Design Report
Budget Period 1

Prepared For:

U.S. Department of Energy
Energy Efficiency & Renewable Energy
Golden Field Office
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Golden, CO 80401

HydroAir Power Take Off System

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Marine and Hydrokinetic (MHK)
System Performance Advancement
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Background

The HydroAir Power Take Off (PTO) design comprises two sets of static guide vanes located on either side of the rotor. These vanes are connected by a shaped duct to provide a route for the (inlet/outlet) airflow. Air enters the duct at a relatively low velocity and acquires a swirl motion as it passes through the inlet guide vanes. The air then accelerates as it passes down the narrowing duct towards the turbine rotor. The air drives the rotor, and then decelerates as it travels back through the expanding duct before passing over the outlet guide vanes. The process is repeated (in reverse) for the next half wave cycle. In comparison with both Wells turbines and conventional impulse turbines, the HydroAir Turbine (HAT) solution offers a step-change in efficiency and operating range. The HAT achieves its remarkable performance with fixed blade geometry, providing the reliability that is critical to the commercialization of wave power.

The HAT has a wide operating bandwidth and high peak efficiency in comparison with competing products. The turbine has only one moving part, the rotor, and hence is much more reliable than turbines that require pitching guide vanes or blades. Unlike the Wells turbine, it is self-starting and there is a significant reduction in noise over the comparable products. The rotor speed is lower than competing turbines which reduces wear but is still high enough to enable the use of off-the-shelf generators without the need for a gearbox.

Dresser-Rand designed and built a unique 10kW test facility, as part of a government funded project in 2005 in order to fully understand the HATs dynamic behavior. The test-stand allows for the fully coupled simulation of the airflow in an Oscillating Water Column (OWC) system which enables validation of Computational Fluid Dynamics (CFD) results with real data. While the facility is under-designed to allow for testing of the 500kW HAT, it will provide the basis for any systems integration testing that can be done at part-scale. The facility is already grid connected and has completed over 10,000 hours in operation. The data from the part-scale testing of the PTO reduces technical risks in the up-scaling process dramatically.

In this project, the main SPA goals have been defined as:

- Power to Weight ratio of 59.01
- Turbine Efficiency of 75%
- Annual Yield of 60%

The efficiency of a wave energy air turbine should be calculated based on a Total-to-Static (T-S) pressure differential across the turbine. In other words, the total pressure at inlet minus the static pressure at outlet will provide the pressure differential across the turbine. This pressure differential is then used to calculate the efficiency of the turbine. Some turbine developers use Total-to-Total pressure differential across the turbine which results in efficiency values to be several percent higher. However, this is technically incorrect because the air pressure at the turbine outlet is exhausted to atmosphere and therefore the remaining energy is wasted. For the purpose of this report, all the efficiency figures are calculated based on Total-to-Static pressure differential across the turbine.

$$Efficiency = \frac{Torque \times Angular\ Velocity}{Pressure\ Differential \times Volumetric\ Flow\ Rate}$$

In Wave Energy industry, Power to Weight (PWR) ratio is a very common Key Performance Indicator (KPI) for the device development. This is usually calculated using the formula below in which the Rated Power is the expected power that the overall system is capable of produce and Capacity Factor is the ratio of the actual power produced at a site to the power produced by the device if operating at the rated capacity.

$$PWR = \frac{Rated\ Capacity \times Capacity\ Factor}{Weight\ in\ Air}$$

However, the formula above is not commonly used in the PTO industry and is not directly transferable to the Power-Take-Off development as it takes into account the buoy characteristics (through the capacity factor ratio); therefore, the result would not be KPI for the PTO. For this reason, and for the purpose of this report, the power to weight ratio is calculated based on maximum possible power production over the weight of the turbine and generator combined. This is also consistent with the SOPO document submitted to the DOE.

