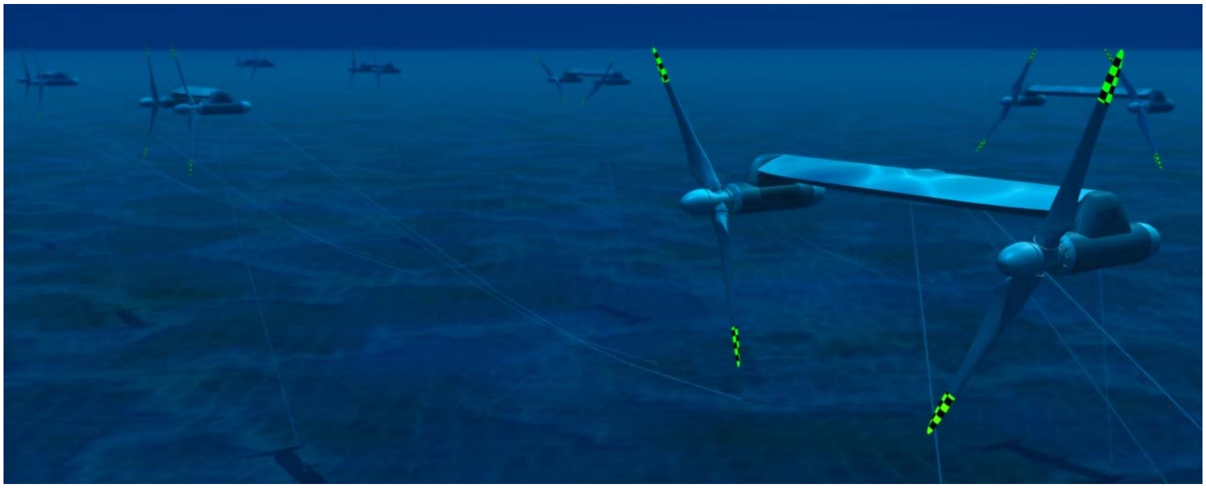

Aquantis Ocean Current Turbine Development Project Report

Innovative Power Generation Technology
February 1, 2010–August 31, 2013

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NOTICE

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1.0 Executive Summary

The Aquantis® Current Plane ("C-Plane") technology developed by Dehlsen Associates, LLC (DA) and Aquantis, Inc. is an ocean current turbine designed to extract kinetic energy from ocean currents. The technology is capable of achieving competitively priced base-load, continuous, and reliable power generation from a source of renewable energy not before possible in this scale or form. A vast area of potential development in the Gulf Stream off of Florida is characterized by a steady flow in the range of 1.2 to 1.8m/s, with seasonal change in direction of +/- 20°, a pronounced velocity shear with greatest velocity at the surface, and ocean floor depths of greater than 200m. It is estimated that >5,000MW of clean renewable base load energy could be extracted from this resource.

The Gulf Stream is a large indigenous power source with high-probability of success for cost-competitive energy extraction and also with premium energy qualities – clean, constant, close to a major transmission system. This displaces the need to import polluting fuels for power generation. A 100MW facility comprised of 40 2.5MW C-planes will produce 532,000,000 kWh/year serving 49,700 households. To generate the same amount of power would require 28 million barrels of oil over the 20-year C-Plane design life. The declining production of America's 500,000 producing oil wells currently average 10 barrels a day or 3,600 barrels per year, per well. A single C-Plane will generate the electricity produced by 24,000 barrels of oil per year. The electricity from a 100 MW C-Plane array will displace CO_2 emissions from fossil fuel generation by 532,000 tons per year.

The key breakthrough and related Aquantis patents are its passive depth stability (PDS) at its prescribed operating depth, which corresponds to a specific velocity (current velocity reduces with depth). The PDS results from the balance of forces acting on the C-Plane between those for rising and those for going deeper. The rising forces are the lift from the hydrofoil platform and the buoyancy factor of the C-Plane. Offsetting this are the forces to drive the C-Plane deeper, namely the weight of the machine (gravity) and the drag of the rotors exerting a downward force due to the tethering to the ocean floor. The operating rotor drag then becomes the determinant of depth since it corresponds to the certain velocity of its operating depth, preventing the C-Plane from migrating deeper, since the current velocity drops, hence less drag, or from rising due to greater drag from higher current velocity.

The aforementioned project has assembled a team of recognized ocean industry and renewable energy experts having on average 30 years experience in their respective disciplines. The major participants include Navy Surface Warfare Center (NSWC) Carderock Division, Applied Research Laboratory (ARL) at the Pennsylvania State University, Straight Forward Systems, Inc. (SFSI), BEW Engineering/ A DNV KEMA Company, Powertrain Engineers, Inc. (PEI), and Bosch Rexroth. The Aquantis project has a calculated cost of energy (COE) at commercialization of <10Cents/kWh which is cost competitive in the Florida electricity market. The device entry point is 1.5MW with a modular/scalable topology that allows expansion to 2.4MW through value added engineering and optimization.

Synopsis

The general use of underwater power generators for producing electricity from water current flow, such as rivers and oceans, is well known. There are two main types of ocean devices:

(1) stationary turbines; and (2) tethered turbines. Stationary turbines utilize stationary towers based on the ocean floor. Electricity-generating turbines are mounted on the towers at a predetermined depth with rotor blades facing the flow upstream or downstream of the tower. This type of design suffers from at least the following disadvantages: relatively high underwater construction and deployment costs; difficult engineering challenges related to installing towers in deep water; reduced or limited current velocity associated with the turbines being located close to the ocean floor resulting in lower power output; and inherent difficulties maintaining and accessing ocean floor systems.

Tethered turbines are anchored to the ocean floor via an anchoring or mooring means, but have a degree of mobility relative to the anchor. In some cases, a wing (hydrofoil) provides lift and/or ballast tanks provide buoyancy in order to keep the turbines at optimal depths. Some of these turbines use a buoyancy chamber to regulate their overall buoyancy, thereby adjusting their operating depths in a current stream. Other devices add movable surfaces that serve as an elevator to control the depth of the device. The elevator surface is adjusted to assist the device to dive or ascend, as needed.

By using both local marine current measurements with known global current patterns, a number of sites in the ocean have been identified for deployment of marine current power generating devices, representing several thousand gigawatts of potential and untapped electricity generation. Many countries throughout the world rely heavily on importing fuel for generating electricity and lack viable renewable energy sources. In view of current population growth (increasing by 1.5 million humans per week), a perilous trend in climate change, growing demand for natural gas and petroleum, and the increasing difficulty in finding and developing new petroleum fields, an urgency has been created for developing and deploying new sustainable and cost-effective technologies to transition energy resources and consumption away from carbon-based fossil fuels.

Most marine current power generating technologies are migrating to the use of submerged systems. Energy can be extracted from the ocean using submerged turbines that are similar in function to wind turbines, converting energy through the process of hydrodynamic rather than aerodynamic lift or drag. These turbines have rotor blades, generators for converting rotational power into electricity, and means for transmitting the electrical current generated to a shore-based electrical grid.

Today, both horizontal and vertical axis turbines are generally considered for producing power from ocean currents. Ocean current power systems are at an early stage of development; only a few prototypes of small scale and a few demonstration units having been tested or shown to date and most devices operate below a 2 MW generating capacity rating.

A number of patents have been issued related to systems for producing energy from ocean currents. Some of the patents describe devices using active stability, depth and rotor control, which generally increases cost and complexity and reduces reliability.

Conventional designs have complex active systems such as control surfaces, variable ballast, variable pitch, winching systems, or mechanical means for raising/lowering the structure. In a moored system subjected to harsh environmental and structural loading from strong ocean tidal currents, gyres (steady ocean currents), and eddies, failures often result in the inability to access the device, or, in the worst cases, complete loss of the structure or hazard to navigation for vessels using the area. In addition to the inherent risks, these controls lack simplicity; they provide many variables and opportunities for failure, along with the additional costs associated with these types of control methods. Lower reliability, lower operating availability for power production and higher maintenance requirements mean that conventional designs are not cost-effective.

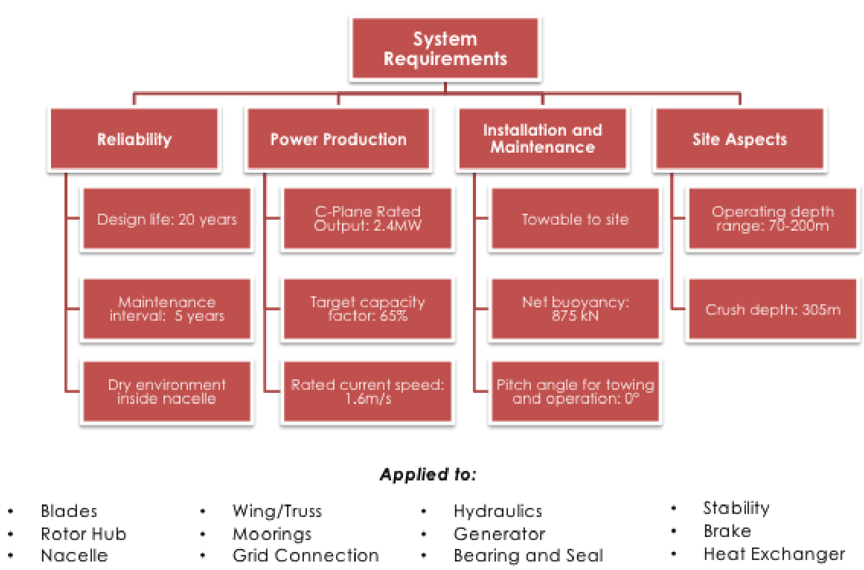
To satisfy customer needs, hydrokinetic devices must provide energy at low cost, high reliability and high operating availability to the applicable power grid, and have a service life greater than 20 years and a high safety factor in operation and maintenance. Maintenance costs for offshore power generating structures require a significantly different mindset than that for onshore power plants. For example, accessing the mechanical components of a submerged system typically requires highly-specialized crews, divers, autonomous underwater vehicles (AUV's), remotely operated vehicles (ROV's) and vessels with related costs for mobilization, demobilization and fuel costs. To be comparable to other power-generating technologies on a cost-of-energy basis, the scheduled visits to perform maintenance should be on the order of once every five years. Sub-systems must, therefore, be simple and reliable, and they should use proven components to achieve low COE targets.

In order to reduce installation and maintenance costs, use of underwater structures and moving parts should be minimized. A method of safely and economically mooring and installing the underwater device in its operational position should be provided, along with a procedure to safely bring the device to the surface for maintenance or for replacement of components.

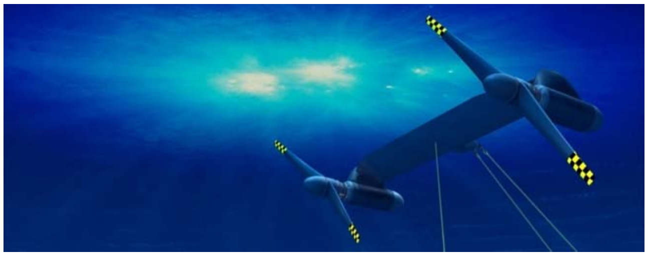
Variable blade pitch should be eliminated to reduce the potential of pitch system failure and other related maintenance issues. Complete emergency shutdown of the device, including stopping rotor blade rotation, should be possible, and the device should have fail-safe depth control which prevents unplanned surfacing.

It is therefore desirable to take a systematic approach to provide a simple and reliable system, one that is easy to maintain and service, with low cost-of-energy (COE) and a service life longer than 20 years. The mission statement for Aquantis is captured in a systems requirement document with flow down requirements for the design to reach the COE and service life goals.

The design achieved under this grant is presented below and meets the mission statement as set out in the system requirements document.



THE AQUANTIS C-PLANE



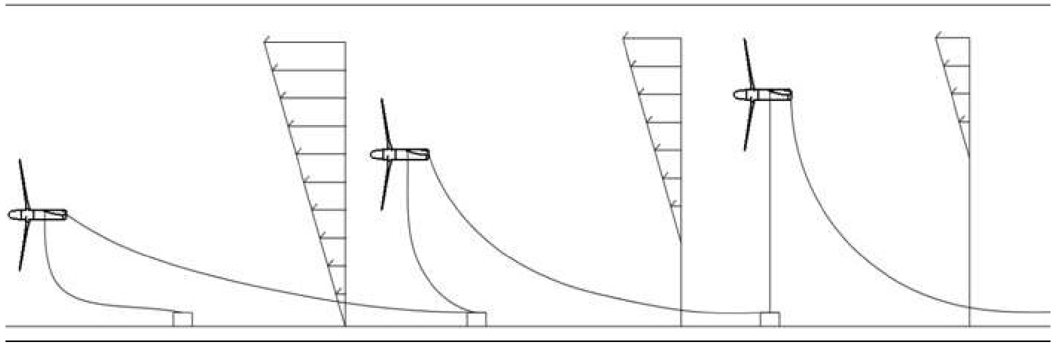
The C-Plane design overcomes the deficiencies and drawbacks of other marine hydrokinetic devices by providing underwater power-generation in which a submersible device, or platform, includes sets of two or more counter-rotating rotor assemblies, each rotor assembly having rotor blades, preferably of fixed pitch. The current-driven rotor assemblies drive hydraulic pumps which, in turn, drive fixed and variable displacement hydraulic motors to drive, preferably at constant speed, electric generators that are all housed in a fluid-tight power pod. The power pods are connected together by a transverse structure, which can be a wing depressor built in the shape of a hydrofoil, a truss, or a faired tube. The submersible device is connected to one or more anchors on the seabed by forward tethers. The device is also connected to an additional anchor on the seabed by a vertical downward tether to limit it to a predetermined depth, to prevent unplanned surfacing, and to counteract buoyancy. A wing depressor generates negative lift to offset buoyancy and ensures that the submersible device seeks/dives down to a predetermined operational current speed along the vertical shear, which corresponds to a specific depth. The same effect can be accomplished with a truss or faired tube, in concert with dominating rotor drag loads. As the flow velocity varies above operational speed, thereby increasing drag loads, the device dives deeper along the vertical shear until it reaches equilibrium (balance of forces) at the corresponding flow speed.

Power Generating System – A power generating system is housed in a water tight, buoyant pressure vessel. The power-generating system includes a drive shaft operatively connected to one or more hydraulic pumps, and at least one hydraulic or mechanical brake. The hydraulic pumps are then connected by hydraulic lines to variable or fixed displacement hydraulic motors, which are directly connected to and drive at least one electric generator. The main shaft rotates within a fluidic (sea water lubricated) bearing that decouples non-torque loads.

The fluidic bearing prevents non-torque loads from entering sealed areas, thus mitigating leak paths, and a flex coupling in the drive line prevents eccentric loads from acting on the hydraulic pump.

The rotor assemblies may be slowed or stopped by hydraulic braking of the pump acting in concert or independent of a mechanical brake for redundancy. The hydraulic brakes use hydraulic pressure to stop the rotor assembly, using energy from a main hydraulic loop that feeds a charged accumulator, which discharges when needed to apply the hydraulic brake to the rotor drive shaft.

Load Shedding – A method of controlling depth of the device underwater is based on the measured velocity shear profile of the current, which said velocity reduces with depth. Controlling depth comprises: anchoring the device in respect of the fluid flow by means of forward tethers connected to forward anchors to resist rotor drag loads; and connecting the device to a rear (aft) anchor, which is fixed in a position under the device, so that the rear anchor limits the device to a predetermined depth below the surface, resists buoyancy, and prevents the device from surfacing.



The aft anchor maintains a predetermined depth of the device at the design current velocity, while increased velocity of the current will cause the device to move to greater depths because of increased drag of the device and/or the negative lift of a wing depressor. Similarly, with a decrease in current velocity resulting in a reduction in drag, the device will rise due to its buoyancy until the predetermined depth is reached and the aft tether line goes taut.

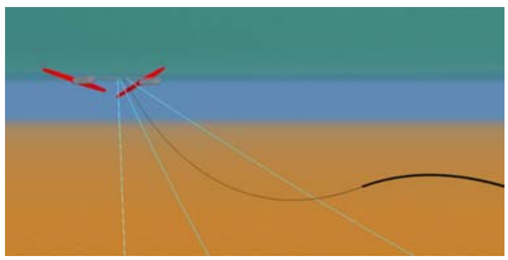
The device is brought to the ocean surface by: (1) braking the rotor assembly to a full stop thereby reducing rotor drag; and (2) releasing the rear mooring line from the platform so that the positively buoyant device will rise to the surface.

The hydraulic drive has the advantage of maintenance intervals that are greater than 40,000 hours (5 years). Due to its robustness, the drivetrain requires few service visits and surfacing. In extreme working environments, this translates into a lower life cycle cost of energy. This outweighs the benefits of higher efficiency of drivetrains based on a standard gearbox and variable speed generators; such systems have documented lower reliability, which is exacerbated in the extreme marine environment.

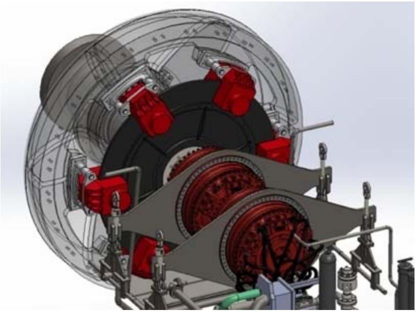
The hydraulic drive also has the advantage that, in eliminating failure-prone gearboxes, variable speed generators and power electronics such as inverters and rectifiers. At the same time, eliminating variable speed generators also reduces the risk of dependency on rare earth magnets, with much higher commodity price risk.

The hydraulic braking on the mainshaft has the advantage of eliminating the need for variable pitch rotor blades to mechanically off-load the rotor/drivetrain system when required. The direct hydraulically-activated braking of the rotor assemblies, coupled with a mechanical brake and passive depth control, provides redundancy for protecting against system overloads and enables the use of simple, reliable, fixed-pitch rotor blades.

The high power density (kW/kg) of the hydraulic systems compared to the heavier, geared powertrains means less volumetric buoyancy, reducing structural size, body drag, and a need for adding low-density materials such as costly syntactic foam to recover platform buoyancy.



Mooring – The C-Plane has the further advantage of a centralized mooring attachment that allows the device to weathervane, or freely pivot into the current direction, reducing the need for active (pitch, roll, and yaw) control of the device. The system relies on the inherent stability of the device in 6 degrees of freedom, and, when subjected to perturbations in the current flow, the device generates the necessary restoring moments for achieving dynamic and static stability. If, for example, a mooring line is lost, the device remains stable and the remaining mooring line supports the tensile-load. Having multiple attachment points would overly constrain the device and would induce instabilities in the system if the device were subjected to changes in the current direction. The five-degree or greater aft rake (rotor coning) of the rotor blades provides a weathervaning effect for additional system stability.



Rotors – At high level flow speeds, the rotor assemblies begin to reach the design limit, at which point most in-stream devices begin to brake or feather the rotor blades by way of variable pitch, much like modern day wind turbines. For tethered devices, the application of the brake to stop the rotor assemblies decreases the drag of the rotor assemblies, resulting in lower tensile forces on the tethers and potential ascent of the device in the water column where even higher flow speeds often exist. Braking the rotor in post-stalled blade states results in momentary spikes in torque loading, the introduction of cavitation, and the introduction of non-linear stability effects. To avoid these issues at high flow regimes, variable ballast control could be used to lower the device to a depth in the water column with lower flow speeds. However, failure of a ballasting control could result in the loss or sinking of the device. Instead, to deal with current velocities higher than the normal operating regime, a fixed wing with negative incidence angle is employed to passively achieve a descent of the device to more appropriate flow regimes within the water column, eliminating the need for ballasting or control surfaces. Since the flow velocity drops off with increases in depth along a vertical sheer, the device can utilize a wing depressor to drive to deeper depths as a means of load shedding on the rotors.

Additionally, the wing depressor provides a means for the device to converge to a predetermined velocity and rotor torque to avoid overload conditions. This approach also avoids exhaustive water tunnel testing used to characterize the non-linear flow regime with rotor assemblies in the post-stall region, which, at present, computational fluid dynamics models are not able to reliably predict.

To install the device when it is floating on the ocean surface, the forward mooring lines are attached, and a weight is used to bring the buoyant device low enough to achieve slack in the vertical line. An ROV brings the slack vertical line with a male end connector to the female connector, which has been pre-attached to the mud-mat (device used to locate and hold the female location of the connector). A mud-mat is used if the anchoring device is an embedment anchor; otherwise the connector may be added directly to the anchor (i.e. a DWT anchor).

