves are unable to toss the Deltawave assembly onto its back, since it is getting pulled through the ocean by its outer layer.

A Deltawave assembly reduces the amplitude of passing waves, but not their wavelengths. If the assembly is curved around a patch of ocean (Figure 11), that circular patch could maintain a level of ocean violence low enough for permanent floating structures to be positioned there. Motions would be limited to heaving (up and down), with much reduced pitching and rolling due to the long wavelength of the passing wave relative to its amplitude. Deep ocean wave energy conversion has been identified as an economic alternative for the remote processing of ores into finished materials: such as bauxite into aluminum, in part because wave power is 60% higher in the deep ocean than it is closer to shore (Ref 1). As it requires no anchorage to function, Deltawave would be ideal for this purpose.

Figure 11: Donut-shaped raft assembly



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Appendix A: Structural Analysis

To provide a construction cost estimate for the economic analysis, a structural analysis was undertaken to estimate the kinds and amounts of materials used in Deltawave.

The Deltawave subrafts are impressed by both static and dynamic water pressures. The subraft transmits these forces to the subraft cylinder as moment and shear loading. In operation, the cylinder remains in the undeployed position. When the moment becomes too extreme, it deploys, and opens the subraft to let water pass through the raft. The cylinder transmits its forces to the two ball-joint housings at its two ends, whence vertical forces are transmitted through the ball-joint and the hinge arm to the hex-axle, then down through the connecting rod to the piston assembly, which extracts the wave energy as pressurized water. If the piston assembly is replaced with a solenoid or hydraulic-electric power conversion system, the energy is extracted as electricity. Each connecting rod is pulled by the water pressure coming from six subrafts (it is directly connected to twelve subrafts, but each subraft is connected to two connecting rods, so the effective force is from six subrafts). These forces were used to calculate the structural requirements of the various parts of Deltawave.

Dynamic Drag Forces, Operational Case (closed-subraft):

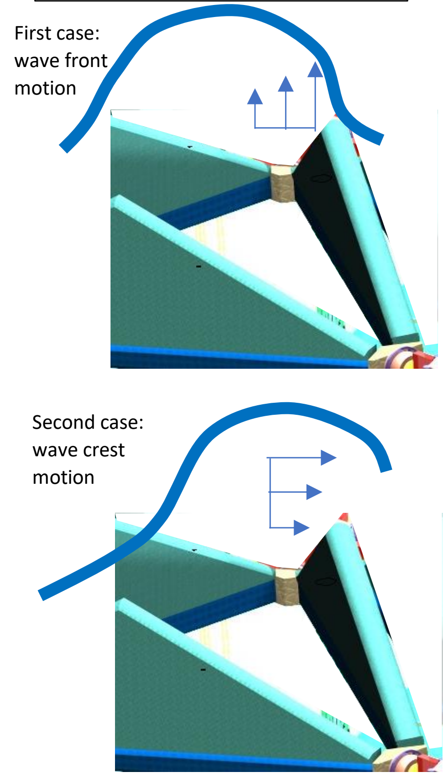
In operation, the subrafts remain undeployed as they are pressed upward by an underlying wave. A maximum wave height of 20 feet is currently baselined for this undeployed condition, since this also is the maximum stroke of the pistons powered by the wave motion, and thus determines their overall size. For wave heights of 5 to 20 feet, the upward forces on the various parts of the current raft design were calculated and added up and translated into an upward force per hexaxle. This is done by assuming the raft travels some percentage of the wave height, called the stroke. For example, in a 5 ft wave, a raft with an 80% stroke goes up 4 feet. To hold down the wave beneath it, it must exert the same force as the 1 ft of water above it would have exerted, which is 1 ft times the density of water. Note that Deltawave is designed to fall, in the trough of each wave, to align with the trough in elevation, and not the average sea level. Thus, any departure from the trough level means Deltawave is being forced upward, and the maximum force coincides with the peak of the wave. For each stroke and wave height, an estimate of the average force per hexaxle, over the wavelength λ was generated, as well as the maximum force at the wave's peak, which is about twice the average force. These forces increase linearly with the factor, 1 - stroke, with a slope of 769.6 kN/ft of wave height (H) for the average force, and twice that slope for the maximum force, for the current Deltawave raft design. For a 20 foot wave and a stroke of 70%, the maximum force on the raft is 2*769.6*20ft*(1-0.7) = 9,235.2 kN. This maximum force is used to structurally size the WEC components, after being multiplied by a safety factor, which is currently 1.8. Divided by the area of the raft, this dynamic pressure (P_D) is 3.0 kN/ft^2.

Dynamic Drag Forces, Nonoperational Case (open-subraft):

In monster waves, Deltawave's subrafts are designed to open to allow large fractions of the water volume to pass through, lessening the force on the raft structure. The strength of the subraft structure is, in this case, based on the dynamic drag force. This force is proportional to the square of the velocity of the water moving past the subraft, and the exposed subraft area. The maximum deployed-case wave size is 60 feet. Waves larger than this are extremely infrequent, especially closer to land surfaces, where a WEC would typically function.

With deployed subrafts, there are two cases to consider, as shown in Figure A1. The first case is with the wave surging upward past the structure at its maximum upward velocity. For a 60ft wave, that is conservatively 17.4 ft/s. A drag coefficient (C_D) of 0.8 is used and a safety factor of 1.8, leading to a dynamic drag pressure (P_D) of 1.9 kN/ft2 on the affected areas of each deployed subraft, which is primarily the cylinder. These values are with a conservative H/λ =3*0.033. The wave period (T) is proportional to the wavelength (λ), which means that for a given wave height (H), less time is available with higher H/λ, so the resulting water velocities are higher. With a less conservative H/λ =2*0.033 the velocity is 14.2 ft/s, and the dynamic pressure (P_D) is 1.3 kN/ft2.

Figure A1: Dynamic drags on open subraft



The second case occurs at the wave crest, as water is surging horizontally over the deployed subrafts. There is a possibility a deployed subraft will not be able to close itself against this surge, which occurs perpendicularly to the open subrafts surfaces. Given a drag coefficient (C_D) of 1.4 and a safety factor of 1.8, a dynamic drag pressure of 2.2 kN/ft2 is expected (under conservative H/λ =3*0.033). With a less conservative H/λ =2*0.033, the horizontal velocity is 11.5 ft/s, and the dynamic drag pressure is 1.4 kN/ft2.

Static Pressure, Nonoperational Case (open-subraft):

In addition to the three dynamic pressure cases, Deltawave must be able to withstand a static pressure proportional to being buried by many feet of water when inundated by a wave of 60 ft wave, in the open-subraft case. Since Deltawave can move upward by up to 20 ft, the maximum depth is 40 ft. Another 10 ft is added for conservatism, so Deltawave is designed to be inundated to a depth of 50 ft. This is 22 psig, or 14 kN/ft2, of static pressure (P_s), in addition to the dynamic loads calculated above. For conservatism, this static pressure was employed in all three dynamic scenarios, including the operational cases.

Subraft cladding calculations

The water presses on the cladding that envelopes the subraft. This cladding is tiled, and each square tile is designed to withstand the water pressure without support up to a certain size, b. Two materials were considered for the cladding: ferrocement and steel. Ferrocement is a form of reinforced concrete, in which the mortar is supported by an embedded steel mesh. The following equation is used to predict the strength of ferrocement beams subject to pure bending (Ref A1):

$M_n/(F'_c*b*h^2*\eta) = 0.005 + 0.422*(V_f*F_y/F'_c) + 0.0772*(V_f*F_y/F'_c)^2$;

Where M_n = nominal moment strength of the ferrocement beam, F'_c = compressive strength of mortar, F_y = yield strength of the steel in the reinforcing mesh, V_f = volume fraction of reinforcement, b = side length of the beam, h = height of the beam (=t, beam thickness), η = the global efficiency factor of reinforcement in resisting tensile bending (usually, $\eta = 0.5$). For a beam that is a square tile under pressure p, the load along the beam, w = p*b, and the Moment, $M_n = w*b^2/8 = p*b^3/8$. Using the definition given above for the nondimensional constant $[M_n/(F'_c*b*h^2*\eta)]$, and rearranging,

$b_{beam} = \text{sqrt}\{[M_n/(F'_c*b*h^2*\eta)]* F'_c*h^2*\eta*8/p\}$,

which gives the size of the square tile b, as a function of the makeup of the ferrocement, V_f, F_y, F'_c, the tile thickness h, the efficiency η, and the water pressure p.

However, the square tiles on Deltawave are not beams secured at two opposite edges, but plates secured at all four of their edges, and such plates can support larger pressures for a given size. For a rectangular beam, $\sigma_{max} = M_n*c/I_z = (p*b^3/8)*12*(h/2)/(b*h^3)$, because, as given above, $M = p*b^3/8$, and also because $I_z = b*h^3/12$, and c = h/2 (=t/2). Simplifying, $\sigma_{max\text{-}beam} = 0.75*p*(b_{beam}/h)^2$. For a plate Reference A2 gives $\sigma_{max\text{-}plate} = 0.75*p*(b_{plate}/h)^2/2.61$. Letting $\sigma_{max\text{-}beam} = \sigma_{max\text{-}plate}$, then b_{plate}/b_{beam} = sqrt(2.61) = 1.61. In practice, we can use the equation above to calculate b_{beam}, and then multiply by 1.61 to get b_{plate}.

The steel cladding option uses a steel skin, currently 0.2” thick, backed up by an array of beams. Given the square tile size calculated for the ferrocement option, b_{plate} above, the pressure offset by the skin itself is calculated as $p_{skin} = 2.61*F_y/(0.75*(b_{plate}/t)^2)$, where t=0.2”. The beam sizing was then done assuming a certain number of beams per width b ($\#_{beams}$), and these beams are resisting the remaining pressure force: $p_{beams} = p – p_{skin}$. Assume the F_y of a plate is 2.61 times the F_y of a beam. Then, using $F_{y\text{-}plate} = 2.61*F_y \Rightarrow 2.61\ M*c/I_z$, where c=h/2, $M = p*b^3/8$, and $I_z = 0.5*h^4/12$ (for a rectangular beam where the width of the beam is half the height h), the sizing equation for the beam is $h^3_{rect} = 1.5*(p–p_{skin})*(b/\#_{beams})*b^2/(2.61*F_y)$. For an I-beam it was found that $h^3_{I\text{-}beam} = h^3_{rect}*\{1/[1 -(1 -2*\{t/h\})^4]\}$. This again assumes the flange width is h/2, and the thickness (of both web and flange) is defined using {t/h}. I-beams use about half as much steel as rectangular beams sized for this application, so the I-beam design is preferred.

Since these calculations assume the yield strength of the beams in a plate is 2.61 times the yield strength of the beams in a beam, it assumes that twice as many beams are found, per tile, then in $\#_{beams}$. That is, the array of beams underlying the cladding skin has $\#_{beams}$ running in one direction under the tile, and the same number running at right angles to the first bunch. Thus, the materials calculation doubles $\#_{beams}$ to determine material requirements for the cladding.

