Marine & Hydrokinetic Technology Readiness Initiative
DE-EE0003636

TIDAL ENERGY SYSTEM FOR ON-SHORE POWER GENERATION

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A. Executive Summary

Addressing the urgent need to develop LCOE competitive renewable energy solutions for US energy security and to replace fossil-fuel generation with the associated benefits to environment impacts including a reduction in CO_2 emissions, this Project focused on the advantages of using hydraulic energy transfer (HET) in large-scale Marine Hydrokinetic (MHK) systems for harvesting off-shore tidal energy in US waters. A recent DOE resource assessment, identifies water power resources have a potential to meet 15% of the US electric supply by 2030, with MHK technologies being a major component. [1]

The work covered a TRL-4 laboratory proof-in-concept demonstration plus modeling of a 15MW full scale system based on an approach patented by NASA-JPL, in which submerged high-ratio gearboxes and electrical generators in conventional MHK turbine systems are replaced by a submerged hydraulic radial pump coupled to on-shore hydraulic motors driving a generator. The advantages are; first, the mean-time-between-failure (MTBF), or maintenance, can be extended from approximately 1 to 5 years and second, the range of tidal flow speeds which can be efficiently harvested can be extended beyond that of a conventional submerged generator. The approach uses scalable, commercial-off-the-shelf (COTS) components, facilitating scale-up and commercialization.

All the objectives of the Project have been successfully met

1) A TRL4 system was designed, constructed and tested. It simulates a tidal energy turbine, with a 2-m diameter blade in up to a 2.9 m/sec flow. The system consists of a drive motor assembly providing appropriate torque and RPM, attached to a radial piston pump. The pump circulates pressurized, environmentally-friendly, HEES hydraulic fluid in a closed loop to an axial piston motor which drives an electrical generator, with a resistive load. The performance of the components, subsystems and system were evaluated during simulated tidal cycles. The pump is contained in a tank for immersion testing. The COTS pump and motor were selected to scale to MW size and were oversized for the TRL-4 demonstration, operating at only 1-6% of rated values. Nevertheless, in for 2-18 kW drive power, in agreement with manufacturer performance data, we measured efficiencies of 85-90% and 75-80% for the pump and motor, respectively. These efficiencies being 95-96% at higher operating powers.

2) Two follow-on paths were identified. In both cases conventional turbine systems can be modified by replacing the existing gear box and generator with a hydraulic pump and on-shore components. On the conventional path, a TRL5/6 15kW turbine system can be engineered and tested on a barge at Project partner TEDEC, facilities in Maine. Alternatively, on an accelerated path, a TRL-8 100kW system can be

engineered and tested by modifying Project partner, Atlantis's, existing MHK Solon turbines. Barge and grid connected test facilities exist for the Solon turbines. On both paths the work can be expeditious and cost effective by reusing TRL-4 components, modifying existing turbines and testing at established sites.

3) Sizing, performance modeling and costing of a scaled 15MW system, suitable for operation in Maine's Western Passage, was performed (Appendix 1). COTS components are identified and the performance projections are favorable. The LCOE is comparable to wind generation with peak production in high demand windows

4) We also determined that a similar HET approach can be extended to on-shore and off-shore wind turbine systems. These are very large energy resources which can be addressed in parallel for great National benefit.

5) Preliminary results on this project were presented at two International Conferences on renewable energy in 2012, providing a timely dissemination of information.

We have thus demonstrated a proof-in-concept of a novel, tidal HET system that eliminates all submerged gears and electronics to improve reliability. Hydraulic pump efficiencies of 90% have been confirmed in simulated tidal flows between 1 and 3 m/s at only 1-6% of rated power (see Figure 1). Total system efficiencies have also been modeled, up to MW-scale, for tidal, and wind, systems. Efficiency projections are between 81% (full rated flow) and 86% (1/3 rated flow). Such high efficiency over a wide operating range compares favorably with conventional systems having a performance range of 87% (full rated flow) to 0% (1/3 rated flow) efficiency. We also identified an accelerated path to commercialization, leveraging conventional MHK system technology and COTS components to meet the urgent need for renewable energy generation.

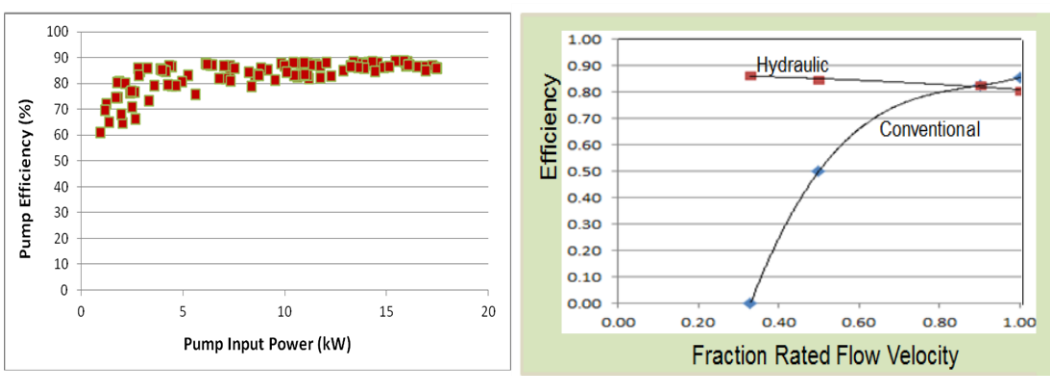

Figure 1. (a) Measured TRL4 pump efficiency in a 1-3 m/s simulated tidal flow (LHS), (b) Predicted performance for HET and actual Conventional Turbine Systems (RHS)

B. Background

In a conventional in-stream tidal energy system, and wind turbines, a fluid flow turns blades, which turn a gearbox to increase the RPM about 100-fold. The high speed output torque is then used to turn an attached generator. For tidal systems, this requires a high-maintenance submerged gearbox, and a submerged generator which are subject to seawater intrusion and failure. Mean-time-between-failure (MTBF) is about 1 year for conventional wind energy systems, and less for tidal energy systems. In contrast, JPL/Caltech has patented a novel hydraulic energy tidal system concept wherein the submerged pump and gearbox are replaced by a long-life mechanical pump (5 years between maintenance). [2] A bio-friendly, high pressure fluid is pumped to shore, where it is converted to electricity, and the low pressure fluid returns to the submerged mechanical pumps to complete the cycle. All gears and submerged electronics are eliminated, thus greatly increasing system reliability and lowering maintenance costs. (see Figure 2.) In addition, total efficiency need not diminish at low tidal flow velocities, as occurs with conventional tidal energy systems, since the hydraulic pump efficiency *improve, and* there are *no* gearbox losses. G*enerator efficiencies can also remain high* by partially engaging an array of on-shore hydraulic generators to roughly match the hydraulic flow, thus maintaining high efficiency and high RPM operation and minimizing the required level of electrical conditioning to be grid ready.

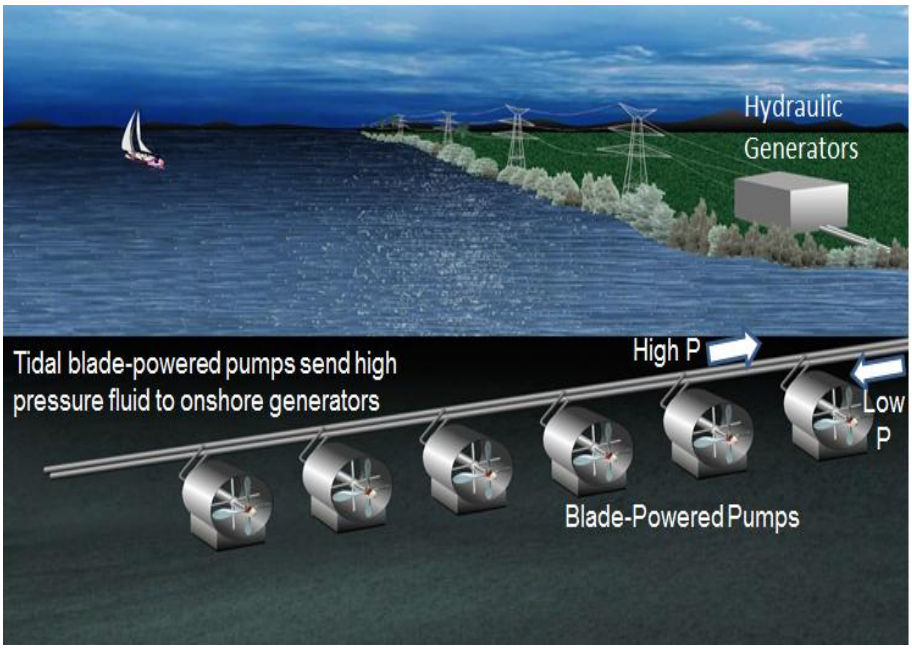

Figure 2. JPL/CalTech HET System Concept

Sunlight Photonics (Sunlight) and JPL/CalTech teamed on this response to the FOA from DOE on the MHK technology readiness initiative. Sunlight is a privately funded, renewable energy company, based in NJ. Sunlight's team of former Bell Labs and NASA scientists and engineers are pursuing efficient and cost effective energy extraction and storage methods from tidal streams.

The proposed Project had two main objectives, first, the demonstration of key aspects of the HET system at TRL4 level in the laboratory, and second, to model the cost, performance and LCOE for a full scale system. The goal was to verify a value added opportunity, which would leverage existing MHK technologies and regulatory developments, rather than plough a new furrow. To pursue this goal and to lay foundations for future commercialization, three entities were recruited as consultants with a view to continuing collaboration on future development stages. The combined team has the experience and capabilities to design, develop, test, manufacture, deploy and operate full scale MHK systems based on the envisioned hydraulic power transfer concept. The consultants are (i) Atlantis Resource Inc. (Atlantis), an international leader in marine turbine technology with experience in large system installation and operation in Europe, Asia and North America. (ii) The Tidal Energy Demonstration and Evaluation Center (TEDEC) at Maine Maritime Academy, University of Maine. TEDEC is one of the few certified US test facilities with established test sites in Maine. The TEDEC are also experienced in testing and evaluation of standard MHK turbines with submerged generators analogous to the TRL4 system developed for this project and (iii) the group of Prof. Kimberly Cook-Chennault at Rutgers University. Laboratory facilities and student assistance were made available at Rutgers for the assembly and testing of the TRL4 system in the Department of Aerospace and Mechanical Engineering. Student participation enabled a cost-effective proposal and was in line with our educational goals. The TRL4 system integration and initial check-out provided the subject material for a Master's Thesis awarded in 2011 [3] and several senior undergraduate projects.

C. Project Goals & Tasks

The initial goals in this project were to

1. Demonstrate in the laboratory, a system with no submerged electronic components or high ratio gears, for MHK power generation.

    a. TRL4 system design, sizing and procurement of COTS components. Modified as necessary, for integration. The HET sub-system to incorporate a positive displacement pump, connected to a hydroelectric generator.

    b. Integration & testing to provide sufficient data for performance evaluation and scale-up planning. Integration at Rutgers University and final testing at Maine Maritime Academy - TEDEC

2. Define preliminary plans for a TRL 5/6 system.

3. Perform performance & cost modeling for large scale systems (TRL3)

4. Provide critical assessments on

    a. The viability of this MHK technology

    b. The potential for accelerated commercialization

    c. The potential to address the critical energy needs and security of the USA.

    d. Identify areas of technical leadership which could lay the foundation for a strong MHK manufacturing industry in the USA

In the course of the Project the following amendments were made.

1. TRL3 tasks were started mid-term because of a delay in the finalization of the NASA/JPL subcontract with DOE. At which time the progress on TRL4 obviated the necessity for separate hardware evaluations.

2. The final testing of the TRL4 system was conducted at Rutgers, including pump immersion testing. Outdoor winter testing at TEDEC was problematic and it was also concluded that there was little added value in relocating the system since only the pump would be immersed and the cost and delay would be substantial. A simulated laboratory immersion test was substituted.

D. TRL4 System Development

System Sizing, Design and Component Selection

Starting from a conceptual design shown in Figure 3, the goal was to develop a 10kW model system which would operate to simulate tidal drive torque and RPM conditions equivalent to identified test sites in Maine, identified by the TEDEC team and the use of a ~2m diameter turbine blade in a maximum flow of 5 knots. The hydraulic pump was identified as the key system component. A COTS component, scalable for a full-sized system and suitable for operation in a submerged marine environment was targeted. The other TRL4 system components were to be sized and selected based on this core unit. It was initially envisioned that this system would use water, with anti-corrosion and lubricating additives as the hydraulic medium.

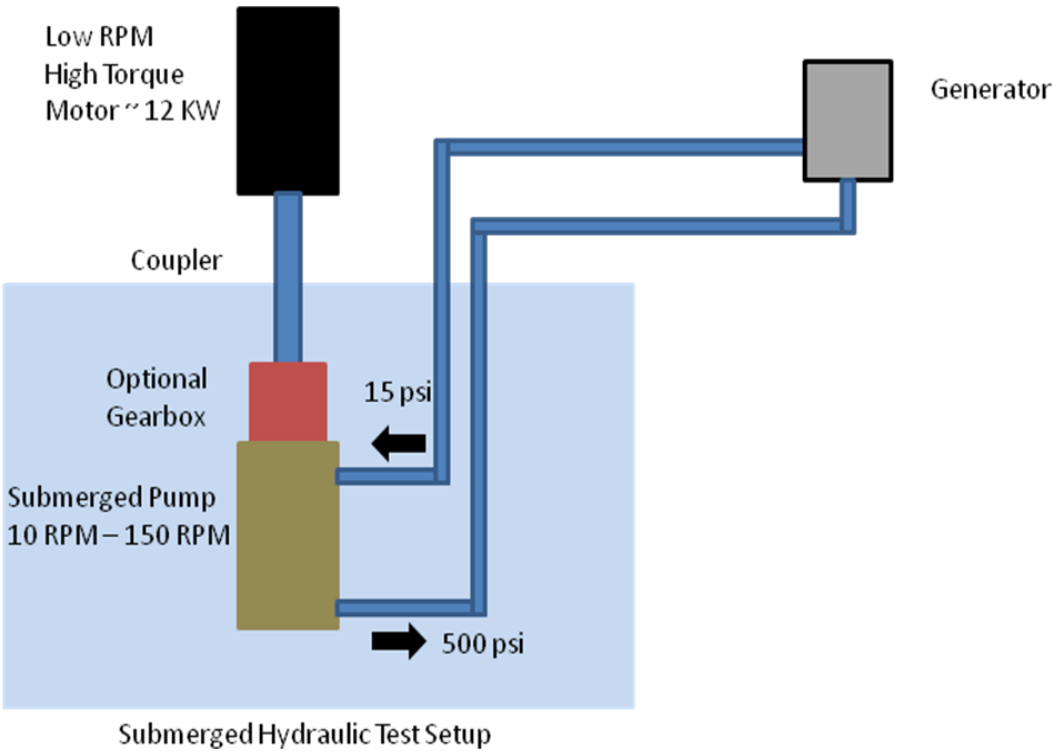

Figure 3. Preliminary Concept design for the TRL4 System

After an extensive survey of pump components and suppliers, two manufacturers were identified for radial/axial hydraulic motors/pumps as leading candidates, Bosch-Rexroth and Hagglunds. Contact was made with Bosch-Rexroth and it was discovered that Hagglunds had recently been acquired by the Bosch Corporation. Separate product lines

were maintained, but the full range of components were available through Bosch-Rexroth in Pennsylvania, USA. After detailed discussions with US and European engineers at Bosch, the CA50 series of hydraulic radial motors/pumps, manufactured by Hagglunds became a focus for the TRL4 system. Units with the same design are available in sizes from smallest (CA50-20) to the largest (MB-4000) with rated powers between 293kW and 1.1MW respectively, at maximum hydraulic pressures of 5000 PSI in each case. Although our targeted TRL4 system was much smaller (10kW), the CA50 units were characterized at >80% efficiency even at <5% of the rated power. With consideration on the availability and scalability of these COTS components for large scale systems and the proven track record for Marine and off-shore oil environments, we locked-in on a CA-50 pump for our system, accepting that we be operating in a lower efficiency window for the demonstration but that >95% efficiencies was achievable with components sized for full scale systems. As a further compromise, we procured a CA50-32 unit, rated at 328kW, instead of a CA50-20 unit, since the lead time was more consistent with the Project schedule. The procured unit was available with “ruggedized” coatings and seals, and suitable for laboratory immersion testing. A “Marinized” grade, suitable for long duration marine immersion is also produced but not available on the schedule of the Project.

Following further analysis and discussions with Bosch Rexroth and other candidate suppliers, the preliminary TRL4 system design migrated to a hydraulic circuit as shown in Figure 4. This system incorporates

1. An electric drive motor and step down gearbox to simulate tidal drive torque and RPM conditions. A CEM2334T, 20HP motor with a 14.63:1 step-down GIF1488E gearbox, both from Baldor were considered appropriate.

2. A Hagglunds CA50-32 Radial Hydraulic Pump

3. A Bosch-Rexroth AA2FM-160 Axial Hydraulic Motor, sized to operate at ~1500RPM with a Hagglunds pump rotation of ~120 RPM

4. An AC Inverter (generator) to convert the mechanical power output of the axial motor. A ZDPM18025C-BV unit from Baldor was selected. A 20HP unit initially targeted but the 25HP unit had a shorter lead time.

As the system design evolved it became apparent that several additional components were required to build an appropriate system.

These included;

1. A Variable Frequency Drive for controlling the electric drive motor. A Mitsubishi F720-00630, 25HP unit was selected as being suitable for both control and power monitoring.

2. A hydraulic "Power Pack", this unit supplies a reservoir for the system and should be sized to provide make-up volume and sufficient fluid pressure to the pump (~50PSI) for normal operation. A Rexroth PP20 unit with a 20G reservoir and a rated maximum flow of 5GPM. This unit is required because the TRL4 system does not have an intrinsic hydraulic head of pressure as would be the case in a deployed system with a submerged pump and hydraulic circulation to shore.

3. A resistive load bank connecting to the generator output to provide a system load for normal operation and to dissipate power in the form of heat. An AVTRON K595 unit with adjustable load settings was identified as suitable.

4. A range of sensors to monitor electrical power, pressure, flow, temperature, mechanical torque and RPM. Also data logging and display for real time monitoring and subsequent analysis.

5. A range of hoses, fittings, relief valves and a check valve to interconnect the system components and support safe operation.

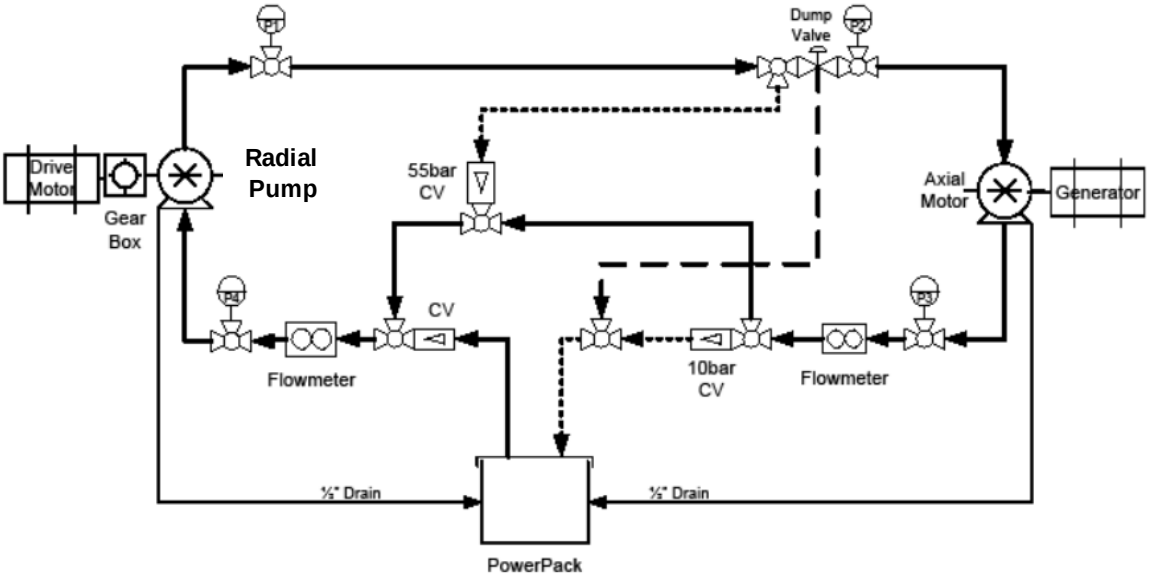

Figure 4. Final TRL4 System Layout.