For servicing, stopping the rotor assemblies by braking results in lower thrust loading, reducing drag on the device and reducing the downward force vector, whereby buoyancy and release of the vertical mooring cable allows the device to rise to the surface where maintenance can be performed or components of the device can be exchanged for servicing.

2.0 Project Parameters

2.1 Project Objectives

The principal objective of the Aquantis Program is to develop technology to harness the vast Gulf Stream energy resource using innovative power generation technology that is projected to be cost competitive with conventional power generation sources in early deployment.

Project Products – The project's effort resulted in five conclusive products:

1. Experimental validation of analytical tools/design: use of scale models and full scale component testing, subsystem integration and global system responses to validate analytical tools and gain confidence in device performance and loads
2. Cost of energy model: robust model factoring in CAPEX and OPEX. This model provided specific design goals for serviceability, maintenance intervals and reliability.
3. Solid models: solid models (Solidworks and Creo) of the full scale design
4. Enabling technology: development of a hydrostatic drivetrain couple with an induction generator
5. Final report: design, trade studies and validation via virtual prototyping using the latest computer-aided engineering tools, including Solidworks, RANDS, WAMIT, ANSYS, ASDS and Orcaflex and leveraging a significant experience base in marine renewable energy conversion and the design of offshore structures

2.2 Project Scope

The Aquantis ocean current power generation device technology is a derivation of wind power generating technology: a means of harnessing a slow moving fluid, adapted to the ocean environment. The Aquantis Project provides an opportunity for accelerated technological development and early commercialization, since it involves the joining of two mature disciplines: ocean engineering and wind turbine design.

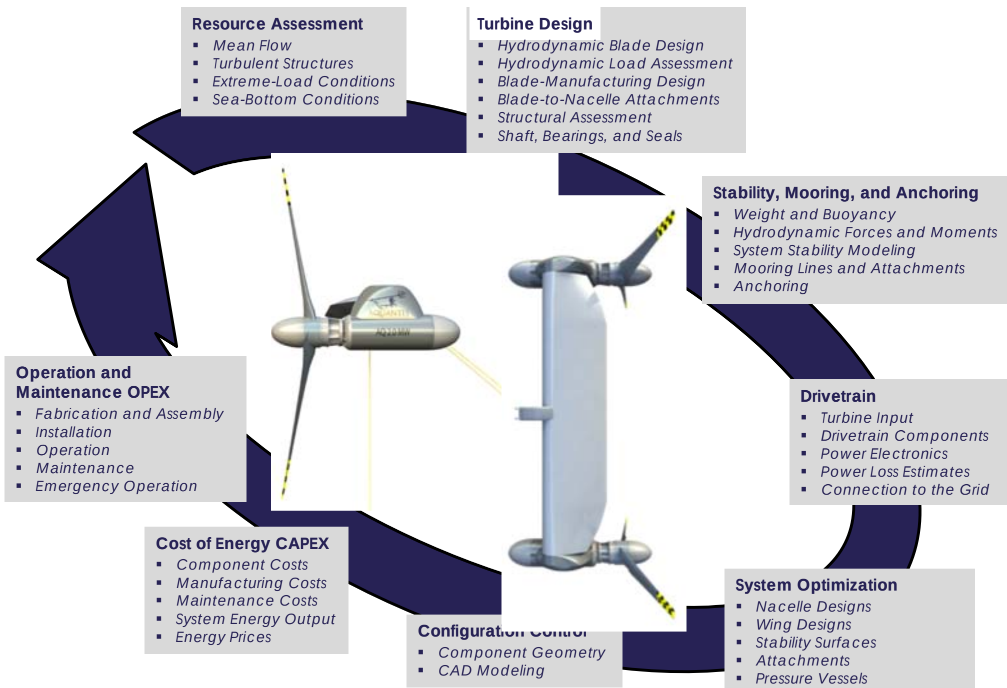
The Aquantis Current Plane technology is an ocean current turbine designed to extract the kinetic energy from the current flow and is capable of achieving competitively priced, base-load, continuous, and reliable power generation from a source of renewable energy not before possible in this scale or form.

Aquantis LLC, formed in 1997, has made substantial progress in the preliminary design and engineering of the C-Plane, a hydrofoil platform with twin, counter-rotating rotors, which drive generators for delivery of electric power to the shore-based substation of the regional electric grid. The C-Plane operates 50m below the ocean surface and is tethered to the ocean floor at in water depths greater than 200m. The design allows for temporary surfacing for periodic maintenance and any required repair of the system.

While C-Plane deployment would be phased with appropriate environmental milestones for expansion to proceed, it is estimated that 6,500 MW of generating capacity could be in operation within ten years from start, a deployment comparable to that experienced by early stage wind power in several European countries with national wind energy programs.

2.3 Tasks Performed

The tasks developed to meet the project objectives when contracted are contained in the following sections. The team used a *design spiral approach* where all design elements and subsystems were developed in unison. These elements presented in the boxes below were executed through organized integrated product teams (IPTs) working towards set system requirements and techno-economic goals.



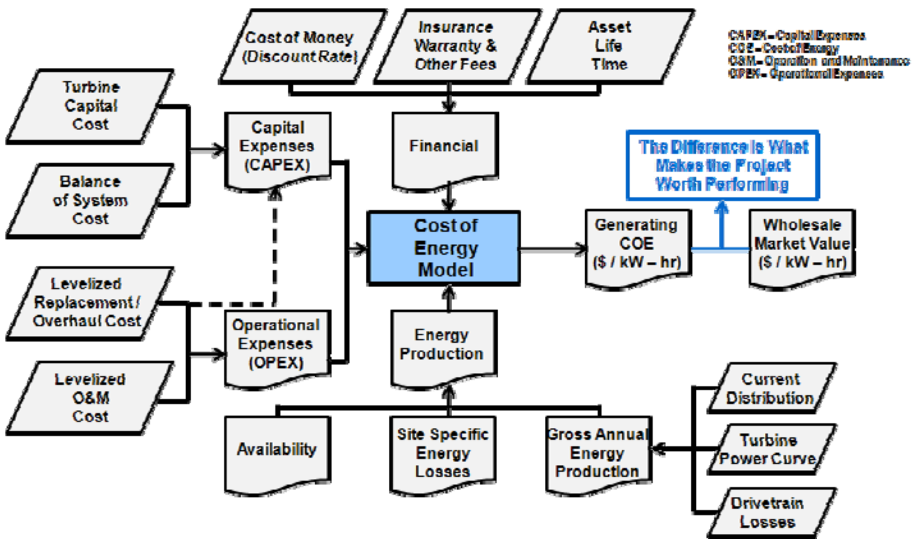
3.0 Project Organization

The Aquantis team engaged best in class domain experts in the respective fields necessary to execute this large system engineering challenge. Legal contracts that included terms and conditions (T&C's), Intellectual Property (IP) rights and financial terms were negotiated and completed with critical team members. Aquantis achieved a balance with the necessary frictions between Naval/DOD marine design expertise and cost conscious techno-economic professionals in an effort to achieve competitive cost of energy (COE) without subsidies. Key team members and their responsibilities are outlined below.

- **Applied Research Laboratory (ARL) at the Pennsylvania State University**
  - Hydrodynamics/structure/marine composites/drivetrain
- **Naval Surface Weapons Center Carderock Division (NSWC-CD)**
  - System stability, mooring, experimental test facilities
- **PCCI, Inc.**
  - Mooring and anchoring component design;
- **Powertrain Engineers, Inc.**
  - Mechanical engineering, design and CAD modeling

- **BEW Engineering (a DNV Company)**
  - Power electronics/conditioning/grid connection; and
- **Potencia Industrial**
  - Generator and power electronics specification and manufacturing insight

Techno-Economics – IPTs design trades and sensitivity studies were run through a comprehensive techno-economic model to drive design decisions (shown below). Given the remoteness of the Aquantis device and required ships of opportunity approaching $75k/day, special emphasis was placed on increased reliability by reduced part count and system complexities.



4.0 Project Evolution

Aquantis is a large system engineering process where subsystems are intertwined with the complexities of a flight vehicle. Unlike a stationary, static, pole mounted, tidal device Aquantis is a deep water tethered flight vehicle involving proper treatment to naval architecture principles. Weight and trim is critical whereby hardware placement is governed by achieving static stability via proper separation of center of buoyancy and center of gravity. Additionally Aquantis must achieve net buoyancy for a fixed wetted area. In order to develop the Aquantis system it was run through the design spiral mentioned above. Each box of the design spiral was executed by their respective IPTs with consistent focus on the aforementioned techno-economics.

5.0 Project Task Activities

This section provides a description of work and relevant outcomes. This material is organized according to the contractual subtasks; any deviations from the tasks are explained herein.

5.1 Florida Resource Evaluation

Dehlsen and Associates, LLC collected initial Florida current resource data between June 2000 and June 2002. In this data collection program an acoustic Doppler current profiler (ADCP) was deployed about 26 km east of Fort Lauderdale, Florida. Approximately 90 range gates were available every 15 minutes while the ADCP was operating. The resource data analysis firm V-Bar was contracted by DA in 2010 to analyze this initial data set.

The following summarizes the methods for processing and quality controlling the data:

- 15-minute time series data from the ADCP was exported from raw data files using WINADCP, a third party software package.
- To facilitate data processing for the ocean current energy resource assessment, hourly average current speed and direction data were produced from the I5 minute samples.
- The data was further simplified by averaging three range gates, which results in average velocity data in 10 m increments or bins. An average for each bin and time step was only produced if there was a value present for each of the three range gates used in the average.

The rotor plane layer of primary interest for this study was the high current speed span of 50-90 meters below the surface. This 40-m span corresponds to the rotor diameter at the time of the proposed turbine, which is a dual-rotor C-Plane.

Florida current resource parameters assessed in this effort included:

1. Maximum average current velocity in the operational depth envelope of the C-Plane
2. Daily mean ocean current speeds
3. Monthly mean ocean current speeds
4. Four average range gates between depths of 50 and 90 meters into ocean current speed and frequency distributions and normalized to a single, 8760-hour year. Gross annual energy simulations were performed for the dual-rotor C-Plane power curve.
5. Ocean current roses directional for the 56.4m and 66.2-m depths. The current within the Florida Straits flows to the north-northeast.
6. Monthly, maximum, hourly, mean current speeds for each depth
7. Simulated mean monthly diurnal gross capacity factors using the available data from the 66.2-m depth

This analysis concluded that with the uncertainties in the depth of the ADCP; the significant shear in the Gulf Stream current in the Florida Straits; the potential for the long-term average current speed to differ from these measurements; and the unknown operational losses of the C-plane – this value is highly uncertain. This net capacity factor value should therefore only be considered indicative of the ocean current energy resource at this location in the Florida Straits.

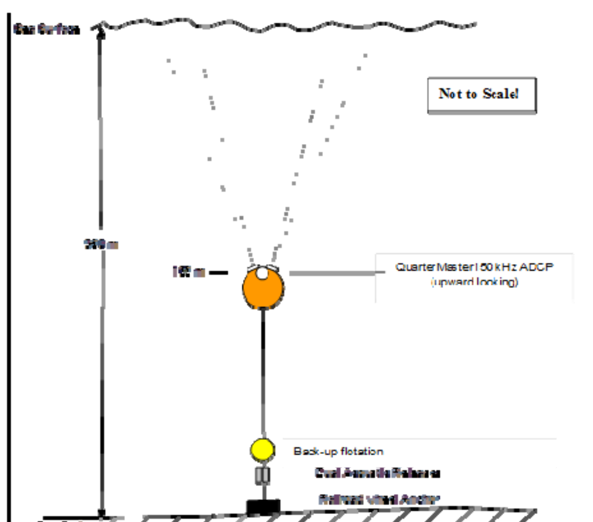
Under contract with DA, Florida Atlantic University (FAU) performed an extensive analysis of the resource using the Hybrid Coordinate Ocean Model (HYCOM) and data obtained from their bottom mounted, ADCP's. The FAU ADCP's were deployed in the resource offshore of Ft. Lauderdale, FL. FAU recovered these ADCP's and used the data in the subsequent analysis. The intent of the analysis was to quantify the resource variables over time and to identify conditions which would impact the design of the C-Plane. The conditions that could impact design and that were studied included flow reversals, eddies, shears and internal waves.

The Florida current resource assessment program was designed to establish a plan for utilizing an ocean instrumented glider that would traverse the Florida current and provide resource information to be used to locate an ideal commercial site location, exhibiting high flow velocities, displaying optimal resource while minimizing cabling and mooring costs, determining anchor suitability, and mapping out a subsea cable route. To accomplish the above, post-processing was required using MATLAB to generate plots for each of the glider sweeps that depict the current cross-section and indicate areas of highest density flow. These data were used in the Florida resource valuation.

Given the selection of a glider, the team laid out a plan to deploy the glider in the area of commercial interest and perform current resource data collection over a finite period. A Liquid Robotics Wave Glider was deployed in conjunction with the period of operation of the FAU ADCP's in the resource offshore Ft. Lauderdale, Florida. The intent of the Wave Glider deployment was to pass over the FAU ADCP's and collect current velocity information for comparison with the data collected by the FAU ADCP's and gather additional data on the resource in the area. Deployment of the Wave Glider was conducted in late 2012, and several passes were made of the FAU ADCP's as well as a general current data gathering mission throughout the resource in the area of commercial interest. Unfortunately, half way through the data gathering process, the team lost communication with the Wave Glider and the subsequent recovery mission failed to locate the system. Due to high levels of commercial and recreational vessel traffic in the area it was postulated that the Wave Glider was struck and sunk or possibly stolen. As a result of this incident, the data gathered during the entire mission was not recovered.

Given the loss of the initial Wave Glider and the subsequent recovery of the FAU ADCP's, DA purchased two, Teledyne/RD Instruments, Quartermaster 150 kHz, ADCP's for resource assessment in the area of commercial interest. DA then planned to deploy a mooring containing one of the ADCP's in early 2013. The data to be gathered was to establish a correlation relationship with a second deployment of a Wave Glider. This ADCP also provided a reference station for the mobile ADCP, so that various regions were evaluated for their suitability for deployment of the Aquantis ocean current turbine. The HYCOM data and spatial planning models were used to determine an ideal location for deployment of the moored ADCP. Additionally, these models and data sets were used to determine a region of interest for deployment of the glider. This provided a solid basis for reference indicative of the flow fields of interest to base the correlations.

The ADCP mooring was installed January 2013 at a location approximately 13 miles east-southeast of Palm Beach inlet as shown. Water depth at the site was approximately 350 meters. The mooring system was designed to position the ADCP approximately 150 meters below the sea surface, looking upward, to quantify current speed and direction variability in the upper layers of the Florida Current.



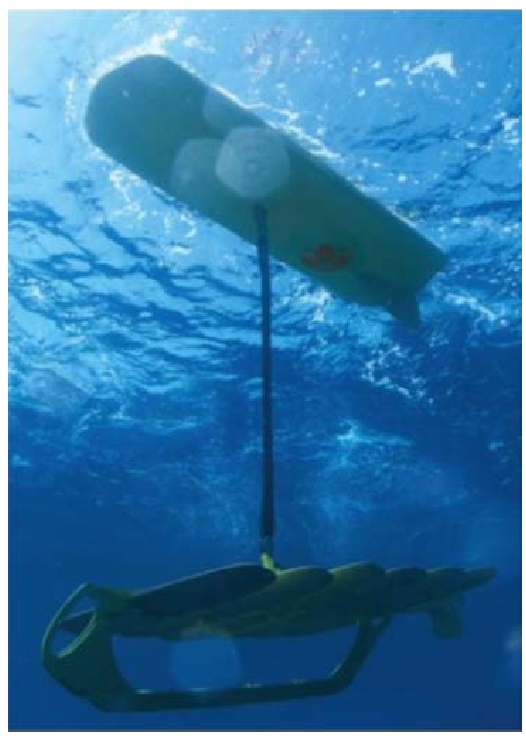
ADCP Mooring Schematic

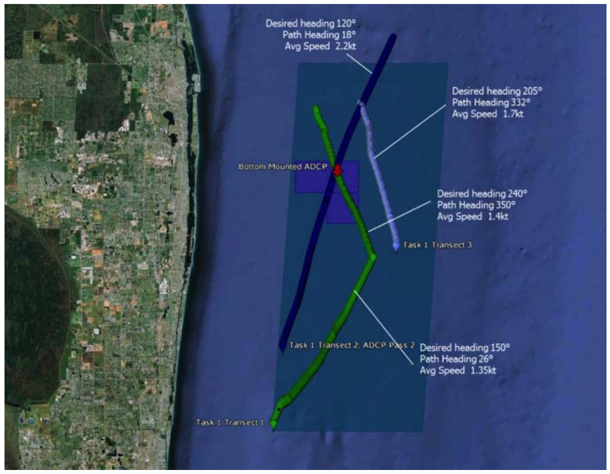
Currents profiles were obtained every 1.12 seconds, and stored in internal memory as single-ping ensembles in beam coordinates. The mooring was recovered successfully in late January 2013. All data were complete and downloaded from the ADCP recorder for use in subsequent data analyses.

Data were transformed from original beam coordinates into an earth coordinate orientation (east-north-vertical). Once in earth coordinates, the data were reduced into roughly 2-minute averages (averages consisted of 107 samples each, approximately 119.84 seconds, due to the 1.12-second sample rate). Invalid data, for example, acoustic beam side-lobe reflections off the sea surface, were obvious in the record and removed from the data set. The ADCP profiles were also adjusted for instrument depth variations; such depth variability occurs due to variations in lateral fluid drag forces on the buoy, causing a vertical layover, and requires each profile be adjusted such that resulting velocities are geo-located to the proper depth layers. These processing steps – coordinate grid transformation, temporal averaging, discarding invalid values, and bin-mapping - were the only manipulations required; otherwise the data were of very high quality.

In the Wave Glider current data gathering program two gliders were used to perform transects across the commercial region of interest. This region spans from 8 – 20 nautical miles offshore. The chief objective was to investigate the ability of the Wave Glider to collect detailed current profiles in the Gulf Stream off the coast of Florida. The mission ran along the coast of Florida between Ft. Lauderdale and North Palm Beach The bottom mounted ADCP collected data approximately 11.2 nautical miles offshore Palm Beach. Multiple Wave Glider deployments in the operational area were coordinated with the Wave Glider operational personnel to maximize at-sea conditions, power management, and coordination with bottom mounted ADCP requirements. The gliders were recovered multiple times to reduce transit time. Post-recovery the ADCP data was downloaded from the Wave Gliders upon each recovery as needed to maintain a back-up of all collected data.

Wave Glider Configuration

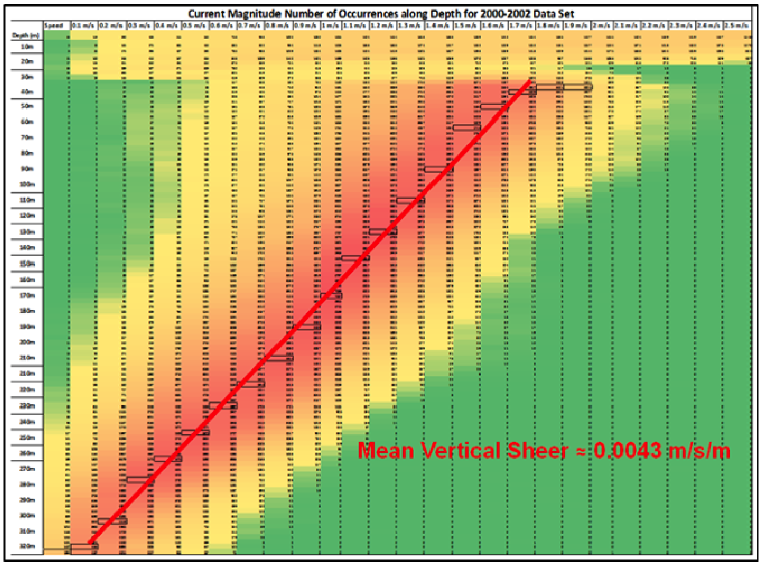




Gliders Transects Within Commercial Area of Interest

The Wave Glider was deployed and recovered repeatedly (10 times) over a 5 nm transect that covered the confirmed location of the bottom mount. DA and Liquid Robotics performed a study comparing the currents profiles measured using the ADCP's mounted in Wave Gliders with profiles generated by the moored ADCP. The goal was to provide DA with statistical comparisons of Glider and moored ADCP data. The two vehicle-ADCP instruments involved in this study were 300 kHz Teledyne Workhorse Monitors. All ADCP's were configured to collect single ping data allowing for maximum flexibility in post processing. For the vehicle to moored comparison, the vehicle data were compared to the moored reference using linear regression resulting in R^2, slopes and intercepts. For the vehicle to vehicle comparison of R^2, average difference in velocity, and standard deviation of the average difference were calculated. The characterization of the resource enabled improved long-term predictability and better understanding of the variability of the Gulf Stream when it comes to forecasting annual energy production and optimal sitting for MHK devices.