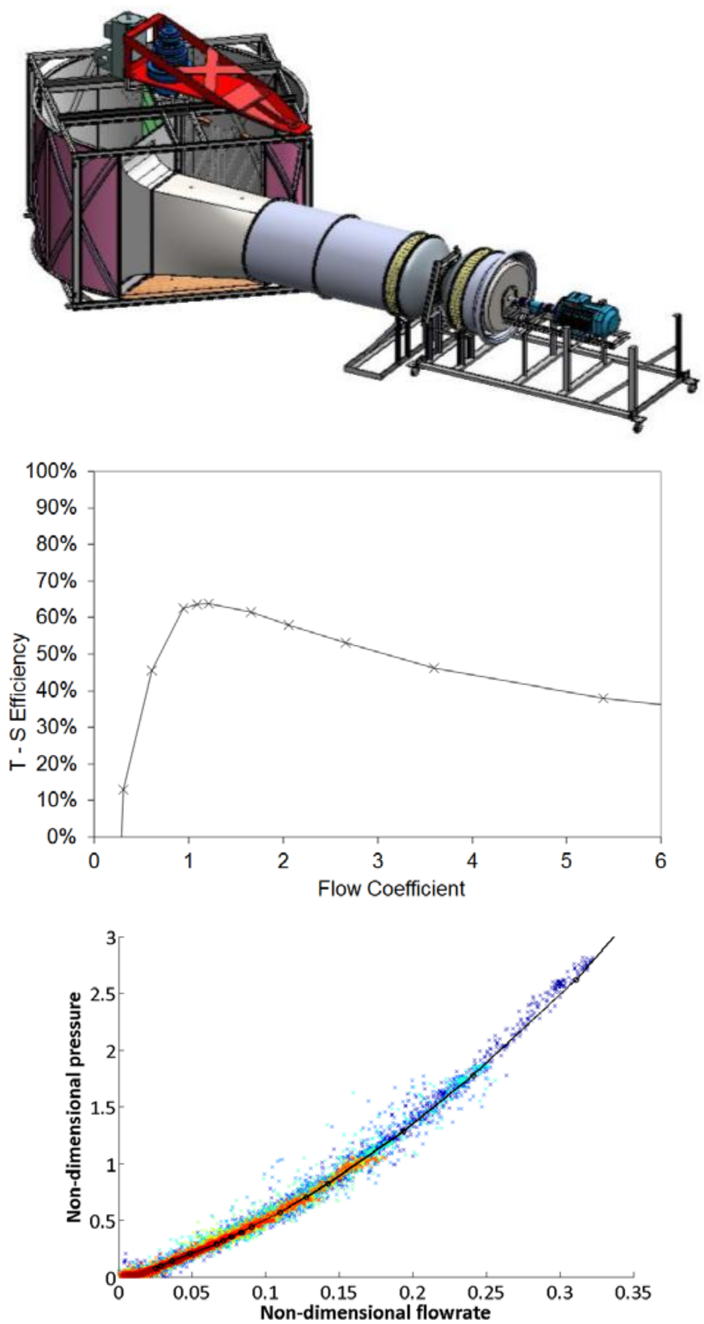


Figure 1 - Test Facility

Task 1.0 - Detailed Implementation Planning

Milestone 1.1 Deliver finalized detailed Integrated Master Schedule (IMS) and Intellectual Property (IP) plan

The detailed IMS was submitted to the Department of Energy (DOE) and Energy Efficiency and Renewable Energy (EERE) on November 26, 2014 for review and comment to ensure agreement with milestone timelines and project objectives. The Dresser-Rand IMS has since been updated to reflect the increased time necessary to complete the generator and VFD evaluation and the down-selection process. As per Dresser-Rand's 1Q RPPR (Research Performance Progress Report) and recent project update meetings, Dresser-Rand is submitting the Go/No-Go deliverables at the end of June 2015 and plan to complete Budget Period 1 design activities in August. The Dresser-Rand Data List was agreed to by the DOE and Dresser-Rand in February 2015; the Data List will be incorporated in the next Modification to the Dresser-Rand Award.

Milestone 1.2 Sub-contractors contracts finalized for work through the duration of the project

We have completed selection of sub-contractors for design engineering with the selection of Dresser-Rand Peterborough and Dresser-Rand Seattle Technology Center (STC) for necessary engineering resources to complete HAT design efforts.

A competitive Request for Proposal (RFP) was issued to prospective suppliers for the Generator and Variable Frequency Drive (VFD) on September 19, 2014. Proposals were received from the following prospective suppliers: DRS Power Technologies, Inc., Gamesa, Curtiss-Wright, Electro Mechanical Specialists, Inc. and Siemens Industry Inc. A compliance matrix was generated by Dresser-Rand to compare offerings, assess key parameters and achievement of SPA goals (e.g. PWR, efficiency, availability)

Dresser-Rand advised the DOE in an email from Gary Pearson to Ryan Sun Chee Fore on December 5, 2014 that Dresser-Rand had narrowed its selection to two (2) suppliers (1 US, 1 non-US) and based on our assessment of each offer may be confronted with the possibility of selecting a non-US supplier that manufactures its generator in Europe but can meet the domestic source requirement for the VFD. In January 2015, Dresser-Rand selected Siemens Industry, Inc. to supply the Permanent Magnetic Generator (PMG) and Variable Frequency Drive (VFD). The PMG offered is based on a standard generator manufactured in Germany and a VFD manufactured domestically. However Dresser-Rand received commitment from Siemens that production generators will be manufactured in the US.

Milestone 1.3 Purchase Orders with sub-contractors placed via engagement with Dresser-Rand Supply Chain Management (SCM)

A Purchase Order was issued to Siemens Industry Inc. March 9, 2015 for Non-Recurring Engineering (NRE) associated with Task 4.0 generator and frequency converter electrical design activities and milestones. Technical discussions commenced in April while commercial terms and conditions associated with the scope of supply for NRE were negotiated. Commercial negotiations were finalized in May.

Statement of Work (SOW) was issued to a number of Composite Consultants for support in the development of a Glass Reinforced Polymer (GRP) material specification and process procedure to support the HAT design. The SOW includes a material study and manufacturing methods report summarizing the work. The first of these was received May 1, 2015 with subsequent updated quotes May 15, 2015. A purchase order was issued to Composites Consulting Group (CCG) June 19, 2015. Within the SOW is the requirement for a Materials Study and Manufacturing Report to include Value Analysis/Value Engineering (VA/VE) activities related to material selection and manufacturing techniques used to aid in the design of the prototype HAT being manufactured at a lower cost.

Subsequent purchase orders will be issued to Dresser-Rand Chula Vista and Re Vision in Budget Period 2 to support construction, installation, commissioning and testing (Tasks 8, 9 and 10 – Dresser-Rand Chula Vista) and Impact analysis (Task 12 – Re Vision).