Parts List for TRL4 System

Simulated Tidal Drive Components:

VFD Motor Control: Mitsubishi F720-00630,25HP [4]
Motor: Baldor CEM2334T, 20HP,1765RPM,3PH,60Hz,230V [5]
Gearbox: Baldor GIF1488E, 2 Stage Helical, 1/14.63 Ratio. [6]

Hydraulic Circuit Components

Radial Pump: Hagglunds CA50-32 [7]
Axial Motor: Bosch Rexroth AA2FM-160 [8]
Power Pack: Rexroth PP20/G2-011/7.5, 20G, 5.5kW, 5GPM [9]

Electrical Generation Components:

AC Inverter: Baldor ZDPM18025C-BV, 25HP, 60Hz,460V 3PH @1800 RPM [10}
Load Bank: AVTRON K595, 55.5kW Max Load @ 240V.

Misc.

Hydraulic Hoses: Parker 451TC-20, 3000PSI, 1.25” ID flexible hose
Pressure Relief Valves: Rexroth DBDS10G1X & 20G1X
Check Valve: Rexroth S20A1.0
Hydraulic Fluid: Schaeffer, “Ecoshield -512”, ISO68 [11]

Sensors:

Pressure Transducers: Bosch Rexroth HM17-11/100, 0-100 bar, 4-20mA
Flow Meters: HEDLAND H800A-100-MR, 0-100G, 0-10V DC.
Thermocouples: Omega, Type K
Torque Gauge:Omega TQ513-2K, 0-2000 in-lbf
Digital Tachometer: Omega PRX102-8N
Generator-output CRM CR6200AC Power Transducer & Current Transformer

Data Acquisition:

DAQ System: National Instruments, NI-Compact DAQ [12]
Software: National Instruments, LabVIEW, Version VI

Hydraulic Fluid:

In was initially envisioned to use water with additives as a hydraulic fluid. However, after extensive discussions with the equipment manufacturers we revised this approach since the preponderance of existing data indicated that internal corrosion effects and wear will occur and severely limit the useful lifetime of the hydraulic components even with anti-corrosion and lubricating additives in the water. As an alternative for the TRL4 demo we turned our attention to a range of manufacturer recommended biodegradable fluids. Specifically, we selected EcoShield™ 512 a Biodegradable Hydraulic Fluid made by Schaeffer. It is blended from a unique combination of high oleic vegetable base oils and biodegradable synthetic polyol ester base fluids. The TRL4 system required approximately 30 gallons of hydraulic fluid for operation. As noted in Section E, attention is focused on using poly-glycols with negligible environmentally risk for a full scale system test.

TRL4 System Integration

The main components of the TRL4 system were laid out on two metal framed skids as shown in Figure 5. The first skid contained the electric drive motor with a directly attached gearbox and the radial hydraulic pump. Coupling from the drive system to the pump was accomplished through a commercial coupler and a custom machined SS spline shaft. An immersion tank was fabricated and configured around the radial pump with a face sealed aperture to the pump body to allow access for the drive shaft.

The second skid contained the axial hydraulic motor attached to the generator. The “power pack” was separate but co-located on this skid. The axial motor and the generator were coupled with two connecters and an intermediate torque sensor as shown in figure 6. A proximity sensor was mounted on the skid and keyed to a post on a coupler unit, for monitoring the shaft RPM.

The hydraulic connections were completed using the hoses and fittings shown above in Figure 5, including casing drain lines from both the Hagglunds pump and Rexroth motor routed to the reservoir in the powerpack unit.

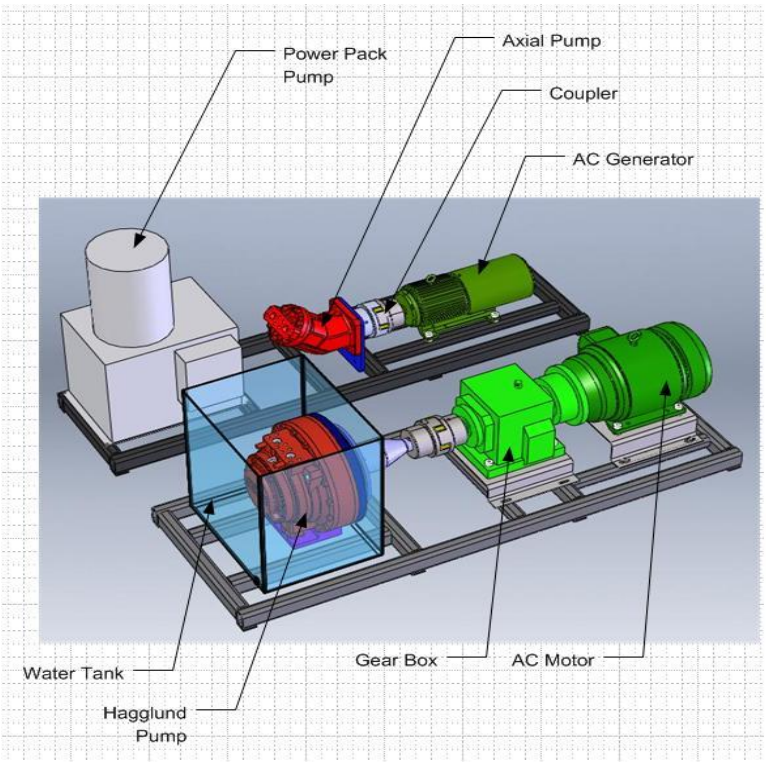

Figure 5. Schematic of Integrated Components on Skids

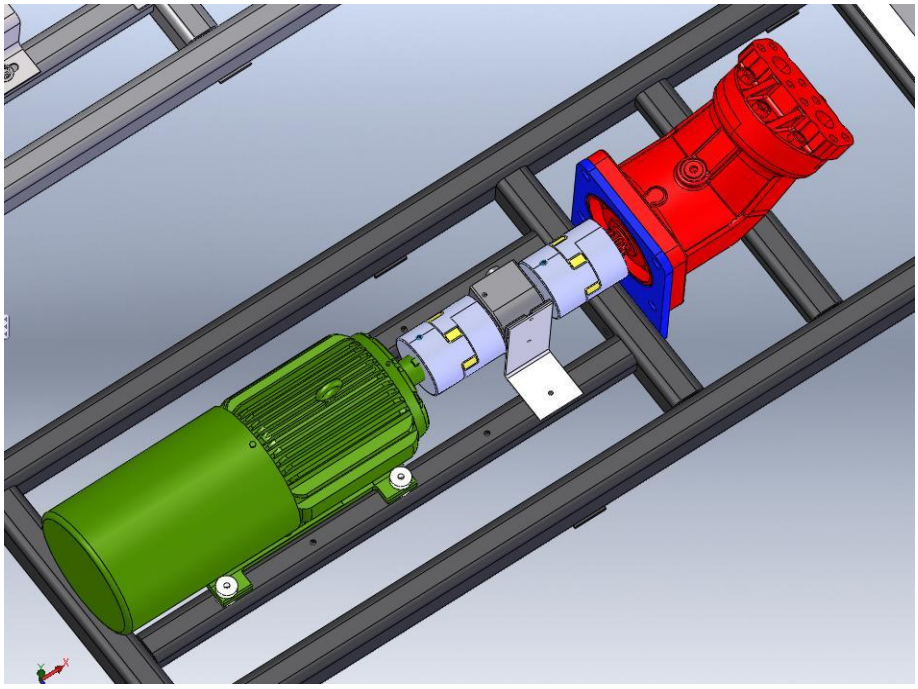

Figure 6. Schematic of Axial Motor to Generator with Torque Meter

Pictures of the components and their integration are shown in Figures 7 to 10.

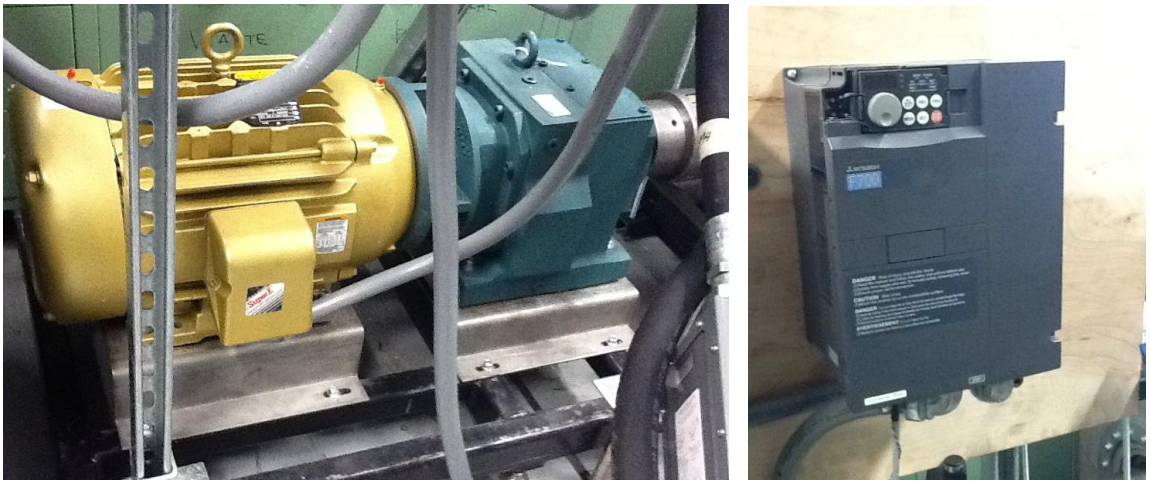

Figure 7.. 20HP, Baldor Electric Drive Motor CEM2334T, Gearbox & VFD Motor Control

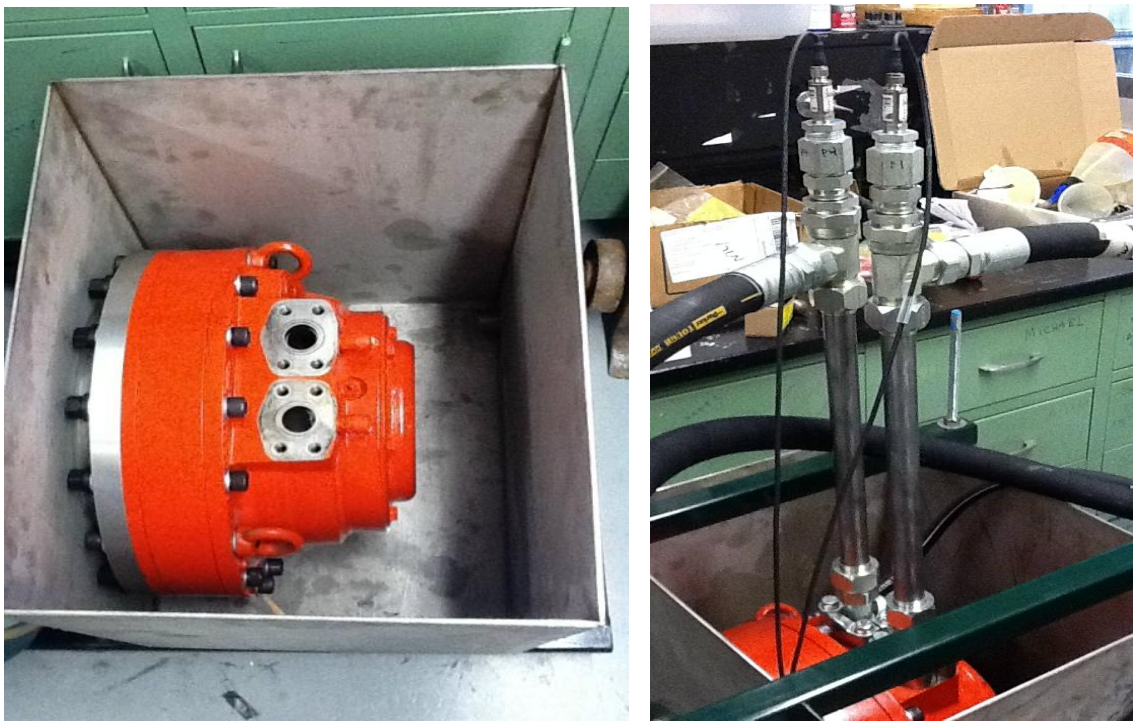

Figure 8.. Hagglunds CA50-32 Pump, Immersion Tank & Hydraulic Connections

Figure 9.. Baldor 25HP ZDPM18025C-BV Inverter & Avtron K595 Load Bank

Figure 10. System Hydraulic Connections

Data Acquisition System

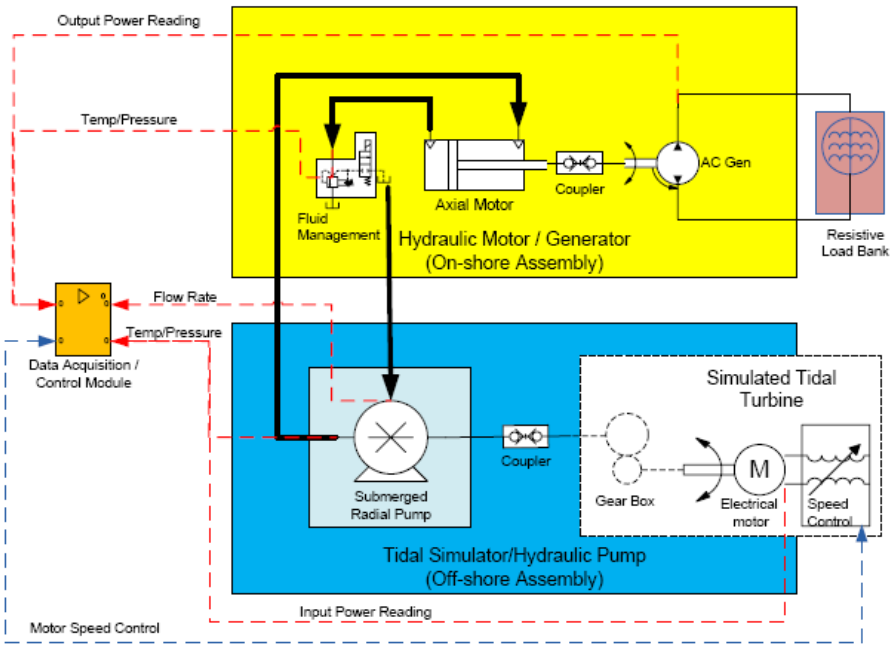

Figure 11. Data Acquisition System (Torque Meter and Tachometer not shown)

The TRL4 system performance was monitored using a suite of sensors. These sensors included analog-voltage and 4-20mA current-loop devices connected through to modules on a National Instruments Compact DAQ system. This system was interfaced to a Windows PC for data logging. A LabVIEW VI program was written for sensor signal conditioning, data display and recording. This DAQ system is shown in Figure 11.

The Mitsubish FR-720 VFD unit provides outputs for monitoring the voltage, current and total power being supplied to the electric motor. System hydraulic pressures were monitored using four BR HM17 type pressure sensors from Rexroth Bosch. These are located at both the pump and hydraulic motor inlet and outlet ports respectively. Hydraulic fluid flow was monitored at the pump inlet and hydraulic motor outputs via remote-sensing Hedland MR Flow meters. An Omega TQ502 shaft torque transducer and an Omega PRX102 Proximity Detector measured the mechanical output and RPM developed by the hydraulic motor. The 3-phase electrical output of the generator was measured by a CRM CR6200 AC Power Transducer. This was wired in parallel with the output resistor load bank. Temperature was measured at several external points on the

components and fittings Type K thermocouples interfaced through a dedicated thermocouple module on the NI-CDAQ.

The LabVIEW VI program accepts multiple digitized inputs from the NI-CDAQ system. It provides for scaling the sensor signals to calibrated values, filtering, and display. Derived parameters such as hydraulic and mechanical powers based on sensor torque, RPM, pressure and flow were also determined. At predetermined time intervals snapshots of the current data were written to a data log file for later analysis. Measurements of the input pump shaft RPM were found to be stable for at given motor drive frequency, and were monitored independently using a hand-held DT-2234C^{+} optical tachometer.

Motor Drive Sub-system

The focus of this TRL4 study is to evaluate the power transfer efficiency between the input of the hydraulic pump and output of the hydraulic motor. The motor drive sub-system and generator-load sub-system are simply employed to simulate and to complete a working system. In full MHK systems these will be replaced by a turbine and a more sophisticated grid matching generator, respectively.

For the TRL4 system we originally planned to calculate the pump input power directly from mechanical torque and RPM measurements on the shaft. However, torque meters of this size were not readily available except as a custom unit with high cost and prohibitive lead times. The electrical power delivered to the motor was therefore used as an alternative in conjunction with a simple motor model to translate the measured electrical input power into mechanical input power at the pump.

In operation, the variable electric motor is driven at controlled speeds provides the input. Understanding sub-system efficiencies relies upon accurately knowing the applied power as the most basic parameter of interest.

The electrical motor is specified by the manufacturer for 3-phase operation at 230VAC. Input current, output power, RPM, electrical power factor and efficiency depend on load. In the Project we operated at a 208VAC line power available in the lab and the parameters therefore require rescaling to represent the reduced output. The output torque for an induction motor follows the square of the voltage. At 208VAC then, the output torque is reduced to (208/230)^2 = 81.8% of the rated 230VAC. The mechanical output power is correspondingly reduced, from the nominal 20 HP to 16.4 HP at 208VAC.

The VFD supplies power to the motor, incorporates the reactive component and motor power factor, and computes the electrical power being used. In conjunction with the electric motor performance data this allows an estimation of mechanical power at the motor output shaft. In the experimental runs, the motor is driven at different speeds by the VFD controller. The drive voltage is constant, but chopped to produce a lower average

voltage to reduce the motor speed. The motor operation is based solely on applied load and can be compared at different drive frequencies. The output voltage corresponding to different frequencies is a chopped version of the 208VAC line.

During testing, we observed motor loading between 25% to 175% of the (adjusted) rated output. From the manufacture's performance curves efficiencies are in the range of 93% at the rated power dropping to ~91% at both extremes. For our subsequent power calculations we assumed a fixed efficiency of 93% for the motor over the range of operation. The manufacturer also describes the gearbox efficiency to be "as high as 98%" per stage. Having a two stage gearbox our net efficiency for the drive sub system is therefore estimated to be 0.89% (0.93 x 0.98 x 0.98).

The drive power used for our system performance evaluation is therefore taken as

(Pump) Drive Power (kW) = (VFD Motor Power (kW)) x 0.89 …… ..(1)

We note that this is a conservative estimate of the sub-system losses and represents an upper estimate for the actual pump drive power, which may be several percent lower.

Rotation speed is not specifically controlled, but observed variations at the output of the 14.63:1 reduction gearbox matched within a measurement uncertainty +/- 0.5 RPM.

Volumetric Flow, Displacement & Casing Drainage:

During the initial check out of the TRL4 system we observed a flow differential of approximately 5GPM between flow meter readings at the inlet to the Hagglunds pump and the outlet of the Bosch-Rexroth motor, the former always being the higher. After checking the flowmeters and sensor circuits we concluded that this difference was real even with the manufacturers specification that the accuracy of the Hedland MR rotatmeter-type flow meters is ±2% of full scale for any reading and also that the temperature of the hydraulic fluid, and consequently it's viscosity, varied between 40-85°C and approximately 70-20 cSt, respectively. As additional points of reference we also calculated the displacement flows according to Equation 2. For the pump and the motor, the displacements per revolution are 0.531G and 0.0424G, respectively. Flow meter (FM) readings and displacement values are plotted in Figure 12 versus the VFD frequency for the drive train.