Additional assessment of the Florida resource was conducted and involved reviewing current speed measurement data taken in 2000 - 2002 and in Q1 2013 and analyzing these for trends in the vertical shear values. DA's consultant V-Bar conducted this analysis. The speed shear or the change in current speed with depth (mps/m) was determined by subtracting the speed time-series of one depth level from the next depth level. This resulted in a speed shear dataset with a depth resolution of 5m. These data were used to quantify the effect and magnitude of passive depth control (PDC) on the C-Plane. The data indicates that PDC is highly dependent on the vertical shear values. DA is currently analyzing the force balance associated with the vertical shear to further quantify all aspects of PDC. The results are plotted in the figure below which identifies the period with the largest instance of speed shear.

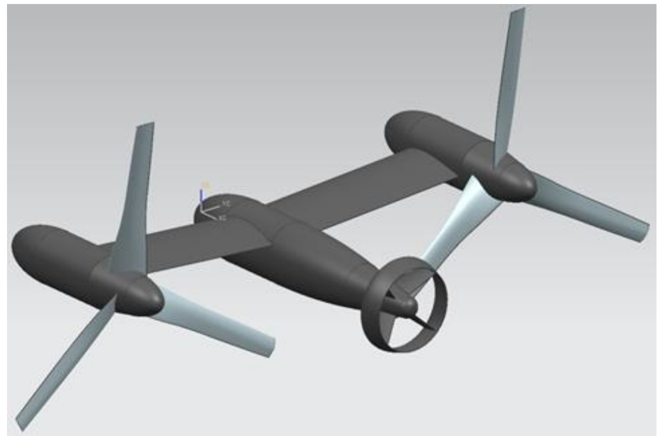


Gulf Stream Vertical Shear Data

Under contract with DA, V-Bar conducted an additional analysis of the 5+ day ADCP data set from the Q1 2013 observations at the highest possible time resolution to find the largest events that may be of interest to C-Plane development. It was found that the maximum time resolution for valid data from the ADCP was approximately 90 seconds in the near range gates. Based on this and previous long-term ADCP analysis it was determined that 10 minute ADCP averages are satisfactory for ocean current energy resource assessment as is similar to modern wind energy resource assessment. ADCP measurements are likely to be sufficient for resource analysis and are a promising technique for use in future commercial analysis for the C-Plane in different project areas. These measurements may or may not be sufficient for specific site suitability analysis depending on the final design criteria for the C-Plane. It was determined that designing the resource assessment and site suitability requirements for the C-Plane be done simultaneously with the design of the machine with commercial site prospecting and development in mind. Refining the methodology and the data gathering has significantly advanced the siting efforts for the C-Plane deployment in the Florida Current: it is a first step in a full understanding of the resource that can be harnessed.

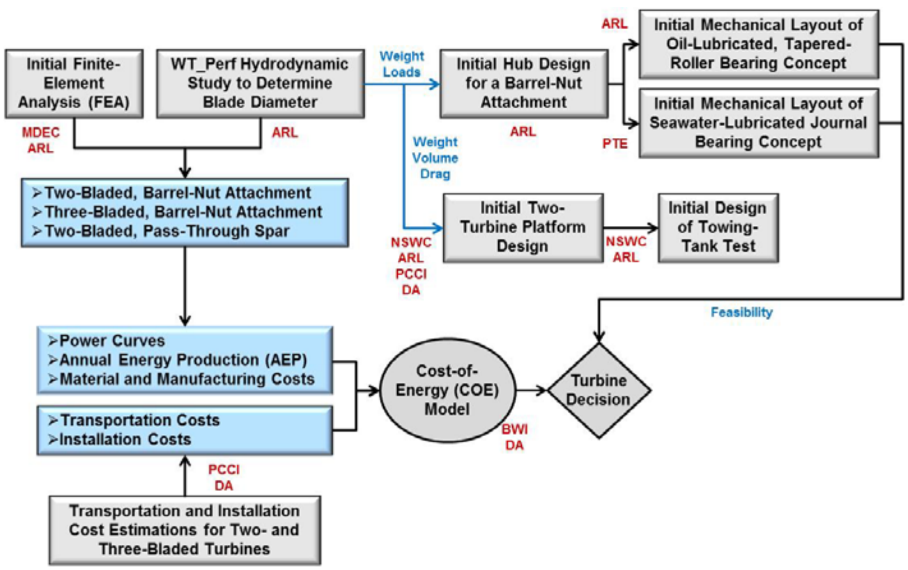
5.2 Steady Computational Fluid Dynamics (CFD) Simulations

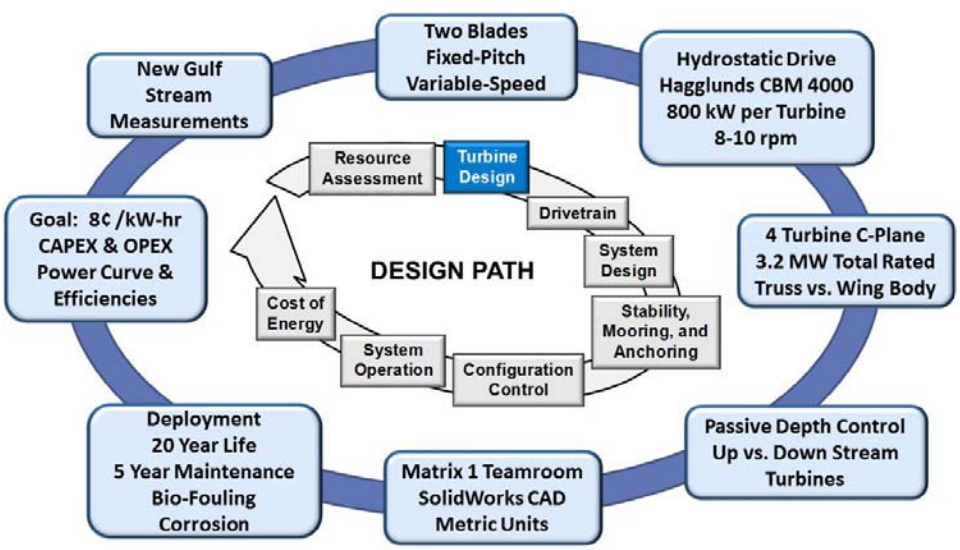
Computational fluid dynamics (CFD) was contracted to ARL Penn State to perform. Because of differences in turbine geometry relative to marine propulsor and pump geometry, ARL Penn State needed to revise some of their post-processing tools in order to accurately determine integrated performance parameters from the CFD results. This revision allowed for comparisons to be made with the NREL design codes HARP_Opt and WT_Perf, and also with XFOIL. The steady CFD results provided input for structural-integrity computations using finite-element analysis (FEA), at different operating conditions of a three-bladed conceptual design at a nominal rated power of 1,300 kW and a rotor-tip diameter of 40 meters. The initial concept involved base lining a 3-Blade Turbine, with two counter rotating 1.3 MW rotors upon which the following trades were conducted:



- Hydrodynamic Trade Studies
- Verification of Design/Analysis Tools (NREL Codes and ARL CFD)
- Hydrofoil Selections
- Fixed vs. Variable Pitch
- Constant vs. Variable Speed
- Tip-Speed-Ratios (rpm)
- Concurrent Engineering
- Blade structural design and hub attachments
- Drivetrain loads
- Platform stability
- Mooring and anchor loads
- Factors of Safety / Stress Knock-Downs

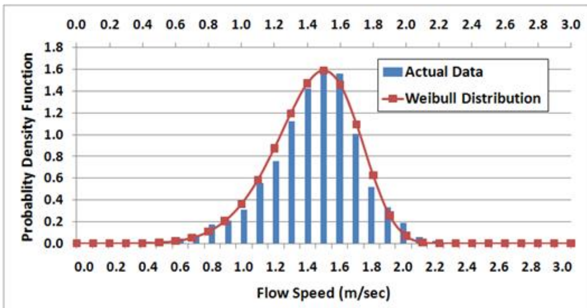
The hydrodynamic workflow and design path are illustrated in the following diagrams:





Fixed Pitch vs. Variable Pitch

Wind Turbines use variable-pitch to maximize efficiency and control structural loads over a large range in wind speeds. An opportunity to simplify Aquantis controls is presented as ocean currents (flow speed) are relatively constant compared to wind speeds in the atmosphere. These speeds are 1-2 mps in the Atlantic Ocean vs. 4-12 mps winds in the Continental U.S. A decision was made shortly after CDR to use fixed-pitch blades because variable-pitch only provides a 1.3% power advantage in an ocean current environment as presented below. By moving to fixed pitch significant mechanical complexity was eliminated from the Aquantis system greatly improving system reliability and adding further benefit proven through our techno economic analysis through OPEX reduction.



Case	AEP (MW-hr/yr)	Power (kW)	Torque (kN-m)	Thrust (kN)	Speed (RPM)
FS/FP	6,400	1,300	6,000	1,300	2.1
FS/VP	6,900	1,300	5,500	1,100	2.3
VS/FP	7,400	1,300	5,700	1,300	3.7
VS/VP	7,500	1,300	3,200	2,300	3.9

Two Blades vs. Three Blades

Theory predicts that maximum efficiency increases as blade number increases. The hydrodynamic benefit of increasing the number of blades becomes greater at low Tip-Speed-Ratios (especially for TSR < 12 for large turbines). In moving from 3 to 2 blades a ~3-6% decrease in annual power would be experienced. And, in moving from 4 to 3 blades a ~1-3% decrease in annual power would be experienced. Increasing the number of blades adds cost and complexity. The higher blade count was used on wind turbines to maintain low torque variation in large wind shears which is not a concern for ocean currents. Using 2 vs. 3 blades increases blade loads but this is not the only factor. Since Aquantis is a flight vehicle, buoyancy, deployment and cavitation margin are additional concerns not present in wind turbines and necessitated further study. A simple blade-centric structural trade-off suggested a two blade approach would be beneficial from a mass and buoyancy perspective. This investigation needs to be expanded to include the impact of fatigue loads and the structural mass of the hub and other structural members.

Fixed Speed vs. Variable Speed

Two concepts of operation exist for modern turbines:

Fixed Speed: Turbine always operates at (or near) a constant rpm

- Pros: Direct grid connection eliminates need for added power electronics
- Cons: Requires a braking (rpm control) system to regulate power because braking can lead to large hydrodynamic loads and surface cavitation

Variable speed: Turbine rpm is proportional to the flow speed

- Pros: Achieves larger annual energy production than fixed-speed designs
- Cons: Excessive flow speeds can result in overloading the blade structure

Variable-depth-control will maintain a 1.6 mps flow speed. In using passive depth control and fixed pitch Aquantis has the AEP advantage of variable speed with the reliability benefit of fixed pitch.

<u>Drive Train: Hydrostatic Drive vs. Gear Drive</u>

A programmatic decision was made to use a direct-drive hydraulic pump. This topology change eliminates a gearbox but impacts turbine hydrodynamics. The hydrostatic drivetrain configuration chosen uses a commercially available hydraulic pump. Two operational scenarios can be used for high pump efficiency:

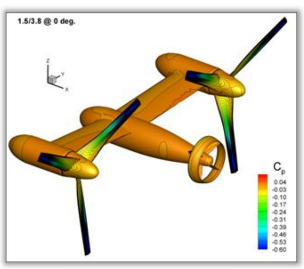
- Running the pump with large internal pressure (service life issue)
- Operating at higher shaft rpm than is hydrodynamically optimal (design decision)

Ideal rpm is in the 4-5rpm range while the present design rpm is 8.7. The pump's rated speed is 8 rpm (12 rpm maximum). Hydrodynamic work was carried out to assess raising RPM to match the desired low speed high torque pump performance of the pump.

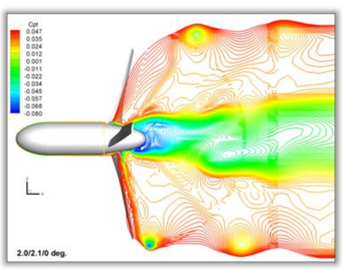
Based on COE analysis, a program decision was made to design the C-Plane using the hydraulic pump with four (4) turbines vs. two (2). Each turbine designed to deliver 800 kW to grid (3.2 MW per C-Plane). COE considering mooring and anchors makes more turbines desirable per C-Plane.

5.3 Unsteady CFD Simulations

In addition unsteady CFD analysis results were provided to the Naval Surface Warfare Center (NSWC) at Carderock for use in developing a stability model for the Aquantis system. This data was used extensively to establish hydrodynamic coefficients for Navy Stability Codes DCAB, BODXYZ and Flightlab.



Full System



Vortex Shedding

Flow Conditions			Mean Forces			Mean Moments		
AoA (deg)	Flow Speed (m/s)	Rotor Speed (rpm)	Fx (kN)	Fy (kN)	Fz (kN)	Mx (kN-m)	My (kN-m)	Mz (kN-m)
0	0.50	1.27	129.8	0.0	0.1	260.1	-2.5	-6.6
5	0.50	1.27	126.0	-0.8	1.7	247.3	-20.3	-7.1
10	0.50	1.27	127.6	-0.6	3.2	258.9	-37.3	-9.4
15	0.50	1.27	126.0	-0.2	4.3	248.7	-55.1	-12.3
20	0.50	1.27	120.6	-0.3	6.0	238.3	-70.2	-14.3
25	0.50	1.27	113.7	-0.3	7.1	212.5	-83.5	-17.0
0	1.00	2.58	504.3	-2.2	0.3	1069.4	-0.2	-23.7
5	1.00	2.58	483.4	-1.3	6.4	1083.7	-91.8	-31.5
10	1.00	2.58	487.2	-0.3	12.4	1054.4	-162.7	-36.2
15	1.00	2.58	489.7	0.7	17.7	1084.1	-233.9	-42.8
20	1.00	2.58	469.3	1.0	22.7	970.9	-308.8	-49.6
25	1.00	2.58	442.1	1.5	27.6	890.0	-358.9	-59.3
0	1.50	3.80	1110.7	-4.4	0.3	2396.6	-6.2	-54.0
5	1.50	3.80	1115.4	-2.2	14.7	2416.6	-196.7	-67.1
10	1.50	3.80	1097.9	0.5	28.6	2373.4	-363.6	-80.8
15	1.50	3.80	1078.1	2.3	39.3	2314.5	-534.4	-94.1
20	1.50	3.80	1036.9	3.2	50.8	2189.1	-691.5	-111.6
25	1.50	3.80	979.5	5.3	61.7	2019.1	-820.3	-134.9
0	1.70	2.80	1289.1	-6.4	2.4	4525.6	-26.6	-28.5
5	1.70	2.80	1277.4	6.7	24.6	4492.7	-171.2	37.9
10	1.70	2.80	1265.6	10.1	40.3	4447.4	-313.8	128.7
15	1.70	2.80	1243.1	17.4	58.5	4352.9	-434.2	204.1
20	1.70	2.80	1196.2	23.8	74.5	4140.5	-552.4	265.1
25	1.70	2.80	1132.1	29.2	89.9	3865.1	-640.3	304.4
0	2.00	2.05	1383.9	-2.9	5.4	7815.8	-88.3	-21.0
5	2.00	2.05	1377.1	22.7	24.7	7757.7	-216.5	448.5
10	2.00	2.05	1350.1	45.9	44.4	7546.5	-343.8	881.5
15	2.00	2.05	1308.2	69.1	63.2	7163.5	-485.0	1259.4
20	2.00	2.05	1224.3	94.3	83.1	6482.7	-792.7	1428.7
25	2.00	2.05	1147.1	107.1	102.1	5873.0	-914.5	1508.3
0	2.50	1.85	1381.7	0.2	4.3	4593.0	-132.2	-27.1
5	2.50	1.85	1408.7	32.6	4.8	4747.9	-100.0	699.8
10	2.50	1.85	1364.9	56.4	14.0	5009.3	-189.5	1177.7
15	2.50	1.85	1277.3	81.9	44.6	5124.1	-506.0	1242.8
20	2.50	1.85	1261.5	127.4	50.8	5581.4	-706.8	1674.5
25	2.50	1.85	1265.2	211.9	46.2	6516.2	-1387.6	2465.1

Matrix of Forces and Moments

The three-dimensional, unsteady CFD simulations consisted of the following cases:

- 36 turbine cases (6 conditions @ 6 pitch angles: 0°- 25°)
  - Input to NSWC Carderock for stability calculations
    - Hydrodynamic coefficients, comparison to FLIGHTLAB
  - Input to PCCI for mooring design
- One full system case

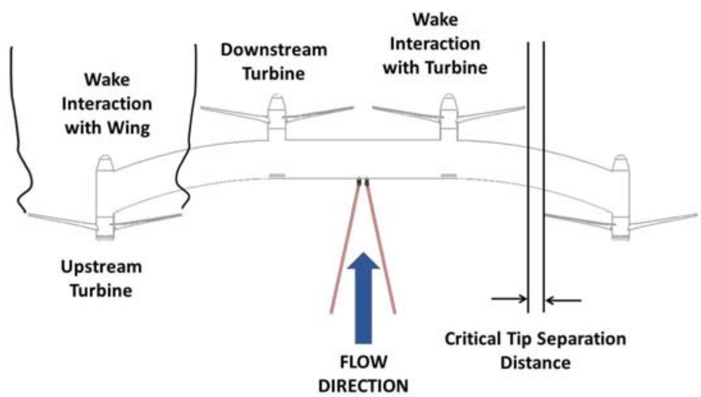
Wake Deficit

The wake generated from the upstream transverse structure is a significant source of fatigue for the rotor and bearings. Unsteady hydrodynamic analysis was performed to understand the wake deficit loads imparted to the composite blades and rotor bearings.

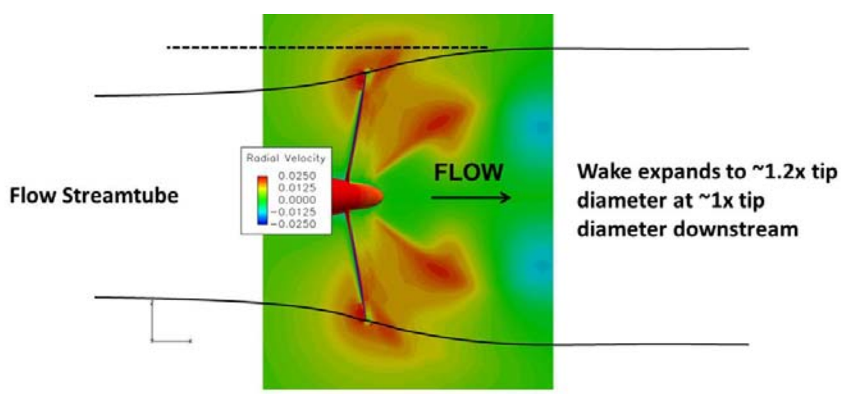
Downstream turbines will experience periodic (unsteady) loading from upstream structures. ARL Penn State conducted force predictions using the UFAM computer code for both an upstream wing (CDR) and the new truss platform structure.

Mean wake deficit from the tower is about 3x greater than that of the wing. In addition, the tower wake will have large unsteady vortex shedding effects. Unsteady forces for the tower are about 2x greater than that of the wing configuration, which may have additional unsteady forces due to Tower vortex shedding effects and potential-flow interaction.