General I-beam structural considerations

For most of the structural calculations below, parts are sized to keep the maximum shear stress in the part, τ_{max}, below the shear limit, S_y, where $S_y = 0.577*F_y$, and F_y is the yield strength of steel = 36ksi. The maximum shear stress is calculated from:

$\tau_{max} = \text{sqrt}\ \{[(\sigma_x - \sigma_y)/2]^2 + \tau_{xy}^2\}$

For an I-beam, τ_{max} occurs at the flange, although in some cases it may occur along the centerline. Both locations are checked. Choosing the x-direction along the I-beam length and the y-direction along its height (i.e. along the web of the I-beam), then

$\sigma_x = M_i*c/I_z$, $\sigma_y = - P_s*b/t_w$, and $\tau_{xy} = V_i*Q/(t*I_z)$

Where M_i = the bending moment (ft-lbf) at section i, V_i = the shear (lbf) at that section, $c = y_{max} = (w+2t_f)/2$ (assuming y = 0 occurs at the I-beam centerline), P_s = maximum static pressure of the subraft (i.e. when buried under a monster wave), b = the spacing of the minor I-beams (generally), t_w = thickness of the web, I_z = second moment of area of the I-beam about the centerline, i.e. where y=0 (in distance^4), and Q = first moment of area of the I-beam flange about the centerline (in distance^3). Note that σ_y is a negative quantity because it's a compressive stress. For an I-beam:

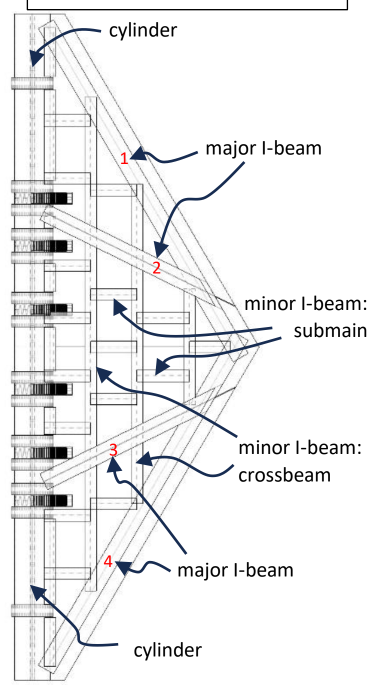
$I_z = t_w*w^3/12 + 2\{f*t_f^3/12 + f*t_f*[(w+t_f)/2]^2\}$, and $Q = f*t_f*[(w+t_f)/2]$

where w = web height, t_w = web thickness, f = flange width, t_f = flange thickness

The equations above are valid for calculating the maximum shear stress at the I-beam flange. Along the centerline, the value of c is zero (i.e. y=0) so there is no bending stress, but shear stress is maximized because Q is increased by adding the first moment of half of the web area about the centerline, which is $(w*t_w/2)*(w/4)$. Bending stresses at the flange generally predominate in I-beam sizing.

Subraft minor I-beam calculations

Figure A2: Subraft I-beams



The minor I-beams frame the cladding tiles for support and are thus spaced according to the tile size b. Half of them run parallel to the cylinder and the other half perpendicular to the cylinder, as indicated in Figure A2. The first class of minor I-beams are called 'crossbeams' (because they 'cross' the major I-beams). They are spaced b feet apart and run parallel to the cylinder from major I-beam to major I-beam. In sizing the crossbeams, the maximum bending moment (lbf-ft) occurs at the crossbeam's center, while the maximum shear (lbf) occurs at the ends of the beam. These are calculated as $M = P_D*b*L^2/8$ and $V = P_D*b*L/2$, where 'L' is the distance between the two major I-beams. Only the dynamic pressure, P_D, leads to bending and shear because it hits one side of the raft preferentially. Since M and V occur in different places, their stresses can be evaluated separately. The limiting condition is typically the bending moment.

To calculate the I-beam web thickness (t_w), the following equation was iterated on:

$$S_y^2 \Rightarrow \tau_{max}^2 = \{\mathbb{M}_{fac}*M*0.5(w+2t_f)/(2*I_z) + P_s*b/(2*t_w)\}^2 + \{\mathbb{V}_{fac}*V*Q/(t_w*I_z)\}^2$$

Where $I_z = t_w*w^3/12 + f*t_f^3/6 + f*t_f*0.5*(t_w+t_f)^2$

and $Q = 0.5*f*t_f*(w+t_f) + \mathbb{CL}_{fac}*t_w*0.125*w^2$

In these equations, w = web thickness, which is the defined thickness of the subraft, f = flange thickness, defined as a set fraction of w for all I-beams, t_f = flange thickness defined as a set fraction of t_w, for all I-beams. Also:

M_{fac} = a switch: =1 if you want to evaluate the bending moment at the beam center,

=0 if you don't.

V_{fac} = a similar switch for evaluating the shear stresses at the beam endpoints. Note that if M_{fac} =1, then V_{fac} = 0, because the two forces are mutually exclusive in this situation.

CL_{fac} = a switch: =1 if the shear is being evaluated at the centerline.

=0, the shear is being evaluated at the flange.

The static pressure force (P_s) is as calculated before.

The second class of minor I-beams are called 'submains' because, like the major I-beams (the 'mains'), they run toward (not parallel to) the cylinder. As with the crossbeams, the driving moment and shear are $M = P_D*b*L^2/8$ and $V = P_D*b*L/2$. Unlike the crossbeams, L = b, because b is the spacing between the crossbeams. The other structural calculations are the same as for the crossbeams.

It is likely that the subraft structure will feature trusses rather than I-beams. To check the material requirements of such trusses, truss calculations were made. Neville trusses were assumed, resulting in isosceles triangles (two sides of equal length) among the truss sections. The upper horizontal struts connected to the upper cladding are called the 'a' struts, the lower horizontal struts connected to the lower cladding are the 'b' struts, and the struts that slant between upper and lower sides are the 's' struts. Assuming a minor truss has a distributed load w along its upper surface, then the dynamic loading is w*L, where L is the length of the minor truss (for a 'submain', L=b, the tile size calculated above), and $w = b*P_D$. Since the load is symmetrical, only half of the truss need be considered, and the load is calculated according to the fractional length, f, where f =0 at one of the pinned truss edges, and f=1 at L/2. For this case, the force on the slanted struts is compressive, and is $F_s = 0.5*w*L*(1-f)/\cos(\alpha/2) + F_{s\ s.p.}$, where α is the internal angle of the slanted struts (the struts form isosceles triangles), and $F_{s\ s.p.}$ is the compressive loading due to static pressure: $F_{s\ s.p.} = P_s*b*h*\tan(\alpha/2)/\cos(\alpha/2)$, where h is the truss height (distance between upper and lower cladding surfaces). The maximum slanted strut force occurs at the pinned edges of the truss, where f=0. By summing moments about the nodes along the bottom strut 'b', the loading of the top struts (a), can be calculated: $F_a = w*L^2/(8h))*(2f – f^2) -F_{a\ s.p.}$, where $F_{a\ s.p.} = F_{s\ s.p.}*\sin(\alpha/2)$. The compressive dynamic loading is a maximum at the center of the truss, where f=1. However, the static pressure places the horizontal struts (a and b) in tension, so the maximum tensile loading on the top struts occurs at the pinned edges where f=0, and $F_a = F_{a\ s.p.} = F_{s\ s.p.}*\sin(\alpha/2)$. The bottom horizontal struts are in tension, both due to the static pressure and the dynamic pressure. For the bottom struts, $F_b = F_a + F_s*\sin(\alpha/2) = w*L^2/(8h))*(2f – f^2) +F_{b\ s.p.} +F_s*\sin(\alpha/2)$, where $F_{b\ s.p.} = -F_{a\ s.p.}$.($F_{b\ s.p.}$ is added to the first term, rather than subtracted, because the bottom strut is placed in tension by both loadings). Since the first two terms are maximum at the center of the truss (f=1), and the last term is maximum at the edge of the truss (f=0), F_b is a maximum somewhere between the edge and the center, and calculations indicate that's at about f=0.25. Since the subraft is symmetrical, the top and bottom struts must be interchangeable, and so the larger size calculated for the bottom strut, F_b, mandates the size of both horizontal struts.

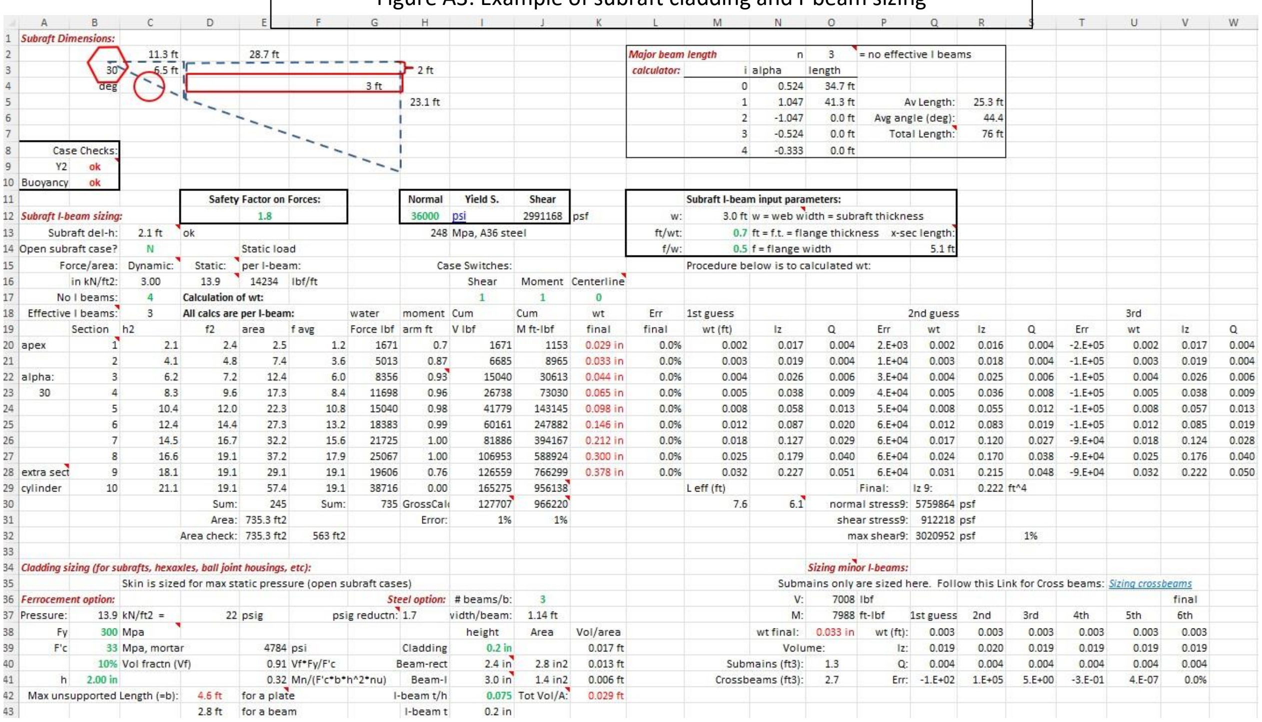
With the strut forces calculated, their sizing is straightforward. It was found that the truss requires about 20% more material than the I-beam, for the same loading. This discrepancy can likely be reduced by further optimizing the truss structure.

Major I-beam calculations

There are four major I-beams that travel from the free apex of the subraft to the cylinder, as shown in Figure A2. These I-beams are loaded by a bending moment and a shear, which determines their sizing. To size these I-beams, they were cut into nine sections, eight equal sections along the triangular part of the subraft, and a ninth rectangular section which interfaces with the cylinder. Each section has a part of the subraft lower cladding transmitting the dynamic water force to it, a force which is proportional to the lower cladding area (A_i) and the dynamic pressure of the passing wave (P_D). The shear force (V_i) and bending moment (M_i) on each I-beam section i is calculated from $V_i = V_{i-1} + P_D*A_i$; and $M_i = M_{i-1} + V_{i-1}*L_i + P_D*A_i*L_{Mi}$; where L_i = the length of I-beam section i, and L_{Mi} = the moment arm of that section. This moment arm would be $0.5*L_i$ if the cladding areas were rectangular. Since they are trapezoidal due to the shape of the subraft, L_{Mi} is slightly smaller than $0.5*L_i$. Note that for the first section, which begins at the free apex of the subraft, V_{i-1} and M_{i-1} are both zero.