Milestone 1.4 Deliver Failure Mode Effect Analysis (FMEA)

A Failure Mode Effect Analysis (FMEA) evaluating all aspects of the turbine has been conducted by Dresser-Rand which involved a comprehensive review of various turbine components, assemblies, and subsystems to identify failure modes, and their causes and effects. For each component, the failure modes and their resulting effects on the rest of the system were recorded in a specific FMEA worksheet. The FMEA worksheet was created by Dresser-Rand with space for failure modes and failure causes to be documented. Failure modes were assessed based on the probability of failure, the likelihood of detection and the severity of the failure. Local effects, higher level effects and end effects were considered and recorded. The FMEA produces a RPN (Risk Priority Number) which will allow failure modes, providing the biggest risk, to be targeted and the compensating measures identified and recorded. The initial FMEA was submitted to the DOE Technical Officer May 5, 2015 to document completion of this milestone.

Task 2.0 Conceptual Design

Milestone 2.1 Definition of parameters i.e. foot print and operating speed range etc.

Ocean Energy USA (OE) provided characteristic data from the Galway Bay ¼ test buoy, which provided the baseline for scaling and creating the optimum damping requirements for the full scale Oscillating Water Column (OWC) chamber. In conjunction with preliminary turbine damping characteristics generated in early Q1 2015, the buoy data suggested a turbine rotor diameter between 1.6 and 1.85 meters was required to obtain ideal chamber damping while operating at high turbine efficiency, with a 1.85 meter diameter corresponding to peak turbine efficiency. In order to satisfy geometric constraints of the target deployment buoy being built by OE, a turbine rotor diameter of 1.65 meters was selected, and as a result ideal buoy damping occurs closer to 86% of peak turbine rotor efficiency. It is expected that operation of the HAT will occur somewhere between the ideal damping and peak efficiency.

Milestone 2.2 Deliver developed damping curve with respect to the expected system performance

Preliminary turbine geometry was developed to meet the required damping of the device. The turbine nominal speed is ~700 rpm with a maximum speed of 1000 rpm. Based on the expected operating condition and power range of the 32 meter hull buoy, an optimum required damping map Figure 2 below was developed for the device.

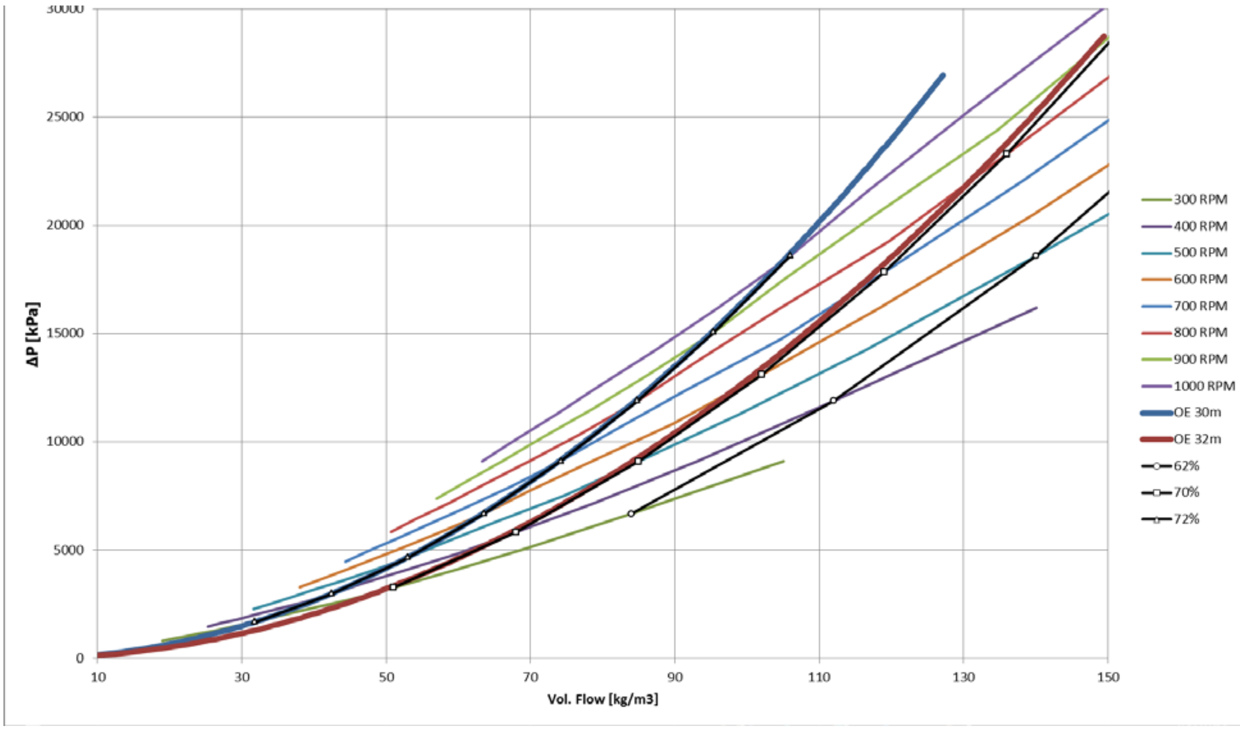


Figure 2 - Damping Map

Milestone 2.3 Fundamental turbine envelope optimization in respect to generator sizing and geometry

The operating envelope of the HydroAir PTO was optimized simultaneously with Milestone 2.2. For the selected geometry of 1.65 meter rotor diameter, the turbine will have an operating band of 300-800 rpm with a maximum speed of 1,000 rpm, and a maximum OWC pressure differential of 10kPa (1.45psia).