Displacent Flow = RPM x Displacement Per Revolution …….(2)

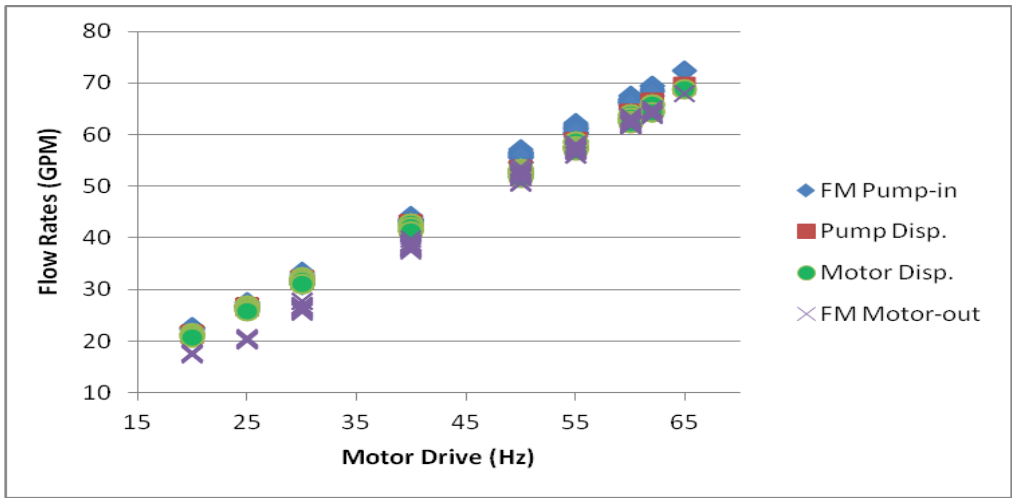

Figure 12. Flow Meter Data & Calculated Displacements vs. Motor Drive Frequency

Notably, the pump and motor displacement values are in excellent agreement within measurement uncertainty. Indeed the motor displacement is considered more precise, since RPM values are higher and measured by a fixed proximity gauge, whereas the slower pump RPM was only periodically verified with a handheld optical tachometer which had a precision of approximately +/-0.5 RPM. It is also observed that the FM2 measurement into the pump exceeds the pump displacement, and more so at high speeds (flows), and that the FM1 measurement out of the motor is lower than the motor displacement, and more so at low speeds (flows).

These differences can be attributed to casing drainage effects in both the pump and motor, which is a normal characteristic in hydraulic components. Basically, fluid seals around moving parts invariably leak to some extent. This is often by design, for example for lubrication. The fluid is typically captured by the machinery case and then returned to the reservoir. The case pressure is kept relatively low to avoid putting strains on the case itself. Since the leakage comes from pressurized fluid, casing flow represents energy lost. The amount of flow depends on the pressure gradient across the seal, the mechanical clearance, and the fluid viscosity. Contributions can come from both the high and low pressure sides within the hydraulic component.

In the TRL4 system the drainage lines are connected to the fluid reservoir in the Powerpack. In a deployed MHK system this could be returned directly to the pump inlet line with appropriate pressure differentials and check valves. Although the drainage from the pump and motor are seen to vary independently depending on the operational speed (or flow), the net drainage remained around 5 +/-1 GPM as shown in figure 13. When

considered as a percentage of the hydraulic flow this becomes much more significant at low flow conditions.

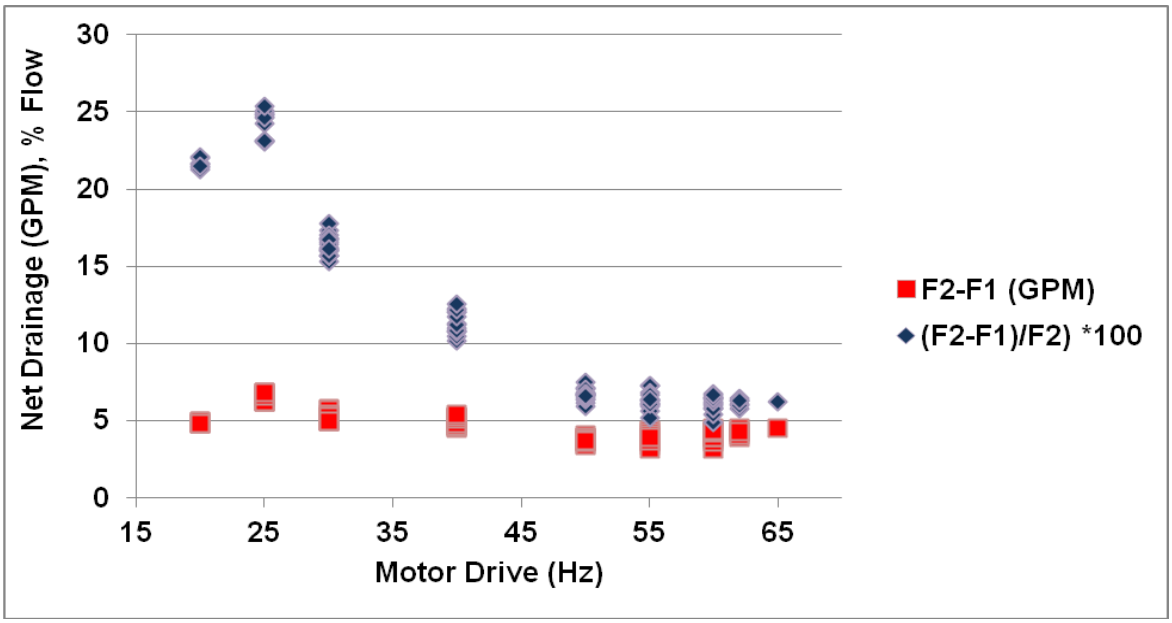

Figure 13. Net Drainage (GPM) and % of Pump Inlet Flow for TRL4 system

This drainage is a factor in the volumetric efficiency, and ultimately the total efficiency, of the pump, motor and hydraulic system. The volumetric efficiencies are captured for the TRL4 system in Figure 14, plotted against the input power, applied to the pump. The basis of the analysis is the ratio of FM to displacement values.

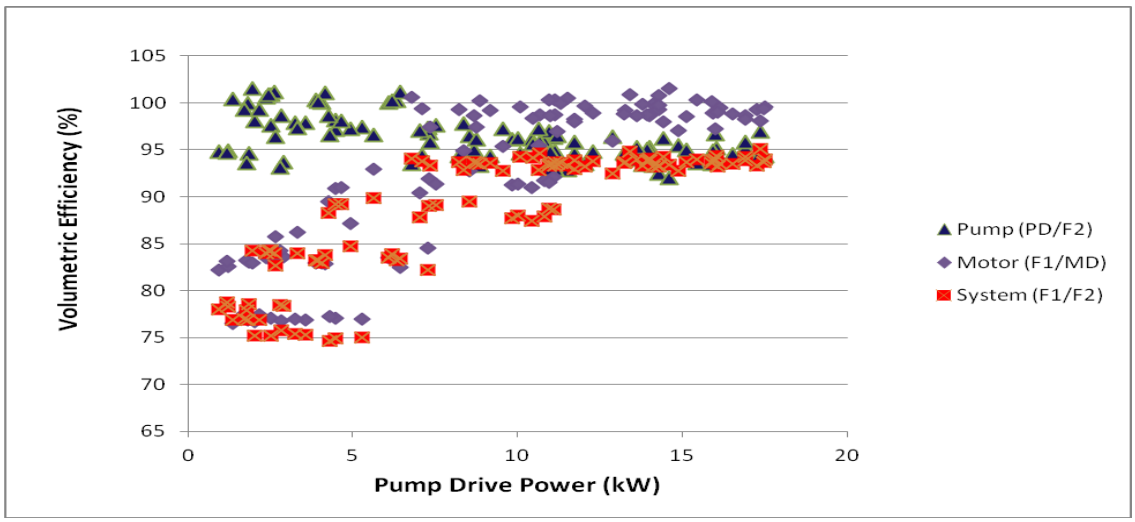

Figure 14. Volumetric efficiencies for the Hydraulic Pump, Motor and TRL4 system.

Consistent with the above discussion, the volumetric efficiency of the pump is calculated to be high (~98%) at low drive powers, dropping to ~95% at higher power. Conversely, the motor has a low volumetric efficiency at low power (~75%) but increases to ~98% at high power. The component behavior is specific to the operating conditions. It appears that the pump is well matched over the whole range while the motor solution may need to be refined for optimized performance at low power. This will be discussed below in the context of large scale and next Phase system designs. The net volumetric efficiency of the TRL4, taken as the combination of pump and motor efficiencies, is seen to be as low as ~75% at low power reaching ~94% for drive powers above 7kW. The volumetric efficiency will be factor in the total efficiency of the components as evaluated below.

It is emphasized that for any deployed system, the COTS component, and especially the hydraulic motor, will be more appropriately sized and will not be run at such low % of rated power regimes. Rather, all hydraulic components are expected to be 95% efficient or better.

In operation, each revolution of the pump delivers a fluid volume to the high-pressure transport line based on the pump displacement, less any pump case flow coming from high-pressure sources within the pump. The high pressure transport flow represents the usable power output of the pump, so the displacement volume flow provides the most direct measurement of the pump efficiency. Similarly at the motor side, the transport flow feeds the motor rotation, and any motor case flow coming from high-pressure sources within the motor. It presents the best measurement of the hydraulic power being delivered to the motor. In the TRL4 system, the pump and motor displacement flows are seen to be identical within measurement uncertainty. Because the motor displacement flow is more precisely determined, as noted above, the motor displacement flow will be used for the output power calculations for the pump.

Other Issues & Uncertainties:

Fluid Temperature: The temperature at the exterior of the hydraulic components was seen to rise during operation to approximately 85°C. This is close to the recommended limit for the hydraulic fluid, pump and pressure sensors. Oil temperature affects fluid parameters such as viscosity and density directly. Air dissolved in the fluid can also be forced out of solution as the temperature rises. The resulting micro-bubbles reduce fluid density and increase compressibility. These factors reduce efficiency and may create measurement errors. However, based on an extensive review the net impact on the TRL4 system is estimated to be a relatively small, (<2%) reduction in hydraulic efficiency. Ultimately, in a submerged system ocean cooling will resolve this issue

Air Entrapment: as configured, is possible that some trapped air remains in the Pressure sensor plumbing after the system has been purged during start-up procedures. Sensor operation is not affected, but the air can compress during operation, changing the operating volume and absorbing energy that would be available. Such trapped volumes

would be relatively small and the potential impact on efficiency is estimated to be <1%. In future systems we will insure the sensor orientation is appropriate to avoid this issue

Test Procedures

The TRL4 system parameters which can set prior, and during, operation are the frequency of the electric drive motor and the resistive load on the generator. Settings are made manually. The frequencies range used is between 20 and 65Hz, or 33.3-108.3% of the rated motor speed. Post gearbox, this simulates a turbine blade rotation between 40 and 130 RPM. The resistive load bank has a nominal setting range between 4 and 52(kW), settable in increments of 4kW. While these values have a relative significance in the TRL4 system, they have no absolute meaning. The actual adjustments are to resistance load and the noted power values correspond to a unique operating condition.

At a given frequency, adjustments to the resistive load, induces a change in the applied torque of the motor drive to maintain a constant drive speed. Increasing the load produces a higher torque demand on the drive system. Low-frequency and/or a low-load, corresponds to a low drive power, while high-frequency and/or high-load corresponds to a higher drive power requirement. In this way we can map the performance for a range of drive speeds and drive powers up to approximately 18kW. The available drive power is limited by the available line power as well as the motor itself. At approximately 208V, 65A 3PH line power we were able to access the full load range up to 52(kW) for drive frequencies up to 40Hz while at 60Hz the maximum usable load is 28(kW). Above this point we exceed available line power and the power breakers disengage.

We implemented the following test procedures.

1. Per component manufacturers recommendations, the TRL4 system was powered and run at 40Hz, 12(kW) load for approximately 30 minutes for the components to stabilize before taking data. This should also ensure flushing of any trapped air from the system.

2. Data was taken at frequency set points of 20, 25, 30, 40, 50, 55, 60, 62 and 65 Hz.

3. At each frequency, data was collected while manually stepping up the resistive load from 12 to 52 (kW), or the highest setting before the line power is exceeded, in 4 or 8 (kW) increments. Starting at the lowest setting, holding for 5-10 minutes at each set point up to the maximum setting. At the end of each data set, with the system still running, the load was reduced to the lowest setting, then the frequency stepped to the next set point and the same data collection procedure repeated.

4. After stepping-up through the frequency settings as indicated in 3. We then reversed adjustments, stepping down in frequency, similarly collecting data at each

frequency while adjusting the load stepwise from minimum to maximum in 4 or 8(KW) increments.

5. The sequence or ramping frequency up and down, collecting data at each frequency while adjusting load, was repeated 3 times during a single test run.

6. Procedures 1-6 represent a typical test run from which we obtain a corresponding suite of data. Each run takes approximately 6 hours to complete. Three such runs were conducted on different days to provide the basis for evaluating the TRL4 system performance. Two of these runs were performed with the system in a dry condition, while one run was performed after filling the immersion tank around the pump with city water. This was the first simulated immersion test.

7. All test data are included in our evaluations below.

A sub-set of exemplary data are shown in Figure 15. This shows a screen plot of measured and calculated power data taken at 40Hz while stepping the load from 12, 20, 28, 36, 44 and 52 (kW). As shown the power values increase with load at the fixed frequency. Data for subsequent evaluations and calculations were taken after the initial hysteresis at each step. While some parameters, including the power as shown, vary stepwise with the load other parameters, including RPM and flow are as expected more dependent on the frequency than load.

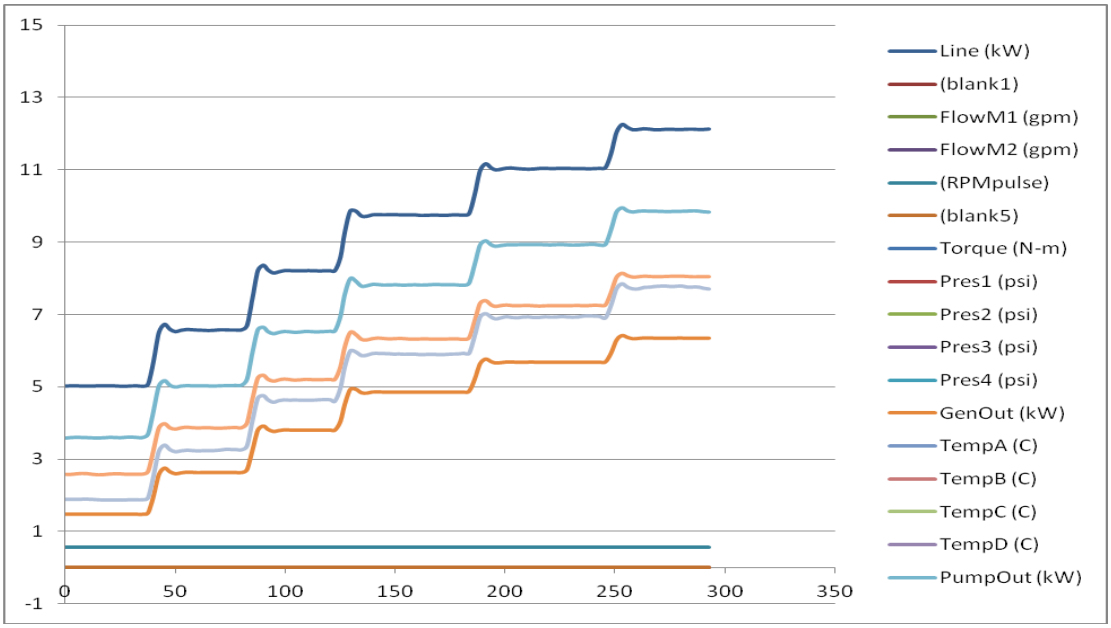

Figure 15. Exemplary plot of power data (in kW) taken at a fixed frequency (40Hz), while varying the resistive load bank settings stepwise, in increments of 8, between 12 and 52(kW)

Hydraulic Power Calculations

With the sensors in place and the LabView DAQ engaged, we can measure and/or calculate the power at a number of points in the TRL4 system for an evaluation of component, sub-system and system performance. The following is the form and rationale of our calculations and estimations.

(i) Pump Input Power (PIP):

As discussed above, PIP is estimated as

PIP (kW) = 0.89 (VFD Power to Electric Motor (kW)) ----- (3)

(ii) Pump Output Power (POP):

POP Is calculated from the product of pump displacement flow (RPM x displacement per revolution) and the pressure differential between the pump outlet (P1) and pump inlet (P4). As discussed above, since the motor displacement flow (MDF) is considered to be a more precisely measured parameter, and identical to the pump displacement it is used in the calculation.

POP (kW) = 0.000435 ((P1-P4) (PSI)) x MDF (GPM))..... (4)

(iii) High Pressure Line Loss (HPLL):

HPLL is calculated from the pressure drop between the pump outlet (P1) and motor inlet (P2) and the flow (MDF).

HPLL (kW) = 0.000435 ((P1-P2 (PSI)) MFD (GPM) ……. (5)

As shown in figure 16, the measured pressure drops (P1-P2) are relatively small (< 4% of P1) but track MFD well. This is consistent with anticipated frictional losses in the hydraulic line, including the effects of T connectors. The magnitude of this loss is expected to depend on the fluid characteristics, hose, configuration and flow. It may be estimated from manufacturer's data. Given the small scale of the TRL4 system this was not done in the present case but is included in larger scale MHK modeling.

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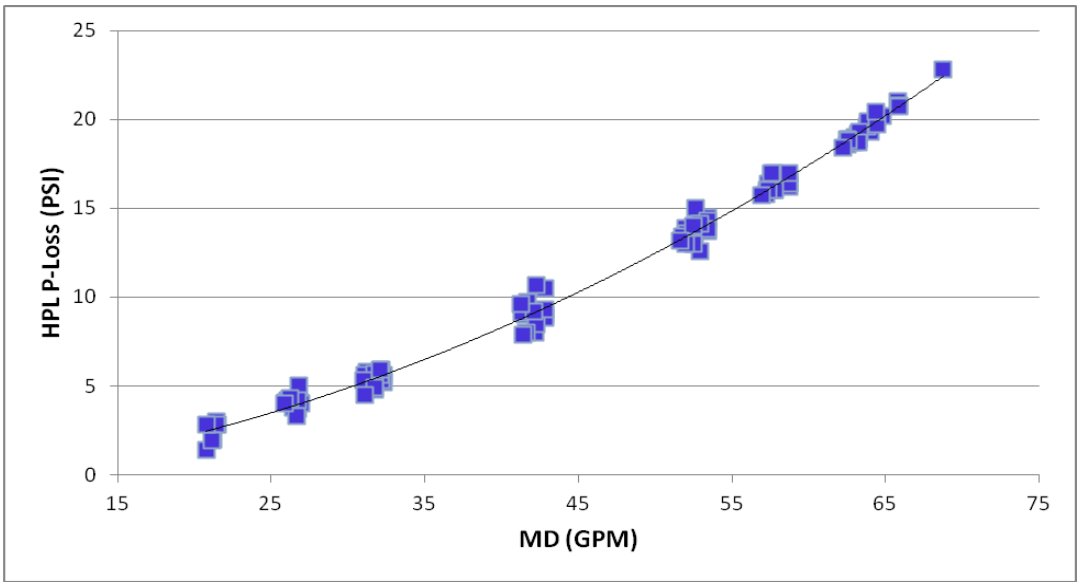

Figure 16. Pressure Loss in High Pressure Line vs MFD

(iv) Motor Input Power (MIP):

MIP can be calculated in two ways, either from the POP, corrected for the HPLL, or from the product of the motor displacement flow and the differential pressure across the motor (P3-P2)

MIP (kW) = (MDF x (P2-P3)) x 0.000435 …. (6a)

or

MIP (kW) = (POP – HPLL) (kW) …. (6b)

In practice it is observed that values calculated according to equation 6a are approximately 92% of those calculated according to equation 6b. We have no definitive explanation for the difference. However, based previous discussion it may be that 6a does not capture the effects of motor casing drainage. In subsequent calculations of motor efficiencies, we elected to use 6b values for MIP. This is the more conservative estimation for evaluating motor efficiency.