There is also the ninth I-beam section, which corresponds to the rectangular subraft area that connects to the cylinder. And there is the cylinder itself (i.e. section 10), for which the summed moments of the subraft become a torque, and the summed shear forces become an upward distributed force when added to the direct force of water pressure on the cylinder itself. An example of the spreadsheet set up to size the major I-beams, minor I-beams, and cladding is shown in Figure A3.

Figure A3: Example of subraft cladding and I-beam sizing



	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	S	T	U	V	W
1	Subraft Dimensions:																						
2			11.3 ft		28.7 ft							Major beam length		n	3	= no effective I beams							
3		30	6.5 ft					2 ft				calculator:	i	alpha	length								
4		deg					3 ft						0	0.524	34.7 ft								
5								23.1 ft					1	1.047	41.3 ft		Av Length:	25.3 ft					
6													2	-1.047	0.0 ft		Avg angle (deg):	44.4					
7													3	-0.524	0.0 ft		Total Length:	76 ft					
8		Case Checks:											4	-0.333	0.0 ft								
9	Y2	ok																					
10	Buoyancy	ok																					
11					Safety Factor on Forces:			Normal	Yield S.	Shear			Subraft I-beam input parameters:										
12	Subraft I-beam sizing:				1.8			36000	psi	2991168	psf	w:	3.0 ft	w = web width = subraft thickness									
13		Subraft del-h:	2.1 ft	ok				248	Mpa, A36 steel			ft/wt:	0.7	ft = f.t. = flange thickness			x-sec length:						
14	Open subraft case?		N		Static load							f/w:	0.5	f = flange width			5.1 ft						
15		Force/area:	Dynamic:	Static:	per I-beam:				Case Switches:				Procedure below is to calculated wt:										
16		in kN/ft2:	3.00	13.9	14234	lbf/ft			Shear	Moment	Centerline												
17		No I beams:	4	Calculation of wt:					1	1	0												
18		Effective I beams:	3	All calcs are per I-beam:			water	moment	Cum	Cum	wt	Err	1st guess				2nd guess				3rd		
19		Section	h2	f2	area	f avg	Force lbf	arm ft	V lbf	M ft-lbf	final	final	wt (ft)	Iz	Q	Err	wt	Iz	Q	Err	wt	Iz	Q
20	apex	1	2.1	2.4	2.5	1.2	1671	0.7	1671	1153	0.029 in	0.0%	0.002	0.017	0.004	2.E+03	0.002	0.016	0.004	-2.E+05	0.002	0.017	0.004
21		2	4.1	4.8	7.4	3.6	5013	0.87	6685	8965	0.033 in	0.0%	0.003	0.019	0.004	1.E+04	0.003	0.018	0.004	-1.E+05	0.003	0.019	0.004
22	alpha:	3	6.2	7.2	12.4	6.0	8356	0.93	15040	30613	0.044 in	0.0%	0.004	0.026	0.006	3.E+04	0.004	0.025	0.006	-1.E+05	0.004	0.026	0.006
23	30	4	8.3	9.6	17.3	8.4	11698	0.96	26738	73030	0.065 in	0.0%	0.005	0.038	0.009	4.E+04	0.005	0.036	0.008	-1.E+05	0.005	0.038	0.009
24		5	10.4	12.0	22.3	10.8	15040	0.98	41779	143145	0.098 in	0.0%	0.008	0.058	0.013	5.E+04	0.008	0.055	0.012	-1.E+05	0.008	0.057	0.013
25		6	12.4	14.4	27.3	13.2	18383	0.99	60161	247882	0.146 in	0.0%	0.012	0.087	0.020	6.E+04	0.012	0.083	0.019	-1.E+05	0.012	0.085	0.019
26		7	14.5	16.7	32.2	15.6	21725	1.00	81886	394167	0.212 in	0.0%	0.018	0.127	0.029	6.E+04	0.017	0.120	0.027	-9.E+04	0.018	0.124	0.028
27		8	16.6	19.1	37.2	17.9	25067	1.00	106953	588924	0.300 in	0.0%	0.025	0.179	0.040	6.E+04	0.024	0.170	0.038	-9.E+04	0.025	0.176	0.040
28	extra sect	9	18.1	19.1	29.1	19.1	19606	0.76	126559	766299	0.378 in	0.0%	0.032	0.227	0.051	6.E+04	0.031	0.215	0.048	-9.E+04	0.032	0.222	0.050
29	cylinder	10	21.1	19.1	57.4	19.1	38716	0.00	165275	956138			L eff (ft)			Final:	Iz 9:	0.222	ft^4				
30				Sum:	245	Sum:	735	GrossCalc	127707	966220			7.6	6.1		normal stress9:	5759864	psf					
31				Area:	735.3 ft2			Error:	1%	1%						shear stress9:	912218	psf					
32				Area check:	735.3 ft2	563 ft2										max shear9:	3020952	psf	1%				
33																							
34	Cladding sizing (for subrafts, hexaxles, ball joint housings, etc):														Sizing minor I-beams:								
35			Skin is sized for max static pressure (open subraft cases)											Submains only are sized here. Follow this Link for Cross beams: Sizing crossbeams									
36	Ferrocement option:							Steel option:	# beams/b:	3				V:	7008	lbf						final	
37	Pressure:	13.9	kN/ft2 =	22	psig		psig reductn:	1.7	vidth/beam:	1.14 ft				M:	7988	ft-lbf	1st guess	2nd	3rd	4th	5th	6th	
38	Fy	300	Mpa						height	Area	Vol/area			wt final:	0.033 in	wt (ft):	0.003	0.003	0.003	0.003	0.003	0.003	
39	F'c	33	Mpa, mortar		4784	psi		Cladding	0.2 in		0.017 ft			Volume:		Iz:	0.019	0.020	0.019	0.019	0.019	0.019	
40		10%	Vol fractn (Vf)		0.91	Vf*Fy/F'c		Beam-rect	2.4 in	2.8 in2	0.013 ft			Submains (ft3):	1.3	Q:	0.004	0.004	0.004	0.004	0.004	0.004	
41	h	2.00 in			0.32	Mn/(F'c*b*h^2*nu)		Beam-I	3.0 in	1.4 in2	0.006 ft			Crossbeams (ft3):	2.7	Err:	-1.E+02	1.E+05	5.E+00	-3.E-01	4.E-07	0.0%	
42	Max unsupported Length (=b):			4.6 ft	for a plate			I-beam t/h	0.075	Tot Vol/A:	0.029 ft												
43				2.8 ft	for a beam			I-beam t	0.2 in														

As before, the I-beams are sized to keep the maximum shear stress, τ_{max}, below the shear limit, S_y, where $S_y = 0.577*F_y$ (F_y is the yield strength of steel, 36ksi). This maximum stress occurs at the I-beam flange, although in some cases it may occur along the I-beam centerline. Both locations were checked. Recall $\tau_{max} = \text{sqrt}\{[(\sigma_x - \sigma_y)/2]^2 + \tau_{xy}^2\}$, where $\sigma_x = M_i*c/I_z$, $\sigma_y = - P_s*b/t_w$, and $\tau_{xy} = V_i*Q/(t*I_z)$, where M_i =

the bending moment (ft-lbf) at I-beam section i, V = the shear (lbf) at that section, $c = y_{max} = (w+2*t_f)/2$ (assuming y = 0 occurs at the I-beam centerline), P_s = maximum static pressure of the subraft (i.e. when buried under a monster wave), b = the spacing of the minor I-beams, t_w = thickness of the web, I_z = second moment of area of the I-beam about the centerline, i.e. where y=0 (in distance^4), and Q = first moment of area of the I-beam flange about the centerline (in distance^3). For an I-beam:

$I_z = t_w*w^3/12 + 2*\{f*t_f^3/12 + f*t_f*[(w+t_f)/2]^2\}$, and $Q = f*t_f*[(w+t_f)/2]$

where w = web height, t_w = web thickness, f = flange width, t_f = flange thickness

The equations above are valid for calculating the maximum shear stress at the I-beam flange. Along the centerline, the value of c is zero (i.e. y=0) so there is no bending stress, but shear stresses are at their maximum, because Q is increased by adding the first moment of half of the web area about the centerline, which is $(w*t_w/2)*(w/4)$. Despite this, the bending stresses at the flange predominate in sizing the I-beams.

Given the extreme static and dynamic forces calculated above, each I-beam section has its maximum M and V calculated, from which these equations are iterated to produce the web and flange thicknesses (t_w and t_f).

As with the minor I-beams, the major beams can also be designed as trusses, so structural calculations were carried out to estimate their material requirements. Neville trusses were assumed, with their repeating units of isosceles triangles, of internal angle α. Since the beams are broken up into sections, each section constitutes its own cantilevered beam: fixed at one end with an incoming shear (V_{i-1}) and moment (M_{i-1}) from the adjoining section closer to the subraft apex, and subject to distributed dynamic ($w = b*P_D$) and static (P_s) pressure loadings along its length. As with the minor trusses, the static pressure places the slanted struts in compression, where $F_{s\ s.p.} = P_s*b*h*\tan(\alpha/2)/\cos(\alpha/2)$. This places the horizontal struts (a and b) in tension, according to $F_{a\ s.p.} = F_{s\ s.p.}*\sin(\alpha/2)$. The dynamic loads are added to these static loads. By summing forces in the y-direction, the dynamic force on each slanted strut is calculated from $F_{sD}*\cos(\alpha/2) = V_{i-1}+w*L_i = V_i$. This dynamic force places the strut in compression for struts slanting downward and in tension for those slanting upward. Since the static pressure places the slanted struts in compression, the maximum strut stress is felt when they are slanting downward and is $F_s = F_{sD} + F_{s\ s.p.} = V_i/\cos(\alpha/2) + P_s*b*h*\tan(\alpha/2)/\cos(\alpha/2)$. The slanted struts were sized according to this force. As for the horizontal struts, for each truss section, a moment sum can be done around one of the nodes running along the upper strut (a), from which $F_{bD} = M_i/h = [M_{i-1} +V_{i-1}*L_i +w*L_i*(L_i/2)]/h$. This places the lower horizontal strut b in compression, but the static pressure places it in tension, so the two effects are subtracted. The limiting stress for the horizontal struts will be the upper strut, strut a, which is in tension from both static and dynamic forces. For strut a, an x-direction force balance shows the maximum dynamic force is $F_{aD} = F_{bD} + F_{sD}*\sin(\alpha/2) = M_i/h + V_i*\sin(\alpha/2)/\cos(\alpha/2)$. Adding the static pressure effect, $F_a = M_i/h + V_i*\sin(\alpha/2)/\cos(\alpha/2) + F_{s\ s.p.}*\sin(\alpha/2)$. This is the maximum tensile force felt by the horizontal struts, and determines the strut size for both upper and lower horizontal struts, since the subraft is isotropic. As with the minor trusses, the major trusses were found to require about 20% more material than the I-beam design. The Neville truss design used here is primarily for calculation of the material requirements. In actual practice, such a truss would be cut down its centerline, and half of it inverted, forming, in effect, a Lattice truss when the two halves are bolted back together.