The conceptual design of the turbine has been completed with the inclusion of all major interfaces and components. The turbine will have a rotor diameter of 1.65 meters and be of a horizontal arrangement.

Figure 3 - HydroAir PTO Concept

A Pugh matrix was used to evaluate different aspects of the turbine with the inclusion of a risk factor to ensure survivability. Some of the most critical items that were evaluated in the Pugh matrix were: disassembly method, rotor shaft interface, inner duct support, bearing type and turbine support. These were evaluated based on criteria such as: mass, practicality, reliability, maintainability and cost. Each concept was evaluated against multiple criteria and was given a total score for comparison purposes.

Pugh Matrix

		Method 1		Method 2		Method 3		Method 4		Method 5	
Criteria	Weighting	Score	Total (WxS)	Score	Total (WxS)	Score	Total (WxS)	Score	Total (WxS)	Score	Total (WxS)
Low Mass	5		0		0		0		0		0
Practicality	5		0		0		0		0		0
Simplicity	4		0		0		0		0		0
Functionality	5		0		0		0		0		0
Reliability	3		0		0		0		0		0
Maintainability	3		0		0		0		0		0
Manufacturability	3		0		0		0		0		0
Cost	4		0		0		0		0		0
Durability	3		0		0		0		0		0
Installation Time	2		0		0		0		0		0
Size	2		0		0		0		0		0
Dis-assemply Time	3		0		0		0		0		0
Low Risk	4		0		0		0		0		0
	Total		0		0		0		0		0

Figure 4 - Pugh Matrix Document

A concept design emerged based on Dresser-Rand's engineering team experience, knowledge and use of the Pugh Matrix. The turbine will utilize single row deep groove ball bearings, a rotor shaft interface using a locking nut and lock washer, and allow the generator side ducting and associated components to move axially on guide rails providing access to the rotor and bearings for maintenance purposes. The generator will be supported on a standalone structure with access for essential maintenance.

The Shut Off Valve (SOV) will be based on the same design principles as was used on the OceanLinx turbine building on prior experience with horizontal blades closing off the annulus, however the details need to be further established due to the large scale of 500kW.

A FMEA is underway with the mechanical side complete. The electrical portion (e.g. PMG and VFD) is in progress with Siemens and will be completed prior to the end of Budget Period 1. Instrumentation FMEA is complete. Compensating provisions resulting from the mechanical side include the use of webs at the inlet to stop large detritus entering the flow path reducing the risk of damage to critical components such as the SOV and rotor blades. Feedback from generator FMEA resulted in adding a parallel electrical path between the generator and braking resistor, independent of the Central Braking Module (CBM), in case of loss of electric grid. Also, a mechanical locking brake would be designed, to lock the rotor shaft, in case of SOV webs failing to close completely.

Milestone 2.4 Computational Fluid Dynamics (CFD) model verifying the expected turbine performance target of 75%

Preliminary CFD work on a conceptual HAT was completed in early Q1 2015, delivering a maximum peak efficiency of 75%. Data from the conceptual design was utilized in milestones 2.2 and 2.3 for sizing of the final turbine. Additional details on CFD work can be found under Milestone 3.2. DOE informed on April 28th that seven (7) iterations of flow path geometry were performed.

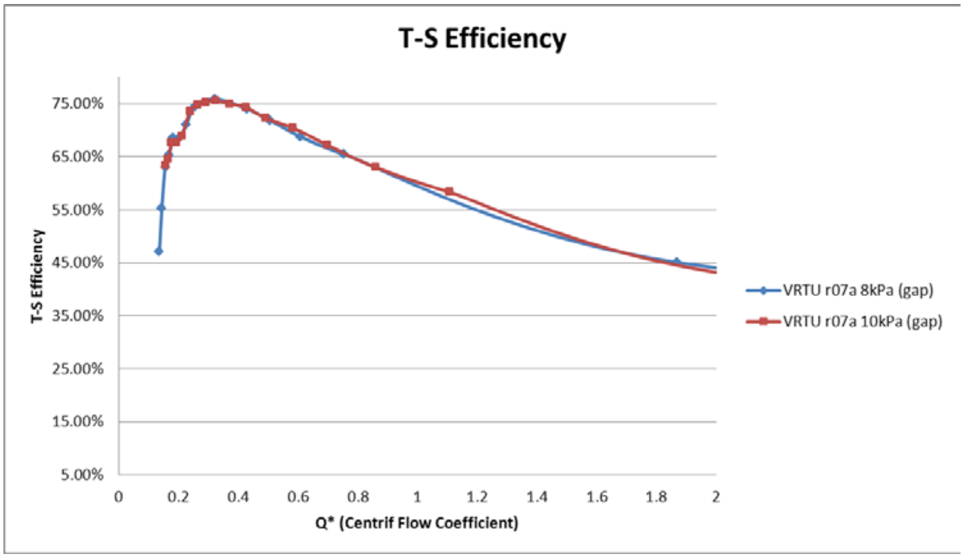


Figure 5 - Isentropic efficiency

Milestone 2.5 Deliver concept design results report

As requested by the DOE Technical Project Officer, Dresser-Rand briefed DOE April 28th on results of the Concept Design during the International Marine Renewable Energy Conference (IMREC) held in Washington, DC April 27-29, 2015. Completion of this Milestone is further documented in the submittal of this Design Report. The CapEx, OpEx and LCOE are still to be finalized as they are dependent on electrical systems and composite component final costing. Performance and reliability quantities from CFD were confirmed with Milestone 2.2 through Milestone 2.4 with additional details of CFD under Milestone 3.2.