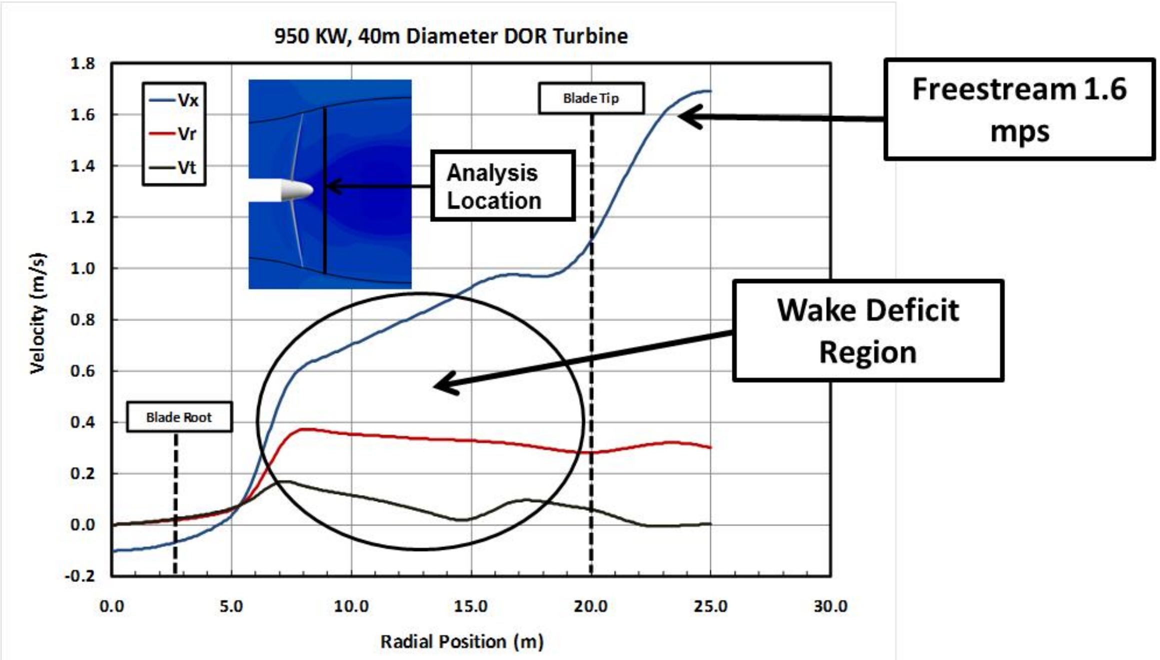
Tip Separation Assessment:



The wake from the upstream turbine expands outward as it proceeds downstream. The recommended tip-to-tip separation distance was calculated to avoid upstream wake interactions with downstream rotor.



The velocity field in the wake of the turbine is significantly less than the free stream velocity.



5.4 Hydrodynamic Design Recommendations

The team formulated a trade matrix taking into consideration control surface actuation versus passive stability tethers, drive topology (upwind/downwind/integrated), grid connection topology, and blade hydro. The following trade studies were carried out with the respective IPT members: (1) the HARP-Opt code provided by NREL was improved upon for blade optimization, structural design, airfoil choices and cavitation margin; (2) blade over speed and locked rotor concept of operations were developed; (3) five promising grid connection topologies were developed in concert with BEW Engineering with a goal of simplifying the power electronics and making them shore based to improve O&M COE; (4) V-Bar reduced previous DA Gulf Stream resource data for application of Boundary Conditions for ARL's initial hydro work; (6) the raw Mat-lab data was parsed by ARL and V-Bar for future BC refinement (this resource assessment effort helped define the next generation instrumentation package to garner the turbulence intensity and spatial resolution to further refine the Aquantis design for power extraction and lifting of components).

The team used the NREL rotor optimization code, HARP_Opt which employs the blade-element momentum (BEM method of rotor analysis), and carried out background investigations of existing concepts for unducted, horizontal-axis hydrokinetic turbines. To address the question of how much energy can the turbine blades extract and still survive structurally, the background investigation focused on energy extraction, aspect ratio, and hub-to-tip ratio. With this information, the Aquantis team began to develop a matrix of parameters for the initial design trade studies. For a given inflow speed, some of the design parameters include tip diameter, hub diameter, shaft rate, and blade number—and the conceptual designs will be evaluated at different inflow speeds. Some of the performance parameters include rated power, structural integrity, surface cavitation, tip-speed ratio, and tip-vortex cavitation.

The team carried out a parametric design study for the Aquantis ocean current turbine. A number of possible foil shapes were evaluated for use as turbine blade sections and the team selected the S816, S817, and S818 foils which were previously designed for NREL for use on large-diameter wind turbines. These foils were analyzed at the proper Reynolds numbers and angles of attack. Then, using HARP_Opt, twelve conceptual designs that varied in rated power and blade number were studied. All of the designs demonstrated efficient turbines, satisfied initial structural-integrity requirements, and avoided blade surface cavitation up to the inflow velocity where the rated power was met.

Some initial analyses of one of the conceptual turbine designs using computational fluid dynamics (CFD) were carried out to primarily gain some experience with this type of geometry and to compare with HARP_Opt, WT_Perf, and XFOIL-outside codes being used for hydrodynamic design.

For computational fluid dynamics (CFD), a new gridding topology was developed in order to use ARL's in-house-developed OVER-REL code to analyze the three-bladed conceptual design at a nominal rated power of 1,300 kW and a rotor-tip diameter of 40 meters. Following this gridding effort, ARL Penn State completed steady solutions of the Reynolds-averaged Navier-Stokes (RANS) equations at six different operating points of the turbine, with each solution repeated for three different turbulence models. Comparisons of these results were made with the WT_Perf results obtained during the design study. The WT_Perf results for the rotor power compare very well with the CFD results for all three turbulence models, until the flow speed increases to the point where the blades approach stall. After blade stall, the flow field becomes very three dimensional and unsteady, and no code can be expected to model the flow accurately. The analysis showed good agreement for the power coefficient. The CFD results do show a slightly reduced power coefficient, where the improved predictions of the viscous fluid losses would yield a decreased turbine efficiency.

While it is difficult to predict the flow behavior while the blades are stalled, it is important to predict the turbine-blade performance prior to and at the stall point. Since WT_Perf utilizes a foil-performance look-up table based on XFOIL computations, the CFD results were compared with XFOIL results, for the foils used for this turbine design. Experience indicates that XFOIL can over predict the maximum lift coefficient at stall, and the Aquantis team needs this information before completing an improved preliminary design of the turbine. In addition, a mesh was generated that allowed for a finite- element analysis (FEA) of the structural integrity of the turbine blades, which was compared with the HARP_Opt results using Bernoulli beam theory during the design study. The FEA loads came from the CFD results. This check on the structural integrity is also required before proceeding with the preliminary turbine-blade design and attachment scheme. In addition work is being carried out that extends the beam theory used in HARP_Opt to include blade deflections due to bending. The next extension included torsional motion of the blades under load.

The Aquantis team targeted a fluidic drive system, and the desire was to use a commercially-available pump which would require a turbine that operated at higher RPM at the rated power of 1,300 kW, rather than at 4 rpm. The major parameters that influence the turbine performance are the number of blades, the tip- speed ratio (TSR), and the hydrofoil lift-to-drag ratio, L/D, of each blade section. If blade chord lengths are maintained, decreasing the blade number will increase the optimal TSR—and, thus, the rpm—but at the expense of a reduced efficiency, or power coefficient. A smaller number of blades can also reduce cost and complexity. At the time, a three-bladed turbine appeared to be the best choice, although a two-bladed turbine was viewed as simplifying deployment and increased rpm about 0.75, but at a reduction in efficiency of about 3-6%. Follow-on analysis showed a two blade fixed pitch variant provided the necessary buoyancy, deployment, and efficiency needs while meeting the structural requirements

The Aquantis team was able to achieve a L/D of approximately 80 for the blade sections through a careful selection of hydrofoils. The optimum TSR is in the range of 4 to 7. All of these concepts are for fixed-pitch turbines. Variable-pitch turbines—as commonly used for wind turbines—allow for more control of the rpm for varying current speeds, which maximizes the output power—but at a higher cost, and more complexity, and less reliability. A fixed-pitch turbine will always operate at variable rpm, proportional to the flow speed squared, unless it is forced to brake or it naturally stalls. While design tools, such as HARP_Opt, have a fixed-pitch, fixed-speed mode, one cannot operate such a turbine at a fixed speed without rpm control.

However, the key issue for the drivetrain is whether or not one can design the turbine for a higher rpm or higher TSR. At an rpm of 10, TSR equals 13.1, which is far above the optimum range for a 40-meter-diameter turbine. To prove this point, calculations were performed at a number of values of TSR for the current design. The power coefficient versus TSR curve showed a plateau around an optimum value of 5.2 for TSR, but the dramatic falloff showed that a value of 13.1 could probably not be achieved. The Aquantis team continued to investigate the feasibility of operating the 40 meter diameter turbine at higher TSR values than 4-7. These TSR were achievable by ganging the hydrostatic pumps in series to increase specific torque capacity.

The next major challenge for the hydrodynamic design of the turbine is the high-current-speed condition. After the turbine reaches its rated power at a given current speed (~1.6 m/sec) the power, thrust, torque, rpm, and blade loading will all continue to increase unless this fixed-pitch turbine is forced to brake or it naturally stalls. Without braking or blade stall the increased blade loading will eventually cause blade structural failure. Stall regulation was investigated as a way to naturally stall the turbine blades. However, stall regulation will be challenging given the predictability of post stall behavior using CFD. Even if successful, the uncertainty in analyzing the flow in a stalled regime will still be an issue. Eventually, testing may be required to check this condition if the testing Reynolds number is sufficient. In any event, the Aquantis team needs to investigate post stall behavior to facilitate the design/use of a braking system while maintaining stability and hydrostatic system pressure.

Another key question that required resolution was whether the turbine rotor blades should be located upstream or downstream of the wing that attaches the turbine nacelle to a center nacelle. Three major technical issues factored into the decision: 1) fatigue; 2) system stability; and 3) mooring. To address the fatigue caused by the downstream turbine rotor blades operating in the viscous wake of the wing, first estimates were made of the total drag from a conceptual wing and then estimated the wake using an empirical Gaussian wake deficit model. The wake depths at the rotor-blade leading edge ranged from 20-35% of the freestream velocity. The 10° aft rake of the conceptual design, added to help system stability, also helped to dissipate the wing wake before it interacted with the tips of the turbine blades. ARL’s in-house proprietary code based on unsteady airfoil theory was used to determine unsteady forces and moments (UFAM). The resulting unsteady forces were provided as input for FEA fatigue computations.

The Aquantis team continued to have discussions regarding the turbine design value for the shaft rate—in revolutions per minute (rpm). For this conceptual turbine design, the NREL code called HARP_Opt was used. Given constraints on the maximum strain from the bending moment and on surface cavitation, this code optimizes the hydrodynamic design of the turbine to yield the best annual energy production. And the best hydrodynamic design for this turbine uses a value of approximately 4 rpm. HARP_Opt uses the NREL code WT_Perf—a blade-element momentum method (BEM)—to perform most of its calculations. But, WT_Perf is an analysis code that requires turbine-blade geometry. Therefore, to evaluate a turbine at a higher shaft rate—(a design not optimized hydrodynamically) a new design code was written.

After comparing the steady, three-dimensional solutions of the Reynolds-averaged Navier-Stokes (RANS) equations to the initial foil calculations using XFOIL, ARL Penn State tabulated a series of steady, two-dimensional RANS computations at different Reynolds numbers and angles of attack, for a more direct comparison with XFOIL. The comparisons are quite good with XFOIL slightly over-predicting the maximum lift, as expected. This comparison is valuable in looking at the stall characteristics of future designs.

Next a series of unsteady, three-dimensional RANS solutions were completed to support the Naval Surface Warfare Center (NSWC) at Carderock for use in developing a stability model for the Aquantis system. These computations, with the three-bladed turbine operating on a turbine nacelle, have been performed at six operating conditions, each at six different angles of attack of the nacelle to the Gulf Stream inflow, yielding 36 significantly-large computations. ARL Penn State tabulated the results for all three components of force and all three components of moment (about an agreed-upon location), giving both the mean values and the root-mean-square (rms) values.

Parametric design studies were revisited using earlier the NREL code, HARP_Opt and two changes were implemented. First, V-Bar provided their assessment of the resource data acquired by Florida Atlantic University, and ARL Penn State fit a Weibull Distribution to the data for use in HARP_Opt. Second, for the hydrofoil characteristics used within HARP_Opt and WT_Perf, the XFOIL computations were replaced with CFD computations, to better represent the maximum lift of the hydrofoils. Then HARP_Opt was used to find new conceptual designs for all the variable-speed cases studied last December. In addition, in response to groups working on the drivetrain conceptual designs for fixed-speed cases using the same parameters were analyzed. These designs do not require continuous mechanical breaking at the higher inflow speeds.

To further the Aquantis design a new in-house design code called WT Size was developed. Using the same blade-element momentum (BEM) equations as the NREL analysis code, WT_Perf, WT_Size solves these equations inversely to design blade geometry for a specified blade-loading distribution. This new code allowed the Aquantis team to perform design studies at higher values of *rpm* or tip-speed ratio (TSR) than the hydrodynamically-optimized designs found using HARP_Opt. Using this code, the team found that designs with higher values of *rpm* and TSR are possible, but the resulting smaller chord lengths and higher aspect ratios need to be checked to see if they are structurally feasible. This further study can focus on either three-bladed turbines—which have a better hydrodynamic efficiency—or two-bladed turbines—which have system advantages for material cost and installation.

The Aquantis team also referred back to the previous calculation performed to determine the fatigue loading on the turbine blades as a result of the blades rotating through the wing wake. Expanding this work, calculation on all three unsteady-force components and all three unsteady-moment components were extrapolated that would be experienced on the shaft. These unsteady forces and moments will be available to the groups working on the drivetrain. The unsteady thrust and the unsteady vertical bending moments are the most significant.

Aside from turbine power performance the team focused on two other performance criteria: cavitation and structural integrity. Surface cavitation occurs when the blade sections operate at off-design angles of attack, which leads to regions of low static pressure on the suction (o4r upper) surfaces of the blades. Operating with surface cavitation for extended periods of time will lead to cavitation erosion of the blade surfaces. Variable-speed turbines will operate cavitation free up to the point where the current speed is high enough to reach the rated power. At higher speeds, some type of braking—which has its own challenges—would lead to blade stall and surface cavitation. Fixed-speed (or nearly fixed-speed) turbines will operate at the on-design angle of attack up to the point of generator engagement. At higher current speeds, the Aquantis team will need to determine when the angle of attack changes enough to lead to surface cavitation and erosion. It is expected that surface cavitation will occur first near the highly-loaded blade roots and then extend radially outward toward the blade tip as the flow speed changes.

During this period, NSWC performed preliminary DCAB analysis of a 2 rotor 45 m diameter monoplane design showing that the overall concept is dynamically stable (stable response to 50,000 lb. vertical load perturbation case and a 100,000 lb. side load (+Y) perturbation case). NSWC also confirmed diving to limit loads in high currents, and developed a theory for adding rotor gyroscopic moments to the C-Plane, which was integrated into the code.

Rotor gyroscopic effects were added to the DCAB software code to determine effect on hydrodynamic stability of the platform. A second mooring line (vertical restraint line) was added to the DCAB software to determine the effect of the restraint line on the platform dynamics (pitch, yaw and roll) during off-axis current events.

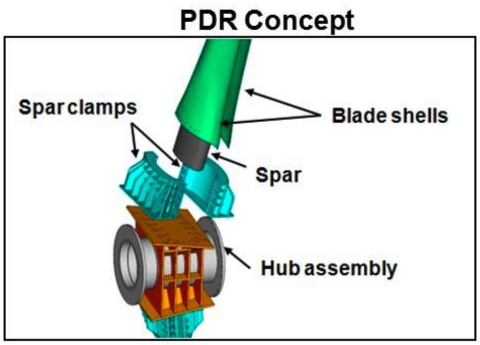
The decision up until the Preliminary Design Review (PDR), as conducted in October 2011, was to move to a 4-rotor/2-blades per rotor platform (3.2MW) rather than the 2-rotor/3-bladesper rotor platform (2.5MW) to help offset mooring and installation costs by reducing the total number of platforms required to produce a 100MW field.

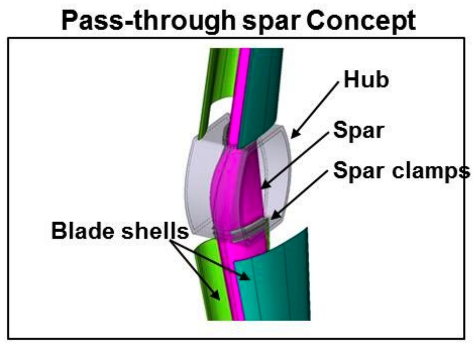
Three basic C-Plane configurations (4-rotors downwind, 4-rotors upwind and 2-upwing/2-downwind) were investigated and the hydrostatic balance points were determined for each configuration at zero speed and at 1.6 m/s current (operational flow).

Two nacelle/rotor support structures (wing and truss) were structurally evaluated based on nacelle weight and buoyancy forces and moments and the hydrodynamic forces (lift, drag and associated moments) from the rotor and support structure. Due to the large torsional moments produced by the nacelle/rotor components weight and buoyancy forces, the truss structure was deemed to be more structurally adequate, lower in weight and less costly than the wing structure. The wing structure could potentially cause adverse pitching and diving forces due to wave turbulence and current eddies.

The Truss/4-rotor platform (no wing) was analytically shown to dive to a deeper depth to limit rotor blade loads in high currents (above1.6 m/s).

Mass properties were updated based on latest information concerning nacelle, rotor, hub and support structure (truss) designs and fed into the hydrostatic stability analysis. After the PDR it was decided to move to a through hub spar to reduce hub weight and better match CP by stacking pumps in series to improve hydrodynamics.





5.5 Finite Element Analysis (FEA) Structural Analysis

Marine and Hydrokinetic (MHK) vs. Wind
When comparing an ocean current MHK blade to a wind-blade of similar power-rating, effects of fluid density and flow speed result in an MHK rotor blade with a span that is 62% of the wind blade.

	Density	Flowspeed @ Rated Power	Power	Efficiency	Area	Radius
	[kg/m^3]	*[m/sec]*	[MW]	[-]	[m^2]	[m]
Seawater	1026	1.6	same	same	0.388	0.623
Air	1.225	11	same	same	1	1

The ratio of blade length also applies to sectional chord and thickness as the rotors are designed for similar solidity. The selection of airfoils has an additional implication resulting from the sectional Reynolds number, given by the operating conditions. The MHK machine operates at extremely high Reynolds numbers due to its large size, speed, and high density operating environment. The 70% blade span Re# is approximately 10 Million, which is more than double what is experienced by some of the largest wind turbines in the world. At these high Reynolds numbers, early turbulent boundary layer transition is inevitable resulting in very high drag, particularly on thicker airfoil sections. The change from air to ocean currents drives the design towards a 37% reduction in blade span while requiring it to be proportionally more slender due to drag effects at high Reynolds numbers. This effect on the shape results in a lower sectional moment of inertia, reducing the beam's ability to resist bending moments.

The MHK rotor experiences torque loading that is **4.3 times higher** than that of a wind rotor of the same power level and efficiency. These ratios can be translated from the rotor to representative blades, where rotor torque is generated via moment summation of lift and drag forces acting on airfoils along the span of the blade. Since only a small portion of the lift vector is aligned with the direction of rotation, the remaining component must be reacted by the blade in the form of sectional bending moments. Bending moment, at any given blade station is a product of the element force (thrust and torque) and the moment arm (radius). Since the thrust load and torque load are simply components of the elemental lift force, the ratio of sectional bending moments generally follows the ratio of thrust. It can be estimated that for rotors of equal power, the hydrokinetic blade has to react 4.3 times higher bending moments due to the change in density and flow-speed for a given power rating.

Salt Water Effects on Material Properties

Composites or fiber reinforced plastic (FRP) have been successfully used in marine structures for over 50 years. High structural performance, light weight, and good corrosion resistance have made composites a preferred choice from small boats to submarines. Long-term material studies have found that FRP materials are subject to water absorption through diffusion, primarily through the matrix structure. Strength of the composite is reduced by corrosion of the fibers themselves, and matrix degradation negatively impacts the micromechanics of the laminate stack. Effects have shown to be dependent on matrix and fiber choice. This effect can be addressed in the design process by applying knock-down factors on material strengths to reach acceptable and safe design allowables. Successful utilization of these materials is possible, but it must be carefully tailored to the environmental conditions. A review of several testing programs has been conducted to arrive at preliminary material properties used for blade allowables. Ifremer France, Montana State University, Thapar University (India), and the WMC Knowledge Center (Holland) have reported on projects aimed at understanding the structural performance of composite materials in a salt-water-saturated state. Additionally, Det Norske Veritas (DNV) gives some guidance on designing for the effects of water in the Composite Components section of Offshore Standard DNV-OS-C501. Further study on the specific degradation of the down selected material used in the detail design phase will be required; however a conservative knock-down was distilled from the review of the aforementioned test programs:

Closed Form Blade Optimization Tool

In order to perform fast hydro-structural iterations, a simplified closed form solution was developed and integrated into a tool, capable of rapid iterations on structural design. The blade is related to a simple sandwich beam with the following assumptions:

- Unidirectional composite sparcaps as facings
- Syntactic or SAN foam as core material
- Baseline material properties (e-glass, syntactic foam)

The ability of the new tool to predict strains in the primary load-carrying components of the blades was checked by running the tool to predict the latest FEA modeled blade from the previous design iteration, and comparing strain predictions. The model was able to predict strains quite well with the exception of the transition area where 3D effects and the large shell component (not modeled in the closed solution) are expected to give reasonable deviations.