Cylinder calculations

The cylinder has two different kinds of forces transmitted to it from the rest of the subraft. To optimize weight and cost, the cylinder incorporates two solutions for these forces. The first force is the torque, T, by the subraft under dynamic loading. The cylinder itself is the optimum shape to resist this torque so was sized to resist it, using $S_y \Rightarrow \tau_{max} = T*r_o/J$, where $J = (\pi/2)*(r_o^4 - r_i^4)$. The second force is the shear force of the subraft, transmitted to the cylinder. A truss, inserted into the hollow cylinder, was designed to resist this shear loading.

Deltawave is sized to meet three design conditions. The first is normal operations, in which the subrafts are closed and pressed from underneath by the ocean swells. The swell pressure is transmitted to the cylinder as both torque and distributed shear loading.

The second is a storm wave condition, in which the subrafts have deployed to the open condition, and a monster wave is surging water past them from below.

In this condition there is no torque on the cylinder, but there is significant drag loading on the cylinder by the water being deflected around it. In this condition, the cylinders internal truss is useless, because it is improperly aligned to resist this loading. However, because of the triangular shape of the subraft itself, and its construction, composed of beams radiating out from the free apex, the subraft itself acts as a truss in this condition.

The third design condition is a storm wave condition, in which a subraft gets stuck in the open position while a storm wave surges horizontally past it. In this condition, the water pushes on the lower side of the subraft, leading to both torque and shear loading on the cylinder, in a manner similar to the 'normal operations' condition.

The internal truss is a Neville truss, with repeating isosceles triangles of internal angle α. The shear force is maximum at the edges of the cylinder, where the cylinder is supported by the ball joint housings. In equation form, the slanted strut inline force is $F_s = 0.5*w*L*(1-f)/\cos(\alpha/2)$, where $w*L$ is the shear force of the subraft on the cylinder, which has length L. This is highest at the edges of the cylinder, where f=0 (f=1 at the center of the cylinder).

The bending moment on the internal truss is the sum of the moment due to cylinder shear force, plus the additional moment transferred to the cylinder by the ball joint housing (to which it is attached) and minus the amount of this bending moment that can be resisted by the cylinder itself. This moment sizes the horizontal struts. There are two horizontal struts, top and bottom (F_a and F_b). They are both sized according to $\max(F_a,F_b) = [(M_{bj} - M_{cyl} + w*L^2/8)/h]*(2f - f^2) + F_s*\sin(\alpha/2)$. In this equation, M_{bj} = the additional moment from the ball joint housing, which is calculated below. M_{cyl} is the moment on the cylinder/truss assembly that can be resisted by the cylinder alone, where $M_{cyl} = (\pi/2)*(r_o^4 - r_i^4)* S_y/r_o$. This is derived from $\sigma_x = M_{cyl}*r_o/I_z$, where $\sigma_x = 2*S_y$ and, for a hollow cylinder, $I_z = (\pi/4)*(r_o^4 - r_i^4)$.

Cylinder posts, ring-bearings and tracks

The sub-raft connects to the cylinder via six ring-bearings, which allow the sub-raft to rotate around the cylinder under monster wave conditions. The inner race of the bearing connects to the cylinder, and the outer race to the sub-raft, and there are 19 roller bearings in-between the two races. The outer race connects to a track upon which a spring-loaded pin runs, and this track is bound to the subraft. This spring-loaded pin holds the subraft in place. This pin is designed to only be compelled to compress inward, into the cylinder, by monster waves, thus allowing the subraft to rotate only in extreme conditions. There is one pin per bearing, so six altogether. These are sized to handle the extreme moment loading (T/6) coming from the subraft. Two pin cross-sections were considered: a solid cylinder, and an I-beam. Because the loading is coming in a single plane, the I-beam shape was found to save considerable material, and was selected.

Ball Joint Housing

The ball joint housing has a complex shape. For structural purposes it was modeled as three square tubes, each of the same length, but of varying widths. The lengths and widths are selected so that the overall area of the housing is the same as its actual area, which influences the overall force and moment of the housing on the rest of the subraft. The first square tube is the housing directly in contact with the ball joint. Along with its two flanges (top and bottom), this first tube has two webs, on either side. The second square tube has three webs, one center web and two side webs. The third square tube, the one in direct contact with the two subraft cylinders welded to the ball joint housing, has four webs, two webs in the center, and two at the side. The square tubes are sized to handle the moment and shear loading of the two subrafts in contact with the ball joint housing. Each subraft is in contact with two ball joint housings, however. Therefore, the net force on a ball joint housing is equivalent to the pressure on just one subraft, $F_{cyl} = A_{subraft}*P_D$. This subraft force, when added to the dynamic water pressure on the ball joint housing itself, $F_{bjh} = A_{bjh} *P_D$, is the force on the ball joint: $F_{bj} = F_{cyl} + F_{bjh}$.

Structural calculations similar to those with the I-beams were performed to size the ball joint housing and estimate its material requirements. In estimating the wall thickness of the cubes, the maximum normal stress was taken at the extreme upper and lower edges of the cube, at $c = (w+2*t_f)/2$, and shear stress was taken at the top of the web, at w/2. Shear stress can also be estimated at the web centerline if requested. As before, the design requirement is to keep $\tau_{max} < S_y = 0.577*36000$psi for A36 steel, where $\tau_{max} = \text{sqrt}\ \{[(\sigma_x - \sigma_y)/2]^2 + \tau_{xy}^2\}$. Choosing the x-direction along the cube length and the y-direction along its height (i.e. along the web of the square tubes), then $\sigma_x = M_i*c/I_{zi}$, $\sigma_y = - P_s*f_i/t_{wi}$, and $\tau_{xy} = V_i*Q_i/(t_{wi}*I_{zi})$, where $I_{zi} = t_{wi}* w^3/12 + 2*\{f_i*t_f^3/12 + f_i*t_f*[(w+t_f)/2]^2\}$, and $Q_i = f_i*t_f*[(w+t_f)/2]$, where w = web height = height of the ball joint, $c = (w+2*t_f)/2$, f_i = flange width, which varies with each of the square tubes i, and t_f = flange thickness. The web thickness, t_{wi}, is the total for all webs (remember there are two webs for the first tube, three for the second, and four for the third), i.e. $t_{wi} = \{\#\text{ of webs}_i\}*t_f$, where, for simplicity, each individual webs thickness is assumed = t_f. The goal of the strength calculation is to determine t_f for each of the square tubes i.

Each of the three tubes has the same length, i.e. $L_1=L_2=L_3=L_{tot}/3$, where L_{tot} is the total length of the ball joint housing. Each tube has a dynamic water pressure force: $F_{wi} = A_i*P_D$, where $A_i = L_i*f_i$. The shear force, V_i, for each of the tubes is: $V_1 = F_{bj}$ (calculated above), $V_2 = V_1 – F_{w1}$, and $V_3 = V_2 – F_{w2}$. The maximum moment for each square tube is $M_i = M_{i-1} +V_i*L_i -F_{wi}*L_i/2$ (where $M_0 = 0$). These shear forces, V_i, and moments, M_i, are used above to estimate each of the square tubes thicknesses, t_f, from which material and cost estimates are based.

Square tube 3, which is welded to two subraft cylinders, transmits its moment to each of them, in amount $M_3/2$. This is used in sizing the cylinder in the discussion above, i.e. $M_{bj} = M_3/2$. Tube 3 also experiences a torque from those cylinders when they are under load. This torque is equal and opposite from the two subrafts and is resisted by two extra metal plates placed over the top and bottom flanges of square tube 3. The plates are subject to normal stress loading, and the thickness of each is calculated by $t = F/(2*S_y*L_3)$, where $F = T/(w+2*t_f)$, and T is the torque on the housing induced by each subraft.

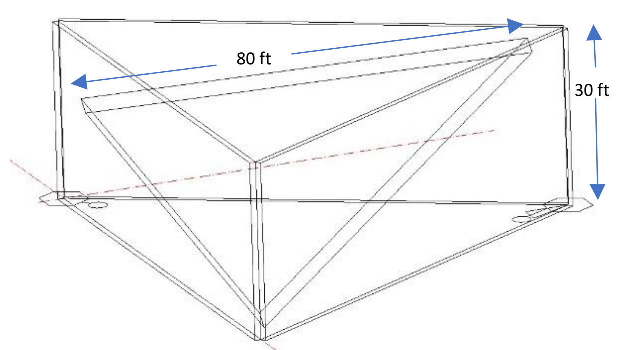
Ball joint sizing

Between the ball joint and the ball joint housing is a layer of Teflon as the bearing interface material. Teflon has a yield strength of 3000 psi, and the ball joint is sized, in part, to accommodate this strength, given the force, F_{bj}, passing across the ball joint. The ball joint is composed of 0.2" thick steel covering 70% of the area of a sphere (the back of the ball joint is open). Inside the ball joint are four plates, vertically oriented to pass the vertical force through the ball. The joint is inside a casing, also of 0.2" thick steel and 70% of the spherical area. In between the joint and the case is the 0.25" thick Teflon interface. The casing is connected to the top and bottom flanges of the first ball joint housing tube via eight vertical plates, four below the ball joint and four above it. The various vertical plates have thicknesses calculated based on the maximum loading, F_{bj}.

Purpose of flexibility in the ball-joint, hinge-arm, and hexaxle design

The rafts are tossed by waves such that their three apices have a variety of heights relative to each other. For the pistons and water-brakes to remain in a constant horizontal location regardless of wave height, flexibility is built into the connectors from the ball joints at the raft apices to the hex-axle center. In this way, deltawave is designed to accommodate waves of up to 30 feet without various parts colliding. Figure A4 shows what an 80ft raft looks like with its apices located at different heights in a 30 ft swell.

Figure A4: Raft with 3 apices at different heights



Each raft has a 2 ft clearance to the edge of its 80 ft envelope. This means the raft apex is 2/sin30 = 4 feet inside of the envelope apex, and the raft edges are 80-4*cos30 = 73.1 ft long. In a 30 ft swell, direct measurements from the CAD model suggest the maximum distance from the raft apex to the envelope apex is 8 feet (in a 20 ft swell, this maximum distance is 5.7 ft). After accounting for the diameter of the ball joint and its minimum distance to the hinge, this means the hinge arm requires an extensibility of 3.9 ft, in a 30 ft swell. This extensibility requirement was increased by 20%, to 4.7 ft, for conservatism. The hinge arm is composed of three sections. Each section is a pair of doors, and each pair slides within the pair external to it. With each door being 3.4 ft long, and the required extension per door being 4.7 ft/2, the minimum overlap between doors is 1 foot. The design challenge is to design roller bearings and railings that meet these geometrical restrictions while transmitting the vertical force involved across the hinge arm. To prevent jamming, the ratio of minimum door overlap to door height was chosen to be 20%. Hence, the hinge arms are 5 ft tall. The door thicknesses were estimated using the maximum force calculated above. The hinge arm doors closest to the hexaxle hinge have the largest moment arm from the ball joint, so have a combined thickness of 2 inches. The second pair of doors have a combined thickness of 1.5 inches, and the third pair a combined thickness of 1 inch.

The ball joint housing iris, through which the hinge arm attaches to the ball joint, must not be too large, or the ball joint housing cannot be strengthened sufficiently to prevent the ball joint from being pulled through the socket. This also relates to swell size. In a 30 ft swell, the maximum vertical angle of the ball joint is $27.8^o = \sin^{-1}(30ft/64.4ft)$, where 64.4 ft is the distance between two ball joint centers. The maximum horizontal angle depends on the maximum raft apex to envelope apex distance, which is 8 ft given above, and the radius of the ball joint (1.5 ft) and distance from the raft apex to the edge of the ball joint (3.5 ft). The maximum horizontal angle is $25.4^o = \tan^{-1} \{8*\sin30/[(8*\sin30-3.5)+(3.5+1.5)]\}$. The maximum ball joint angle in its socket is therefore $37^o = \cos^{-1}(\cos27.8*\cos25.4)$. The hinge arm connects to the ball joint through a rod with a 0.56ft radius, which adds 20^o to this, so the actual half-angle of the ball joint iris is 57^o. There is thus $33^o = 90^o-57^o$ of socket overlapping the ball joint on all its edges, an area sufficient to ensure strengthening will prevent the socket from being damaged in large waves.