(v) Motor Output Power (MOP):

MOP is calculated directly from the measurements of RPM and Torque on the motor output shaft in-line to the generator.

MOP (kW) = (Shaft RPM x Torque (N-m)) x 2Pi/60 ….. (7)

Hydraulic Efficiency Estimations and Benchmarks

Component and sub-system efficiencies are calculated based on ratios of the above calculated powers, specifically ;

Pump Efficiency (%) = (POP/PIP) x 100 ……. (8)

Motor Efficiency (%) = (MOP/MIP) x 100 …… (9)

Hydraulic Sub-system Efficiency (%) = (MOP/PIP) x 100 ……. (10)

The pump, motor and generator efficiencies are benchmarked against manufacturer's performance specifications for the specific operational range investigated.

In terms of benchmarking**, t**he manufacturer's performance curve for a CA50 unit is shown in figure 17. The unit is has a wide high-efficiency operating zone. Overlaying the TRL4 operating range the expectation is that the CA50-32 pump should operate between 85-90% total efficiency (including hydraulic and mechanical components).

For the AAF2M-160 motor, a master curve is not available but in discussion with the Bosch-Rexroth engineering team, we were able to bound the expected performance between 75-82% total efficiency for the TRL4 operational range as captured in Table1.

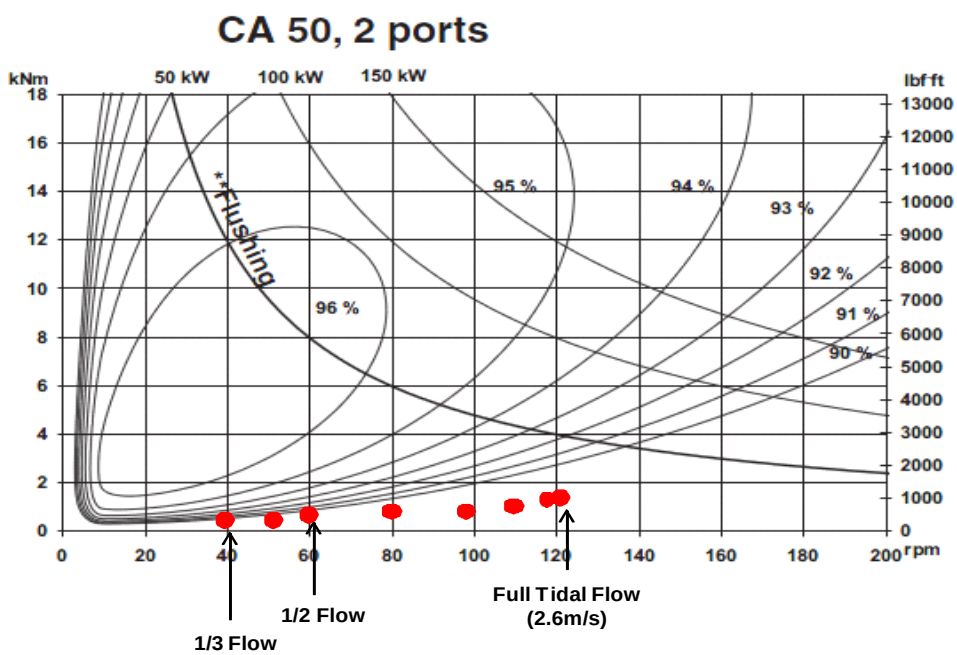

Figure 17. Manufacturer's Performance Curves for CA50 Pump

Table 1. Manufacturer's Performance Est. for AAF2M-160 motor (in TRL4 Operational Range)

Motor RPM	Motor Delta P (bar)	Estimated Total Efficiency
500	5	79%
500	17	81%
750	8	78%
750	29	82%
1000	13	78%
1000	34	82%
1250	15	75%
1250	37	80%
1500	18	76%
1500	33	79%

Results & Analysis

The aforementioned power measurements and calculations for the hydraulic components and sub-system are presented graphically in figures 18 to 20. These are plotted vs. the reference input powers which are used for efficiency calculations. In all cases we observe approximately linear relationships with zero offset intercepts. Although, this is an oversimplification, the linear slopes approximate to an operational efficiency while the intercept offsets are attributed to operational thresholds for the components

The POP is observed to be approximately 88.5 % of the PIP with an offset of 0.28kW

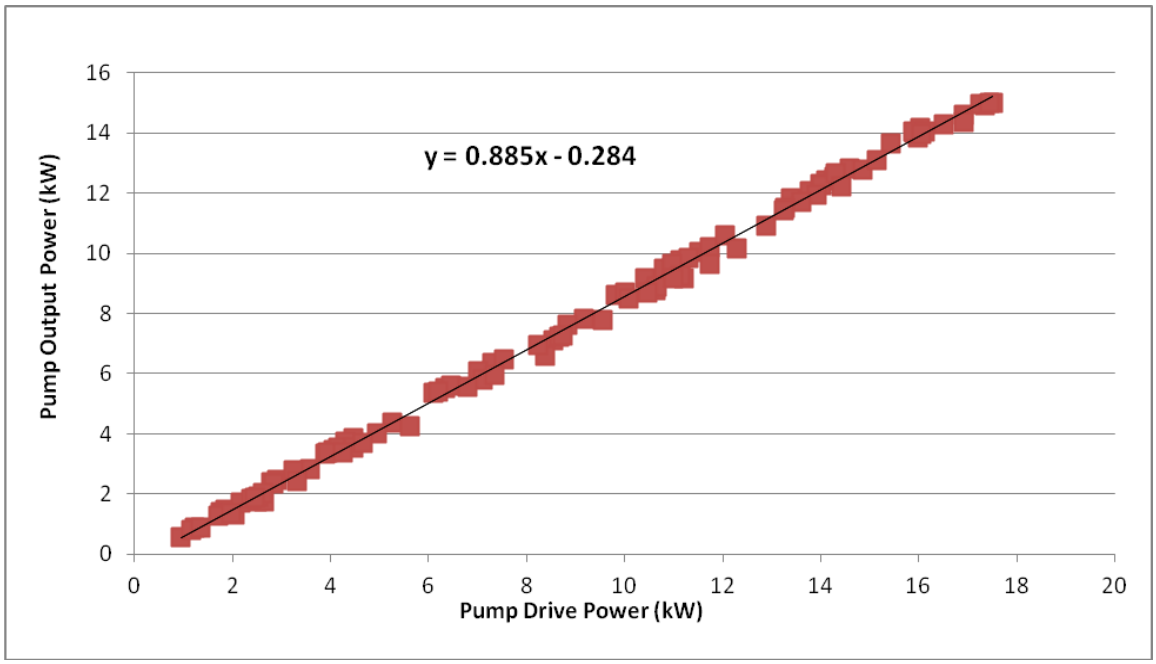

Figure 18. Pump Output Power vs. Drive Power

The MOP is observed to be approximately 75% of the MIP with an offset of 0.55kW

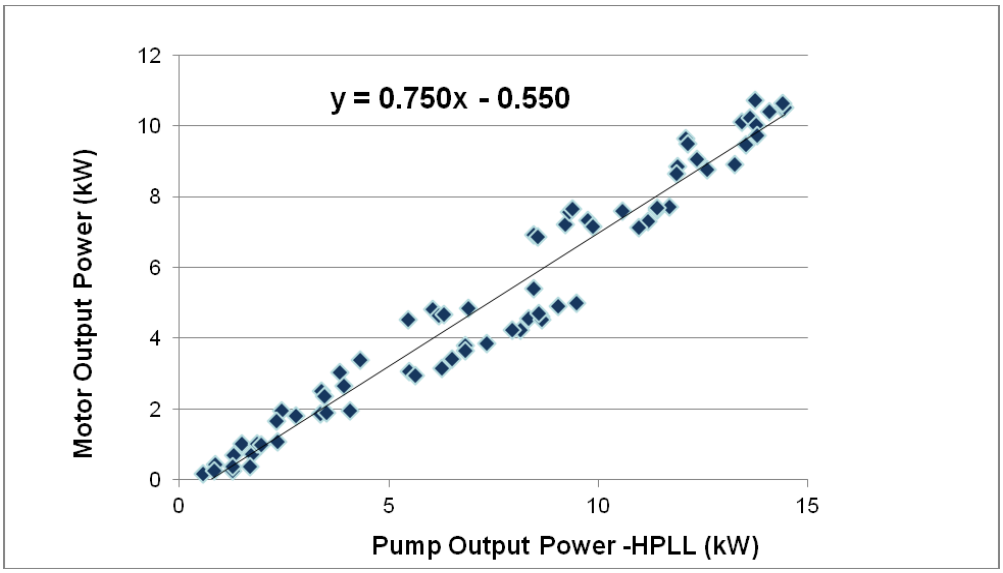

Figure 19. Motor Output Power vs. (Pump Output Power - HPL Loss)

The MOP is also observed to be approximately 63% of the PIP with an offset of 0.72kW

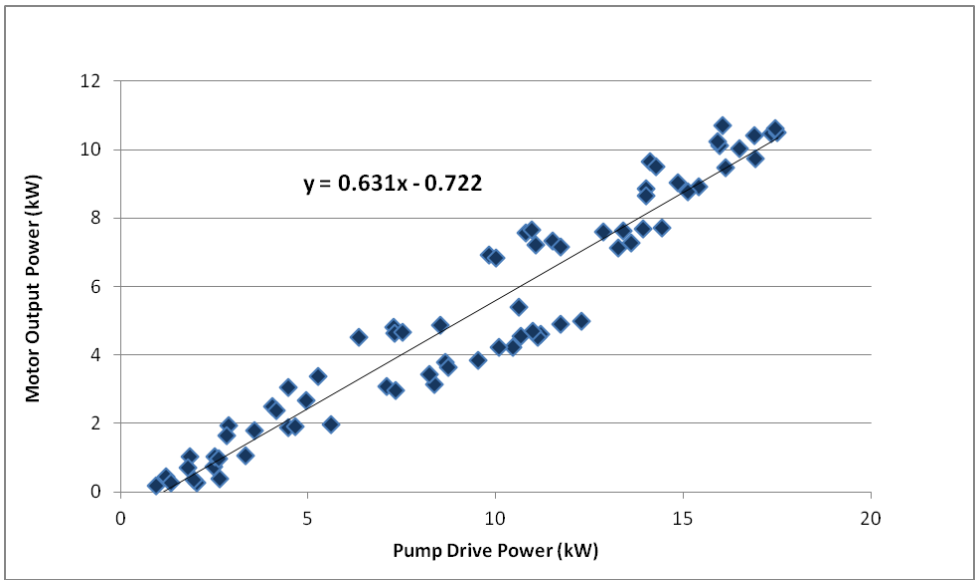

Figure 20. Motor Output Power vs Pump Drive Power)

Based on the above power evaluations the total efficiencies of the components sub-systems and overall TRL4 system can be estimated from the power output/input ratios. These are presented graphically in figures 21-23 versus the corresponding input power.

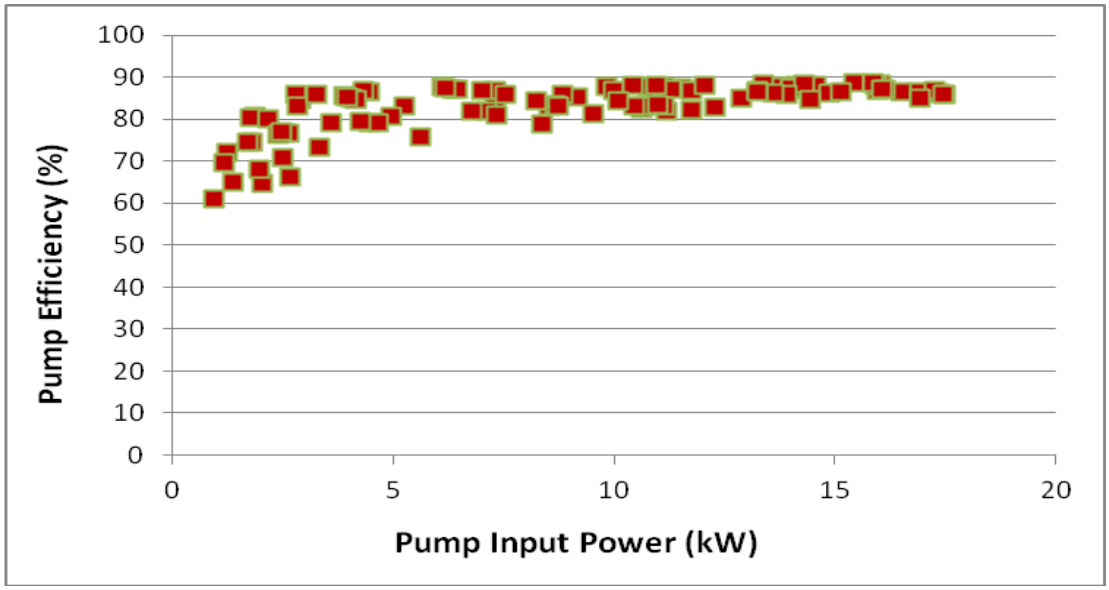

Figure 21. Pump Efficiency vs. PIP

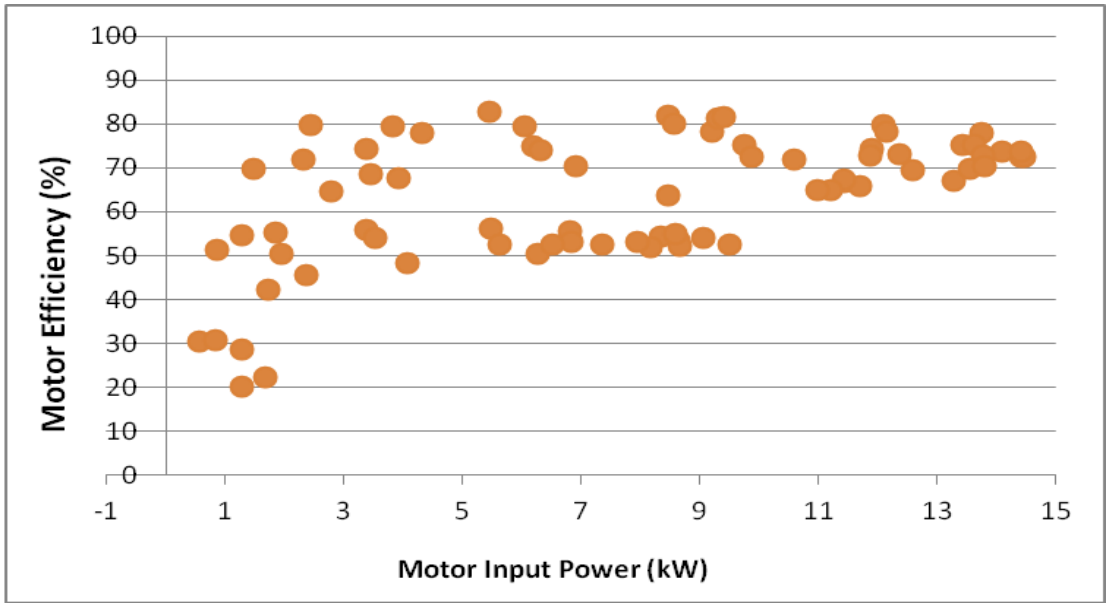

Figure 22. Motor Efficincy vs. MIP

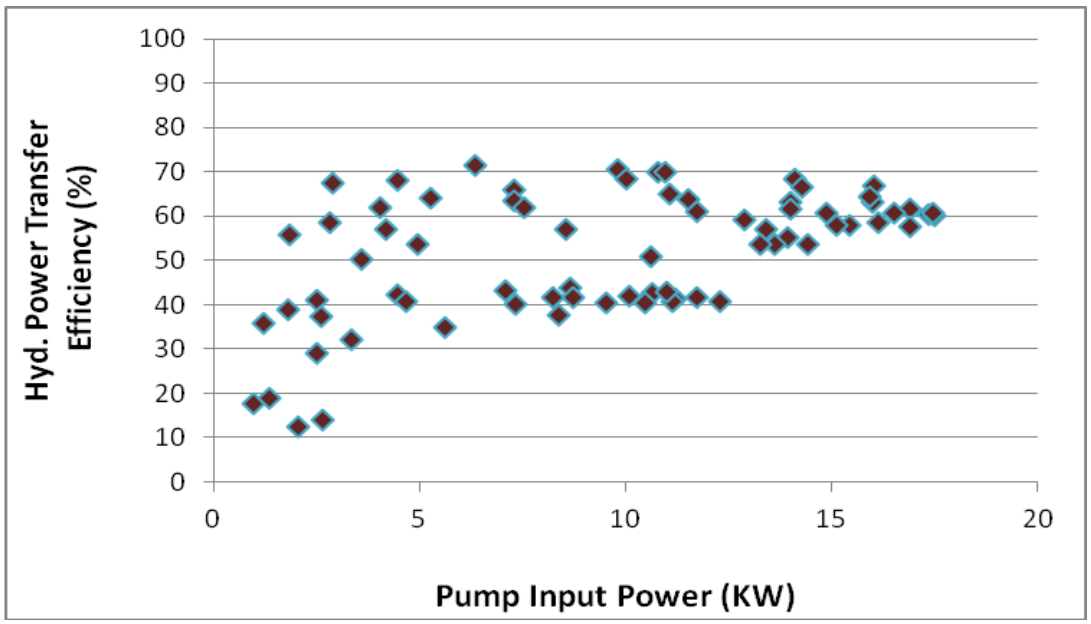

Figure 23. Hydraulic Power Transfer vs. PIP

Analysis and Benchmarking

From Figures 18 and 21, we observe that the Pump output has an approximate power dependence of 88.5% dependence on the drive power above a threshold of 0.284kw.

When the efficiency is calculated, low input power values are obviously impacted by the threshold but attain 85-90% efficiencies at higher power. This is in good agreement with the manufacturer's performance curves shown in Figure 17, where the red dots show the operating range of our TRL4 system.

Similarly from Figures 19 and 22, we observe that the motor has an approximate power dependence of 75% above a threshold of 0.55kW. There is more scatter in the motor data and the impact of the offset is more apparent up to higher input power. The motor performance is also in line with expected manufacturer's performance data in Table 1.

When the subsystem power and efficiency is estimated as shown in figures 20 and 23 the combination of pump and motor performance is apparent. The sub-system has a power dependence of 63% above a threshold of 0.722kW. This is in good agreement with the simple product of the pump and motor dependencies (66%) if we allow for a loss efficiency impact of the high pressure line. The form of the sub-system efficiency data in figure 23 closely mirrors that of the motor data as would be expected. With a significant impact of the relatively high threshold, evident at lower input power.

This analysis is specific for the present TRL4 system, it is emphasized that in any deployed system, the COTS components will be more appropriately sized and none will not be run at such low %'s of their rated power. Rather all hydraulic components are each expected to be 95% efficient or better.

TRL4 Summary and Conclusions

Based on the above results and discussions, we can state the following.

An operational TRL4 system was designed constructed and evaluated to validate the proposed MHK system approach. This system was operated and evaluated under a simulated tidal flow of approximately 1-3 m/s, and between 2-18kW. The radial pump and axial motor used were COTS components, scalable to MW size. In the TRL4 system the specific units employed were operated at only 1-6% of their rated power.

In this regime the radial piston pump performs extremely well over the wide TRL4 operational range and matches the manufacturer's estimated efficiency.

The axial motor also conforms to the manufacturer's estimated efficiencies in the operational range but well below its optimal performance. Volumetric inefficiency of the unit at low powers (see Figure 14) is considered a determining factor.

It is re-emphasized that in a deployed system, the COTS components will be more appropriately sized and none will not be run at such a low % of their rated power. Rather all hydraulic components are each expected to be 95% efficient or better.

The subsystem power and efficiency follow predictable trends based on the pump and motor performance with a small reduction attributed to the line losses in the system.