The closed form solution runs the following routine:

1. Global variables including material stiffness, strain allowables, densities, costs and geometric calibration factors to compensate for 3D shape effects are input.
2. A proposed blade definition is entered as station-wise inputs for radius, thickness, chord, thrust/length and torque/length. Additional manufacturing inputs are entered to define and place a linearly tapering spar-cap into the planform of the blade.
3. Sectional bending moments are calculated and for each section a simplified beam station is created.
4. Moments are applied to the stations and an optimization routine calculates required spar-cap thickness at each station.
5. The simplified structural design is related back to a full blade layout. Component volumes mass and cost estimates are calculated and summed.

A unique feature of this model is the ability to drive maximum spar thickness and use this to drive back to a thickness distribution. This is an important feature since the sections are very highly loaded compared to a wind turbine blade, and spar-cap thickness becomes very high, reducing sectional efficiency, and presenting challenges to manufacture and quality inspection.

A composite part is only as good as the manufacturing process in which it was created, and the multitude of manufacturing problems in current wind-blades and MHK blades alike highlights the need for manufacturability included in a holistic design process.

Syntactic Foam

Various foams were investigated to reduce wet weight, prevent corrosion, and reduce void spaces. Syntactic foams achieve low density by incorporating glass microspheres into the formation. They are pourable down to a density of 445 kg/m3. For the depths being considered for Aquantis, expanded-plastic foams were also investigated, which have both a lower density and a lower cost. One cannot pour these types of foams, but they can be machined and bonded into place.

FEA Analysis

FEA analysis and a three-dimensional lamination code was used to analyze the structural integrity of the same conceptual turbine design used for the CFD and UFAM analyses, using solid E-glass/epoxy composite blades. The blade-to-hub attachment at this point was modeled explicitly. Worst case environmental material properties were applied to account for water uptake, temperature exposure, and other effects. At a single condition, the FEA computations essentially gave the same results using either the line loads from the design codes or the distributed static-pressure loads from CFD. Using the CFD input, the structural integrity of the blades was analyzed at six different operating conditions (six different current speeds). Note that both the CFD and FEA results assumed either braking or natural stall to control the rpm when the current exceeded 1.6 m/sec and the rated power was met. The results showed large factors of safety and small tip displacements. For the fatigue analysis, a quasi-steady analysis was performed by simply adding the peak fatigue load to the steady load from the CFD static-pressure distribution. Again, even for fatigue, the factors of safety were large, indicating a fatigue life that would exceed 50 years, which suggests that a downstream turbine should be acceptable. FEA computations were performed to analyze the effect of filling part of the blades with metal spars and syntactic foam and with reducing the blade chord. Finally natural frequencies of the various blades were estimated.

None of the hydrodynamic blade designs have merit if they are not structurally viable, so the Aquantis team checked the structural integrity of each design using a preliminary finite-element analysis (FEA). This analysis is especially important at higher values of *rpm*, where it is desirable to reduce the blade chord lengths to retain efficiency. As a result of the Conceptual Design Review and in preparation for the FEA, a blade manufacturing concept and a blade-to-nacelle attachment concept were selected.

Underwater (UW) composite applications require a different set of paradigms and design requirements than in-air structures. Critical considerations in undersea environments include:

- Moisture absorption and effect on material properties
- Pressure loads and ingress of high pressure water
- Corrosion, calcareous deposits
- Degradation of adhesives
- Inaccessibility for inspection
- Inability to re-torque barrel nuts

Attachments: Sub-modeling Barrel Nut vs. Through Hub
Sub-modeling was carried out develop for a section of the spar with a single screw. This sub-model included a metal insert and mounting plate, adhesive layer, and glass composite spar section. Displacement restraints applied to the top of the model were based on displacements from the global model. The boundary conditions and assumptions were based on 1.6 m/s loads and highest loaded screw in tension. Eight cases were considered:

1. No flange on insert
2. No preload
3. Preload based on grade 2 screw
4. Preload based on grade 8 screw
5. Flange on insert
6. Preload based on grade 8 screw
7. Preload based on grade 8 screw, reduced adhesive at top
8. Preload based on grade 8 screw, reduced adhesive at top and bottom

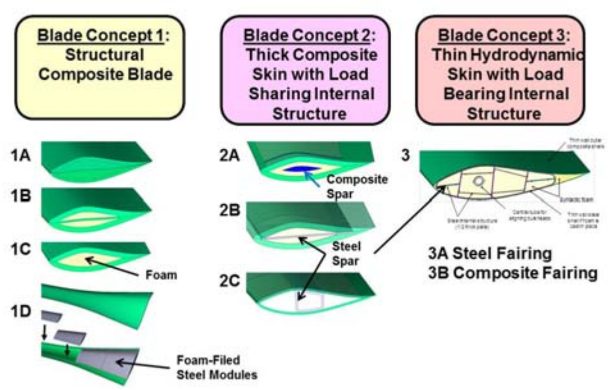
Bonded-in inserts results
High screw preloads were required to prevent lift-off of the spar from the base. A flanged insert is required to reduce stresses on the laminate and adhesive due to high screw preload forces. With a “perfect” adhesive joint, laminate fatigue margins of safety of less than zero for transverse shear failure mode resulted in an adhesive margin of safety less than one. Typical adhesive joints also exhibit multiple laminate failure modes.

Barrel nuts results
High screw preloads were required to prevent lift-off of the spar from the base. Margins of safety of less than zero for all laminate fatigue failure modes were calculated. Hub weight and ovalization of the attachment was analyzed at different rotor diameters to determine the impact to the seawater bearing.

Pass-through spar summary:
The pass-through spar design was determined to be feasible. A simple model was developed to assess feasibility with a constant cross-section spar that passes straight through hub. The spar was prevented from radial displacement. Preliminary calculations indicate positive margins of safety for the composite laminate and the screws at the joints..

5.6 Structural and Material Design Recommendations

The blade-to-hub attachment remains a challenging aspect of the program. The traditional Naval propulsion pin-and-socket scheme has two issues: physically fitting such a large blade into a hub socket with a close tolerance and the lack of hub space to install the bearing-and-seal package radially under the turbine blades. The traditional method of attaching wind-turbine blades to a hub has issues as well. Therefore, the Aquantis team came up with several other concepts, some of which were variants of other concepts, and held creativity sessions to discuss the pros and cons. The attachment concepts included a tang on the outside of the hub, a hub with stepped pockets, raised sockets, a bolted flange and spline, and segmented shoes while alternative blade concepts included the use of metal spars and the use of syntactic foam. The various concepts were evaluated for material, structural integrity, complexity, ease of manufacture, ease of assembly, number of fasteners, robustness, dry weight, wet weight, cost, and space allocation for the bearing and seal package. Various coatings for resistance of bio-fouling, cavitation, and corrosion are also being discussed. All the concepts are applicable to either three-bladed or two-bladed turbines, although a separate pass-through concept was considered that is applicable only to a two- bladed turbine.



A number of blade-manufacturing concepts and blade to nacelle attachment concepts were developed. The various blade manufacturing concepts all incorporate some combination of composite material, metal, and syntactic foam. Then more quantitative evaluation was initiated of the various concepts. Using loads determined from the hydrodynamic design and analysis of the conceptual turbine, finite-element analysis (FEA) was performed of the various blade and attachment concepts. After determining if the concepts were structurally viable, cost was used as the primary metric for evaluating concepts, with weight being the secondary metric. The cost metric includes both acquisition and life-cycle costs. As a result of these evaluations attempts were made to simplify the concepts to reduce both cost and weight, while still being structurally viable. Some of these simplifications will affect the hydrodynamics, so the effect on power-extraction performance was analyzed.

The initial blade-manufacturing concept involved a steel spar with steel bulkheads, using low-carbon steel and as simple pieces as possible. A composite skin would attach to the bulkheads to achieve the hydrodynamic shape, but the steel structure will take all of the loads. Foam will be used to increase the buoyancy. Different types of foam were investigated, including syntactic foam and silicon foam, which can possibly be poured and is much less expensive. However, it cannot take any load. Different types of foam were ordered for testing at higher pressures, to mimic depth, and determine the characteristics of the various foams to crushing at higher pressures and to absorbing water, which could change the blade buoyancy over time. The primary reason for selecting this blade manufacturing concept is to reduce material cost. The FEA, using brick elements, provided the factor of safety for bending stresses and strains, along with the tip deflection and the torsional twist of the blades.

The blade-to-nacelle attachment concept also involves steel, using simple plates and/or tubes. The attachment will be outboard of the outer surface of the nacelle, to simplify the attaching procedure. The redundant joint will involve both a male/female joint and bolts. A centerline steel tube will enclose the bearing-and-seal package. Again, a composite skin would form the outer surface of the nacelle, if necessary, and foam could be used to increase buoyancy and help protect the steel from the seawater. FEA shell models were used, for the attachment scheme. To move forward with both the blade manufacturing and blade to nacelle attachment concepts, a consultant was brought in with experience with building steel structures that exist for many years subsea. The consultant helped with issues regarding manufacturing methods (including items such as welding), corrosion (including corrosion protection), and manufacturing costs.

Primarily for cost savings, a blade manufacturing concept developed involves a steel spar with steel bulkheads, a composite skin, and foam to increase the buoyancy. The Aquantis team has a great deal of expertise and experience with regard to marine composite materials. However, the team had less expertise and experience with regard to marine structures with steel and foam. For steel structures in seawater, Newport News Shipbuilding (NNS) was contracted to discuss material selection, manufacturing processes, corrosion protection, and costs.

Various foams were investigated to reduce wet weight, prevent corrosion, and reduce void spaces. Syntactic foams achieve low density by incorporating glass microspheres into the formation. They are pourable down to a density of 445 kg/m^3. For the depths being considered for Aquantis, expanded plastic foams were also investigated which have both a lower density and a lower cost. One cannot pour these types of foams, but they can be machined and bonded into place. Four types of expanded plastic foams were procured of various densities and were tested by immersing them in saltwater at high pressure and checking them weekly for water absorption.

To help support the structural design, the Aquantis team made some minor compromises to the hydrodynamic design: (1) reduction in blade aft rake from 10° to 7°, (2) reduction in root blade chord, and (3) reduction in root blade pitch angle (resulting in less blade twist).

The Aquantis team performed design work looking at a wind-turbine-type blade/hub joint, with a round-ended spar with 23 Grade 8 screws within 23 barrel nuts. This six-inch thick spar would have to be 6.5-feet in diameter. This spar is significantly larger than the current through-the-hub spar design, so it was necessary to adjust the shape of the rotor blade from the hub to some distance outboard. The updated hydrodynamic design would either have to use much thicker blade sections or transition from a cylinder at the hub to a hydrofoil shape some distance away. This design would likely show that the efficiency will decrease, leading to an increase in blade tip diameter so to obtain the same power which, in turn, would increase the moment arm for the structural design and, thus, increase the spar diameter. This entire iteration process would take some time. However the hydrodynamic efficiency would decrease and the weight and cost would increase, driving the up the cost of energy (COE). In addition, this design could very well have a cavitation issue with the thick blade or cylinder at the blade root.

The Aquantis team completed a preliminary structural design of the turbine using finite-element analysis and some in-house lamination codes. The structurally-robust design had acceptable factors of safety. The turbine hub would be built around a center mold of syntactic foam. This model would be surrounded by a through-the-hub spar fabricated from an e-glass/epoxy composite material. Then, additional molds of syntactic foam for the leading- and trailing-edges of the blade would be applied to the spar. Then, e-glass/epoxy skins would be attached to the spar, one for the suction surface of the blade, and one for the pressure surface. The whole blade assembly would be installed in a hub. The primary blade loading is in the spanwise direction due to the bending moment on the spar. The joint included a steel clamp with Grade 2 screws.

A listing of composite material properties for use in determining knockdowns for the marine environment was provided by ARL. Using information provided by Mechanical Design Engineering Consultants (MDEC), a conceptual hydrodynamic design of a turbine with a cylindrical root section, transitioning to a hydrofoil at 25% span was developed. This design provided more blade surface area and, especially, more blade volume. Thus, this design showed more favorable buoyancy characteristics, at the cost of some powering performance. This type of blade/hub joint would require significantly more work to determine the structural integrity, weight, and cost. Except when using this type of joint, which one can also use for a two-bladed turbine, the three-bladed turbines demonstrated only one real advantage over the two-bladed turbines, a reduction in loading and flap moment per blade with a corresponding increase in cost.

As part of our parametric blade studies a close eye had to be kept on wet weight to achieve static stability. The two bladed variant is much easier to deploy from an installation and maintenance perspective whereas the three blade rotor experiences much less root bending moments. These were analyzed through a techno economic process to down select. While the blade wet weight is important, an over-riding criteria is the attachment method employed which is parasitic to the wet weight. As this is the preferred attachment method in wind turbines the knockdowns have to be evaluated for fatigue in saltwater, creep, preload, hydrogen embrittlement, KT- stress concentrations, corrosion, inspection, etc. to develop the proper design criteria.

5.7 Small-Scale Water-Tunnel Testing

After careful evaluation of the options and facilities available it was decided to substitute a tow tank test for the proposed water tunnel testing program.

5.8 System Stability Analysis

Work was conducted at the Naval Surface Warfare Center (NSWC) at Carderock on developing a stability model for the Aquantis system. The key to this model is representing the turbine rotor blades. To develop the necessary hydrodynamic coefficients, it is required to know how the forces and moments on the blades change as a function of flow angle. ARL Penn State provided this information by using their in-house-developed unsteady RANS code, TCURS (based on OVER-REL). Therefore, a computational grid was generated for the same three-bladed conceptual turbine design used for the steady RANS computations (described previously) and an unsteady RANS computation was started at a flow angle of 0° relative to the turbine nacelle. Later computations included 5°, 10°, 15°, 20°, and 25°.

Navy Carderock was engaged to determine the stability coefficients/ equilibrium of the c-plane. For use with DCAB or other coefficient based simulation tools used in the Navy a process was outlined for establishing these coefficients and inputs needed from other team members. Tow tank test preparation was begun to garner experimental stability coefficients.

As part of the Aquantis C-Plane turbine modeling effort rotorcraft analysis tools were applied to predict forces and moments on an isolated rotor/nacelle from the C-Plane design. Studies by the National Renewable Energy Laboratory (NREL) have demonstrated that rotorcraft analyses can be successfully applied to predict loads on wind turbines. The Flightlab code was applied to the NREL validation test case and was found to match well when predicting on-blade angle of attack distribution and turbine power under normal operating conditions.

To date, the Aquantis C-Plane analysis effort has focused on generating force and moment predictions for use in a stability derivative analysis. The Aquantis C-Plane analytical model consists of a blade-element rotor model with nonlinear unsteady aerodynamics and dynamic wake, and a potential-flow panel model of the nacelle. For the stability analysis, forces and moments at the turbine hub are predicted for a nominal current flow velocity of u = 1.5 m/s and pairwise variations of in-plane velocities (v, w) and rotational rates (p, q, r). The Flightlab code was modified to allow automated parametric input of these velocities and the predictions were compared against nonlinear transient analyses to assure their accuracy. Preliminary results from the v-w sweep have been transmitted and additional results are forthcoming. In addition, the Flightlab model is being exercised to predict forces and moments over ranges of flow velocity, yaw angle, and rotor rpm for comparison with ARL predictions. Consideration is being given to eventual expansion of the current isolated rotor/nacelle model to a complete model of the C-Plane system for future analysis efforts.

A spreadsheet analysis was conducted to estimate the in-air and in-water weights of the C-Plane platform and to determine the platform center of gravity (CG) and center of buoyancy (CB) for hydrostatic stability assessments. The analysis was conducted using the single generator, fluidic drive configuration for the power generation system. The platform concept being analyzed was per the preliminary concept design as defined in layouts furnished by DA. The concept design shows a blunt nose nacelle of approximately 14.6 meters in length including the rotor and spinner cone. To date, the analysis has been concerned with the nacelle and rotor system and has not included the wing and mooring attachment.

Pitch plane static stability was investigated using the weight assumptions above and the forces and moments developed by the nacelle/rotor system as calculated by ARL computational fluid dynamics (CFD) runs for various current velocities and pitch angles of attack to the flow. For static pitch plane stability the static (net buoyancy) and hydrodynamic (flow velocity dependent) forces and moments summed about the C-Plane longitudinal mooring point location must be zero and the pitch trim should be maintained at 0°. A wing connecting the 2 nacelle/rotor assemblies at the forward end of the nacelle was used to obtain hydrodynamic lift and additional buoyancy.

With the mooring line attached to the forward end of the nacelle nose the C-Plane could not achieve static stability due to the large negative pitching moment produced by the net buoyancy of the nacelle located so far aft of the nose. Calculations were performed to determine longitudinal mooring point locations to satisfy static pitch plane stability throughout the current velocity range. Due to the high net buoyancy of the nacelle/rotor assembly, the mooring point location required for static stability balance was found to be aft of the nose. This was not acceptable since tow points (mooring points) located this far aft on towed bodies typically exhibit yaw plane instability.

Using computational fluid dynamics, ARL Penn State completed a series of unsteady, three-dimensional RANS solutions to support the Naval Surface Warfare Center (NSWC) at Carderock for use in developing a stability model for the Aquantis system. NSWC Carderock completed unsteady CFD results at six pitch angles, each performed at six operating points of the variable-speed, three-bladed, 1.3 MW conceptual turbine design resulting in 36 computational solutions. In addition, ARL Penn State completed a larger case that includes a wing, center body, and ring wing for added stability.

NSWC has been using its Flightlab rotor dynamics code to predict rotor forces and moments and to compare with that produced by ARL. The correlation was very good (typically within 10% of the CFD predictions) for the forces predicted as a function of pitch angle of attack. There is less correlation with the moments due to some modeling inadequacies with the rotor wake and airfoil look-up tables that were investigated to bring these correlations more in line with CFD predictions. The Flightlab analyses produce result files in the order of minutes for the run cases instead of requiring hours, as is the case for CFD. Thus, analysis cases could be run in a short period of time with a reasonable amount of correlation with the CFD modeling. Flightlab was utilized to determine the effects of uneven flow in the vertical and horizontal planes of the C-Plane platform to determine the effect on stability.

Work has been done to modify the NSWC hydrodynamic stability prediction software code (DCAB) to incorporate multi-body and multi-line parameters for analysis.

5.9 Moorings and Attachments System Definition

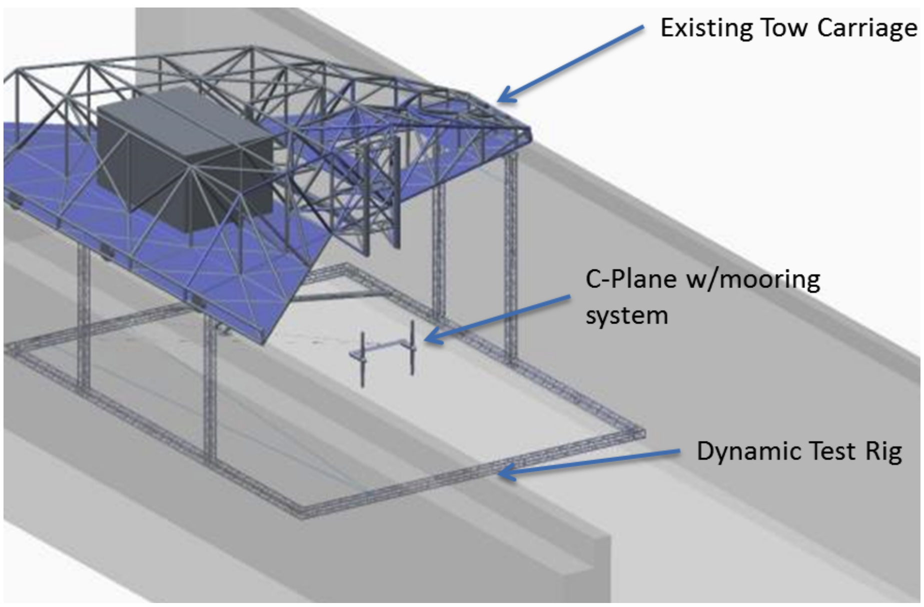
Definition of the moorings and attachment system was contracted by DA to PCCI. PCCI is a mooring systems designer and installer who specializes in mooring systems for US Navy vessels. PCCI completed a mooring top level requirements (TLR) document for the initial definition. This document defined the requirements governing the design, installation, maintenance and repair of the mooring system required to maintain the Aquantis C-Plane ocean current hydrokinetic energy system in the correct operating position. The TLR also provides an overall system description of the Aquantis C-Plane System to be moored, it's design operating and survival environmental conditions, the allowable moored C-Plane motions, the conceptual mooring and power cabling arrangement, and mooring systems inspection, maintenance and repair operational requirements.