<u>Hinge arm sizing.</u>

Figure 5 shows the arrangement of the hex-axle. The hex-axle is six sided and at each of its apices is a hex-hinge. The hex hinge allows rotational flexibility in one dimension only, much like a door hinge.

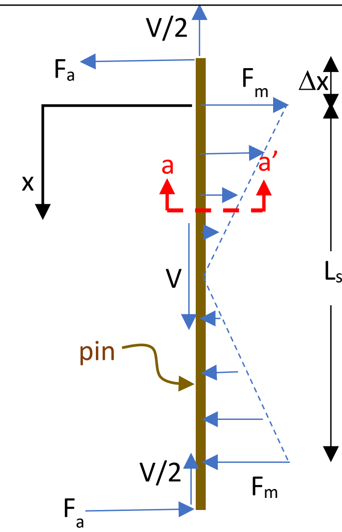
Each hex-hinge is connected to a hinge arm which in turn connects to a ball joint. See Figure 6. The hinge arm must expand from a minimum to a maximum length (currently 3.4ft to 8.2ft), while transmitting vertical loads from the ball joint to the hinge. To perform this function, the hinge arm is composed of three pairs of sliding doors on runners. The minimum overlap between the doors is 1ft, so the maximum length is 3.4ft + 2*(3.4ft-1ft) = 8.2ft. To prevent jamming of the doors, the minimum overlap per unit height of the doors is kept above 20%, so the doors are 5ft high (h =1ft/0.20). The thickness of each pair of doors is dependent on the maximum moment felt by that pair, which is the maximum force at the ball joint multiplied by the maximum lever arm of the extended pair, $M_{max\ i} = F_{bj}*L_{pair\ i}$. The door thickness was calculated from $S_y > 0.5\sigma_x = 0.5[M_{max\ i}*(h/2)/I_{zi}]$, where $I_{zi} = t_i*h^3/12$. Thus $t_i = 3M_{max\ i}/(S_y*h^2)$. This is for each pair of doors, so each door is half this thickness. There is also shear loading on the doors, that is maximum at the centerline. It was found that this was not the limiting stress situation, so the thickness calculated here is conservative.

To protect the hinge arm and ball joint from the ocean environment it is likely that a rubber boot will be needed to surround these parts and flex with them as they move. This part has not yet been designed.

<u>Hex hinge sizing</u>

Each hexaxle has 6 hinges. The hinge pin is sized to handle shear and normal loading transmitted to it from the hinge arm. Assuming the pin is restrained at its two endpoints by the hexaxle structure, the normal loading at each end is half the hinge arm shear load V. The normal stress, σ_x, is this loading divided by the pin area, $\pi D^2/4$. The maximum shear stress is due to the Moment of the hinge arm and is $\tau_{xy} = F_M Q/(t*I_z)$, where $F_M = M/L$, L is the length of the pin (6ft), $Q/t = (D^3/12)/D$, and $I_z = \pi D^4/64$. By keeping $S_y > \tau_{max}$, where $\tau_{max} = \text{sqrt}\ \{[(\sigma_x - \sigma_y)/2]^2 + \tau_{xy}^2\}$, and $\sigma_y = 0$, then $D^4 => [V/(\pi S_y)]^2 + [16F_M/(3\pi S_y)]^2$.

Figure A5: Forces on the hex hinge pin



The pin diameter is sufficient to withstand loading at the endpoints of the pin, however, the maximum moment on the pin occurs internal to the endpoints, and at that location the pin diameter must be jacketed by the hinge stud so that their combined diameter is sufficient to withstand the moment at that location. The hinge pin loading has the profile shown in Figure A5. At the endpoints, the hexaxle structure imposes two forces, F_a. The distributed loading in the center (F_m) is from the hinge arm. The vertical loading from the hinge arm is V. From the top of the pin, the moment at location a-a' is $M_{aa'} = F_A(x+\Delta x) - \{F_m*x*(x/2) - 0.5(F_m* - F_{m@aa'})*x*(x/3)\}$, where $F_{m@aa'} = F_m*[1-x/(L_s/2)]$, and L_s is the height of the hinge stud. Thus $M_{aa'} = F_A(x+\Delta x) + [F_m/(6L_s/2)]*x^3 - (Fm/2)*x^2$. The maximum moment, M_{max}, along the pin length is where $dM_{aa'}/dx = 0$, and the pin+stud casing diameter, at this location, is calculated from $S_y > 0.5\sigma_x = 0.5[M_{max}*r/I_{zi} + (V/2)/(\pi D^2/4)] = 16M_{max}/\pi D^3 + V/\pi D^2$. The calculated diameter at this location is about 60% greater than the diameter at the pin endpoints. With the extra jacketing of the hinge stud, the pin will withstand this maximum moment load.

<u>Hexaxle structure sizing</u>

The hexaxle is composed of a hexagonal structure holding at each of its apices a hex-hinge. The hexaxle has a maximum diameter. This is currently set to 7 feet. This means each of its six spoke panels, and six side panels, has a length, $L_{spoke} = 7/2 = 3.5$ ft. The hexaxle has the same height as the hinge, L (=6 ft). To estimate the material in the hexaxle, the spoke panels were sized for their loading, and the total material doubled to account for extra material in the side panels and in the center rod, which is connected via a ball joint to the connecting rod.

If the spoke panels are I-beams, their thickness (t) is calculated from $S_y > 0.5\sigma_x = 0.5M_{max}(L/2)/I_z$ = where $M_{max} = M + L_{spoke}*V$, where M and V are the moment and shear at the hinge, as calculated above. Since $I_z = t*L^3/12$, thus $t > 3M_{max}/S_yL^2$. The calculation was repeated for maximum shear in the I-beam, but the thickness is 1/4th the thickness calculated for moment loading.

The spoke panels can also be simple trusses, composed of a frame and one diagonal. In that case the moment, M, impacts the top and bottom struts by the couple composed of two horizontal forces, F_a = M/L (also discussed above). Including the vertical shear force, V, the three major struts (top, bottom, diagonal) can be sized according to their expected loading. This was then doubled to account for the material in the side panels, and the center rod of the hexaxle.

It was found that the structural material requirement of the hexaxle when composed of trusses was half the requirement when composed of I-beams, so this is the structural architecture selected.

<u>Connecting rod and pistons</u>

The connecting rod is sized to handle the dynamic water loading on the hex-axle and the six rafts surrounding it (as transmitted to the hex-axle through the six ball-joints it is in contact with), that is: $F_{cr} = 6*F_{bj} + F_{ha}$ (the hex-axle has an area of 32 ft^2 which is also pressed up by the dynamic water loading). Currently, the connecting rod is designed to be cylindrical with a diameter (D_o) of 3 ft, and to be 150 ft long. It connects to the hex-axle through a ball-joint, allowing the hex-axle to pitch in various directions with wave action. The rod thickness was calculated from $2*S_y > \sigma_x = F_{cr}/A_{cr}$, where $A_{cr} = (\pi/4)*(D_o^2 - D_i^2)$.

Between the connecting rod and the waterbrake is the power conversion chain (PCC), which converts the relative motion between them into energy, which is then transmitted to shore via a cable. Two different systems were evaluated: a piston pump system that outputs pressurized seawater and a more traditional PCC that converts the relative motion into hydraulic power that turns a turbine driving an electric generator. The second system was only evaluated for the economic analysis, allowing a comparison with competing marine energy technologies.

Herein are the calculations used to size the seawater pressurization system: In sizing the pistons, the average upward force of a wave on an object trying to constrain it is needed. It was found earlier that the maximum force for a 20ft wave, constrained to 70% of its free height, is 9,235.2 kN, and the average force throughout the upward half of the wave is half of this, or F_y = 4,618 kN. Assuming the seawater is pressurized to p = 459psi (which corresponds to a water column of 1000 ft), the total piston area required is $A = F_y/p$. Assuming there are $\#_{pumps}$ pumps capturing this upwardly directed energy, the diameter of each calculated from $A = \#_{pumps}*\pi D^2/4$. Currently, $\#_{pumps}$ =6, so D =1.8ft. Once the diameter and number of pumps are assigned for the maximum wave height, the number of pumps engaged with lower wave heights will be brought down, to keep the pressure constant. For example, a 4 ft swell would only engage 1 pump, a 6 ft swell 2 pumps, and a 13 ft swell 4 pumps, etc. This can be done mechanically.

Similarly, the downstroke of the connecting rod can also pressurize seawater. Here the downward force is that of the weight of the raft assembly per hex-axle. Currently, the single piston assigned to capture this energy has a diameter of 2.7 ft, to produce seawater at 459psi.

The pump cylinder wall is sized to handle the hoop and long stresses that occur under pressure, i.e. such that $S_y => p*D/(4*t) + (p/2)$, where p = internal gage pressure of 459psi multiplied by a safety factor, which is currently 1.8, and t is the cylinder wall thickness. This was checked against the axial stress that occurs in the cylinder when the piston tops out under a monster wave. In that case the pressure is zero, but the axial stress is large. The design case is the one under pressure. Similar calculations were made to size the pump pistons.

The pump assembly would be surrounded by floatation tanks, such that its buoyancy would be neutral.

Waterbrake sizing

The rafts pull upward on one end of the connecting rod. The other end is anchored to the waterbrake. The waterbrake has the shape of an upside-down umbrella. Each umbrella has six spokes, as shown in Figure 8. The slanted spokes are supported by six horizontal braces. The spokes are overlain by a tarp material bonded to a fish-net-like support. Each umbrella-like assembly also opens and closes like an umbrella, so that the waterbrake and connecting rod can be pulled out of the ocean without dismantling neighboring rafts. In the current design, each waterbrake has two layers of umbrellas, a top and bottom (N_L=2).

The brake is subject to both upward and downward forces within a single wavelength. However, the arrow-head shape of the umbrella means it will move downward with greater ease. When forced upward, the umbrella forms a 'cup' shape that encloses a certain volume of water that will have to be accelerated to allow the brake to move upward. When forced downward, water easily deforms around the umbrella. On average, the water is not moving vertically, since the waterbrake moves up and down repeatedly in one wave period. This means that the upward motion of the raft, as a wave swells under it, is resisted by the weight of the water trapped in the 'cup' shape of the umbrella, water that has no initial vertical velocity. In a 20 ft swell, the average force on the waterbrake is half the maximum force, i.e. F_{avg} = 1038 klb_f. The umbrella 'cup' has a volume, V = L^3*tan(η), where L = (80ft/2)*cos(30°) and η is the vertical angle of the spokes on the umbrella (currently η = 26°). Thus V = 20.3 kft^3. With two layers of umbrellas per waterbrake, the mass of water that must be accelerated by F_{avg} is M = 2*V*62.4 lbm/ft^3 = 2532 klb_m. The average acceleration experienced by this water is F_{avg}/M/32.2 = α = 0.013 ft/s^2. For a 20 ft wave, the period is 11 sec in deep water, so from trough to crest takes t = 5.5s, and in this time the waterbrake moves a vertical distance d = 0.5*α*t^2 = 0.2 ft, or about 1/100th the vertical distance moved by the hex-axle, which is 20 ft. This ensures that the waterbrake is properly anchoring the raft in the ocean.