We are most encouraged by the radial pump performance in the TRL4 system over a wide range of operating conditions. This is a key result which provides a strong validation for the proposed MHK approach, using scalable COTS components. The pump and motor were extremely oversized for the TRL4 system and the motor performance exhibited a significant impact. However, based on the results and manufacturers ratings we can expect pump, motor and sub-system efficiencies to exceed 90% in more appropriately sized systems.

Based on the observed motor performance, it would be advantageous to run these components at higher power. In order to efficiently operate the MHK system over the widest possible range of tidal flows in also be advantageous, not only from a power generation standpoint as will be discussed later from MW systems, but also from the hydraulic efficiency standpoint, to multiplex several smaller motors in the circuit which can be selectively engaged in response to flow conditions.

Although not explicitly discussed, the TRL4 system data obtained from runs when the immersion tank was filled was indistinguishable from the data obtained for the dry unit. Further testing of the reliability in marine environments is obviously required and would be part of a subsequent TRL5/6 syste

E. Preliminary Designs for Higher TRL Systems

TRL 5/6 System @ 15-25 kW:

The direct path forward from the present TRL4 system will be a similar, or slightly larger sized, TRL5/6 system, complete with bladed shaft and nacelle. This assembly could be built using COTS components or alternatively engineered by modifying an existing MHK turbine, replacing the submerged generator with a radial pump. The radial pump and other components from the TRL4 system could be reused. However, based on our observations we would replace the current axial motor and generator with two smaller units which could be multiplexed to achieve higher performance at low flow conditions. The proposed system would be configured for barge mounting and testing at TEDEC..

TRL 8 System @100 kW:

An accelerated path to TRL8 is also open through the modification of Atlantis's Solon 1 (uni-directional) and/or 2 (bi-directional) systems. Basically,replacing the electrical generator with a radial pump and adding multiplexed axial motors and generators. These Solon systems are available. They have been tested in barge mounted (see figure 24) and pedestal mounted configurations. The outputs have also been grid connected.

The most cost effective and expeditious would be to modify and test these systems at their current location. Approvals could also be expedited since it could be presented as an extension of previously approved work. Pictures of the units are shown in Figure 25 and their specifications are given in Table 2.

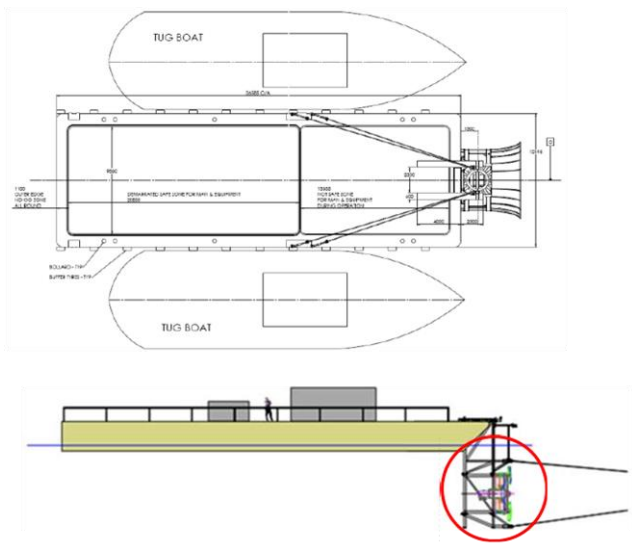

Figure 24. Schematic of barge-mounted Solon 1 System

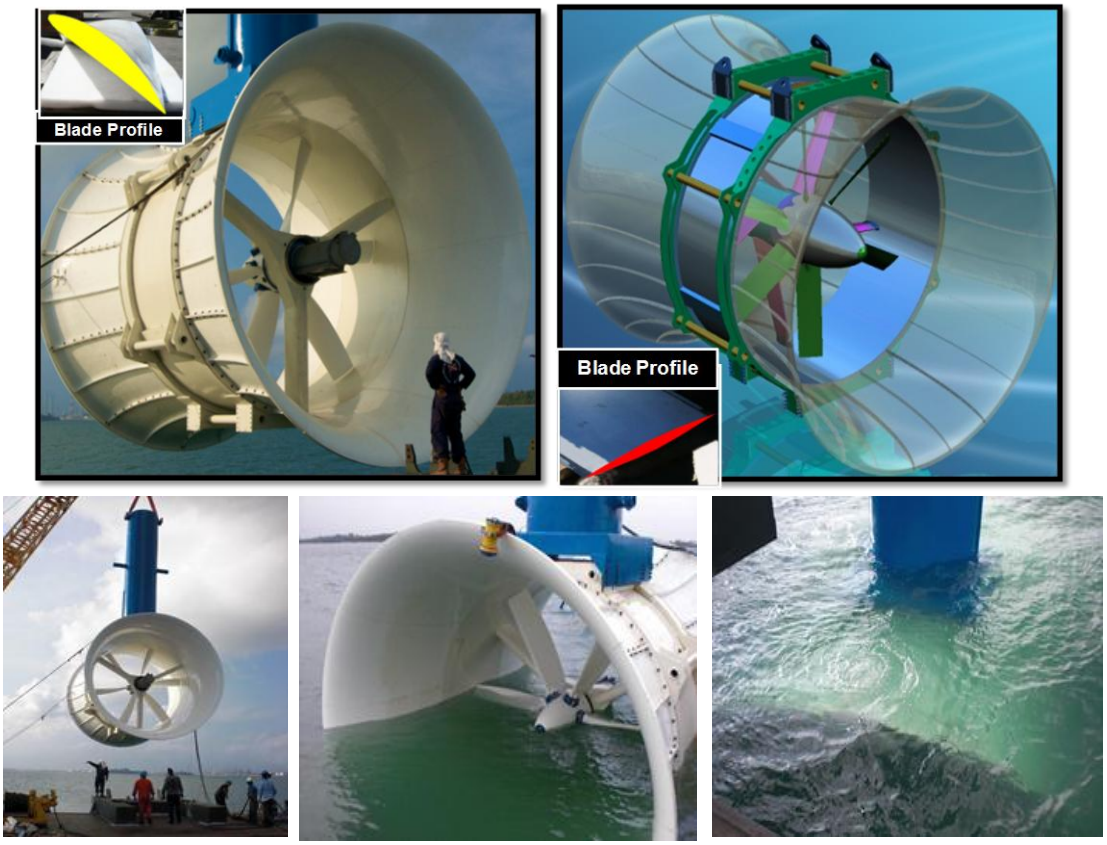

Figure 25. Solon 1 (Top LHS) and Solon 2 (Top RHS)
and the submersion of the Solon 1 unit during barge testing.

Table 2. Specs. For Solon 1 and Solon 2 Systems

Water Speed		
Maximum operational	3.00 m/s	3.0 m/s
Maximum velocity during life	4.12 m/s	4.2 m/s
Capacity	500 kW @ 4 m/s (125 kW @ 2.5 m/s)	100 kW @ 2.5 m/s
Shaft Speed	18-25 rpm @ 2-4 m/s	18 rpm @ 2.5 m/s
Turbine Dimensions		
Length	7.8 m	7.8 m
Diameter at mouth	7.1 m	7.1 m
Diameter at throat	5.5 m	5.5 m
Weight (excluding blades)	20,985 kg	26,654 kg
Shaft diameter	325 mm	300 mm
Shaft length	1.1 m	2.3 m
Nominal diameter of hub	1.0 m	1.2 m
Blade	Mono-directional	Bi-directional
Material	FRP	FRP
Length	2.1 m	2.1 m
Chord length	520 mm	558 mm
Tip diameter (i.e. swept area)	5.4 m	5.4 m
Number of blades	3 or 6	3 or 6
Profile	Patent pending	Patent pending
Generator		
Number of motors per turbine	2	1
Manufacturer	SME with 1:27.7 gearbox from Bonfiglioli	SME's direct-drive PMG
Rating	100 kW + 40 kW	100 kW
Voltage	500 V	500 V
Efficiency	87%	86%
Weight	1,200 kg	5,500 kg
VSD		
Number of VSDs per turbine	2	1
Manufacturer	Vacon	Vacon
Rating	100 kW + 40 kW	100 kW
Operating Conditions		
Site location	TBA	TBA
Site conditions		
Design depth of operation	15 m	15 m
Minimum clearance from se	5 m	5 m
Expected life of turbine	5 years (structural)	5 years (structural)
Maintenance cycle	5 years (structural)	5 years (structural)
Deployment and Support		
Method	Gravity-based	Gravity-based

F. TRL3, 15MW System Cost & Performance Modeling

This task is covered in a report prepared by sub-awardee NASA/JPL which is attached as Appendix I. In summary, it is proposed that, the reliability, maintainability, and efficiency of large tidal energy systems can be significantly improved by using hydraulic energy transfer designs. All submerged electronics and gears being replaced by high efficiency COTS, radial piston pumps which pump environmentally friendly, water-miscible polyethylene glycol (HEPG) to onshore hydraulic generators. The projected pump performance is shown in Figure 25

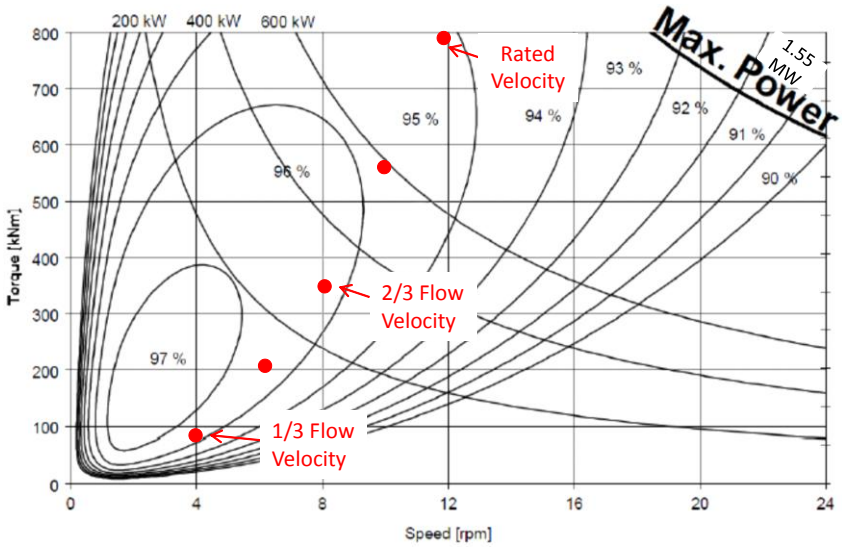

Figure 25. Projected performance of MB240 (1MW) Radial Hydraulic Pump

By multiplexing an array of on-shore hydraulic generators and selectively disengaging them during slow tidal conditions, it will be possible to maintain a nearly constant generator rpm with a high-efficiency power output that requires little power conditioning. Total tidal energy fractional efficiency actually increases from about 0.81 to about 0.86 when rated velocities decrease to 1/3, while conventional tidal efficiencies decrease to zero at 1/3 flow speeds (see Figure 1).

The total cost for hydraulic tidal power production is estimated to be $0.15/kW-hr, which is currently somewhat higher than wind power but lower than PV solar. However, these costs do not include, the added advantage of MHK generation at peak demand times. Conventional and cross-flow turbines MHK systems were considered. The estimated cost to modify cross-flow turbines to a HET system is approximately $0.50/Watt. This will be offset by the savings from the elimination of expensive multi-pole generators.

It is also pointed out that similar gearless hydraulic energy transfer designs can be used to harness a broad range of tidal energy, ocean current energy, river current energy, offshore wind energy, onshore wind energy, and ocean wave energy.

G. Project Accomplishments

With the support of the DOE Wind & Water Program under this FOA we have been able to accomplish the primary goals of this project.

1. We have in-place, an exceptional team with the combined experience and capabilities to take our MHK technology, rapidly through to commercialization and ocean deployment

2. We designed, integrated and evaluated a TRL4 level system in the laboratory which was operated in a simulated range between 1-3 m/s tidal flow and 2-18kW power. The system demonstrated the viability of a HET approach for a MHK turbine system based on COTS components, which eliminates high ratio gearing and submerged electronics, both common points of failure in conventional MHK turbine systems

3. System design enhancements were identified from project observations, including the use of multiplexed on-shore hydraulic motors and generators to extend the high efficiency range of operation to lower powers.

4. We identified two paths forward towards commercialization. One is a fully configured 15-25kW TRL5/6 turbine system which can be Barge tested at TEDEC in Maine. The second track is to modify an Atlantis Resources Solon platform to include a HET system. This has the advantage of accelerating progress towards a grid connected TRL8 demonstration using existing equipment and test facilities, with COTS HET components. Both paths can be expedited in a cost effective manner.

5. We modeled and costed a large scale (15MW) system suitable for implementation in conventional and cross flow turbines. Projected performance is exceptionally high over a much wider range flow conditions than conventional systems. The LCOE is currently in-line with other renewable resources and the approach has the added advantage of potentially providing most power at peak demand times.

6. We identified that this HET technology can be implemented in other tidal and wind power systems. This greatly expands the harvestable resource and the ability to address the national needs.

7. We have reported the results of this project at two international forums [13,14]

H. Glossary

COTS: Commercial-off-the- shelf

HET: Hydraulic Energy Transfer

MHK: Marine Hydro-Kinetic

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Appendix I: TRL 3 Task, Final Report

Hydraulic Tidal and Wind Power System Sizing

JPL Publication 12-1

JPL Publication 12-10

Hydraulic Tidal and Wind Power System Sizing

Final Report

Jack A. Jones

Prepared for

U.S. Department of the Energy

Through an agreement with

National Aeronautics and Space Administration

by

Jet Propulsion Laboratory
California Institute of Technology
Pasadena, California

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This research was prepared at the Jet Propulsion Laboratory, California Institute of Technology, for the U.S. Department of Energy through an agreement with the National Aeronautics and Space Administration.

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JPL is grateful to the Department of Energy for funding this theoretical and experimental development for hydraulic energy transfer of tidal energy. The theoretical part of this effort was carried out at the Jet Propulsion Laboratory, California Institute of Technology. The experimental portion of this task will be submitted under separate cover by Allan Bruce of Sunlight Photonics.

The cost information contained in this document is of a budgetary and planning nature and is intended for informational purposes only. It does not constitute a commitment on the part of JPL and/or Caltech.

Table of Contents

1 Executive Summary

Tidal energy, offshore wind energy, and onshore wind energy can be converted to electricity at a central ground location by converting their respective energies into high-pressure hydraulic flows that are transmitted to a system of hydraulic generators by high-pressure pipelines. The high-pressure flows are then efficiently converted to electricity by a central hydraulic power plant, and the low-pressure outlet flow is returned. All gears and submerged electronics are completely eliminated (JPL/Caltech patents granted and pending). The Department of Energy (DOE) is presently supporting a project led by Sunlight Photonics to demonstrate a 15 kW tidal hydraulic power generation system in the laboratory. Sunlight Photonics will issue a separate report on this experimental phase, which has successfully integrated and demonstrated all major hardware components.

Another portion of this DOE project involves sizing and costing a 15 MW commercial tidal energy plant, which is the subject of this Final Report. For this task, Atlantis Resources Corporation's demonstrated 18-m diameter tidal blades operate in a nominal 2.6 m/sec tidal flow to produce one MW per set of tidal blades. Fifteen blade units are submerged in a deep tidal area, such as Maine's Western Passage. Each set of blades is attached to commercial-off-the-shelf (COTS) Hagglund radial piston pumps, and all pumps are connected to a high-pressure (20 MPa, 2900 psi) line that is 35 cm ID. High-pressure HEPG fluid is transported 500 meters to a parallel series of onshore, COTS axial piston hydraulic generators. HEPG is an environmentally-friendly, biodegradable, water-miscible fluid. The total cost of producing energy with this tidal power plant is estimated to be $0.15/kW-hr, which is between the cost of wind energy and solar energy.

Hydraulic adaptations to Ocean Renewable Power Company's (ORPC's) cross-flow tidal turbines are also discussed. Costs to convert a submerged ORPC tidal system to a hydraulic device with onshore power generation are about 50 cents per watt, minus the cost of ORPC's expensive submerged generators, which would be entirely removed.

Although not originally planned, applications of Hydraulic Energy Transfer (HET) for wind energy have also been added to this report. For wind energy that is onshore or offshore, a gearless, high-efficiency, COTS, radial piston pump can replace each set of troublesome, top-mounted gear-generators for conventional wind turbine systems. Environmentally friendly HEPG fluid is then pumped to a central system of easily serviceable ground generators, which consist of a parallel series of axial piston hydraulic generators. Total hydraulic/electrical efficiency of 81% is close to that of conventional wind turbines at full-rated wind speeds. Total HET efficiencies increase at slower speeds, however, while conventional wind turbine efficiencies decrease significantly. In addition, all troublesome gears are eliminated for HET wind and tidal energy systems.

2 Introduction: State of the Art for Tidal and Wind Energy

There are numerous ways to obtain non-carbon-emitting, renewable electrical power. One of the objectives of this paper is to briefly review some of the state-of-the-art for tidal energy and wind energy systems. A new hydraulic energy transfer (HET) design will be discussed that allows centralized, ground-based power generation for onshore and offshore wind energy, as well as for tidal and river current energy.

2.1 Tidal Energy

There are many different types of tidal power technologies. A partial list of categories includes the following (Reference 1).

Barrage or dam: A barrage or dam is typically used to convert tidal energy into electricity by forcing the water through turbines, activating a generator. Gates and turbines are installed along the dam. When the tides produce adequate difference in the level of water on opposite sides of the dam, the gates are opened. The water then flows through the turbines. The turbines turn an electric generator to produce electricity. Small power plants using this technology are now functioning in France, Russia, and Canada. The dams have been criticized, however, for resulting in the accumulation of silt and other material behind the dams.

Tidal Fence: Tidal fences look like giant turnstiles. They can reach across channels between small islands or across straits between the mainland and island. The turnstiles spin via tidal currents typical of coastal waters. Some of these currents are 5-8 knots and generate as much energy as winds of much larger velocity. Tidal fences also impede boat traffic, as well as sea life migration.

Horizontal Axis Tidal Turbines: There are many types of horizontal axis tidal turbines. The most common of these tidal turbines look like wind turbines. They are arrayed underwater in rows, as in some wind farms. The turbines functions best where coastal currents run at between 3.6 and 4.9 knots: In currents of that speed, a 15-m diameter tidal turbine can generate as much energy as a 60-m diameter wind turbine. Ideal locations for tidal turbine farms are close to shore in water depths of 20 to 30 meters. This type of tidal turbine generally does not impede sea life migration or result in slit buildup.

The first tidal generator actually attached to a commercial grid in the United States was operated by Verdant Power in New York City's East River. Verdant's Roosevelt Island Tidal Energy (RITE) Project was initiated in 2002 and was operating on-grid intermittently until 2009. The project consisted of six 35-kW horizontal axis turbines that were fully bidirectional and accumulated over 7000 hours of operation. A simple operational schematic of a horizontal blade tidal turbine system is shown in Figure 1. The tidal flow turns a blade at about 15 rpm, which is increased to about 1500 rpm by means of a gearbox. The higher rpm is then used to generate electricity by means of a submerged generator, and the energy is sent to shore with submerged power lines. The project was plagued by a number of problems, including blades breaking off and salt water leakage into the generators. Reinforced turbines

were installed in September 2008 (Reference 2), but they all eventually failed due to salt-water leakage into the submerged generators. Verdant has recently received a license to attempt to reinstall 30 generators by 2015.

Atlantis Resources Corporation, which is a partner in this DOE-sponsored task, has an 18-m diameter tri-blade system (Figure 2) called AR-1000. It is a powerful and efficient single-rotor turbine expressly designed for offshore ocean use. The AR-1000 combines a fixed pitch blade operation, a single stage gearbox, and a flexible coupling to the highly efficient permanent magnet generator with a rating of 1.0MW at 2.65m/s. The complete tidal turbine system is fully UK-grid-compliant, employing features developed over the past 10 years.