Under the direction of DA, the Naval Surface Warfare Center – Carderock Division (NSWC-CD) also worked with PCCI to develop mooring concepts for the C-Plane platform. The effort was focused on developing the lowest cost option from material, installation and maintenance aspects. The dependence on platform net buoyancy was a key and critical factor. Attachment of the mooring lines at the platform was also of major concern to prevent platform yaw and lines fouling with the rotors when off-axis flow occurs. Mooring from the nose of the nacelle is the most desirable location, and efforts were conducted to develop a C-Plane layout that would provide static stability for that case.

DA and NSWC collaborated with PCCI to develop possible low cost and low risk mooring concepts for the C-Plane that utilized 4-leg and 3-leg mooring lines. These concepts provided good pitch-plane stability but constrain the C-Plane in yawing due to off-axis flow conditions. The need for internal braking and load shedding and to reduce blade cavitation at higher than rated power current flow was addressed.

5.10 Moorings and Attachments Model Basin Test Assistance

A scaled mooring design was prepared for the C-Plane dynamic basin testing program. This work involved taking the full scale, 2.4MW, two-rotor C-Plane system and applying the appropriate scaling factors to arrive at the values and specifications for the 1:25 scale basin test system. Lines were chosen that simulated the appropriate scaled parameters. A section of scaled chain was added to the aft mooring to keep the line from fouling in the rotors in the same manner in which it acts in the full-scale concept. Springs, float and masses were incorporated in the system to simulate the appropriate stiffness factors.



Basin Test Mooring System

5.11 Analysis of Mooring Systems

Analysis of the C-Plane mooring system was initiated with PCCI delivering a Systems Requirements Document (SRD). The SRD was developed to add further details to the previously prepared TRL. PCCI also prepared a mooring installation storyboard, a cost estimate for a typical C-Plane mooring installation, and conducted initial OrcaFlex modeling of conceptual mooring designs. OrcaFlex is a commercial software tool used in the dynamic analysis of offshore systems, surface craft and subsea mooring systems. It is widely accepted in industry and well proven for this type of work. PCCI conducted discussions with subsea geotechnical survey firms, equipment suppliers and vessel operators for specification and sourcing of mooring hardware and installation options to quantify the practical aspects of the analysis work.

As a precursor to further mooring system analysis in OrcaFlex, a concept of operations was defined for deploying, installing and maintaining the C-plane mooring system. Moorings were conceived to provide station-keeping in a constant high-current environment where the direction and magnitude of the current varies with depth and over time. Presently there are no existing design standards for mooring of large hydrokinetic turbine arrays. The C-plane, with passive depth control, was modeled where the water depth and resulting applied current load and direction may change over an undefined timescale. Design iterations included:

- Apply current direction, speed, and water depth.
- Apply rotor loads (based on early unsteady CFD)
- Select components to meet maximum calculated mooring loads and other operating requirements.
- Consider installation and maintenance requirements.
- Use of best available materials and technology.

The initial C-Plane configuration analyzed in OrcaFlex consisted of a four rotor system. The two outermost rotors were positioned forward (up current) of the truss structure, and the two inner rotors positioned aft (down current) of the truss. All four rotors were joined by a truss structure. The truss structure had a small contribution of drag however it did not add lift to the system. A connection point is used to reference and connect the C-Plane components; this point is given negligible mass. For the purpose of these simulations towed fish were used for horizontal cylindrical bodies, while lumped buoys were used for the connection point and wing. Both buoys have six degrees of freedom and may have wing attachments. Wing attachments are used to model the rotor and the nacelle connective structure (wing or truss). The rotors within the model do not rotate axially, but distribute the loading across twelve wings. The lift and drag coefficients for the stationary wings reflect the thrust loadings. These loadings were calculated using WT_Perf and provide the rotor thrust loading as a function of current speed.

Design standards for offshore renewable energy devices are still in development. However, there are numerous standards available from classification societies and associations for the design of offshore oil and gas structures. For determination of an adequate factor of safety for design of the C-Plane mooring system the American Petroleum Institute (API) standards were referenced. The recommended factor of safety for offshore moorings per API-RP2SK is 1.67 for the intact case, and 1.25 when analyzing a mooring system with a damaged line. Due to the multiple unknowns at the time of the analysis utilizing a standard safety factor, such as that recommended by API-RP2SK, would not be conservative, therefore it was recommended that a higher factor of safety be utilized at this stage in the design. The decision was made to utilize a factor of safety of 3.0 for the intact mooring line tension. To determine an appropriate factor of safety for the damaged line analysis, the ratio of the agreed upon intact factor of safety (3.0) to the API intact FoS (1.67) was taken and multiplied by the API factor of safety for the damaged line analysis.

Based on the conceptual system and the stated assumptions, OrcaFlex runs were conducted to determine the mooring loads, mooring lengths, line clashing, anchor loading, and C-Plane offsets. The C-Plane moorings permit the system to weathervane with the prevailing current direction. The mooring system was analyzed for both the intact and one line damaged cases. For analysis of the intact mooring system the cases described previously were solved and the results extracted. The mooring system was also analyzed for the case of a damaged mooring line. For this analysis two cases were analyzed, the intact case which resulted in the highest tension for a forward line, and the intact case which resulted in the highest tension for the aft line. The second most loaded line in each case was broken during the analysis, and the maximum tension after the break was recorded. It should be noted that this analysis did not consider dynamics. For each of the cases analyzed, the damaged line was disconnected from the C-Plane and the static solution was solved.

Multiple environmental conditions (26) were evaluated for their impact on the C-Plane position, orientation, and maximum line tension. The position offsets are referenced from the position of case #1, which is the base operating case. The Z offset position is referenced from 0 at the water surface, negative downward.

The maximum mooring line tension for both the intact and damaged cases is shown, along with the required minimum breaking strength (MBS) for the mooring legs based on the factor of safety. The results of the analysis show that the highest tension observed in the forward lines results from the one line damaged case. However, the intact mooring case is the limiting case for the aft line.

Maximum Mooring Line Tension

Maximum Tension			Environment	Speed at C-Plane (m/s)	Intact or Damage	FoS	Required MBS (kN)
Forward Line	3769	kN	10 Deg, 1.8 m/s @ 70m	1.62	Intact	3	11306
Forward Line	6523	kN	10 Deg, 1.8 m/s @ 70m	1.68	Damaged	2.25	**14677**
Aft Line	1597	kN	No Flow Condition	0	Intact	3	**4791**
Aft Line	1648	kN	No Flow Condition	0	Damaged	2.25	3708

Following this initial analysis work and the shift to a two rotor, 2.4MW system, a revisit to previous mooring design and costing work was conducted in the interest of maximizing adaptability and minimizing cost. These activities centered on the role that the mooring system and its installation plays in the cost of energy. Scaling parameters were utilized to arrive at the 2.4 MW, two rotor, C-Plane mooring system component sizes.

With the later decision to design the system for the 1.5MW, two rotor, C-Plane prototype configuration, DA completed a revised cost analysis of the moorings and attachments to determine the cost of a complete mooring system. Installation scenarios were also reviewed in light of the reduced system size to arrive at a revised operational cost, as well as maintenance and repair of the system. The 1.5MW system minimizes risk in the prototype development stage and allows a margin for mooring system loads and design development. The mooring and attachments costs were applied to the cost of energy model to allow a full evaluation of the 1.5MW system.

Future activity will include detailed design of the mooring and attachment system for the 1.5MW C-Plane system. This task will entail using an Orcaflex analysis to determine the mooring system loads and sizing the mooring line, attachments and anchors based on these loads. Further work will investigate shorter forward lines that comply with anchor requirements to minimize spacing between C-Planes. Detailed objectives consist of:

- Perform a feasibility study on reducing the forward mooring lines from two lines to one.
- Update the OrcaFlex model to reflect new hydrodynamic loads for wing/truss structure, and C-Plane platform and rotors.
- Perform OrcaFlex modeling to define mooring forces and dynamics of the 1.5MW C-Plane.

5.12 Anchor and Mooring Component Design

Based upon the Orcaflex simulation of the 3.8MW, four rotor, C-Plane mooring system preliminary specifications were developed for the components including line, chain and anchor sizes. These components were later scaled based on rotor thrust and system drag to arrive at the component sizes required for the 2.4MW, two rotor system.

5.13 Enabling Technology and Direct Drive

The Aquantis C-Plane is an innovative downstream, dual rotor, counter rotating, fixed pitch, tethered ocean current energy device for harnessing ocean current kinetic energy for competitively priced utility scale electric power. C-Plane arrays will be deployed and operate at a depth of 50 meters to optimize power extraction in the strong flow regime while staying below marine traffic. The rotor, drivetrain, structure and control system have all been optimized for both loads and hydrodynamics to ensure economically feasible deployment, stability and durability in the ocean environment. The C-Plane's array plans take advantage of shared mooring points and close proximity for reduced costs and minimized footprint on the environment. In this way, power density is optimized and deployment, transmission and maintenance costs reduced.

The Aquantis design also features four innovative methods of load shedding that enable cost effective utilization of various different flow regimes from ocean currents, tidal flows and island power including:

1. Passive depth control – Depth seek to maintain near constant flow through the rotors.
2. Hydrostatic Braking – Divert flow to increase system pressure and slow the rotor.
3. Mechanical Braking – Fail-safe shutdown in unanticipated load conditions.
4. Stall Regulation – Torque control of the rotor to reduce hydro performance and decrease loads.

These features enable the Aquantis C-Plane to deliver base-load energy, at a cost competitive with other renewables such as wind and solar.

The emerging marine hydrokinetic (MHK) energy sector presents new and complex challenges for Power Takeoff -PTO design. The slow resource flow speeds drive ocean current MHK systems toward large rotors, low input rotational speeds and high input torque demand. In addition the device is required to work at deep ocean depths where hydrostatic pressure attempts to drive corrosive saltwater into drivetrain components. These systems must also provide maximum reliability due to the high cost of maintenance and repairs compounded by high cost of the vessels required to access and service a device at sea. With support of the DOE and the AWP grant, Aquantis has developed an ocean current energy device with an integrated PTO that addresses all these challenges at a competitive COE.

Conventional technology of a multi-stage gearbox and rolling element main bearings are significantly challenged to operate reliably in this environment. The low rotational speeds and high loads are drivers for common wind turbine failure modes.. An expensive, large diameter, compliant seal is required to keep seawater out of the bearing lubrication. A conventional drivetrain is not the answer for robust and reliable operation of an MHK energy device. The Aquantis Next-Gen PTO developed under this project was designed to address all these design challenges.

The proposed Aquantis PTO uses an innovative selection of components to achieve a robust and reliable transmission capable of withstanding the intense demands of a subsea ocean current energy device. The technical criteria imposed on the design of the Aquantis PTO are: a) maximize reliability b) minimize maintenance c) minimize weight. The proposed drivetrain is an integrated package comprised of a variable speed hydrostatic transmission, seawater lubricated main bearings and an oil compensated seal.

Hydrostatic Drivetrain Concept

Over the course of the project, the Aquantis team made a detailed review of the drivetrain design, and settled on the combination of reliability, torque density and performance of a hydrostatic drivetrain. After review of the initially proposed direct drive concept it quickly became clear the mass and cost of an extremely low speed generator would not prove cost effective for the Aquantis C-Plane. As such the majority focus of the drivetrain portion of this project was toward advancing the hydrostatic drivetrain.

The Aquantis PTO uses an extremely torque dense hydrostatic drivetrain to address the high torque, low speed MHK environment. The proposed transmission is based on a low-speed high torque (LSHT) pump such as that available from hydraulic industrial suppliers such as Hägglunds (Bosch Rexroth) or MacTaggart Scott (Eaton). This pump weighs less than 1/2 the weight of a traditional gearbox (17,000 kgs) or 1/8 of a direct drive arrangement (~50,000 kg). In addition the pumps and motors allow for very large transmission ratios increasing output speed and reducing both the size and weight of the generator. The proposed hydrostatic drivetrain also features a variable ratio that allows for a constant speed hydraulic motor-generator (HMG) and the elimination of subsea power electronics both reducing weight and increasing reliability. The reliability of the system is also enhanced by the ability to damp transient events through pressure relief and a fluid coupling of the low and high speed components.

Validation of the Hydrostatic System

Hydrostatic transmissions are a proven technical solution for many large power transmission applications including large winches, cranes and other systems. Hydrostatic drives have a long history of reliable service in the harsh offshore oil and gas industry and submarine propulsion. Hydraulics are also unmatched in torque/power density and allow distributed system architecture. These features are all directly applicable and desirable in the design of MHK system drivetrains.

Using conventional analysis techniques a spreadsheet was developed to aid in preliminary sizing of components and identifying commercially available units and areas where hardware gaps exist. The calculations cover a broad range of pump input speeds (rotor speed) and provide estimates of the available power over a range of system operating pressures. These estimates incorporate real-world efficiencies that have been verified in various, large-scale hydrostatic transmissions developed for power-generation and other applications.

In addition, our commercial partner Bosch Rexroth has supported the testing of a conceptual MHK transmission in collaboration with IFAS and Aachen University. Bosch has also developed proprietary software for detailed analysis and dynamic modeling of the hydrostatic system. The dynamic modeling is primarily required to evaluate generator control system schemes and to test the system response to uneven flow resulting from rotor dynamics, and system start-up and shut down.

Seawater Bearing Concept

Over the course of the project, the Aquantis team investigated multiple main bearing designs to support the C-Plane rotors. Conventional rolling element bearings would operate at low rotational speeds and high dynamic loads which are drivers for common wind turbine failure modes such as micropitting. In addition a conventional bearing system is dependent on the effectiveness of the seal to keep out seawater of the grease or oil lubrication. In this configuration the deflections of the shaft (>4 mm) must be accommodated by a large, expensive and compliant seal. Considering these factors and including maintenance and reliability the lowest total lifecycle cost system is the seawater bearing concept.

In the proposed Next-Gen PTO the downstream rotor hub is directly supported by large diameter (~4m) seawater lubricated journal bearings. The bearing system comprises high performance non-metallic bearing surfaces mounted in a compliant arrangement to maximize contact area and minimize wear. Through elimination of bearing seals and use of seawater lubrication the bearings can be optimally placed in the load path to minimize the loading on the bearings and protect the downstream components and seals. The large diameter of the bearings allows for convenient packaging of a diaphragm type coupling that compliantly absorbs any deflections of the bearing support system. These features further isolate the seal and hydrostatic drivetrain from unanticipated loading, reducing the need for heavy supporting structures and increasing reliability. Also, the seal is no longer required to accommodate large shaft deflections eliminating a potential leak path which could impact buoyancy, static stability and drivetrain life. Additionally the bearing arrangement utilizes the material lubricity and the surrounding seawater as a lubricant eliminating the need for oil or grease replenishment reducing the requirement for service and maintenance.

Description of Bearing System

The bearing system is integrated into the rotor hub and support structure in an effort to minimize the load on the bearings. The result is the large diameter spread bearing arrangement as shown above.

Two separate design paths are being considered for the seawater bearing arrangement. The first uses a standard low modulus elastomer and relies on elasto-hydrodynamic lubrication (EHL) which is the formation of a fluid film to minimize friction and wear. This technology is well proven in the application of ship's stern tubes; however, the C-Plane presents new design challenges such as low rpm and high off-axis moments for this technology. EHL relies on the compliant nature of the bearing material and high relative velocities between the bearing and runner to generate a fluid film. The low elastic modulus of the elastomer bearing material results in fairly low contact pressure requirements around 4MPa. The requirements of low contact pressure and high velocity to generate a fluid film drive the C-Plane's low-rpm bearing architecture to larger diameters. The image at left shows an exploded view of a proposed thrust and radial bearing package designed using this approach.

The second design path addresses the risk of potential fluid film breakdown by utilizing materials such as ultra-high-molecular-weight polyethylene (UHMPE) or existing composite materials such as Orkot and assumes little to no fluid film. These materials are capable of providing very low wear rates and low friction, even without a fluid film. Hydrodynamic lift pockets, which promote fluid film formation, can also be incorporated into the bearing design to improve performance and reduce wear. These advanced materials have a much higher elastic modulus than standard elastomer bearing materials which means the bearing diameter can be minimized to reduce cost and torque loss.

Validation of Bearing System

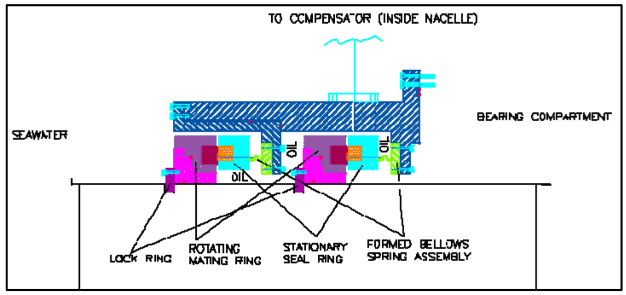
Seawater bearings are a proven technology in naval architecture for systems such as stern tube bearings supporting large ship propellers. Currently seawater bearings are also being used in small scale tidal prototypes. Aquantis and its University of Florida partner have performed preliminary calculations and analysis of the contact pressures, friction and wear rates to validate the feasibility of the system. University of Florida has also performed preliminary material performance tests and classifications for wear and friction coefficients. These first analyses give a high level of confidence to feasibility and implementation of the design.

Oil Compensated Seal Concept

Another critical component integrated into the PTO is an oil compensated seal used to keep the high pressure seawater from entering the nacelle. The seal is a specialized design that uses an accumulator charged by the ambient seawater to pressurize the internals of the seal. This design allows the system to operate at variable depths to maximize energy capture and minimize loading on the rotors and structures. The seal design features virtually no wear or leakage over years of operation in similar applications.

Description of Seal System

The concept of the oil compensated seal system is similar to oil lubricated main shaft seals for commercial shipping applications, i.e. oil under a slightly higher pressure than sea pressure keeps seawater out of the ship. For a submerged application, the oil pressure has to be considerably higher to keep the seal compartment at a higher pressure than surrounding sea pressure. This is accomplished by using a solid oil system connected to an accumulator whose opposite side is connected to the sea.



The seal ring assembly consists of two oil compensated mechanical seals. The primary sealing interface is perpendicular to the shaft axis between the rotating mating ring and the stationary seal ring. The seal rings have a formed metal bellows behind to provide net closing force on the mating ring face. In addition to providing this closing force, the change from the standard multiple spring approach to the continuous single convolution bellows is to primarily permit greater angular deflection. A secondary benefit is to eliminate the dynamic O-ring which will permanently deform under the long life and high loads. The bellows also provides a hermetic seal and eliminates a potential pitting location.

Validation of Seal System

Oil compensated seals were first employed on an autonomous underwater vehicle for a classified project for the US Navy. This vehicle was delivered in 1992 and had a maximum leakage specification of 1cc/day. The seal/bearing system exceeded all expectations and had no discernible leakage. Another example is the photonics mast (digital periscope) for the Virginia Class submarines, which is a similar application to the Aquantis project. The periscope operates at very slow speed (0- 20 RPM), and has a shaft diameter of approximately 14 inches. These periscopes employ an oil compensated seal system with a maximum leakage specification of 1 cc/day. These seals have been in operation for over 10 years with no discernible leakage. The seal system does not require a specific oil type and can be pressurized with miscible bio oils with no loss in performance or increase in leakage. The seal for the Next-Gen PTO arrangement is within the scope of current technology with simple advancements to meet the life, performance and displacement requirements of the C-plane.

5.14 C-Plane Optimization

General arrangements and mechanical-design issues including bearings, seals, shafting, blade attachments, wing attachments, mooring-line attachments, weight, and buoyancy were optimized within the design spiral. This work also included working with other organizations (Potencia Industrial and BEW) particularly with regard to the drivetrain and power electronics, and in developing an overall solid model of the system using computer-aided design (CAD).

Another key component required before proceeding with the preliminary turbine design is the size of the hub. The result of parametric design study essentially ruled out the possibility of using a direct-drive generator. As a result, Dehlsen Associates studied either the use of a gearbox between the turbine blades and the generator or the use of hydrostatic transmission. Preliminary estimates of the shaft size and the bearing loads lead to initial conversations with Timken Corporation and the Schaeffler Group regarding bearings and the use of their bearing software design tools. Initial weight estimates, including an investigation of the weight of the tailcone on the bearing reaction forces were made. During this preliminary mechanical-design effort updates to the design spreadsheet were made to assist in estimating weight, buoyancy, and loads.