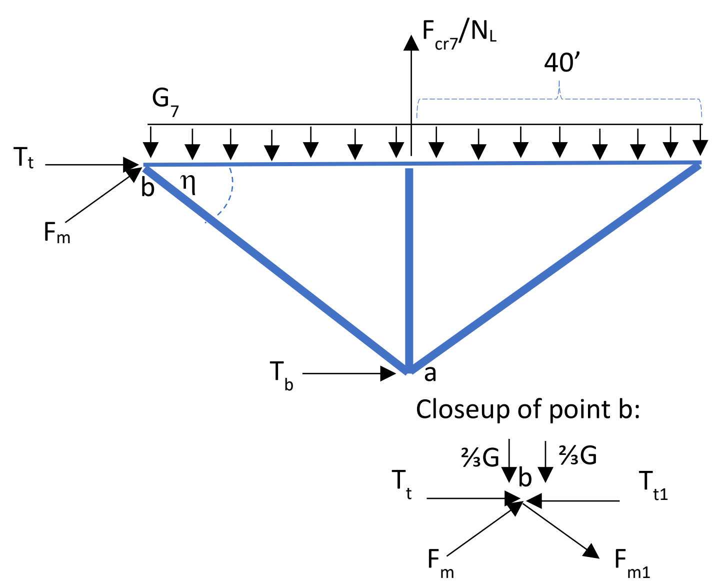
The pressure on the waterbrake tarp material is F_{cr}/N_L, where F_{cr} is the upward force on the connecting rod, and N_L is the number of umbrella layers in the waterbrake. There are six spokes per umbrella, so the force G on each spoke is G = $F_{cr}/N_L/6$. The downward pressure p_r on the tarp material is $F_{cr}/N_L/A$, where A = 6*L_{spoke}*(L_{spoke}*cos30°). L_{spoke} is the horizontal length of each spoke, which currently is 40 feet. Another way to calculate the force on each spoke is G = p_r*(A/6). Each spoke member is supported at its two ends, but the pressure force G is not split evenly between them. The tarp over each spoke has the approximate shape of an equilateral triangle, meaning the pressure force per unit length, w, along the spoke increases as you travel from the inner end to the outer end. For this triangle of inner angle 60°, w = F/Δx = $p_r A_x/\Delta x$ = $p_r L_x \Delta x/\Delta x$ = p_r1.155x. By summing the moment at the inner end, its determined that 2/3rds of the pressure force falls on the outward spoke end, and the remaining 1/3rd falls on the inner spoke end. Since there are 6 spokes, all meeting at their inner ends, the pressure force at that inner joint is 6*⅓*G = 2G.

By summing the forces on the central pin of the umbrella, the Force on the spoke, F_m, can be calculated using F_{cr}/N_L = 2G + 6$F_m$sin(η). The spoke is in compression on the upstroke of the connecting rod. The horizontal brace (T_t) is in tension, calculated by T_t = F_m cos(η). The brace cross-sectional area A = $T_t/(2S_y)$. As before, S_y is the material shear limit. This is likewise used to estimate the cross-sectional area of the spoke. However, while the brace is only subject to axial loading, the spoke is also subject to bending loads, because the tarp presses down throughout its length. If the spoke has length P, it is determined that the maximum moment from this bending load is at 0.58P, with value M = 0.1283GP. This bending load is added to the axial load to determine the required spoke size: 2*S_y = σ_{max} = Mc/I + F_m/A (where c = h/2, h is the height of the beam). If the beam is rectangular (height h, base b), then h as a function of b is h = {F_m + sqrt[F_m^2 + 4(2S_yb)(6*0.1283*GP)]}/(4S_yb). For an I-beam an iterative solution is required. This indicates an I-beam would use 80% less material than a rectangle, so is the shape of choice for the spoke. The brace should probably also be an I-beam, but since it theoretically only endures axial loads, a rectangle is currently baselined.

The calculation above assumes each waterbrake is working alone to resist the movement of its associated connecting rod. This design simplifies insertion and removal of waterbrakes, since they are not connected to each other, except vertically in pairs. However, another option is to shift this load partially to neighboring waterbrakes, which reduces their structural requirement overall. This requires that, after deployment, the brake spokes can make a structural connection. Once made, the waterbrake assembly is like a single huge underwater truss structure and more capable of resisting localized loads along its surface. However, to make and unmake that structural connection without diver support, which would be expensive, the brake spokes need to make it automatically, with self-guiding structures.

For this second case, the tarp pressure is lower because it is being spread among surrounding waterbrakes. Instead of one waterbrake per level resisting the connecting rod's force, there are seven per level. That's an approximation, of course, but the waterbrakes most likely to be engaged when one connecting rod is pulling upward, are the central waterbrake, and the six surrounding waterbrakes, so for the purposes of designing the structure, this is the assumed engagement. Therefore $G_7 = (^1/_7)*G_1$. As before, summing vertical forces about the central pin of the umbrella, $F_{cr}/N_L = 2G + 6F_m\sin(\eta)$. Since F_{cr} is the same as before, and G is much lower, F_m is higher than before. The surrounding umbrellas are symmetrical in their responses, so analyzing one of them is appropriate for all six. Figure A6 shows how one of these umbrellas is impacted by F_m, and T_t, as well as by the other forces on the umbrella. Note that, because it is assumed only the surrounding 6 umbrellas are involved in the response, and due to symmetry between them, these are the only forces present. F_{cr7} is the force on the neighboring umbrella from its connecting rod, and T_b is the T_t force from the second row of waterbrake umbrellas. Taking the moment about point 'a', $0 = F_m\sin(\eta)*40' + F_m\cos(\eta)*40'\tan(\eta) – T_t*40'\tan(\eta)$, from which $T_t = 2F_m\cos(\eta)$, which is twice the value for the single case. Summing vertical forces about point 'b' yields $F_{m1} = F_m - (^4/_3)G/\sin(\eta)$. As mentioned previously, ⅔G is the water pressure force on the umbrellas concentrated at the outer end of the spoke. Summing horizontal forces about point 'b' yields $T_{t1} = T_t - F_m\cos(\eta) - F_{m1}\cos(\eta)$. Then, considering the horizontal forces over the umbrella overall, $T_b = T_t - F_m\cos(\eta) = F_m\cos(\eta)$.

Figure A6: Forces on the waterbrake



For the lower layer, the primary umbrella has the same forces on its struts as the upper layer did. However, for the surrounding 6 umbrella's the forces are slightly different. To evaluate the forces, use the same diagram from Figure A6. All the forces are subscripted to indicate they are for the bottom layer, and there is one additional horizontal force, which is the reaction force to T_b: same magnitude but opposite direction, located at the center of the six horizontal braces. Meanwhile, the force T_b shown in figure _ is now $T_{b②}$. Note that $F_m = F_{m①} = F_{m②}$ since the connecting rod's force is split between the two waterbrake layers and the force of water-pressure, G, is equal everywhere. Taking moments about point 'a', $0 = F_m\sin(\eta)*40' + F_m\cos(\eta)*40'\tan(\eta) - T_{t②}*40'\tan(\eta) - T_b*40'\tan(\eta$, from which $T_{t②} = 2F_m\cos(\eta) - T_b = F_m\cos(\eta)$. As before, summing vertical forces about point 'b' yields $F_{m1②} = F_m - (^4/_3)G/\sin(\eta)$, hence $F_{m1②} = F_{m1①}$. Likewise, $T_{t1②} = T_{t1①}$ since, as before, $T_{t1②} = F_{m1②}\cos(\eta)$. Summing horizontal forces about the lower umbrella, $T_{b②} = T_b + T_{t②} - F_m\cos(\eta) = F_m\cos(\eta)$.

Joining the waterbrake umbrella's into one connected unit does not substantially change the axial forces felt by the structure, however due to the much lower water pressure (G), the bending force felt by the strut (upon which the tarp is lain) is much lower, and where material savings is realized. Recall that, for a rectangular beam (height h, base b) as the spoke shape, $h = \{F_m + \mathrm{sqrt}[F_m^2 + 4(2S_yb)(6*0.1283*GP)]\}/(4S_yb)$. Thus h should be $\mathrm{sqrt}(^1/_7)$ as high as before, or roughly 40% the previous height (for the same base b). This 40% material savings remains true for other shapes, such as an I-beam. Due to the increased material required for the braces (T_t), however, the advantage of joining the waterbrakes falls to a 23% material savings, and therefore may not be worth it.

The waterbrake umbrella is covered in a layer of tarp material underlain by a nylon fish-net. Each umbrella is composed of six triangular tarp sections, each connected at its edges to the spokes (F_m). Each triangular tarp has the water pressure, G, on its surface. The force on the tarp is highest at the edges, where they connect to the spokes, and the tension per unit length along the edge of the tarp can be calculated from $Ft/l = (G*d/2)/\sin(\delta)$, where d = the distance across the tarp from one spoke to the other, which increases along the spoke as you travel from the center of the umbrella to the edge, and δ = the angle the tarp makes with the horizontal at the spoke, which is currently considered to be 15°. From this, given the yield strengths of the tarp material and the netting, the number of netting layers could be calculated. This number increases as the overall force increases from the umbrella center to the edge. The calculation was repeated using simple nylon rope as the backing material, with a substantial cost savings. The ideal solution may involve, for backing material, a combination of netting and rope.

Pipeline sizing

The power conversion chain (PCC) discussed in this Appendix involves capturing the wave energy as pressurized seawater, using an array of piston pumps. The pressurized water is then piped to land, where further power conversion occurs, ending in electric current. A 4 ft diameter seawater pipeline, 1 mile long, about 100 ft below the surface of the ocean, was considered. The large diameter helps keep internal frictional losses low. As with the pistons, the pipeline wall is sized to meet hoop and long stresses, so that $S_y \Rightarrow p*D/(4*t) + (p/2)$, where p = internal gage pressure of 459psi multiplied by a safety factor, which is currently 1.8, and t is the wall thickness. As before, consider $S_y = 0.577*F_y$, where F_y = Yield strength. A36 steel has a yield strength of 36,000psi, so the thickness of steel needed would be 0.5 in. Polypropylene has a yield strength of 7,450psi, so the thickness needed would be 2.6 in. Currently, the polypropylene option is baselined due to lower cost, corrosion resistance, and neutral buoyancy. In the case of polypropylene, or nylon, strength would be achieved through a series of windings of various sizes of tarp material, netting, and rope. This pipeline would be suspended in the ocean by a series of floats. The pipeline is close to neutral in buoyancy because both inside and outside the pipe is water, which is incompressible, and if the pipe walls are made of plastics, they will be neutrally buoyant as well. The advantage of a split PCC is that electric generator parts that require more maintenance are located on land, but this option becomes prohibitively expensive if the WEC is located more than a mile out to sea. Although this mechanical analysis discusses the split PCC option's mechanical parts, like the pipeline, the economic analysis assumes a more traditional PCC for Deltawave, in which the motive power is input to a hydraulic-fluid based loop driving an electric generator locally, next to the WEC. The electricity is then cabled to shore. A HVDC cable can carry electric power many miles. For most locations, the deep water capable of transmitting unattenuated open-ocean waves to the WEC is located more than 5 miles offshore.

References for Appendix A:

1. 'Bangladesh National Building Code 2012: Chapter 12: Ferrocement Structures'. June 2012.

2. 'Rectangular Plate Uniform Load Simply Supported Equations and Calculator'. ©2000-2024, Engineers Edge, LLC.