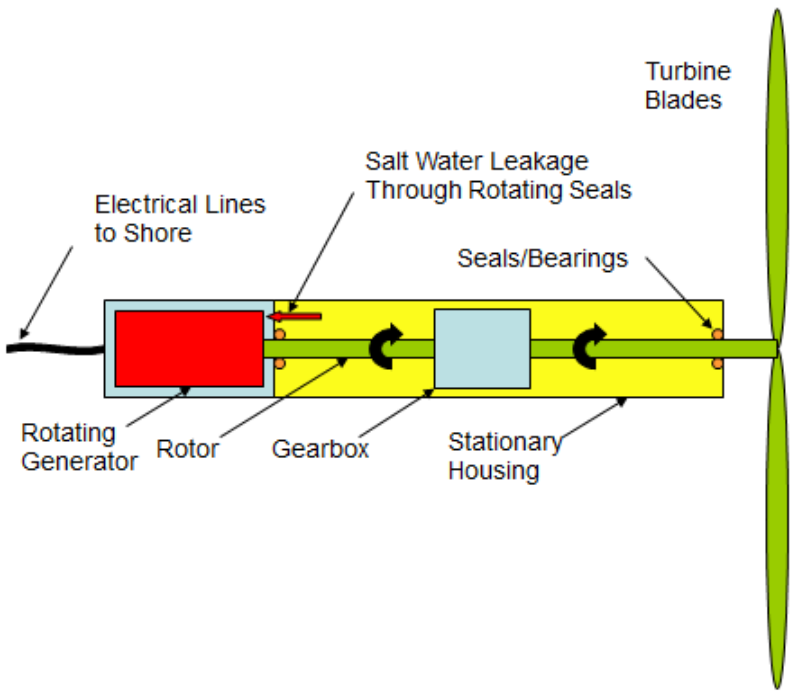


Figure 1. Typical Horizontal Axis Submerged Tidal Turbine

Figure 2. Atlantis Resource Corporation's 18-m Tidal Turbine

The AR-1000 was first connected and generated to grid in June 2011. This was achieved at the EMEC test facility using the same existing gravity-based structure and connection platform developed for the revision generation AK-1000 turbine deployed in 2010. Testing has proved that the water-to-wire efficiency of the device is in excess of 42%, as predicted by theoretical modeling. Testing of the system to refine the control system and improve the overall system reliability is still ongoing.

There are numerous other versions of horizontal axis turbines, including Marine Current Turbines, which use counter-rotating blades on a common tower (Figure 3) This design became the world's first operational in-stream tidal turbine in 2008, generating 1.2 MW on Strangford Lough in Northern Ireland.

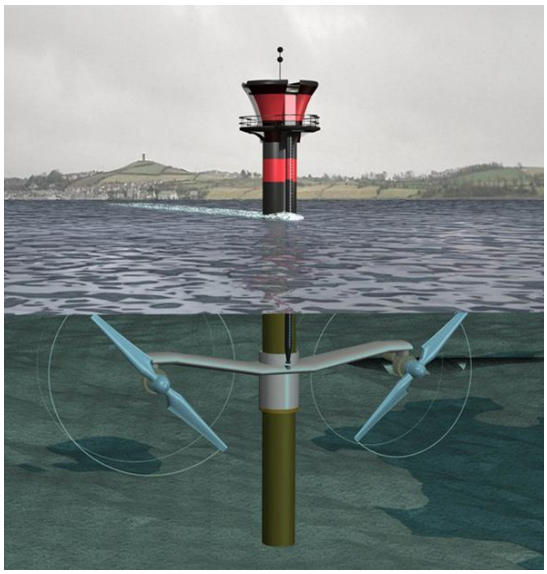


Figure 3. Marine Current Turbine Arrays

Ocean Renewable Power Company (ORPC) produces an alternative type of in-stream tidal generation system, which uses unique cross-flow turbines that drive a permanent magnet generator on a single shaft (Figure 4). These units can be stacked vertically and horizontally into much larger units. A 4x4 unit can produce about 763 kW in a 2.6m/sec tidal flow. The units use a permanent magnet generator and do not require a gearbox. Also, they turn in the same direction regardless of incoming or outgoing tide.

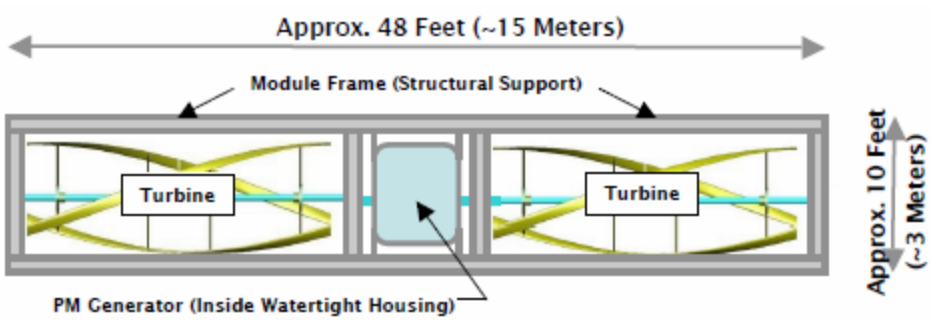


Figure 4. ORPC Cross-Flow Tidal Generator

2.2 Conventional Wind Turbine Systems

Wind turbines have been used to generate electricity since the late 19th century, although the modern wind power industry did not begin until about 1979, when several European countries began commercially producing small wind turbines. Worldwide, wind power now has the capacity to generate 430 TWh annually, which is about 2.5% of worldwide electricity usage (Reference 3-4). In terms of potential wind generating power, as stated by DOE, offshore wind energy could potentially supply 4,000 TWh/yr in the US alone, and onshore wind energy could potentially supply an additional 37,000 TWh/yr (Reference 5). This total potential U.S. wind energy production is about ten times more than the entire 2010 U.S. electricity demand of about 4,000 TWh/yr (Reference 6). Wind energy costs are significantly lower than natural gas, solar power, or coal with carbon sequestration (Reference 7).

The operating theory behind most present wind turbines is shown by the schematic drawing in Figure 5. Typically, three blades are used to harness the wind and transfer their slow-revolving torque to a gearbox. The gearbox increases the rpm to about 1800 rpm, and the attached alternator then generates electricity. At high wind speeds, total efficiency is close to about 80% of theoretical values, but this drops greatly at slower wind speeds. Not only does the total available power drop off according to the cube of velocity, but the relative efficiency of the gearbox and alternator greatly decrease with wind speed.

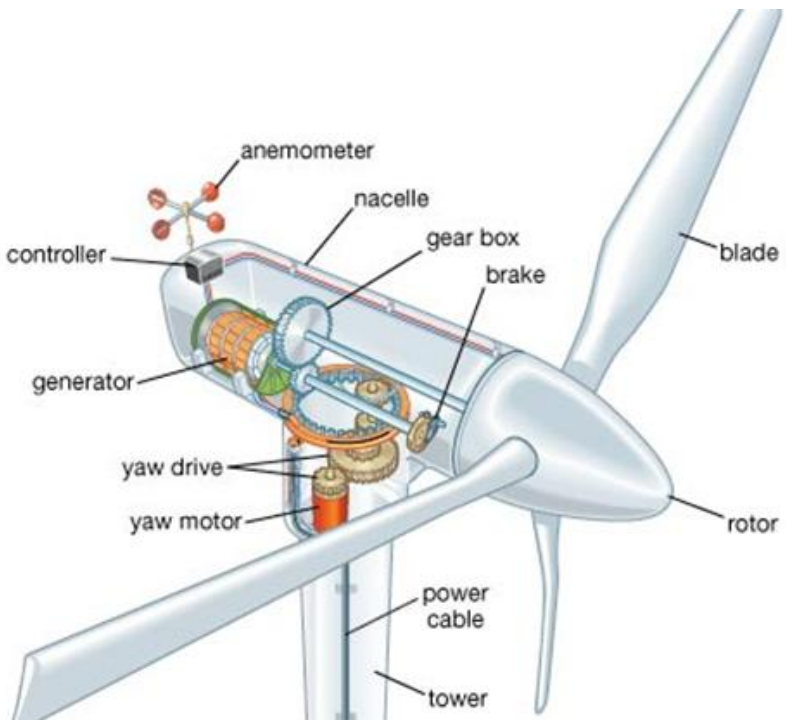


Figure 5. Typical Wind Turbine Components

The gearboxes are the primary failure mode for wind turbines, and servicing the generator or gearbox at the top of the wind mast is their primary maintenance cost. Three European companies (Chap Drive/**Norway**, Artemis/**Scotland**, and Voith Turbo WinDrive/**Germany**) have replaced the top-mounted wind turbine gears and generator with a top-mount hydraulic

pump. The hydraulic pump then sends oil to a hydraulic generator at the bottom of the mast. The two big advantages to this type of design are that the gearbox is completely removed, and maintenance on the ground-level generator is greatly facilitated.

Delft University in the Netherlands has taken hydraulic wind power generation one step further for offshore wind power generation. They use a hydraulic oil loop on the mast to power a seawater pump that sends high-pressure seawater to a generator on a remote platform (Figure 6). Electricity from hydroelectric generators is then transferred to shore by means of a buried cable (Reference 8). Total electromechanical efficiency is about 80%.

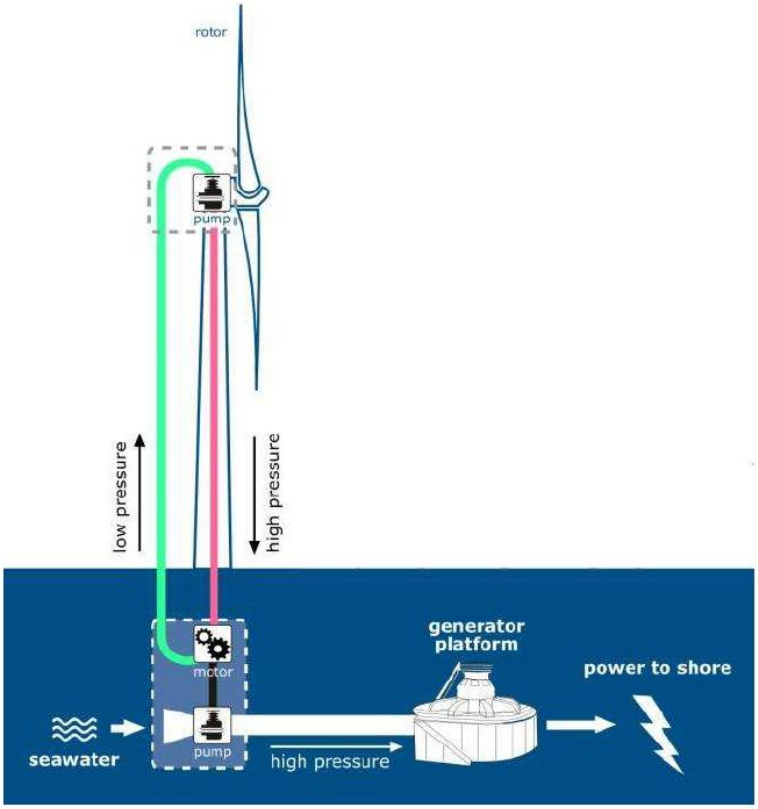
2. 

Figure 6. Delft University Offshore Wind Hydraulic Energy Concept

2.3 Direct Drive Train (DDT) Wind Turbine

A direct drive train is one that takes the power coming from a motor without any reductions (such as a gearbox). This type of system has been used occasionally in offshore wind turbines, where the expense of repairing a gearbox is more expensive than the gearbox itself. Although DDT offers increased efficiency, reduced noise, and longer lifetimes, all this comes at an expensive price. The main disadvantage of the system is that it needs a special, very expensive motor. The slow, high-torque motor needs to be physically much larger than its faster counterpart, and all this mass must be supported at the top of the turbine tower. Finally, direct-drive mechanisms need a more precise control mechanism. Low-voltage variations on a high-speed motor that are reduced to low rpms can go unnoticed, but in a direct-drive, those variations are directly reflected on the rotational speed.

3 Tidal and Wind Hydraulic Energy Transfer (HET) Designs

Unfortunately, none of the European hydraulic wind systems noted above in Section 1.2 have excelled, primarily because the pumps and/or hydraulic generators are complicated, custom machines, and each tower has only one ground generator. In addition, the European HET designs lose a potentially valuable means to increase efficiency, as described below.

JPL/Caltech has recently patented a new means to generate power for tidal energy and wind energy systems (Ref 10, 13) utilizing wind or tides to power a series of ***off-the-shelf radial piston pumps*** (typical efficiency ~0.95), which send a bio-friendly fluid to a series of ***off-the-shelf, high efficiency, axial piston hydraulic generators***. As the wind speed decreases, the pump efficiency increases, the pressure drop decreases, and the generator performance can be maintained at an optimum rpm by shutting off some of the generators. Other wind generators, both conventional and European hydraulic systems, suffer large losses as the rpms decrease. Wind energy and tidal energy can both be used to turn pumps instead of generators, and the pumped fluid can be transferred remotely to generate electricity. There are numerous advantages to using HET technology which will be explained in the following sections for both tidal energy and wind energy.

3.1 Tidal Hydraulic Energy Transfer

The conventional in-stream tidal turbine shown in Figure 1 is somewhat similar to the wind turbine shown in Figure 5: Turbine blades spin slowly due to the flow of a river, tidal flow, or ocean current. The rotor's rotational speed is increased through a gearbox, which then drives a turbine generator. Each turbine's output is then conditioned and transferred to shore by means of a buried electrical cable. As mentioned in Section 1.1, this submerged electrical design is subject to salt water corrosion of electrical components due to all-too-common leakage of salt-water through its rotating seals. Furthermore, the submerged cable and power conditioning are both expensive and dangerous, and the gears are subject to failure.

The JPL/Caltech HET design for this type of horizontal axis in-stream tidal turbine is shown in Figure 7, with blades and pump rotated 90° for simplicity. For this design (Reference 1), turbine blades spin slowly due to water currents, like the other systems. The rotor's rotational speed is transmitted directly to a commercially available, high-pressure fluid pump, without using any gears. The high-pressure fluid, such as environmentally friendly polyethylene glycol-based synthetic hydraulic fluids (HEPG), is transported in small flexible lines to a shared stainless steel pipe and then to an efficient, onshore hydroelectric power plant. This all-mechanical design is less expensive (Reference 9), more efficient, and eliminates all gears and all submerged electrical component corrosion. A 500-m long, 0.35-m inside diameter pipe at 200 bar (2900 psi) can efficiently deliver 15 MW of hydraulic power to shore.

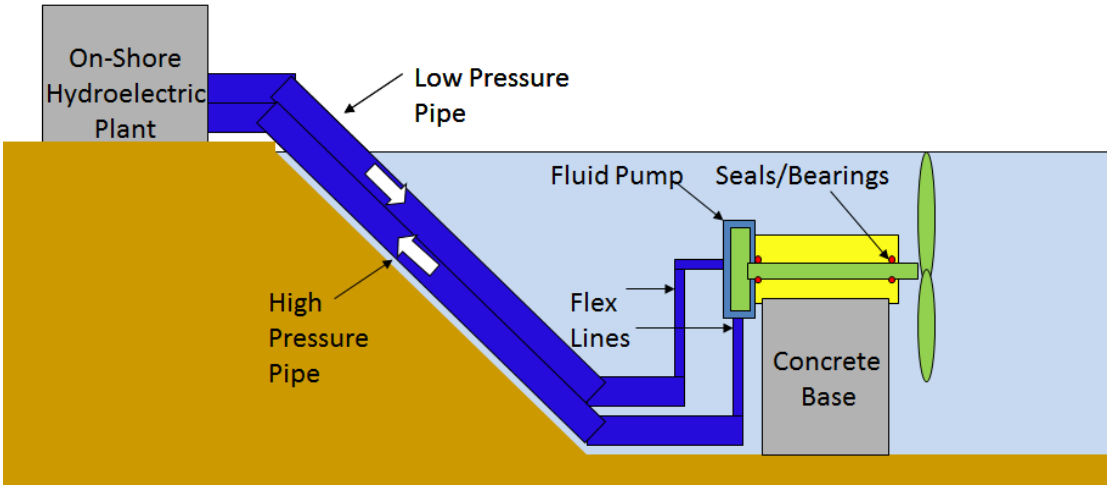


Figure 7. JPL/Caltech Hydraulic Tidal Concept

A series of HET tidal blade units can be combined such that they are all attached to a common high-pressure line that delivers high-pressure fluid to an onshore power plant (Figure 8). A separate line can then deliver low-pressure fluid back to each of the HET tidal blade units.

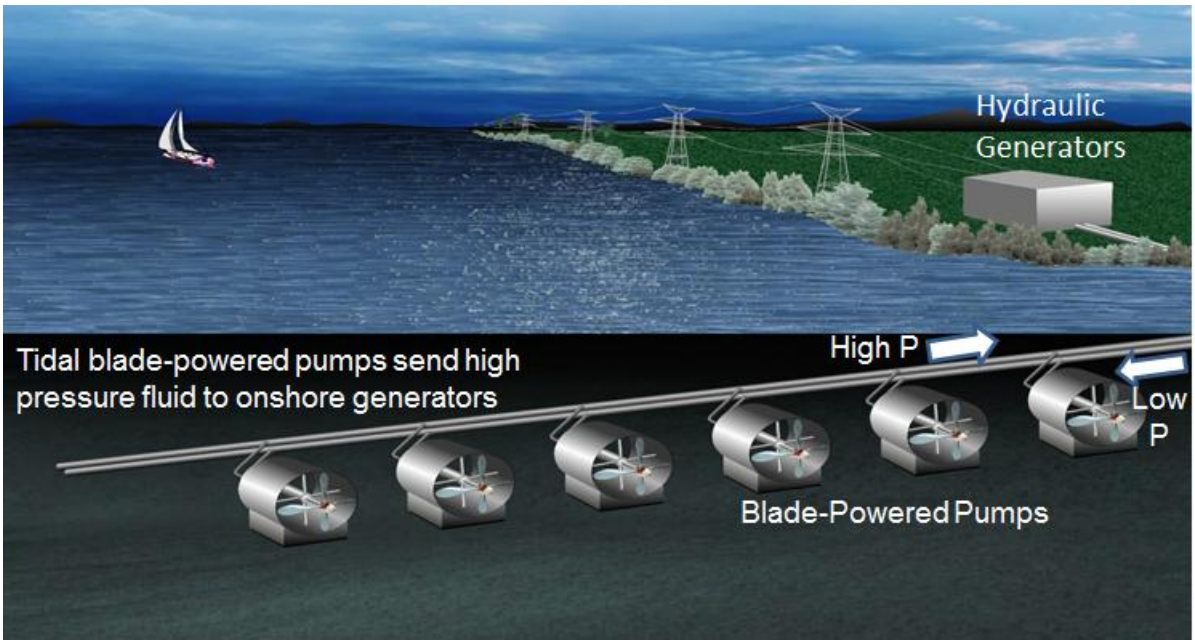


Figure 8. JPL/Caltech Hydraulic Energy Transfer Schematic

For this DOE funded project, a preliminary design has been made for a 15-MW hydraulic tidal power system in Maine's Western Passage tidal area. Both hydraulic systems in Figures 7 and 8 eliminate all gears and submerged electronics, thus greatly improving tidal energy reliability. Caltech has recently been granted patent rights to use this closed cycle hydraulic transfer design for tides, ocean currents, ocean waves, offshore wind, and onshore wind (Reference 10). In a parallel task, the Department of Energy is also funding Sunlight Photonics (FY'11–'12) to design, integrate, and test a 15-kW hydraulic energy transfer tidal

energy system that pumps environmentally friendly, biodegradable fluid to a remote hydroelectric generator (Reference 15). Both the experimental 15-kW system and the analytical design of the 15 MW systems are described later in this report.

3.2 Wind Hydraulic Energy Transfer

JPL/Caltech has taken the wind hydraulic energy transfer systems described in Section 1.2 an additional step further. For both offshore and onshore wind energy, the top-mounted pump is used to pump environmentally friendly hydraulic fluid from a series of wind blades directly to a series of generators that are remotely located. For the case of offshore wind, the generators can be located onshore or on a platform offshore (Figure 9).