Preliminary discussions have also started with Hoffman Kane regarding potential mechanical seals. A trade study was also conducted of turbine blades forward or aft of the wing.

The mechanical design of the shaft, bearings, and seals were evolved. Based on estimates of torsion and bending, a desired 20-year life, and the use of 17-4 PH stainless steel at H900 condition, initial estimates were calculated with a center bore for electrical wires, if needed. The use of fluid-film bearings seemed problematic because of the large loads and, even more importantly, the low rpm, which would not allow adequate hydrodynamic film lubrication. This finding led to the need for a seal. The Aquantis team has experience with both lip seals and mechanical-face seals. Lip seals have a short life, on the order of days to months, which would be inappropriate for Aquantis. Hoffman Kane was contacted and they indicated that they could deliver a mechanical-face seal at this diameter that should last 20 years, although they recommend scheduled changes of the O-rings and faces every couple of years. These seals work well in seawater, have almost zero leakage, and have no memory issues. However, they are expensive, typically $2,000 per inch of shaft diameter. ETTEM USA was also engaged for their mechanical-face seals.

Roller bearings and ceramic ball bearings were investigated. Roller bearings can provide tremendous load-carrying capabilities at slow shaft speeds, and a single bearing can support both radial and thrust loads. A representative from Timken Corporation discussed with the Aquantis team the possible use of their bearings and their analysis software. Later, after using this software, it was found that some spherical roller bearings should work at this size and for these loads, although their total life may not be adequate. However, the conservative loads are based on dry weight of solid E-glass/epoxy composite blades. Reducing the weight of the tailcone using syntactic foam, which adds buoyancy, or simply removing the tailcone will help to some level. A conscientious effort was initiated to reduce the weight of the blades. The most significant improvement, however, would come from moving the bearings as close to the center of gravity of the turbine as possible, radially beneath the rotor blades. Another permutation that was strongly considered was using this design condition in determining how to attach the composite turbine blades to the hub.

The mechanical design work continued with both the Timken Corporation and the Schaeffler Group to identify bearings to support both the radial loads and the thrust loads of the turbine. Although several bearings have been considered, the focus has been on spherical roller bearings, which primarily support radial loads, but can also support some thrust loads. The use of additional thrust bearings was also being considered. The bearing package will also require a large mechanical-face seal. The location of the bearing-and-seal package will strongly impact the radial loads that the bearings must support and thus the size and cost of the bearings. After initially discarding fluid-film bearings because of lubricating problems at very low shaft rates, a novel mechanical design was developed that would use Duramax EHL to incorporate a water-lubricated bearing concept. This design will reduce the size of the shaft and the mechanical-face seal, which will reduce costs, in addition to further developing all of these bearing and seal concepts to evaluate them on the basis of acquisition and life-cycle cost, as well as how to better handle wear and radial sagging of the bearings.

For the hydrodynamic design of the turbine, ARL Penn State has been working with BEW Engineering, and Straight Forward Systems (with additional input from Powertrain Engineers) to provide the best overall system drivetrain, from the turbine through the generator and the transmission lines to the grid. As a result of the Conceptual Design Review, it was decided to move forward with a two-bladed turbine, primarily to reduce the material cost of the blades and to simplify shipping and handling. The Aquantis Team looked at increasing the blade-tip diameter, in order to extract enough extra power to overcome the expected power losses in the drivetrain and still provide 1.3 MW to the grid.

Four different drive train topologies were evaluated:

(1) Hydrostatic drive with a variable-speed turbine and a permanent-magnet generator,
(2) Hydrostatic drive with a fixed-speed turbine and an induction synchronous generator,
(3) Gear drive with a variable-speed turbine and a permanent-magnet generator, and
(4) Gear drive with a nearly fixed-speed turbine and an induction asynchronous generator.

The Aquantis team explored the concept of similitude, in hopes of providing a simple method to optimize the Aquantis turbines. An appropriate derivation of similitude for this problem was developed. The key point is that one can only estimate the performance of a turbine if one compares it to a similar turbine with known performance. Thus, the tip-speed ratio (*TSR*), as well as the number of blades, must be the same between the two turbines. As it turns out, one cannot use similitude to perform an optimization in this case. Nonetheless, this similitude analysis did highlight some interesting points. First, the best power density (power/mass) is for smaller turbines. However, these small-diameter turbines are limited in the power that they can extract from the current. More power requires larger diameter turbines, which reduces the shaft rate and increases torque, which emphasizes the challenges in matching the turbine to the drivetrain. Also, in examining the similitude relationships for the structural integrity of the blade, it was determined that the centrifugal forces only contribute about 1% of the total maximum stress for these low-shaft-rate turbines.

The mechanical-design aspects of the turbine work was continued with possible vendors that could help with two possible options for a bearing-and-seal package. The first option is a sealed, oil-lubricated package with a large shaft, spherical-roller bearings, an additional thrust bearing, and a mechanical-face seawater seal. The second option involves a seawater-lubricated bearing assembly—combining radial and thrust bearings—and a smaller shaft and a smaller mechanical-face seawater seal than used in the first option. The Aquantis team also began looking at possible mechanical rotor brakes. Vendors provided initial cost estimates, which proved extremely high for the second option. However, given that updates for the bearing loads based on the turbine-rotor weight it is likely much lighter than the original large, conservative assumptions.

The mechanical design of the bearing-and-seal package was challenging due to the changes in the designs of the turbine and the drivetrain. When the focus was still on the 45-meter-diameter, two-bladed turbine, progress was made on the first option: an oil-lubricated design with a shaft, spherical-roller bearings, a thrust bearing, and a mechanical-face seal. Different material options were pursued for the shaft, which will affect the cost. Design work for the seals continued, including review of a mechanical-face seal for the main seawater seal and a couple of lip seals for oil retention. The details of the design of this bearing-and-seal package also depends on a decision on how much of the turbine nacelle might be flooded.

A second design option of a seawater-film bearing, a thrust bearing, a smaller shaft, a coupling, and a smaller mechanical-face seal was pursued. The higher torque levels for the 45-meter-diameter, two-bladed turbine were a concern for both design options, but primarily for this second option and its coupling. Also, with the overall length of the coupling the C-plane should not need intermediate torque transmission. Finally, this *friction-style* coupling should be able to connect directly into the rotor-end flange, providing much easier installation and removal.

With the new focus on the 35-meter-diameter, two-bladed turbine, work was initiated to consider the design of both options for the bearing-and-seal package at this smaller size. In the discussions with the potential component vendors, the potential for two variants of turbine design at different sizes, with the larger turbine being a future upgrade to the smaller turbine was evaluated. In addition, the capabilities of different machine shops to fabricate components at this size were assessed. Future assessments should include machine shops in Florida.

The second full design iteration involved a preliminary design that had a variable-speed, 35-meter-diameter, two-bladed turbine with a rated power of 800 kW out of the generator at a current speed of 1.6 m/sec that would drive the hydraulic pump. The use of passive depth control on the system would eliminate the need to use a brake during normal operation of the turbine keeping the turbine blades on design and mitigating surface cavitation. However the pitch of the blades had to be changed to increase the shaft rate and better match the pump. The alternative would be to increase the static pressure of the pump, which would significantly reduce its operational life, so that option was rejected. The Aquantis team was able to increase the shaft rate while holding the axial induction factor and, thus, the wake losses near their optimal value, but the blade losses increased, resulting a lower hydrodynamic efficiency (or power coefficient, C_P). As a result, we also had to increase the turbine diameter to 40 meters to achieve the desired rated power. Next, the blade design was evaluated using computational fluid dynamics (CFD) as well as the blade structure and its attachment to the hub.

Two possible drive train concepts for the bearing-and-seal package were evolved. The first option is an oil-lubricated design with a shaft, spherical-roller bearings, a thrust bearing, and a mechanical-face seal. The second option is a seawater-film bearing, a thrust bearing, a smaller shaft, a flex coupling, and a smaller mechanical-face seal. As the preliminary design phase proceeded, Dehlsen Associates, LLC asked Powertrain Engineers, Inc. (PEI) to focus on the first option and ARL Penn State to focus on the second option. In addition, ARL Penn State has worked on the nose of the nacelle, the aft endcap of the nacelle, the tailcone, and the structure of the nacelle itself, which forms a pressure vessel, and the nacelle joints. For the bearing-and-seal package, as well as the wet brakes, work was performed with possible vendors such as Duramax, Timken, Ameridrives Couplings, Altra Industrial Motion, John Crane, and Ettem USA. The focus on this preliminary design is to estimate weight, volume, cost, and power losses (when appropriate).

The large CP and AEP penalty of matching the pump performance with hydrodynamics led to the possibility of ganging two pumps in series being explored. Ganging the pumps together allowed for reductions in shaft rpm and increase turbine efficiency by 25%.

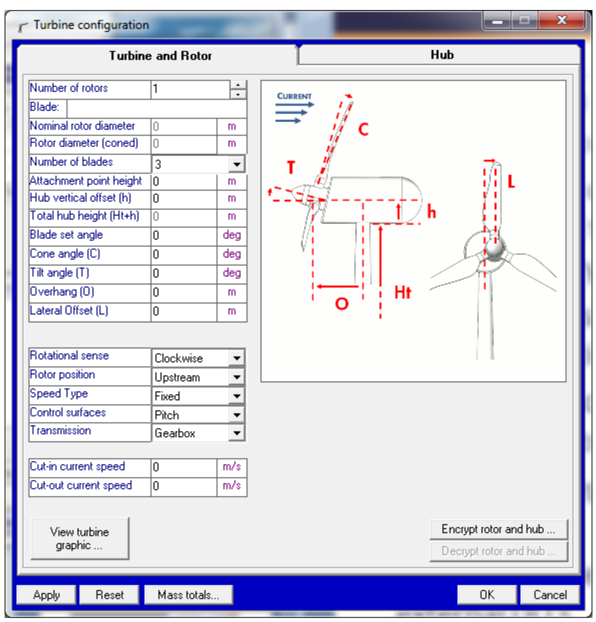
The ability to incrementally expand torque through stacking of pumps and power by HMG expansion provides a market entry point of 1.5MW whereby power and torque could be expanded by value added engineering to arrive at 2.4MW for the Florida market.

Global Integration Model

In working through the C-Plane stability analysis using codes such as DCAB and Flightlab it became apparent that there was not an integrated analysis solution for solving for all motions of, and loadings on, the C-Plane. There was a possibility to integrate several software packages together to solve for these parameters, however this was deemed to be a potentially risk prone and time consuming process. DA began looking to other software solutions to provide a global integration model of the C-Plane. Ideally this model and the resulting analysis using it would provide a fully integrated set of motions and loads that the C-Plane would experience under different operating conditions.

TIDAL BLADED from GL Garrad Hassan (now known as DNV-GL) was chosen to produce and analyze the global integration model. The global integration model has been and will continue to be used to model the dynamics of the C-Plane, incorporate advanced control algorithms and a full 6DOF model of the hydrostatic drivetrain. TIDAL BLADED is the one of the few commercial tools available for modeling dynamics of tidal or ocean current devices. Many of the modules of TIDAL BLADED are derived from BLADED, wind turbine modeling software with almost 20 years of history.

TIDAL BLADED is used to perform time-domain simulation with full hydro-elastic modeling based on multi-body dynamics. It includes validated models to describe added mass effects on both the rotor and support structure. The ability to model multiple counter-rotating rotors, custom drivetrains and incorporate an external controller makes it ideal choice for the Aquantis C-Plane global integration model. Moorings are represented as quasi-static entities although there is ongoing effort to include dynamic mooring capability. One of the TIDAL BLADED input screens is shown in below.

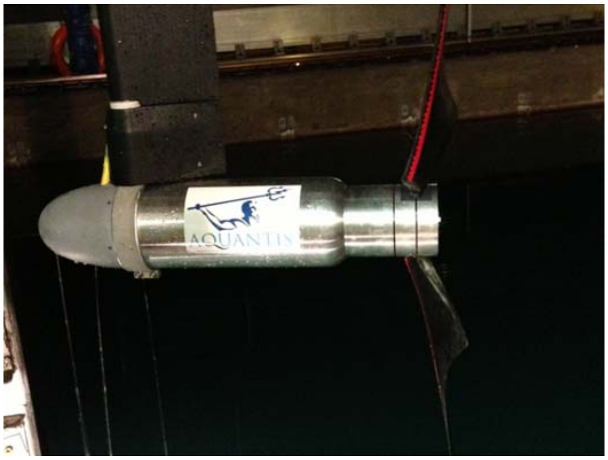


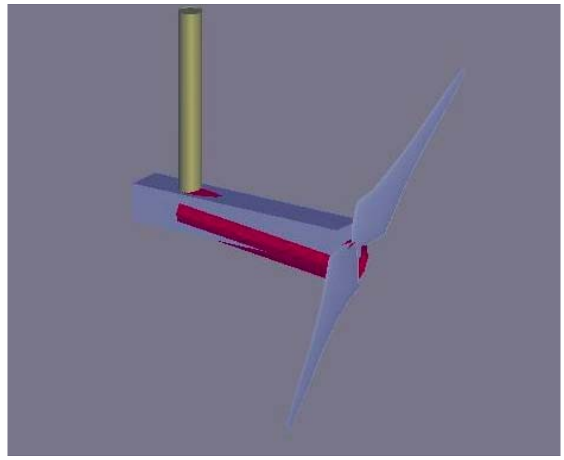
Tidal Bladed Input Parameters

The overall plan for global integration model development is as follows:

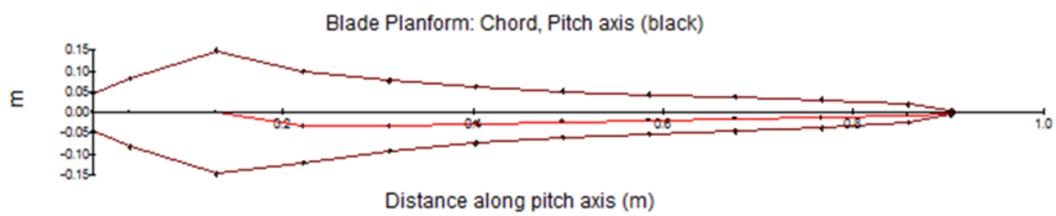
1. Develop a TIDAL BLADED model for tow tank captured test.
2. Validate the model against results from tow tank captured test.
3. Build a dynamic test scaled model and compare predictions against measurements to validate stability and loads for a floating system.
4. Build a full scale model after validation against scaled model tests.
5. Incorporate an external controller to evaluate load mitigation strategies.
6. Use the global integration model to optimize mooring location and design.
7. Perform platform fatigue and extreme loads calculation using site specific sea state conditions and stability analysis.

The figure shows a 1/25th scale of Aquantis C-Plane single pod that was used for the captured test and the corresponding model in TIDAL BLADED. Masses and buoyancy parameters were tuned according to the actual test article. In this first analysis, the blades, nacelle and strut are assumed to be infinitely stiff. The blade planform as modeled in TIDAL BLADED is shown.

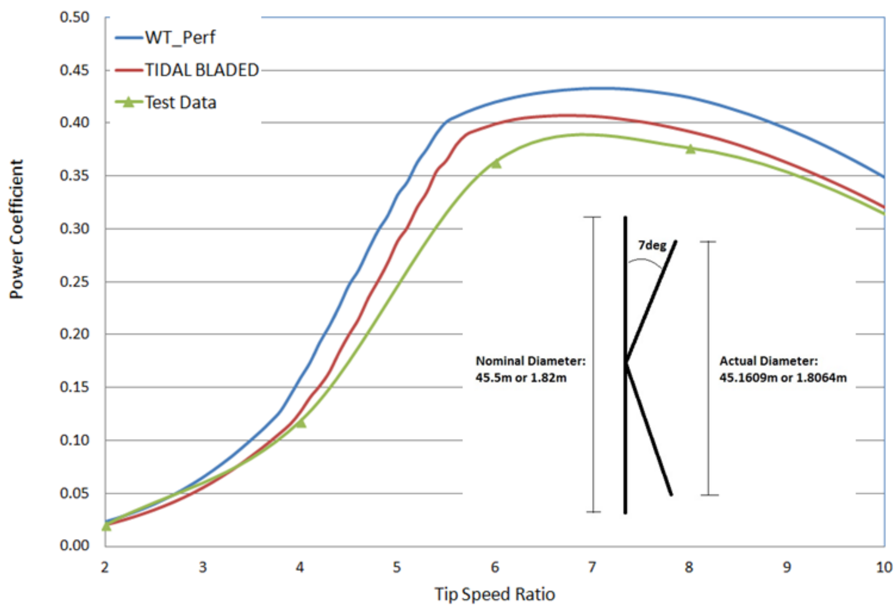




Capture test 1/25th scaled model and TIDAL BLADED representation

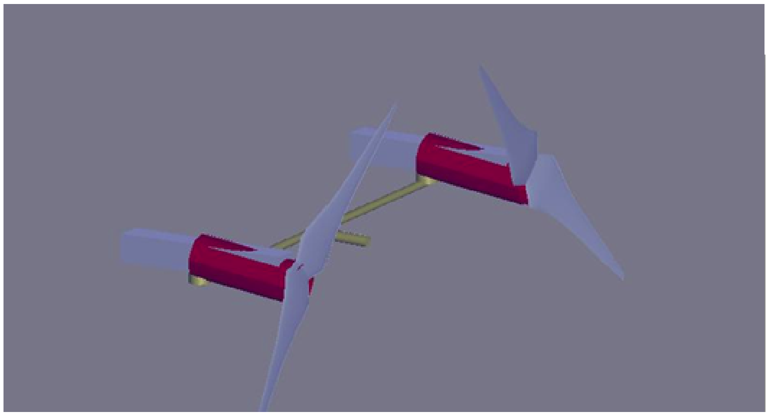


Blade chord distribution for 1/25th scaled model



Comparison of TIDAL BLADED power coefficient versus tip speed ratio predictions against test data and WT_Perf

To validate TIDAL BLADED, simulations were conducted for various tip speed ratios and yaw error sweep angles and the results compared to the test data. For validation of hydrodynamic performance, results were also compared against another blade element momentum (BEM) simulation tool from NREL called WT_Perf as shown.



Preliminary representation of dynamic model for Aquantis C-Plane in TIDAL BLADED

5.15 Baseline Design Concepts of the Power Conversion and Transmission

The Aquantis C-Plane power conversion and transmission efforts centered on developing the conceptual architecture and component usage. DA leveraged the significant experience base of its partner DNV-KEMA BEW Engineering in these developments.

The use of a hydrostatic transmission (HST) in conjunction with an induction generator in the C-Plane eliminates the need for significant amount of power conversion components, namely the converter. The variable ratio of the HST maintains the induction generator at synchronous speeds and thus delivers consistent voltage and frequency power production to the grid. Therefore, the major focus of the work done was on the grid connection and transmission architecture.

The objective of the grid connection and transmission architecture work was to demonstrate the feasibility of moving traditional device resident utility interconnect equipment from the C-Plane to an onshore substation point of interconect. The advantages of this relocation are substantial considering the installation and operational economics and environmental impact of being offshore. The objectives were studied in terms of a 100 MW offshore facility, returning power to shore at the highest possible distribution voltages (34.5 kV). This voltage level eliminates the need for expensive and maintenance-intensive offshore substations or subsea transformers. Furthermore, because of the size of these facilities and the anticipated distances to shore, AC distribution is the economical (and reliable) first approach. A second objective was to document,

through computer simulation, the performance of hydraulic drivetrains during electric utility fault conditions and confirm the efficacy of such hardware under these severe conditions. The grid simulation software tool PowerSim (PSIM) was used in this phase of the study.
In consideration of connecting the C-Plane or array of C-Planes to the utility grid it was necessary to evaluate the electric utility interconnect impacts. The processes used to determine these impacts are as follows:

1. Establish a dynamic model of the hydraulic drivetrain system, including hydraulic pump, hydraulic motor, AC induction generator, rotor dynamic coefficient of performance (CP) characteristics and all compliant mainshaft connections.
2. Develop a preliminary model of the parasitic parameters of 20 km underwater cables.
3. Establish impedances and dynamic characteristics of onshore substation and transmission lines.
4. Analyze steady-state load flow to determine voltage swings as a function of load and necessary onshore compensation needs.
5. Determine utility interconnect test case matrix to exercise the MHK device and facility validating substation approach. A subset of the final test case matrix to be determined was the IEC 61400-21 test conditions as listed below.
6. Iterate the substation apparatus, i.e. static volt ampere reactive (VAR) compensation, dynamic voltage restorer, etc. necessary to meet utility requirements
7. Document drivetrain mechanical loads and fold extreme load conditions into drivetrain requirements.
8. Identify non-electrical system impacts of the utility interconnect requirements and factor these impacts into the drivetrain and turbine design.