Appendix B: Economic Analysis

The LCOE (levelized cost of electricity) is the amount a company would have to charge customers for the electricity from their device, and still make a modest profit. 'Methodology for Design and Economic Analysis of Marine Energy Conversion (MEC) Technologies' (Ref 3) was used to develop an estimate of Deltawave's LCOE. Equation 2-1 in that paper is LCOE = (CapEx*FCR + OpEx)/AEP, where LCOE is the levelized cost of electricity, CapEx is capital expenses (first costs), OpEx is operating expenses (annual costs), FCR is the fixed charge rate (what capital expenses cost annually, as a fraction), and AEP is annual energy production, in kWhr. FCR is a value provided by DOE for MEC-type technologies and includes the effect of rate-of-return, depreciation, inflation, tax rates, and other key financial variables. An FCR of 11.3% was used in Ref 3 on the four MEC's studied there, RM1: tidal turbine, RM2: river turbine, RM3: buoy wave-energy-converter (WEC), and RM4: ocean current turbine.

"Reference Model 5 (RM5): Oscillating Surge Wave Energy Converter" (Ref 4) was also used. It discusses specifically the economic analysis of a WEC designed to be operated under the same conditions as Deltawave. Most MEC's tap a steady or slowly varying hydrological flow, like a tide or ocean current, but WEC's requires structures unlike other MEC's, because the energy resource is the oscillations at the interface of two fluids of very different densities in a gravity field. The resisting structures of WEC's result in comparatively high structural costs. Note that Ref 4 used a more current FCR of 10.8%, and this was also the FCR used in this Deltawave analysis.

For the economic analysis, a Deltawave WEC composed of 20 linked hexrafts is considered. A hexraft is six triangular rafts (i.e. eighteen subrafts) arranged in a hexagon. The hexrafts in this WEC are linked 10 across and 2 deep. It's assumed the entire WEC can be pivoted around one end to match the prevailing wavefront, if there is any, and thus maximize power production.

The width of a WEC that is 10 hexrafts wide is W = 10*80ft*cos(30^o) = 1386 ft. The annual average wave resource is largest in the Pacific Northwest, so an incoming wave power per wave width of 9.1 kW/ft will be used, to match conditions there. Note that the annual average resource in California is 6.5 kW/ft and is below 2 kW/ft in most other locations in the United States.

The incoming wavefront is, for this WEC, subjected to a depth of 2 hexrafts, which is about four hexaxles. Calculation indicates that for most waves the mechanical absorption efficiency of Deltawave will be over 70%. The efficiency of turning this into electricity in the PCC is assumed to be 85%, so the overall efficiency of power conversion is 70%*85% = 60%.

For MEC's the 'device availability', that is its useful life outside maintenance, is usually assumed to be 95%. And the efficiency of electricity transmission is assumed to be 98%. Therefore, the annual energy production of Deltawave is AEP (in kWhr/year) = 9.1kW/ft*1386ft*60%*98%*95%*8766hrs/yr = 6.2x10^7 kWhr/yr. Since this AEP is similar to that produced by a 100-unit array of the other two WEC's being compared to, their 100-unit cost estimations can be used for Deltawave where other cost data was lacking (for example, in estimating cabling costs).

Figure B1 can be referred to in the discussion that follows. It highlights specifically the costs associated with RM3, RM4, and RM5. That is the two WEC's and the ocean current turbine MEC. References 3 and 4 examined these costs for a variety of power plant sizes. Generally, costs fall dramatically with power plants composed of more devices. Figure B1 summarizes power plants composed of 10-units and of

100-units. The Deltawave WEC under study here produces 88% of the AEP of the 100-unit option of RM3, the buoy WEC, and 70% of the AEP of the 100-unit option of RM5, the flap WEC.

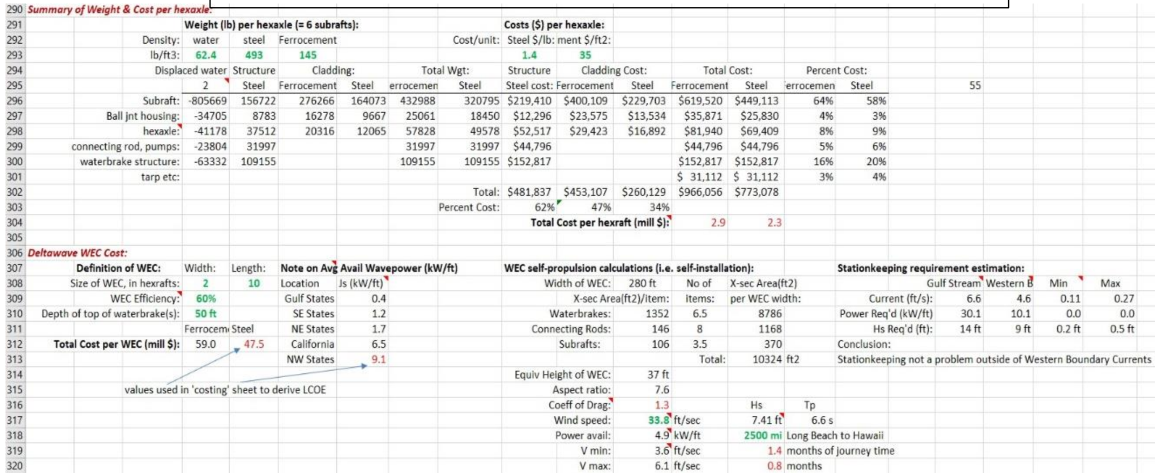
Figure B1: Costs for RM3, RM4, RM5, and Deltawave

1			RM3				RM4				RM5			Deltawave	Mooring Option:				
2			Buoy WEC			Ocean current turbine					Flap WEC			20 hex-raft WEC		0	0=no moorings, 1=moorings		
3		10 units		100 units		10 units		100 units		10 units		100 units		20 units					
4																			
5	wave pwr/ft:	10.2	kW/ft	10.2	kW/ft					9.1	kW/ft	9.1		9.1	kW/ft		power/ft	Ave wavesize	
6	unit width	65.6	ft	65.6	ft					82	ft	82		1386 ft	ft	0.26 mi	kW/ft	ft	
7	Wave Pwr	670	kW	670	kW					750	kW	750		12673	kW		9.1	9.1 ft	
8	Power out	85.8	kW	85.8	kW	2800	kW	2800	kW	108	kW	108		7540	kW				
9	Efficiency	13%		13%						14%		14%		60%					
10	Rated Pwr	286	kW	286		4000	kW	4000	kW	360	kW	360		25135	kW		18.1	11.5 ft	
11	AEP	700226	kWhr	700226	kWhr	22851209	kWhr	22851209	kWhr	881404	kWhr/unit	881404		61538928	kWhr for all 20 units				
12	FCR	11.3%		11.3%		11.3%		11.3%		10.8%		10.8%		10.8%					
13	CapEx:	$/kW	% CapEx	$/kW	% CapEx	$/kW	% CapEx	$/kW	% CapEx	$/kW	% CapEx	$/kW	% CapEx	$/kW	% CapEx		Initial Cost:	155	mill$
14	Structure	7200	34%	6200	46%	1450	16%	1200	19%	7400	35%	6400	46%	2165	35%				
15	PCC	1700	8%	1400	10%	3750	42%	2700	43%	1300	6%	1090	8%	1390	23%				
16	Moorings	1651	8%	1651	12%	459	5%	459	7%	3189	15%	2896	21%	0	0%				
17	Installation	3000	14%	1050	8%	800	9%	413	7%	2500	12%	1050	8%	242	4%				
18	Environmntl	2300	11%	300	2%	210	2%	25	0%	1604	8%	159	1%	159	3%				
19	Other	5200	25%	2999	22%	2300	26%	1423	23%	5007	24%	2205	16%	2205	36%				
20			% Envrnmt		% Envrnmt		% Envrnmt		% Envrnmt		% Envrnmt		% Envrnmt			$/kW, avg for 100 unit designs			
21	OpEx:	1150	34%	350	26%	360	14%	170	14%	1283	45%	207	19%	207		200			
22																			
23		$/kWhr		$/kWhr		$/kWhr		$/kWhr		$/kWhr		$/kWhr		$/kWhr					
24	LCOE:	1.44		0.77		0.24		0.15		1.45		0.69		0.36					
25	LCOE CapEx	0.97	67%	0.63	81%	0.18	74%	0.12	81%	0.93	64%	0.61	88%	0.27	76%				
26	LCOE OpEx	0.47	33%	0.14	19%	0.06	26%	0.03	19%	0.52	36%	0.08	12%	0.08	24%				
27																			
28		$/kW		$/kW		$/kW		$/kW		$/kW		$/kW		$/kW		$/kW, avg for 100 unit designs			
29	Cable Installtn	800		308		365		74		846		175		242		142			
30	Cable cost	308		308		90		90		244		244		276		307			
31	PCC/capacity																		
32	factor:	5667	$/kW	4667	$/kW	5357	$/kW	3857	$/kW	4333	$/kW	3633	$/kW	4633	$/kW				

Capital expenses (CapEx) for MEC’s includes all the costs associated with getting the item in the water. In Figure B1 this is broken down into six major subcategories: structure, power conversion chain (PCC), moorings, installation, environmental assessment, and other. ‘Other’ is what is left after the other items are counted, but includes things like design cost, miscellaneous infrastructure, contingency, and profit.

The structural design in Appendix A allows an estimate of Deltawave’s structural cost. At the top of Figure B2, a weight estimate is made of Deltawave’s structural parts. Both steel and ferrocement cladding options are considered separately. Assuming ferrocement costs $35/ft2 and steel costs $1.4/lb, the cost breakdown is shown. The steel-cladding option is lowest in cost. 60% of the structural cost is in the subraft, 23% is in the waterbrake, and the rest distributed among the hexaxle, connecting rod, piston pumps, and other items. For this economic analysis, which considers a Deltawave WEC composed of 20 hexrafts, the structural cost is $47.5 million, of which $1.1 million is for the 1 mile long, 4ft diameter pipeline carrying seawater under pressure to shore, where the PCC is located. Regarding the steel cost per weight, an average of such costs for RM1, RM3, RM4, and RM5 is $1.2/lb, whereas for RM2 its $2.2/lb due to the use of more expensive alloys. For RM5, the flap WEC which is most like Deltawave, a steel cost of $1.2/lb is used, and likewise for RM3, the buoy WEC.