Milestone 2.6 Decision point – Confirmation of Testing Site

Currently the decision is on hold pending direction/guidance from DOE/EERE. Recipient is aware of Ocean Energy's pending award under DE-FOA-0001081 Marine and Hydrokinetic (MHK) Demonstrations at the Navy's Wave Energy Tests Site (WETS) and its desire to test the Recipients HydroAir PTO on the OE buoy at WETS. Recipient's first priority is to test at WETS.

Subtask 2.1: Conceptual Turbine Design Risks and Mitigation Measures

Extensive use of CFD was utilized to produce a final turbine size and validate performance targets in Milestone 2.2 through Milestone 2.4. Dresser-Rand has extensive experience with the validation of CFD and used an approach that's been validated through physical testing at the Cranfield University test facility. Building on previous experiences with similar projects Dresser-Rand performed a FMEA to evaluate the concept design using the document created under Milestone 1.4.

Task 3 – Detailed Turbine Design

Milestone 3.1 Production of detailed HydroAir radial turbine design

The detailed design of the HydroAir PTO is being undertaken by Dresser-Rand. The ducts, guide vanes, shroud and rotor blade geometry have been optimized through CFD in Milestone 2.2, 2.3 and 2.4.

The main interface of the HydroAir PTO with the OEL buoy is an interface frame, the frame consist of two rings which are connected via webs with the SOV fixed to the outer ring. A bolted connection provides a nonpermanent joint allowing removal. The HydroAir PTO consists of four sets of ducting, the generator side outer and inner, and the chamber side outer and inner. The outer chamber side ducting will be fixed to the SOV, and the inner chamber side ducting will be fixed to the inner ring of the interface frame.

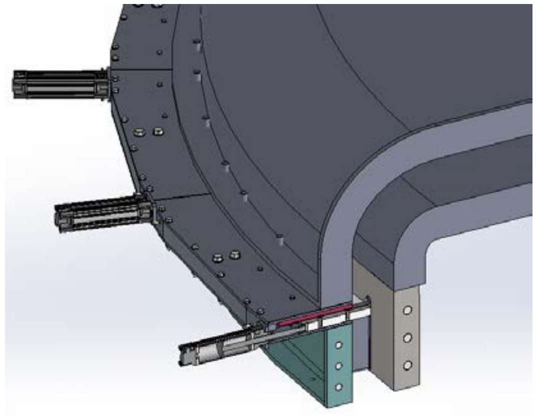


Figure 6 - SOV, interface frame and ducting

Bearing selection has been completed with life calculations performed. Single row deep groove ball bearings were chosen as a result of Pugh matrix. The reason for this is their simplicity, a key factor of the HydroAir PTO. There are three bearings on the Hydro PTO, two as part of the rotor assembly, with one thrust bearing taking thrust loads in both directions and one floating bearing. The generator side ducting bearing is incorporated into the generator side ducting to position the ducting and maintain concentricity, ensuring peak efficiency can be achieved (see Figure 7). Calculations for mean time between failures were based on maximum rotor operating conditions and worst case loading. This approach was taken to be conservative; as a result bearing life is satisfactory to allow planned preventative maintenance, reducing unplanned downtime and helping to achieve the target annual yield of 60%.

Magnetic bearings were assessed in line with the SPA goal; a detailed evaluation showed that single row deep groove ball bearings were in this case more appropriate. The use of magnetic bearings would add weight to the HydroAir PTO resulting in a lower PWR and provide a limited advantage. Magnetic bearings also present more complexity with the need for calibration. A main focus of the design of the HydroAir PTO has been to reduce downtime with regards to maintenance, single row deep groove ball bearings can be changed rapidly to get the turbine back on line. The cost of single row deep groove ball bearings is significantly lower than that for magnetic bearings impacting the LCOE. While magnetic bearings present advantages with regards to condition monitoring, dealing with irregularities in mass distribution, low friction and dealing with high speeds, the duty of the bearings on the HydroAir PTO do not necessitate such a high specification bearing. Magnetic bearings also necessitate a power supply which adds the complexity of routing cables through the air flow.

Other bearings evaluated where hydro dynamic, hydro static and lubricant free bearings. The two former were discounted due to cost and complexity associated with a dedicated lube oil system. All three bearings will be sealed to prevent the ingress of water and salt. As added protection the use of rotary lip seals is also incorporated to stop any water pooling in the bearing housings. The seals lips are manufactured from a material that is not susceptible to a marine environment to ensure they perform as expected.