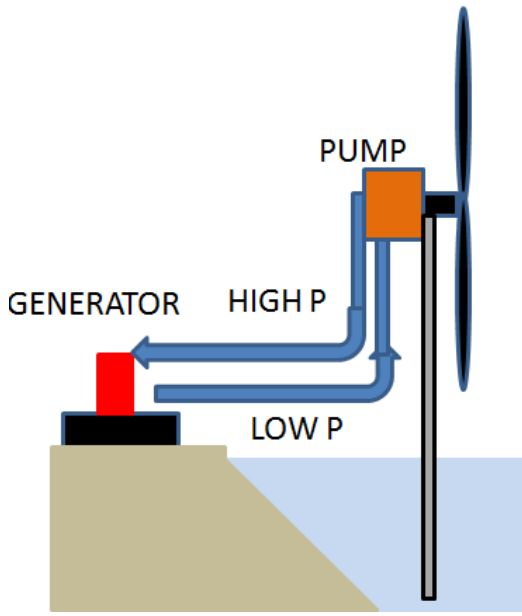


Figure 9. JPL/Caltech Offshore Wind with Hydraulic Energy Transfer

For onshore wind generation, the pumps can connect to a common high-pressure line that goes to a series of remotely located generators. The returning low-pressure fluid then joins a common line to return to the blade pumps. This type of design not only eliminates all troublesome gears, it allows all ground-based generators to be in a common location. As the wind decreases, various hydraulic generators can be shut down to produce a nearly constant high rpm. Also, if there is a failure of one or more generators, they can easily be taken off-line (Figure 10). For offshore wind or for tidal energy, the oil cooling unit can be eliminated, since the pipes themselves will be cooled.

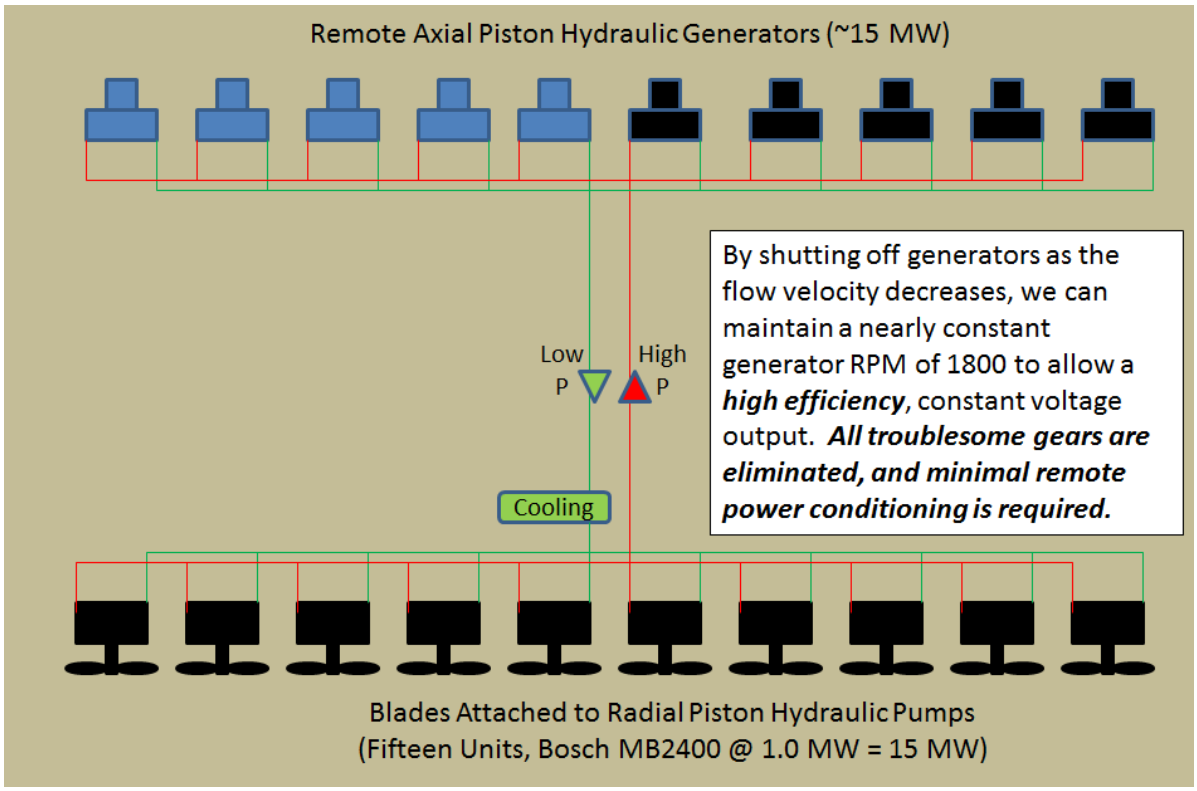


Figure 10. Wind or Tidal Energy Hydraulic Flow Schematic

4 Sizing Full Scale Tidal and Wind Hydraulic Transfer Systems

4.1 15 MW Hydraulic Tidal System

Preliminary sizing has been performed for a 15-MW tidal energy plant that would be located in Maine's Western Passage from Dog Island to Deer Island Point (Figure 11). We would plan to take 15% of available power, or 15 MW, from the tidal stream in the Western Passage. The blade units we propose to use are the 18-m diameter tidal blades (Figure 2) that have been demonstrated by Atlantis Resources Corporation off the coast of Scotland. Each set of blades are capable of generating 1 MW in a tidal flow of 2.6m/sec, which is the mean maximum tidal flow in the Western Passage. We would use 15 sets of blades that are gravity-placed on the channel floor by means of reinforced concrete foundations. The depth of placement would be at least 30 meters, thus allowing 10 meters of navigable channel above the blades at low tide.

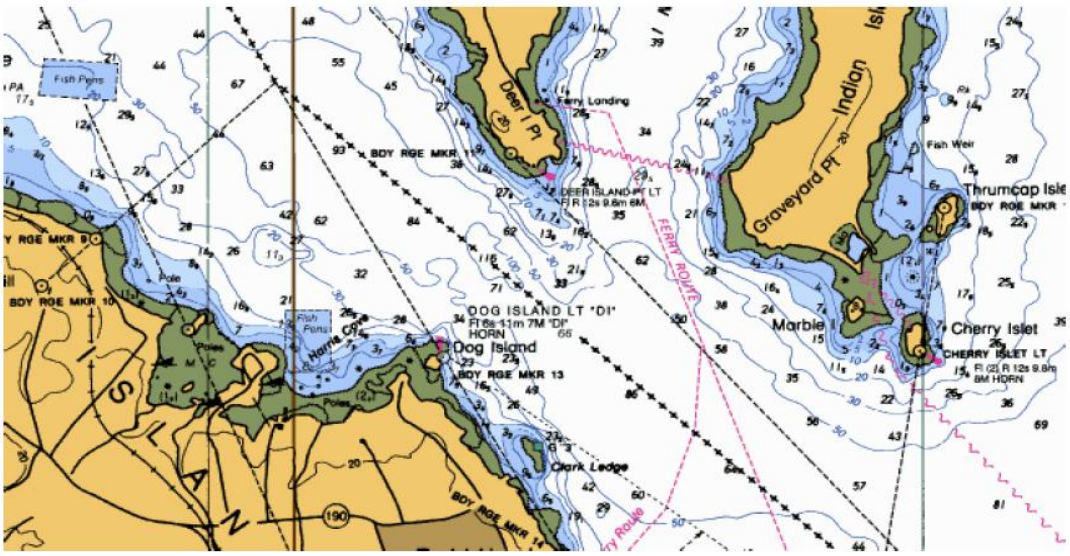


Figure 11. Maine's Western Passage Tidal Area

The corresponding rpm of the Atlantis blades at a tidal flow of 2.6 m/sec is approximately 12 rpm. The rpm and torque corresponds very well with Hagglund's #MB2400-2400 radial piston pump. This pump has no gear reduction, thus avoiding major energy loss and maintenance problems. The MB2400 has a maximum allowable speed of 24 rpm and maximum power output of about 1.55 MW, although these conditions are only meant for short transient operation. There are a variety of onshore hydraulic motors and generators than can generate 1800 rpm of electricity at a combined efficiency of about 90%.

High- and low-pressure hydraulic lines are both attached to the vertical strut of each blade unit. Flexible high-pressure outlet lines and low-pressure inlet lines connect the Hagglund pumps to the permanent lines (Figure 12). The preferred hydraulic fluid is HEPG (polyethylene glycol), which is a non-toxic, environmentally friendly, biodegradable oil that is used as a food additive and is fully miscible with water. Polyethylene glycol has been

shown to have long-term, low toxicity to aquatic organisms with amounts below about 1% (Reference 11).

The average ID of the high-pressure (3000 psi or 207 bar) stainless steel pipe is 35 cm and the average ID of the low-pressure (150 psi or 10.3 bar) reinforced fiberglass pipe is 40 cm. The pipe diameters would be somewhat larger near shore and smaller further away from shore to account for the varying amount of HEPG flow that is carried in the pipes. Total pressure drop in the high-pressure pipe is eight bars, and two bars in the low-pressure pipe, or 5% loss total of the entire flow for the 500-m × 2 roundtrip length.

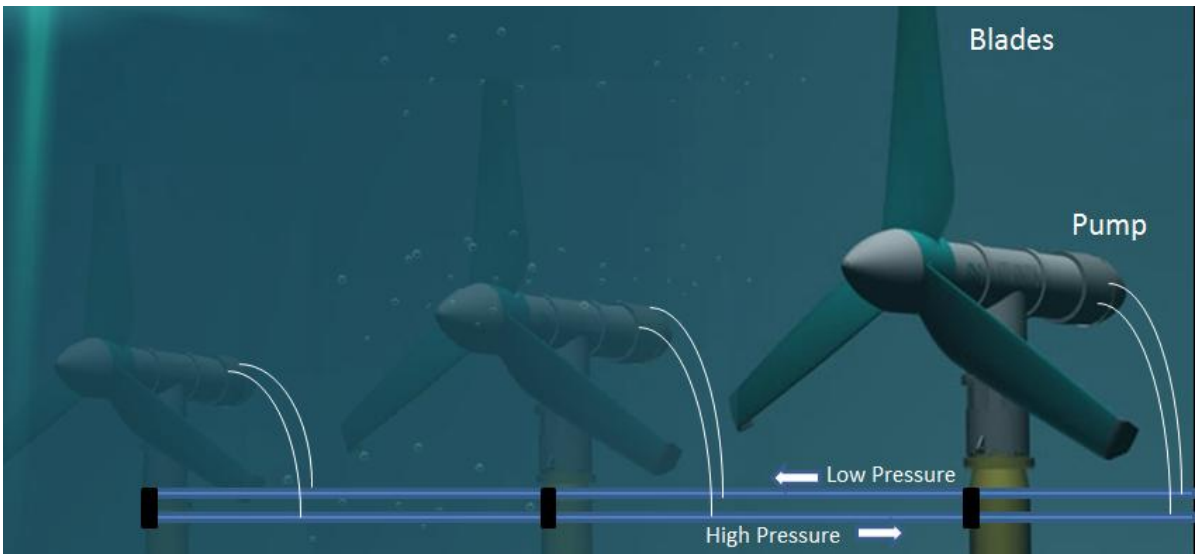


Figure 12. Atlantis Resource 18-m Blades with Hydraulic Energy Transfer

Recent costing of various energy sources for plants entering service in 2016 has been performed by the Energy Information Administration for the Department of Energy (Table 1, Reference 14). Some total system levelized costs include $0.063/kW-hr ($63/MW-hr) for advanced cycle natural gas plants, $0.095 for conventional coal, $0.140 for coal with carbon sequestration, $0.097 for conventional wind, and $0.211 for solar photovoltaic.

Table 1. Estimated Levelized Cost of New Generation Resources, 2016.

Plant Type	Capacity Factor (%)	U.S. Average Levelized Costs (2009 $/megawatthour) for Plants Entering Service in 2016				
		Levelized Capital Cost	Fixed O&M	Variable O&M (including fuel)	Transmission Investment	Total System Levelized Cost
Conventional Coal	85	65.3	3.9	24.3	1.2	94.8
Advanced Coal	85	74.6	7.9	25.7	1.2	109.4
Advanced Coal with CCS	85	92.7	9.2	33.1	1.2	136.2
Natural Gas-fired						
Conventional Combined Cycle	87	17.5	1.9	45.6	1.2	66.1
Advanced Combined Cycle	87	17.9	1.9	42.1	1.2	63.1
Advanced CC with CCS	87	34.6	3.9	49.6	1.2	89.3
Conventional Combustion Turbine	30	45.8	3.7	71.5	3.5	124.5
Advanced Combustion Turbine	30	31.6	5.5	62.9	3.5	103.5
Advanced Nuclear	90	90.1	11.1	11.7	1.0	113.9
Wind	34	83.9	9.6	0.0	3.5	97.0
Wind – Offshore	34	209.3	28.1	0.0	5.9	243.2
Solar PV[1]	25	194.6	12.1	0.0	4.0	210.7
Solar Thermal	18	259.4	46.6	0.0	5.8	311.8
Geothermal	92	79.3	11.9	9.5	1.0	101.7
Biomass	83	55.3	13.7	42.3	1.3	112.5
Hydro	52	74.5	3.8	6.3	1.9	86.4

[1] Costs are expressed in terms of net AC power available to the grid for the installed capacity.

Source: Energy Information Administration, Annual Energy Outlook 2011, December 2010, DOE/EIA-0383(2010)

3.

Total system levelized costs have also been estimated for the HET tidal energy approach (Table 2). All marine hardware and installation costs have been estimated by Atlantis Research Corp., based on actual fabrication and servicing costs for the Atlantis Research 1-MW tidal turbine. Piping and fluid costs assume stainless steel pipes with HEPG fluid. Hydraulic motor and pump costs have been provided by Bosch-Rexroth Incorporated, and generator costs have been provided by Baldor Motors. Using equations at the bottom of Table 2, the 2012 costs for HET tidal energy are estimated to be about $0.150/kW-hr, or about $0.170/kW-hr, based on estimated 2016 costs. This amount is about midway between coal-fired plants with carbon sequestration and energy from solar photovoltaic plants.

Table 2. Cost of Energy (COE) for Hydraulic Tidal Energy

Component	Initial Cost for 15 MW Plant ($K)		5 Yr Maintenance Cost ($K)	
Ocean Components				
18-m Blades/Rotor		$800		$400
Nacelle		$2000		$1000
Ancillary		$480		$240
Pumps	300*15=	$4500		$2250
Fluid		$600		$600
Pre-assembly		$7500		$250
Ocean Installation	800*15=	12000	600*15=	$9000
EPA Approval		$3000		--------
Pipes		$500		$250
Subtotal		$31,380		$13,990
Land Components				
Generator	50*15=	$750		$375
Hyd Motors	50*15=	$750		$375
Assembly		$1000		$300
Power Conditioning		$1000		$500
Grid Connection		$2000		------
Outdoor Gen Housing		$1,000		$200
Subtotal		$6500		$1,750
TOTAL		**$37,880**		**$15,740**

COE = [(DR + IWF) * ICC + LRC + O&M] / AEP
DR = 20-year Discount Rate = 0.07
IWF = Insurance, Warranty, and Fees = 0.01
ICC = Initial Installed Capital Cost = $31,380 K
LRC = Levelized Replacement Costs = 31,380/20 = $1569 K
O&M = Levelized Operations and Maintenance Cost = (15,740/5)*0.6 = $1889 K
AEP = Net Annual Energy Production = 15,000 KW * 0.33 * 8760 hrs * 0.95 = 41.20 MW-hrs
(where Average Capacity Factor = 0.33 times Max Power, Availability = 0.95)
Thus, COE = $0.149/KW-hr = $149/MW-hr

Assumptions:

Major service required every 5 years
Replacement required at 20 years
Atlantis Research Corp. 18-m blades with gravity-based system (15 units @ 1 MW max, 0.33 MW aver)
Bosch pumps and hydraulic motors with Baldor generators
Sinusoidal tides with peak velocity at 5.1 knots (2.6 m/sec)

4.2 Hydraulic Tidal System with ORPC Cross-Flow Blades

An alternative to Atlantis Research Corp.'s 18-m tidal blade design is the ORPC cross-flow turbine system (Figure 4). The ORPC design's turbine rotates in one direction only, regardless of current flow direction, and the generator does not require a gearbox—a major advantage.

In lieu of the ORPC submerged generators, it appears possible to install submerged pumps, similar to those used by the Atlantis Research Corp.'s 18-m blades. This design would still use a 4 × 4 matrix for each tidal generation unit, but the four submerged generators would be replaced with four gearless, Hagglund CM 280 pumps. In all, six 4 × 4 units, one of which is shown in Figure 13, would produce a total of 4 MW peak power.

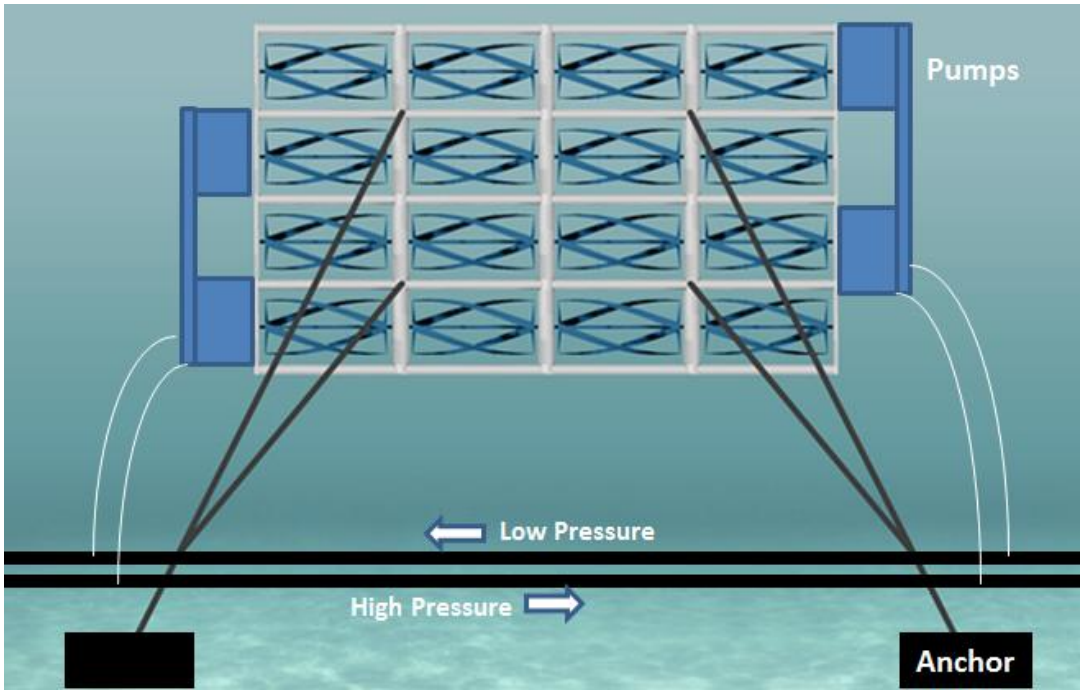


Figure 13. ORPC Cross-flow Blades with Hydraulic Energy Transfer

Total hydraulic conversion cost for the ORPC system has been estimated to be about 50 cents per watt (Table 3), less the cost of the expensive submerged gearless generators, which would be eliminated. Costs assume stainless steel pipes for the high-pressure fluid, FRP pipes for the low-pressure fluid, Bosch-Rexroth axial piston hydraulic motors and radial piston hydraulic pumps, and Baldor generators.

Table 3. Cost of Energy (COE) for ORPC hydraulic Conversion

Component	**Costs ($K) for 4 MW ORPC Hydraulic Conversion**	
Hi Press Steel Pipe (20-cm ID x 300-m)		75
Low Press FRP Pipe (30-cm ID x 300-m)		15
HEPG (10,000 Gal)		200
Hydraulic Motors		300
Generators		200
Radial Piston Pumps	50*24=	1200
TOTAL		**$1990 K**

For the ORPC hydraulic design, the high-pressure stainless steel pipes would average 20 cm ID, and the low-pressure reinforced fiberglass pipes would average 30 cm ID, resulting in a total system pressure drop loss of 5%. The total amount of non-toxic, environmentally friendly, biodegradable polyethylene glycol (HEPG) would be about 10,000 gallons. Since it is fully miscible with water, if the entire quantity of glycol leaked in a single tidal flow, the total mixed content of glycol with seawater would be about 30 parts per **b**illion, assuming complete mixing.