Figure B2: Deltawave structural cost breakdown



290	Summary of Weight & Cost per hexaxle:																		
291		Weight (lb) per hexaxle (= 6 subrafts):						Costs ($) per hexaxle:											
292	Density:	water	steel	Ferrocement			Cost/unit:	Steel $/lb:	ment $/ft2:										
293	lb/ft3:	62.4	493	145				1.4	35										
294		Displaced water	Structure	Cladding:		Total Wgt:		Structure	Cladding Cost:		Total Cost:		Percent Cost:						
295		2	Steel	Ferrocement	Steel	errocemen	Steel	Steel cost:	Ferrocement	Steel	Ferrocement	Steel	errocemen	Steel		55			
296	Subraft:	-805669	156722	276266	164073	432988	320795	$219,410	$400,109	$229,703	$619,520	$449,113	64%	58%					
297	Ball jnt housing:	-34705	8783	16278	9667	25061	18450	$12,296	$23,575	$13,534	$35,871	$25,830	4%	3%					
298	hexaxle:	-41178	37512	20316	12065	57828	49578	$52,517	$29,423	$16,892	$81,940	$69,409	8%	9%					
299	connecting rod, pumps:	-23804	31997			31997	31997	$44,796			$44,796	$44,796	5%	6%					
300	waterbrake structure:	-63332	109155			109155	109155	$152,817			$152,817	$152,817	16%	20%					
301	tarp etc:										$ 31,112	$ 31,112	3%	4%					
302							Total:	$481,837	$453,107	$260,129	$966,056	$773,078							
303							Percent Cost:	62%	47%	34%									
304										Total Cost per hexraft (mill $):	2.9	2.3							
305																			
306	Deltawave WEC Cost:																		
307	Definition of WEC:	Width:	Length:	Note on Avg Avail Wavepower (kW/ft)				WEC self-propulsion calculations (i.e. self-installation):						Stationkeeping requirement estimation:					
308	Size of WEC, in hexrafts:	2	10	Location	Js (kW/ft)				Width of WEC:	280 ft	No of	X-sec Area(ft2)				Gulf Stream	Western B	Min	Max
309	WEC Efficiency:	60%		Gulf States	0.4					X-sec Area(ft2)/item:	items:	per WEC width:			Current (ft/s):	6.6	4.6	0.11	0.27
310	Depth of top of waterbrake(s):	50 ft		SE States	1.2				Waterbrakes:	1352	6.5	8786			Power Req'd (kW/ft)	30.1	10.1	0.0	0.0
311		Ferrocem	Steel	NE States	1.7				Connecting Rods:	146	8	1168			Hs Req'd (ft):	14 ft	9 ft	0.2 ft	0.5 ft
312	Total Cost per WEC (mill $):	59.0	47.5	California	6.5				Subrafts:	106	3.5	370		Conclusion:					
313				NW States	9.1						Total:	10324	ft2	Stationkeeping not a problem outside of Western Boundary Currents					
314									Equiv Height of WEC:	37 ft									
315	values used in 'costing' sheet to derive LCOE								Aspect ratio:	7.6									
316									Coeff of Drag:	1.3		Hs	Tp						
317									Wind speed:	33.8	ft/sec	7.41 ft	6.6 s						
318									Power avail:	4.9	kW/ft	2500 mi	Long Beach to Hawaii						
319									V min:	3.6	ft/sec	1.4	months of journey time						
320									V max:	6.1	ft/sec	0.8	months						

Deltawave's power conversion chain (PCC) cost was estimated from four of the five PM's given in References 3 and 4. Although widely differing in design (RM1 uses basically a wind turbine PCC sealed against water, while RM3 uses a hydraulic loop with accumulators to even out oscillating wave impulses), the various designs were found to have similar costs per unit average power produced per year (AEP, i.e., output power). Typical capacity factors, like for the two WEC designs, are 30%, meaning the PCC is designed to handle about three times the expected average output power. However, for the ocean current turbine (RM4), the capacity factor was 70%. Except for RM2, the river turbine, the PCC cost per output power was between $4000/kW and $5000/kW for all designs, so a value of $4600/kW was used for Deltawave. This corresponds to a value of $1390/kW of rated power (assuming a capacity factor of 30%). For the other two WEC designs, the corresponding PCC costs are $1400/kW and $1090/kW. Note that, as currently configured, Deltwave's PCC will be located on shore, which should lead to lower maintenance costs. If it is decided to retain a localized PCC, and port energy to shore as electricity, note that the MEC's in References 3 and 4 are designed with redundant systems (turbine rotors, cooling systems, controls, sensors, etc) to reduce maintenance. Therefore, the estimated PCC cost should not increase substantially if an ocean-positioned PCC is baselined.

Deltawave is designed to be self-installing, positioning, and mooring. This is made possible using GPS positioning, and by siphoning a fraction of the incoming wave energy for these purposes. The WEC would also be remotely monitored for position, should station-keeping go awry. Mooring is 12% and 21% of the Capital cost of the two WEC's studied in References 3 and 4 (RM3 and RM5), so attacking this cost is important. The waterbrake stabilizes Deltawave against wave action, and the low profile of the rafts ensures stability against wind action. This WEC is designed to operate in 'deep water' which for a 20 ft wave is 300 feet of depth or more. For 'deep water' waves, items positioned within the water column are mostly exposed to purely oscillatory action, especially when most of their profile is located 40 ft or more below the surface. To estimate the station-keeping power requirement, Reference B1 reports that the strongest ocean currents are along western boundaries of the oceans, such as the Gulf Stream and the Kuroshio current. However, the highest wave resource is on the eastern boundaries of the oceans, like the Pacific Northwest and the North Sea, so this is where Deltawave is most likely to be located. The wave resource required by Deltawave to maintain its position against the expected current can be calculated by estimating the drag coefficient and the area exposed to drag. These calculations are shown in the bottom half of Figure B2. For the WEC under consideration, which is two hexrafts wide, the width is 280 ft and the area exposed to drag is 10,324 ft^2. For such an item, the C_D (coefficient of drag) is approximately 1.3. The station-keeping power required per unit WEC width is $P/w = F_DV/w$, where V is the velocity of the current, and the drag force $F_D = ½\rho C_DV^2$ (ρ is the density of water). This power should be increased by the conversion efficiency, η, so that $P_{req'd} = P/\eta$ (Deltawave's efficiency is expected to be about 60%). From the required power, the needed wave height to supply that power to Deltawave can be estimated. From linear wave theory (Ref 1), $P/w = [\rho g^2/(64\pi)]*H_s{}^2*T_e$, where g is the acceleration of gravity and H_s and T_e are the height and wavelength of the predominant waveform. It can be approximated that, in deep water, T_e (in seconds) = 0.89 H_s (in ft). This was used to estimate the wave height, H_s, needed to keep Deltawave stationary in a variety of ocean currents. Outside the currents in the western boundary waters, the required predominant wave height needed for station-keeping is below half a foot. It thus seems likely that, properly designed, Deltawave could be self-mooring. This has the added advantage of its not being restricted to operate in relatively shallow waters.

A similar calculation was done to estimate how long it would take Deltawave to move itself from its construction site to its installation site. As shown in Figure B2 (bottom-right), the NE Tradewind's have an average wind speed of 33.8 ft/s (Ref 1), which in deep water provides approximately 4.9 kW/ft of power to the Deltawave WEC, corresponding to a minimum speed of 3.6 ft/s, and a speed about twice that if the long axis of Deltawave is aligned with the predominant wavefront during travel. Thus, a journey of 2500 miles (for example, from Long Beach, CA to Hawaii) would take Deltawave about a month and a half, if self-propelled.

The installation costs of Deltawave are assumed limited to the cost of installing cabling. To estimate the cost of electrical cabling and cable installation, the costs per kW of rated power for the two WECs, RM3 and RM5 (100-unit options) averages out to $242/kW for cable installation and $276/kW for the cable itself, so this was used for Deltawave. The averaged corresponding costs for the five MEC designs studied in Ref 3 and 4 yields a similar cost estimate for these two items.

Assuming Deltawave is self-mooring and self-installing without further cost, it has an LCOE of 36 cents/kWhr. This would rise to 46 cents/kWhr if mooring were required, including its increased installation costs. The higher estimate is based on using the 100-unit option of RM3, the buoy WEC, values for moorings and installation, and assuming those of Deltawave would be the same. The buoy WEC (RM3) was used for this estimate, rather than the flap WEC (RM5), because the flap WEC has a higher mooring cost needed to increase the efficiency of its wave energy conversion. Deltawave's mooring costs are assumed, if needed, to be more in line with those required by the buoy WEC.

It's difficult to estimate the environmental assessment, mitigation and monitoring costs of Deltawave, so the capital cost for this item reported for the 100-unit option of RM5, the flap WEC, was used as an estimate. Underwater structures tend to provide fish with greater places to hide from predators, so fish populations near this WEC may increase, rather than decrease. Also note that when Deltawave is aligned parallel to a coastline, there is the possibility of swinging the WEC 180° around whichever end the cable is attached to. In this way, a stretch of coastline 'shaded' by Deltawave one year, can be exposed to wave action the next year to reduce detrimental impacts on nearby beaches.

The capital expenses previously discussed are for Structure, Power Conversion Chain (PCC), Installation, and Environmental assessment. Other capital expenses, in Figure B1, are placed in the 'other' category.

Despite being miscellaneous, these still add up to a large share of total capital expenses. As an example of the types of costs in this category, for the 10-unit option of RM5, the flap WEC, 'other' constituted 24% of total CapEx, of which 2% was for design, 0.4% for site assessment, 9.5% for miscellaneous infrastructure, 4% for subsystem integration and profit margin, and 8.5% for contingency.

As was done in assessing the environmental capital costs, Deltawave's 'other' capital cost will be assumed to equal those of the 100-unit option of RM5, the flap WEC. It should be noted that this may overestimate this cost, since it makes 'other' 36% of the total CapEx for Deltawave, whereas a typical 'other' cost is 21% of total CapEx, for the other MEC's. However, if Deltawave requires moorings, and their cost is taken from the 100-unit option of RM3, then this 'other' cost for Deltawave is only 26% of total CapEx, so the estimate remains.

Operational expenses, OpEx, include the cost of annual maintenance, insurance, and post-installation monitoring (including for environmental impact). Maintenance requires an assessment of part failure rates, replacement part costs, and operations costs, both marine and shoreline. To reduce maintenance costs, Deltawave is designed to have its hexaxles easily separable from its rafts at the raft ball-joints, so that all parts can be towed to shore for shoreline maintenance. Avoidance of marine maintenance operations is key to keeping costs down. Also, the baseline Deltawave design is to place the critical electrical parts of the PCC onshore, to also reduce maintenance costs. However, this requires a hi-pressure pipeline from the WEC to the shore, which may have maintenance issues of its own.

Deltawave's rafts are designed to be able to be turned upside down, which means barnacle growth from one year could be discouraged the next year by turning the raft upside down. Meanwhile, little barnacle growth occurs on surfaces lower than 40 ft below the surface, so the waterbrake surfaces will be spared.

For decommissioning, Deltawave is designed to be made of materials that, abandoned at the bottom of the ocean, would be chemically benign. Steel and concrete structures fit this description. Exceptions are the plastics used in the waterbrake and the Teflon bearing used in the ball joints, which would be recovered before sinking. The desire is to reduce decommissioning costs as much as practicable. On some shorelines, concrete and steel structures are intentionally scuttled to form artificial reefs. Such a fate for Deltawave would require further study.

In Figure B1 the OpEx cost of Deltawave is assumed equal to that of the 100-unit option of RM5, the flap WEC. At $207/kW of rated power, this is close to the average OpEx of all 100-unit options studied (RM1-RM5), which was $200/kW. For RM5 (100-unit option), the OpEx breakout is 16% on marine operations, 11% on shoreside operations, 1% on replacement parts, 20% on consumables, 33% on insurance, and 19% on post-installation environmental monitoring.

This economic evaluation leads to an expected LCOE for Deltawave of 36-46 cents/kWhr, depending on whether mooring and installation costs are included. It doesn't include an elevated land-based reservoir that would be used to store energy from the WEC to ameliorate the highly-variable nature of the wave resource. By way of comparison, Lazard in Reference B2 reports the 2021 cost of unsubsidized wind power in America to be 26-50 cents/kWhr.

References for Appendix B:

1. ‘Fundamentals of Physical Geography, 2nd Ed’. Chapter 8: Introduction to the Hydrosphere. Section q: Surface and Subsurface Ocean Currents. Pidwirny, M. (2006)

2. ‘Lazard’s Levelized Cost of Energy Analysis - Version 15.0’. Lazard Financial Services Company. © 2021