The rotor assembly consists of a rotor wheel, 32 blades and 8 shroud pieces. The blades will be attached to the rotor wheel with the shroud fixed to the blade tips. The blade and shroud geometry were optimized through CFD; Milestone 2.2, 2.3 and 2.4 and will be manufactured from composite. The rotor wheel will be manufactured from Aluminum to provide corrosion resistance and reduce weight increasing PWR. The rotor has been sized preliminarily based on worst caseloads (with an estimated mass for the blades and shroud). The rotor wheel will see loads primarily due to loading from the blades and shroud, dependent on mass and angular speed. Dresser-Rand and CCG are currently working to produce a material specification that will allow an accurate mass to be calculated allowing optimization of the rotor wheel assembly.

The rotor assembly will be supported by a stub shaft that is housed in a bearing housing on the chamber side of the turbine; allowing the generator side of the turbine to be pulled back axially leaving the rotor exposed for maintenance. The bearing housing has been sized with the use of FEA to reduce weight but maintain structural integrity.

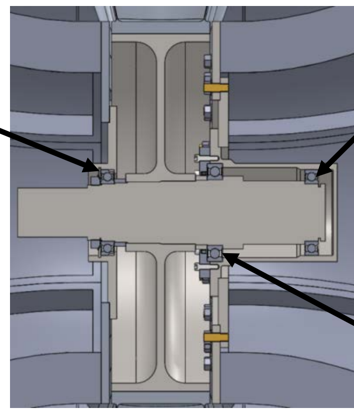


Figure 7 - Bearing Arrangement

The bearing housing will be supported by the main interface frame (see Figure 8) so that it can remain in place if the chamber side ducting needs to be removed. An interface plate will tie the bearing housing to the inner duct to allow concentricity to be fixed.

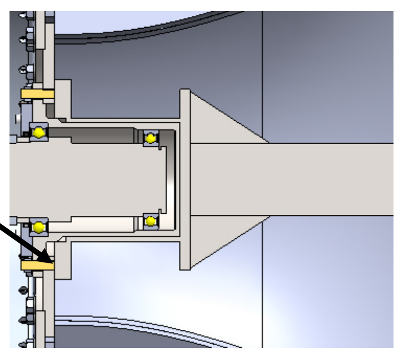


Figure 8 - Bearing housing interface

The rotor shaft will be manufactured from stainless steel to give the corrosion resistance properties needed for a marine environment but also the structural properties desired. The shaft has been sized based on maximum torque derived from the turbine power. Due to an overhung arrangement the rotor will see shear stresses, bending stresses, and both tensile and compressive stresses as the thrust load is reversed. The shaft is stepped to facilitate the fitting of bearings and the rotor; fillets have been used at changes in diameter to reduce stress concentrations. These have been evaluated based on a worse case load as the exact mass of the blades, shroud and rotor wheel have not yet been finalized, due to pending consultations with CCG. FEA will be performed on the finalized design.

A locking nut and washer will be used to secure the rotor wheel to the shaft. This allows the rotor shaft to be left in situ but the rotor assembly (wheel, blades and shroud) to be removed quickly. A tip spacer ring is used to join both the outer chamber and generator side ducting together and maintain the correct clearance between the rotor and the inner ducts.

The generator will be supported on a fixed standalone structure. The structure will be manufactured from hollow tube to reduce weight but maintain structural integrity (similar to a wind turbine). The interface requirement between the buoy and the support has been provided to OE. The generator will be bolted to the frame with the option to use shims when aligning the generator with the rotor shaft. Access to the generator will be possible to allow essential maintenance to be carried out.

The inner generator side ducting will be positioned by the third bearing with its housing bolted to the ducting; this allows concentricity between the chamber side ducting to be maintained. The generator end of the duct will be fixed to the movable frame. The frame will allow the generator side half of the turbine to be pulled back. The outer duct will be positioned using the tip spacer ring and the guide vanes.

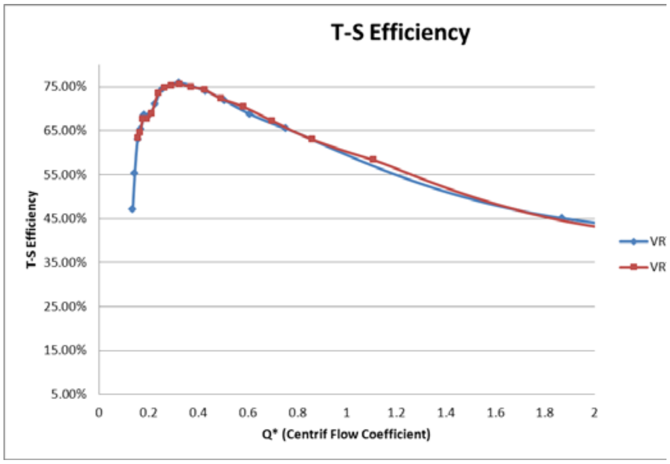
A drive train will be placed between the generator and rotor shaft with the use of multiple flexible couplings to compensate for any misalignment. This will need to be removed before the generator side of the turbine can be pulled back for maintenance operations.

Figure 9 - Generator Stand

The HydroAir PTO has two sets of guide vanes, one at the inlet and one at the outlet. The guide vanes will be manufactured from composites with CCG's input to develop a material specification. The guide vanes will be bolted between the inner and outer ducts to ensure positional accuracy. The guide vanes direct the flow of air so surface finish on these components is a key factor regarding material specification and or any proprietary coatings. CCG will advise regarding these matters.