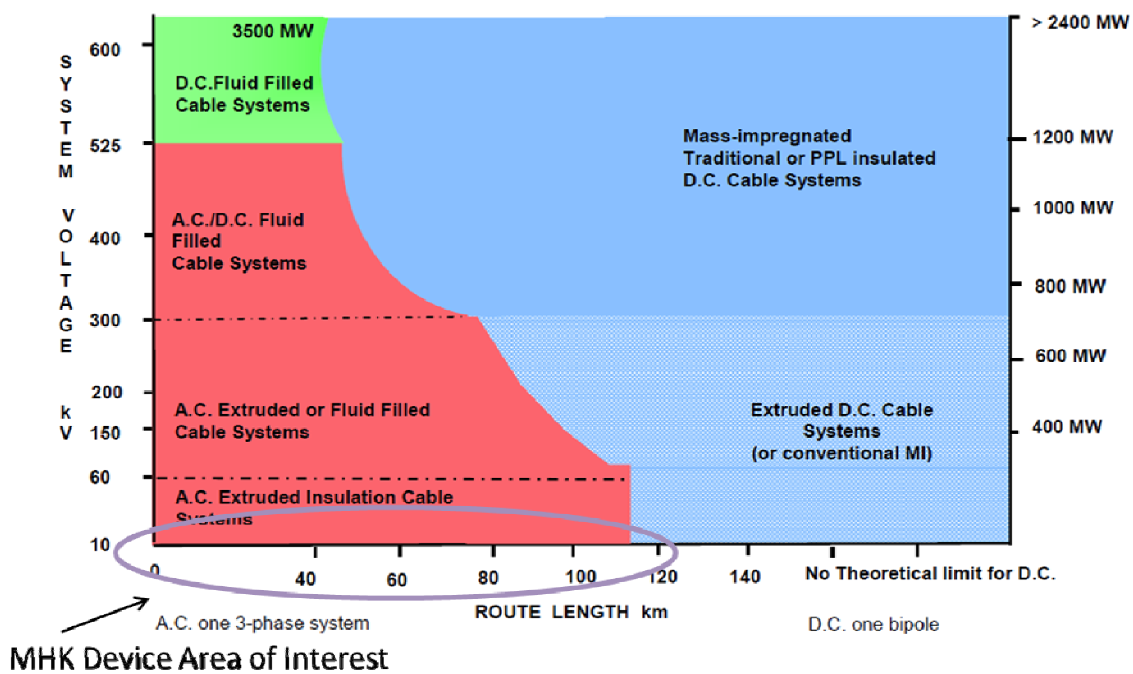
The processes to determine the optimum hardware selections, costs and cost of energy were as follows:

1. Review of current commercial offerings, development level, selection and costing of main shore export cable(s), array cable(s), junction boxes, dynamic riser power cables, review of dynamic cable loads packaging, MHK device power cable connections, cable anchoring, hardware necessary for the cable management in the water column and subsea power connectors.
2. Identify availability and preliminary specification of all hardware at the proposed voltage of 34.5 kV and adjust collection system concept to meet acceptable voltage levels.
3. Prepare a cost of energy (COE) analysis on the selected bill of material.

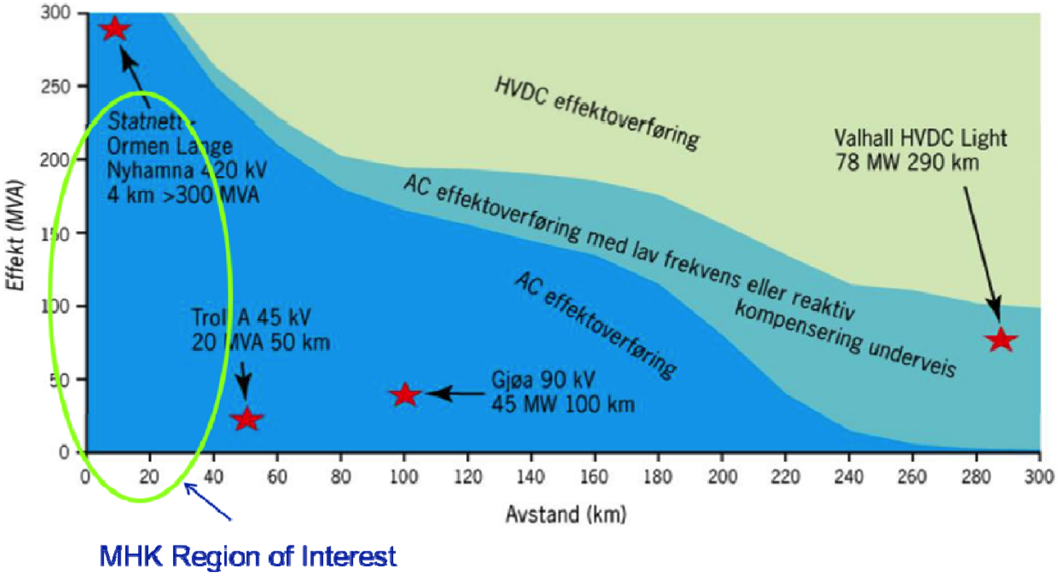
The scope of the C-Plane grid connection effort were the investigation and technical-economic trade studies aimed at identifying an on-shore approach to meeting electric utility interconnection requirements (e.g. FERC 661B) for the C-Plane. This approach allows the movement of sophisticated and complex electrical and electronic components off of the C-Plane, thereby improving the system reliability, in favor of transferring complexity onshore where installation, operation and maintenance is significantly more straightforward and less expensive. Included in this investigation was the use of static VAR compensators, dynamic voltage restorers and other apparatus at the onshore substation point of interconnect. In this system level (rather than device level) approach to interconnect, a further point of investigation is the performance of C-Plane hydraulic drivetrain under electric utility fault conditions such as symmetrical and non-symmetrical low-voltage-ride-thru (LVRT) events.

In addition to the investigation of the onshore interconnect approach, a second aspect of the work was to determine and identify a class of low cost underwater collection system hardware. This includes reducing underwater cable costs, as well as underwater junction boxes, terminations and collection points.

For purposes of specificity, a commercial, practical focus for early adoption of the C-Plane was used in the proposed investigation. Specifically, the investigation performed was focused on a 100 MW offshore C-Plane facility. The study also contemplated distances to shore of less than 25 kM, although this is easily extended without limitation to distances of 100kM and beyond.
In the figures as provided below, a presentation of typical AC and DC distribution (transmission) for various MW capacities over different distances is shown. The figures are in agreement that the need for high voltage DC does not exist for early stage C-Plane adoption in the 100's of MW scale at the distance contemplated at this time. Later stage, commercial adoption may change this thinking, but for now, AC distribution is the most cost effective approach to bringing C-Plane power to shore. There is also existing standard AC equipment which provides a good starting point for future cost reduction activities. With the understanding that distribution is to be AC, attention must turn to large facility interconnect requirements as determined by FERC Order 661B, 2006. While traditionally these requirements have been met at the device level, for example at the wind turbine level in a windplant, the offshore operating environment provides a strong incentive to move this traditional approach from the C-Plane device and allow it to reside at the shore line; co-located with onshore substations. The appeal of eliminating this equipment from the C-Plane and its environment is apparent and there is an improvement in cost-of-energy by reduced O&M costs.



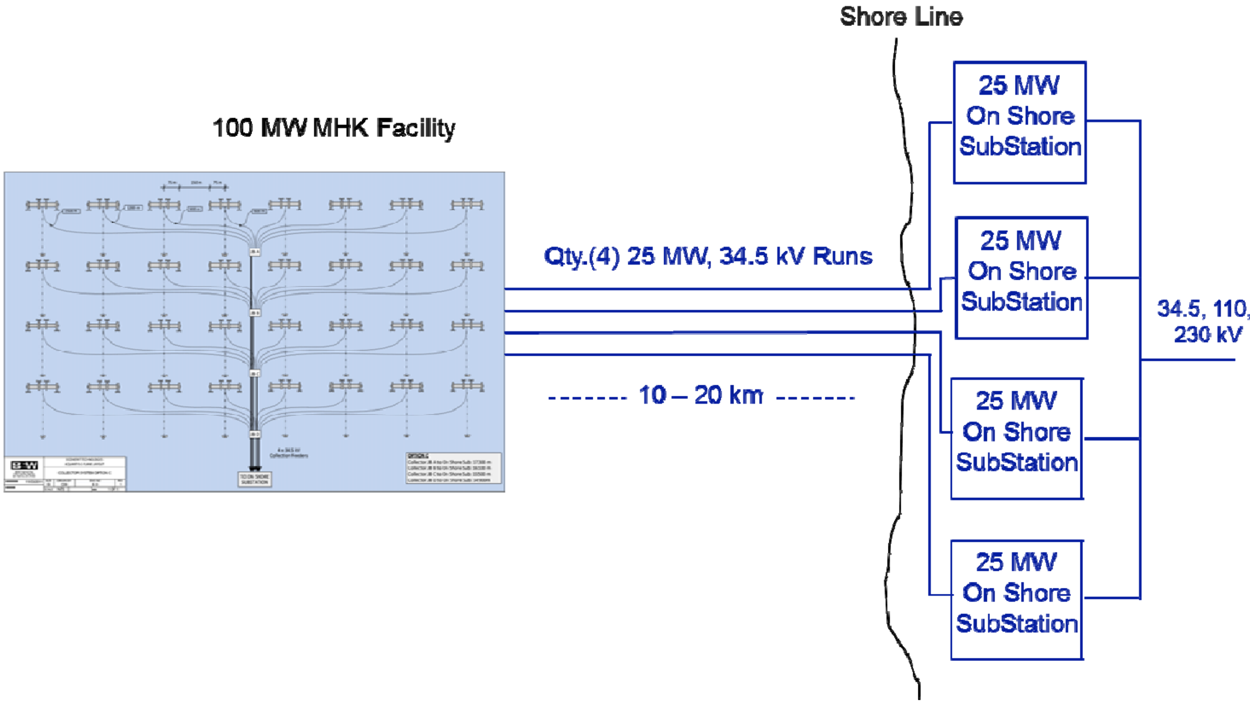
C-Plane AC and DC Grid Connection Methods



Representative Grid Connection Projects and Methods

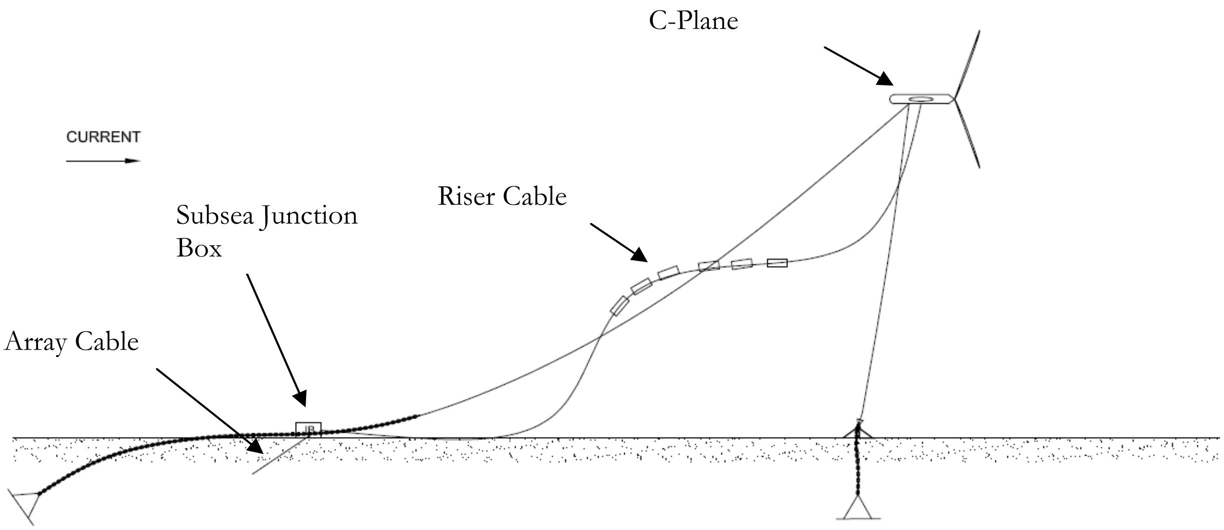
As part of the interconnect study, a baseline AC collection system was concepted. Specific hardware in the collection system includes cabling, underwater junction boxes, mateable underwater connectors and termination points. While all of these currently exist in one form or another, the existing base of equipment has mostly been deployed in offshore oil and gas fields, or in military applications, neither of which are cost pressured to the extent of independent power producers with low electrical cost-of-energy requirements. A review of the existing, commercially available electrical hardware leveraged from earlier studies by DNV and others served not only as a starting point for costing and component concepts.

The design relationship shown below between a 100 MW collection of MHK devices and the onshore substation(s) incorporates grid interconnection apparatus such as static VAR compensation, dynamic voltage restorer, or other appropriate hardware. In the example shown, a 20 km run to the shoreline is shown at 34.5 kV with no offshore substation. Multiple runs to shore in 25 MW segments are shown although it has not yet been confirmed that this is the cost optimal configuration.



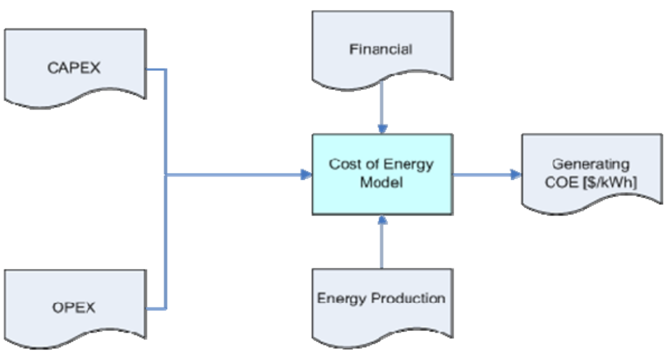
Conceptual Grid Connection Arrangement

The MHK electrical collection system will consist of a main shore export cable(s), array interconnect cables, a subsea junction box, junction box splices or connectors, riser cable and a riser cable to the C-Plane connector. An illustration below shows the arrangement of a majority of these components used with the C-Plane and the mooring line arrangements can be found in. Additional subsea junction boxes may be required to connect the array interconnect cables. The power cable and mooring lines that secure the system to the seafloor must be managed in a manner that prevents them from tangling with each other during service. These cables also require proper management during launch and recovery of the systems. It is of great importance to the success of the installation that these factors are considered and the components are reliable and cost effective.



C-Plane Offshore Grid Connection Components

5.16 Cost of Energy and CAPEX



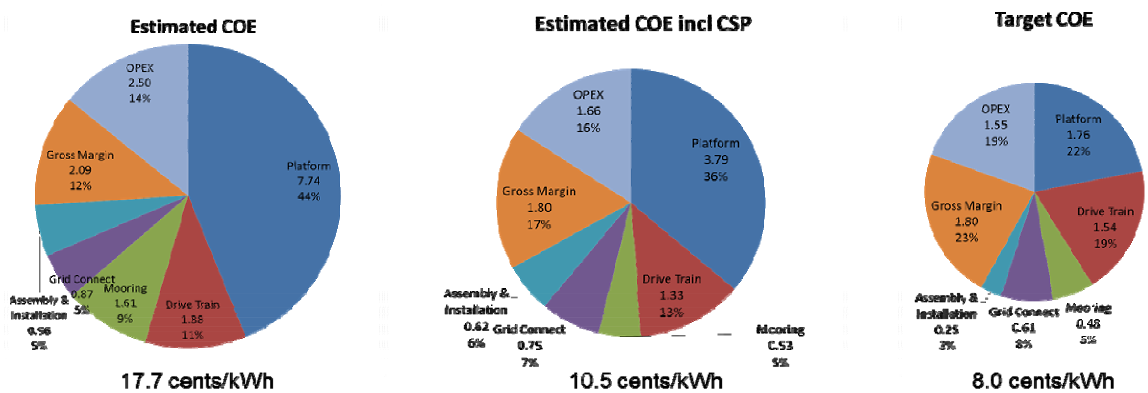
Over the course of the project, the Aquantis team developed a detailed Cost of Energy (COE) model to investigate and quantify the C-Plane optimization. The goal of the Aquantis project is to produce reliable base load power at a cost-of-energy (COE) goal of 8 ¢/kW-hr. As reported at the Aquantis Ocean-Current Turbine PDR, the preliminary design had cost-of-energy (COE) of 17.7 ¢/kW-hr.

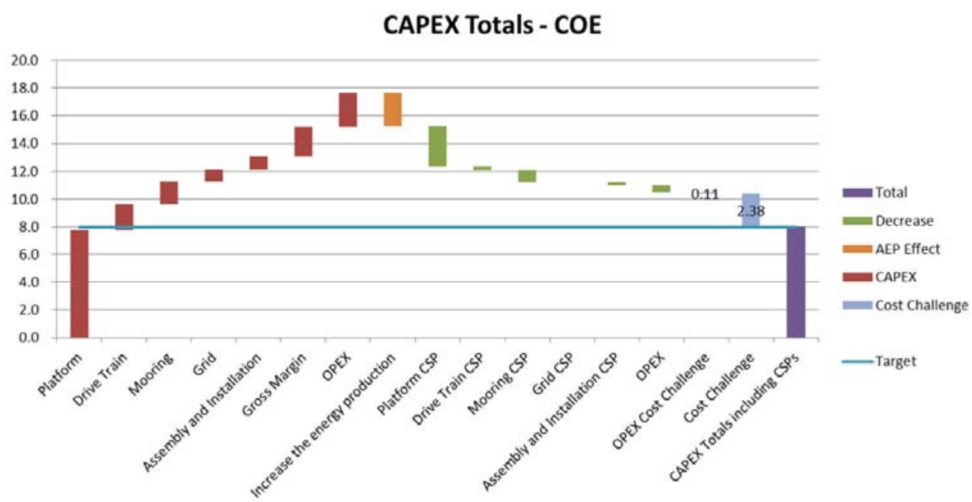
The Aquantis team updated the preliminary turbine design to (1) improve the hydrodynamic performance, (2) reduce the weight, and (3) reduce the cost. One key optimization was stacking multiple hydraulic pumps in series to increase the torque capacity. This development lowers the shaft RPM to the hydrodynamic optimum and allows a larger rotor diameter able to capture more power, increasing the name plate rating. In addition, the Aquantis team completed a preliminary hydrodynamic and structural design of the turbine using the blade-element code, WT_Perf' computer-aided design (CAD)' finite-element analysis, and some in-house lamination codes — resulting in a structurally-robust design with acceptable factors of safety.

In parallel to the optimization of the C-Plane configuration, a considerable effort was made to develop an Operations & Maintenance (O&M) model for the Aquantis project. The O&M model is currently using the component and cost data from the COE model, and using the probabilistic modeling tool Crystal Ball to calculate an estimated O&M cost for the 20-year lifetime of the C-Plane. Assumptions have been made on the design life, maintenance intervals, and failure rates for the components. The next step is to modify the assumptions based on knowledge from the different subject matter experts both within in the Aquantis group and internally in DNV/BEW.

These system updates allowed the Aquantis team to project driving the COE from 17.7 ¢/kW-hr to 10.47 ¢/kW-hr.

Design Parameters and Costs for Preliminary Turbine Designs





BOM

The BOM cost and weights were developed through detailed CAD design, structural analysis and the manufacturing experience of PEI, BEW, ARL and the Aquantis team.

Mooring Costs

The mooring cost were developed using the best knowledge of PCCI and quotations from various marine equipment suppliers.

Grid Connect Costs

The Grid connections cost for the C-Plane were scaled using an NREL report "Electrical Collection and Transmission Systems of Offshore Wind Power"

O&M: Operation and Maintenance Modeling

BEW Engineering has been tasked by Dehlsen Associates to develop an Operation and Maintenance model for the Aquantis Ocean Current Turbine concept. The main objective of the model is to use a Monte Carlo simulation in the Crystal Ball for Excel software to produce a probabilistic estimate of the costs and reliability parameters of the Aquantis concept.

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This report provides a step-through guide of the model and some insight into the processes involved: 1) defines the inputs required for the model, 2) describes how the model carries out the analysis and how to run an analysis in Crystal Ball, 3) defines the outputs from the analysis, and 4) provides the summarized assumptions as discussed in the preceding sections of the report.