4.3 Preliminary Hydraulic Transfer Wind Energy Sizing

For onshore and offshore wind, a small 15-MW system has been sized, although much larger power systems can be scaled up. We have selected 1.0-MW, 60-m diameter blade sets that rotate at 20 rpm, for a maximum velocity wind of 12 m/sec (27 MPH). Each set of blades is connected to a Bosch-Rexroth/Hagglund radial piston pump (#MB2400-1950).

For this particular example, we assume the generators are located 500 meters away from the wind pump units, and thus an identical 15-MW fluid transfer system as the Atlantis Resource Corp. hydraulic tidal system described in section 3.1. The average ID of the high-pressure (3000 psi or 207 bar) stainless steel pipe is 35 cm and the average ID of the low-pressure (150 psi or 10.3 bar) RFP pipe is 40 cm. total pressure drop is again 5% of the entire flow for the 500-m x 2 roundtrip length. Distances longer than 500 meters would require larger diameters in order to maintain the same 5% total pressure drop loss. These percentage losses are the same as the hydraulic tidal energy system described in Section 4.1, which operates at 12 rpm using a different version of the Hagglund radial piston pump (Hagglund #MB2400-2400).

4.4 Hydraulic Transfer Efficiency for Tidal or Wind Power Systems

The total efficiency of the wind or tidal HET system is as follows for full rated flow velocity:

Total Efficiency = Pump Effic*Pressure Effic*Hydraulic Motor Effic *Generator Effic

~ 0.95 * 0.95 * 0.95 * 0.95

~ 0.814

This total electromechanical efficiency compares well with the Delft University calculation of 80% for an offshore hydraulic wind energy system (Section 2.2, Figure 6)

At 1/3 of the full rated flow speed, the JPL hydraulic pump efficiency increases to about 0.963 (Figure 15) and the pressure drop efficiency increases to at least 0.99. The hydraulic motor and generator efficiency both stay at 0.95 by means of shutting off generators. Thus, total efficiency ***increases*** to about 0.860 (Figure 14). It should be noted that the efficiency curves of the MB2400-1950 for wind turbines are shifted for the somewhat higher rpms and lower torques corresponding to wind turbines. The resulting total efficiency numbers for both hydraulic wind and tidal energy systems are thus very similar and are shown in Figure 14. This is a large improvement over other wind turbine systems, which suffer significantly ***lower*** efficiency at low wind speeds. Conventional wind turbine combined gear/electronic efficiencies (Reference 12), excluding power conditioning, vary from about 0.89 (full rated wind velocity), to about 0.5 at ½ rated velocity, and to zero (1/3 rated velocity).

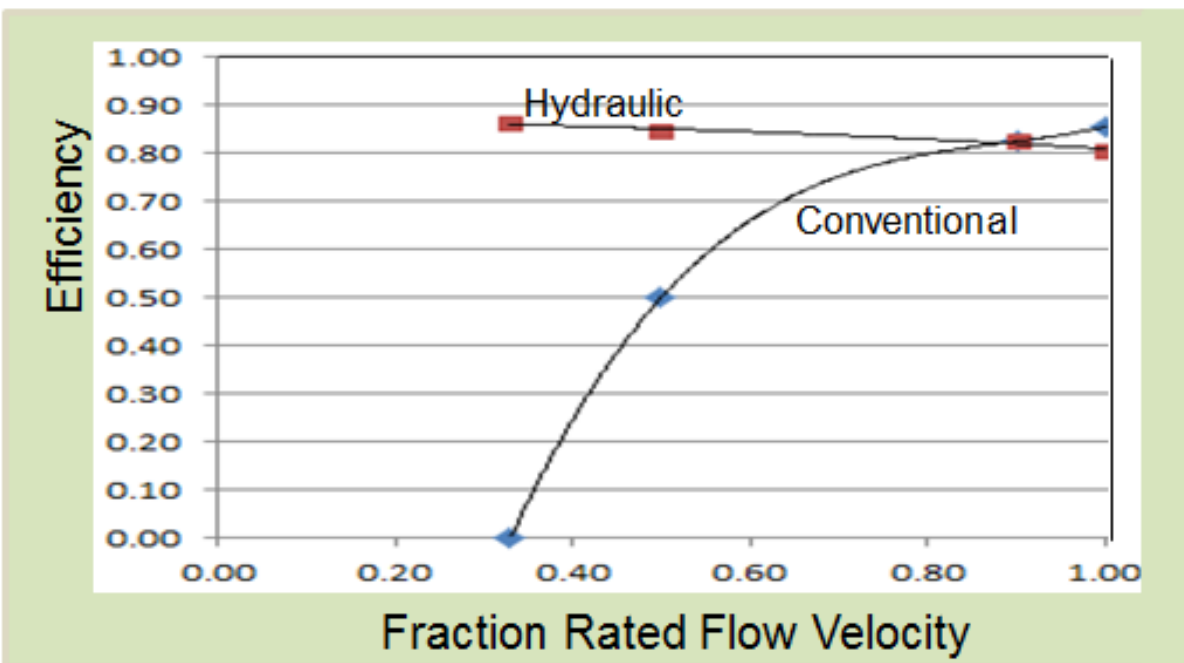


Figure 14. Blade to Grid Total Efficiency for Wind or Tidal Energy

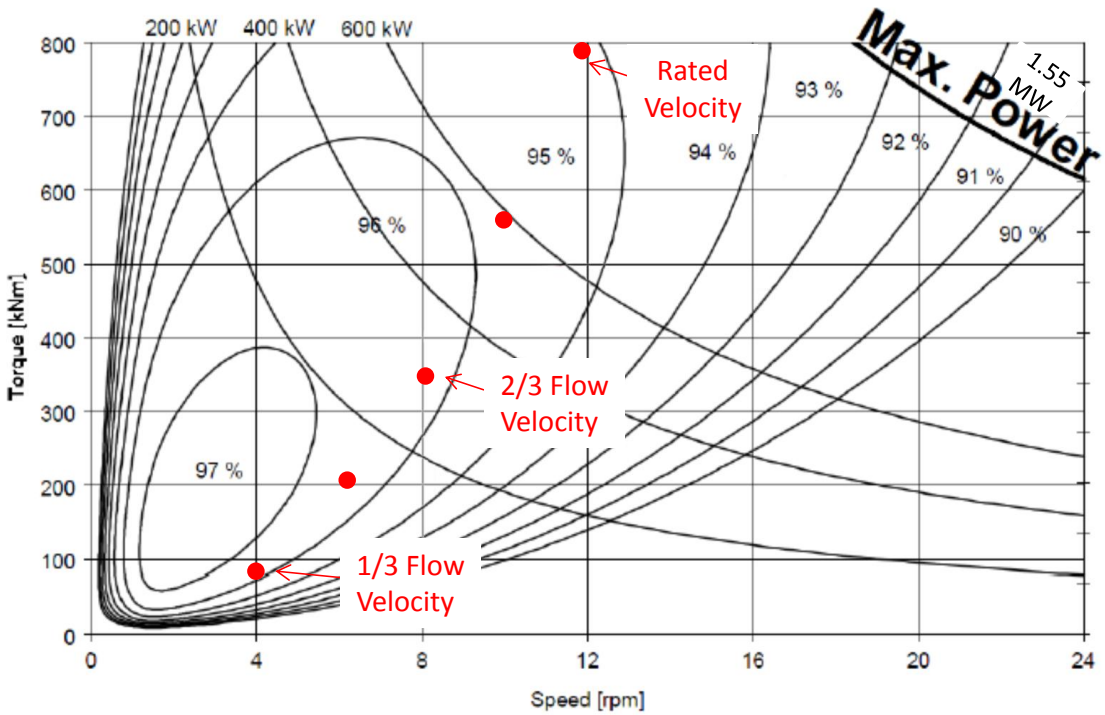


Figure 15. Efficiency Curves for MB 2400 Hydraulic Pump (1 MW)

5 Hydraulic Tidal Simulation Test Set Up

In addition to sizing tidal and wind hydraulic transfer systems, another part of this DOE project is to demonstrate a proof-of-principle hydraulic energy transfer design. As shown in Figure 16, a 15 kW AC motor drives a gearbox to simulate the torque and rpm of a 3-m/sec tidal flow on 2-m diameter tidal blades. The gearbox is then connected to a Hagglund pump, which drives an environmentally friendly fluid to a hydraulic generator. It should be noted that the gearbox is only used to simulate the correct tidal blade torque and rpm, and would thus not be in an actual tidal or wind operating hydraulic transfer system. Sunlight Photonics will issue a separate report on this experimental phase, which has successfully integrated and demonstrated all major hardware components.

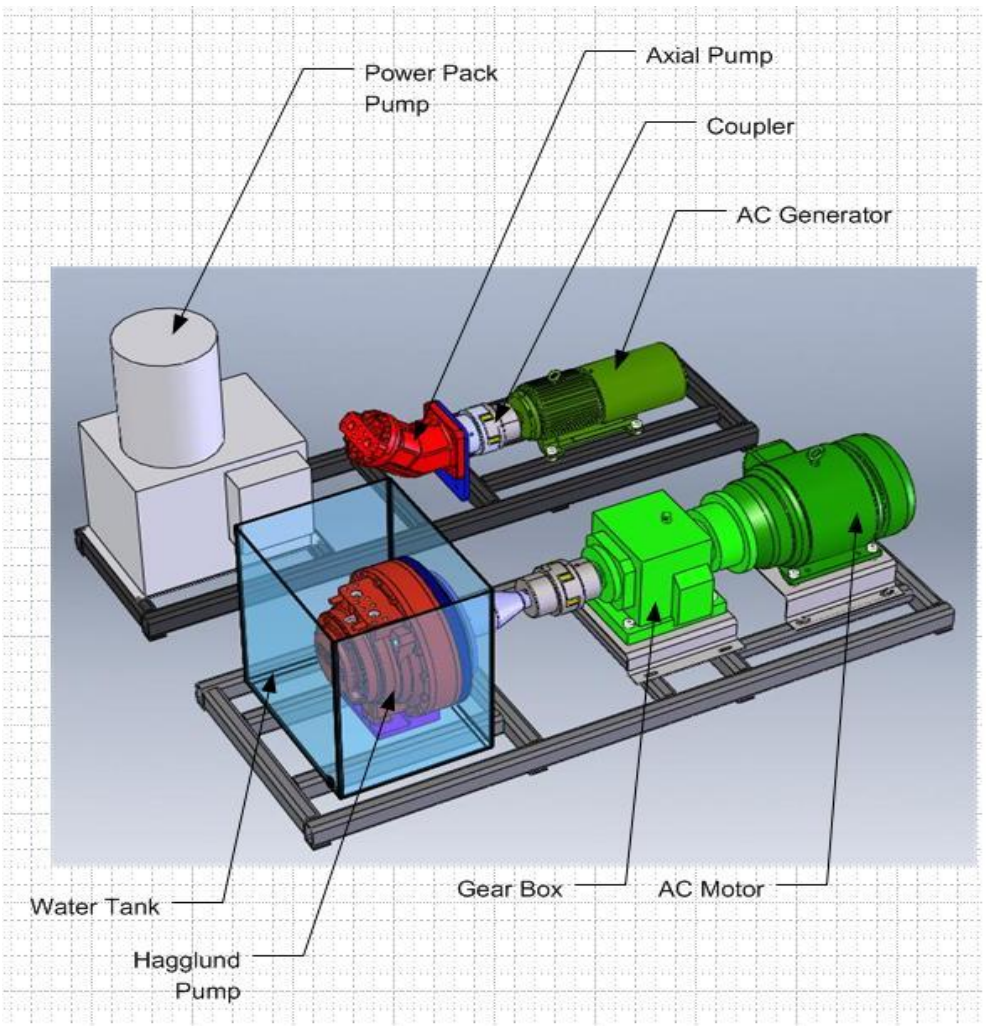


Figure 16. Hydraulic Energy Transfer Test Setup

6 Summary and Conclusions

The reliability, maintainability, and efficiency of wind energy and tidal energy systems can be significantly improved by using hydraulic energy transfer designs. In both instances, all failure-prone gears are eliminated, and the electronics are moved to a convenient, more easily maintained, hydraulic power generating station. For tidal energy, all submerged electronics and gears are replaced by off-the-shelf, radial piston pumps, which pump environmentally friendly, water-miscible polyethylene glycol (HEPG) to onshore hydraulic generators. For wind energy, the complex, top-mounted gears and generators are replaced by off-the-shelf, gearless, radial piston pumps, which pump the same HEPG fluid to a central, ground-located series of hydraulic generators, which are much more easily maintained.

By closing off some of the hydraulic generators during slow tidal or wind conditions, it is possible to maintain a nearly constant generator rpm with a high-efficiency power output that requires very little power conditioning. Total wind or tidal energy fractional efficiency actually increases from about 0.81 to about 0.86 when rated velocities decrease to 1/3, while conventional wind and tidal efficiencies decrease to zero at 1/3 flow speeds. Similar gearless hydraulic energy transfer designs can be used to harness tidal energy, ocean current energy, river current energy, offshore wind energy, onshore wind energy, and ocean wave energy (Reference 10).

Total cost for hydraulic tidal power production has been estimated to be approximately $0.15/kW-hr, which is larger than wind power costs, but less than costs for solar power. Total costs to modify the ORPC cross-flow turbines to a hydraulic energy transfer system are approximately $0.50/watt. The net cost is less after the cost of the expensive multi-pole generators, now used in the ORPC process, is deducted from the total cost of the HET conversion.

7 Acronyms and Abbreviations

COE	cost of energy
COTS	commercial-off-the-shelf
DDT	direct drive train
DOE	Department of Energy
EMEC	European Marine Energy Centre
FRP	Fiberglass reinforced pipe
HEPG	hydraulic polyethylene glycol
HET	hydraulic energy transfer
ID	internal diameter
JPL	Jet Propulsion Laboratory
NASA	National Aeronautics and Space Administration
ORPC	Ocean Renewable Power Company
RITE	Roosevelt Island Tidal Energy
rpm	revolutions per minute

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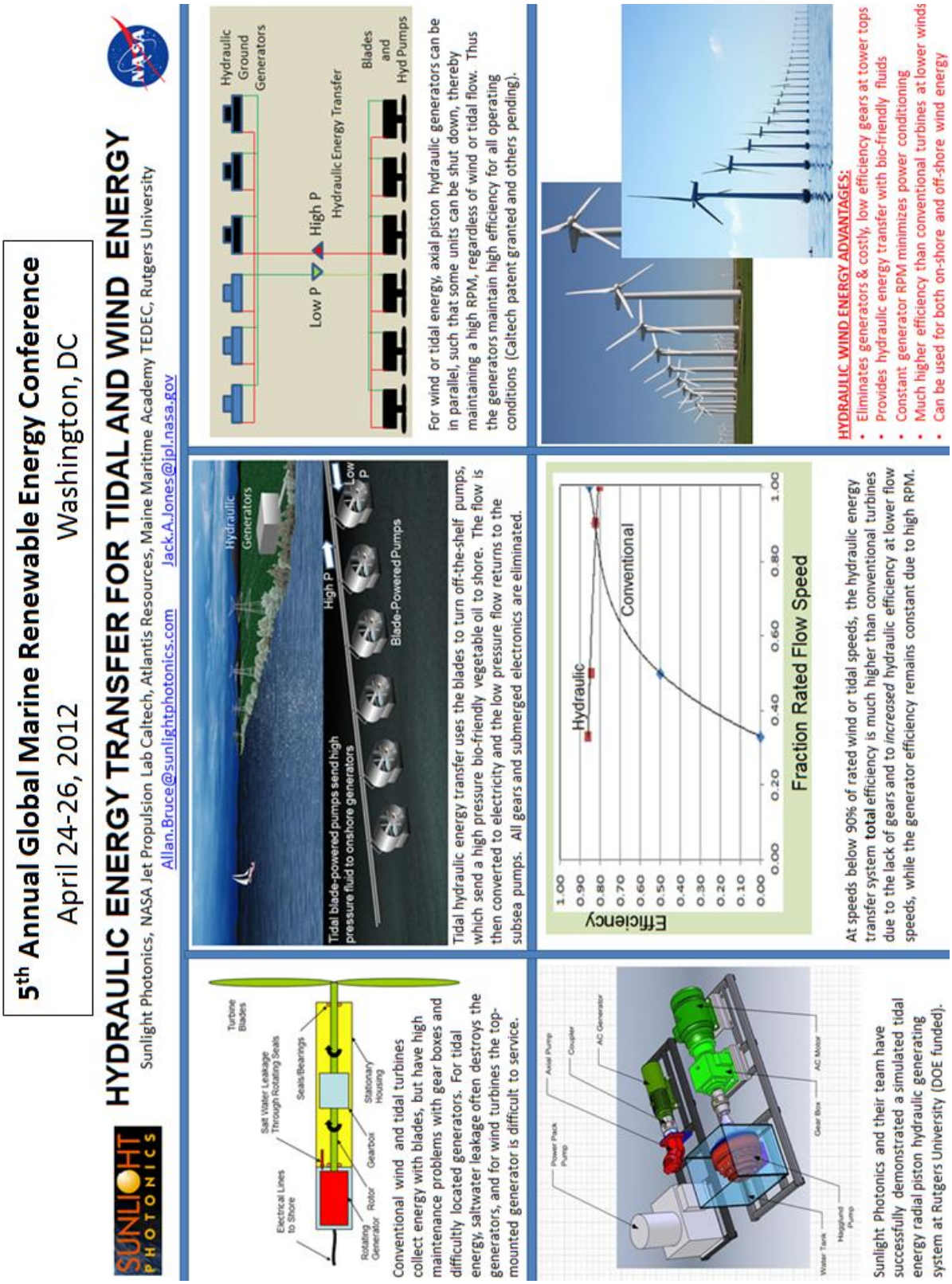
APPENDIX A: GLOBAL MARINE RENEWABLE ENERGY POSTER



APPENDIX B: RENEWABLE ENERGY WORLD CONFERENCE ABSTRACT

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On-Shore Central Hydraulic Power Generation for Wind and Tidal Energy

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Abstract

Tidal energy, offshore wind energy, and onshore wind energy can be converted to electricity at a central ground location by means of converting their respective energies into high-pressure hydraulic flows that are transmitted to a system of generators by high-pressure pipelines. The high-pressure flows are then efficiently converted to electricity by a central power plant, and the low-pressure outlet flow is returned. The Department of Energy (DOE) is presently supporting a project led by Sunlight Photonics to demonstrate a 15 kW tidal hydraulic power generation system in the laboratory and possibly later submerged in the ocean. All gears and submerged electronics are completely eliminated.

A second portion of this DOE project involves sizing and costing a 15 MW tidal energy system for a commercial tidal energy plant. For this task, Atlantis Resources Corporation's 18-m diameter demonstrated tidal blades are rated to operate in a nominal 2.6 m/sec tidal flow to produce approximately one MW per set of tidal blades. Fifteen units would be submerged in a deep tidal area, such as in Maine's Western Passage. All would be connected to a high-pressure (20 MPa, 2900 psi) line that is 35 cm ID. The high-pressure HEPG fluid flow is transported 500-m to on-shore hydraulic generators. HEPG is an environmentally friendly, biodegradable, water-miscible fluid. Hydraulic adaptations to ORPC's cross-flow turbines are also discussed.

For 15 MW of wind energy that is onshore or offshore, a gearless, high efficiency, radial piston pump can replace each set of top-mounted gear-generators. The fluid is then pumped to a central, easily serviceable generator location. Total hydraulic/electrical efficiency is 0.81 at full rated wind or tidal velocities and *increases* to 0.86 at 1/3 rated velocities.