Milestone 3.2 Verify aerodynamic performance

Computational fluid dynamics (CFD) modeling of the HAT began in early Q1 2015 with preliminary work focused on providing characteristic performance curves; these curves were then utilized for geometry and envelope sizing tasks detailed in milestones 2.2 and 2.3 above. Once the final rotor diameter of 1.65 m was selected, work shifted to detailed modeling and optimization of the complete HydroAir flow path. Current predictions indicate a peak efficiency of 75% (see **Error! Reference source not found.**10).


T-S Efficiency
75.00%
65.00%
55.00%
45.00%
35.00%
25.00%
15.00%
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T-S Efficiency
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Q* (Centrif Flow Coefficient)
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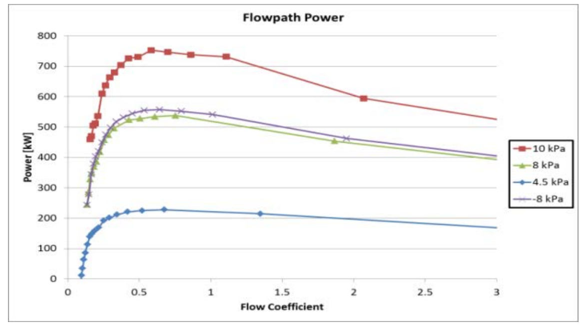
Flowpath Power
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Power [kW]
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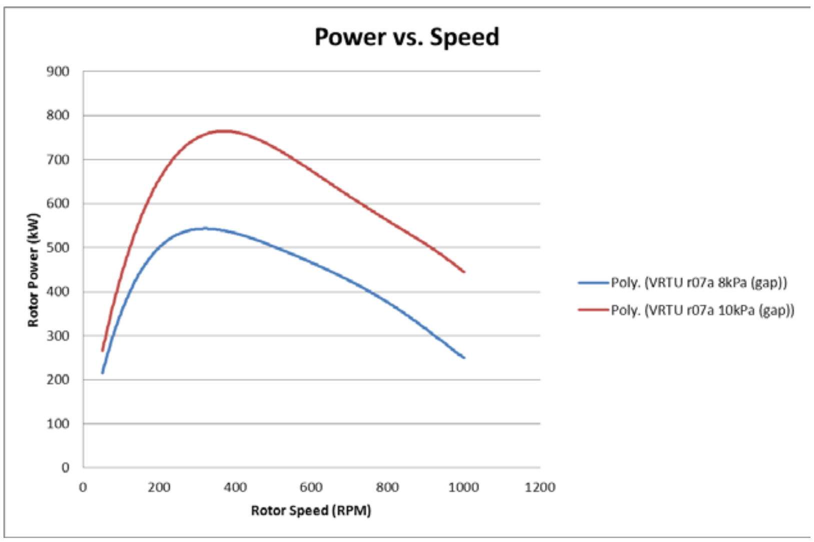
Power vs. Speed
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800
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Rotor Power (kW)
0
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Poly. (VRTU r07a 10kPa (gap))

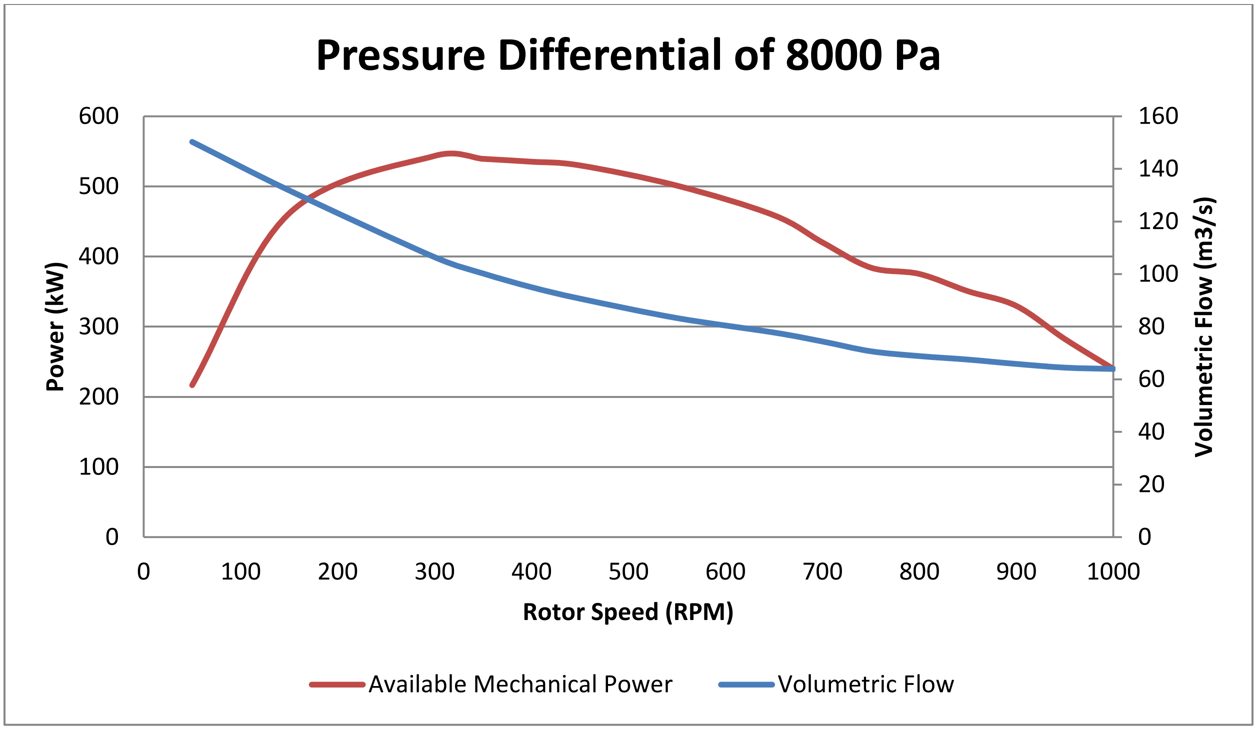
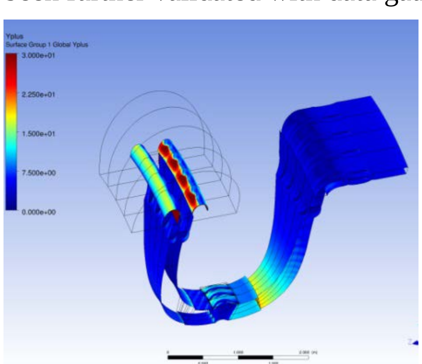


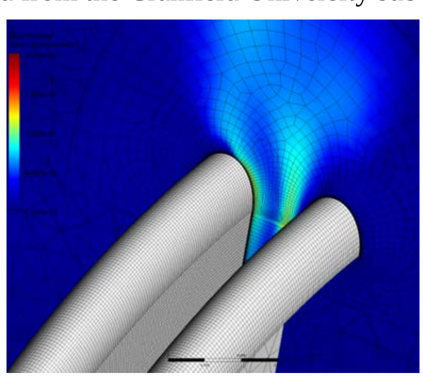
Figure 10 - predicted turbine performance and power as a function of flow coefficient for various OWC pressure differentials

Simulations have been run across a variety of operating conditions, ranging across the full speed range of 0-1000 rpm and various pressure differentials across the turbine (ΔP) up to the maximum operating pressure of 10kPa.

CFD runs utilized for performance prediction include modeling of all relevant flow path geometry from inlet to outlet, including structural support components and gaps between rotating and non-rotating surfaces. Additional sub-scale models have been run for detailed component design and investigation of specific flow phenomena, including optimization of specific aerodynamic components; investigation of various cross-flow conditions on the exterior bell-mouth inlet; and detailed modeling of the proximity effect of the buoy on the flow entering and exiting the turbine.

Over 120 CFD runs have been completed in support of turbine design and pretest predictions, utilizing a computational cluster of 128 cores and expending over 40,000 CPU hours. Models utilized the 3D viscous RANS equations solved by ANSYS CFX with the SST two-equation turbulence model and scalable wall-functions, and used a stage mixing-plane approach when coupling rotating and non-rotating domains. Global Y^+ values were less than 30, and less than 10 within critical flow regions; these values were found to sufficiently capture the boundary layer effects within regions of adverse pressure-gradients encountered in the turbine, based on various grid dependence studies performed on the current HydroAir turbine geometry. The employed CFD techniques have been further validated with data gathered from the Cranfield University sub-scale test rig.





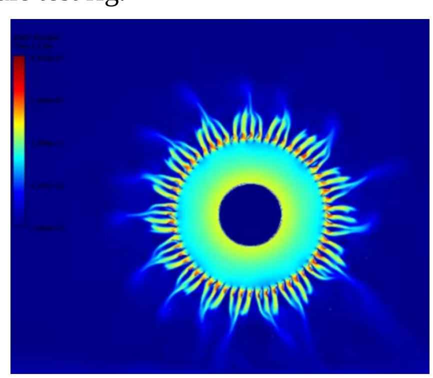


Figure 11, 12 & 13 – examples of CFD results

Milestone 3.3 Reduce weight

As the detailed design progresses and individual components are finalized Dresser-Rand will verify the weight of the overall HydroAir PTO and compare this with the targeted reduction in weight. FEA has been used to optimize components with regards to structural integrity and create a more efficient design. Consultations are ongoing with CCG to develop the material specifications for composite components with Power to Weight Ratio (PWR) a key driving factor. CCG will perform the necessary structural analysis to optimize material specification and increase PWR. As it stands, it is estimated target of the weight of turbine and generator would be 11,334 kg and 3,650 kg respectively. Presuming a maximum nominal power of 1MW, which both generator and turbine are able to generate if fed with corresponding pneumatic wave power, the estimated target PWR would be 66.74 (watt/kg).

Milestone 3.4 Increase annual yield

Various elements have been optimized to achieve the annual yield of 60%. These include:

- Turbine Efficiency - High efficiency of 75% (increased from 68%)
- Effective servo control strategy based on torque and speed sensing
- Meantime between failure 25 months
- Maintenance Downtime Maximum 3 days
- Planned Maintenance – once every 12 months, reducing to once every 24 months
- Severe weather conditions necessitating shut down of turbine

CFD analysis has been completed under Milestone 3.2 with the turbine achieving a peak efficiency of 75%. Based on Wave Rider Data for the WETS in 2013 (supplied by OE) the total pneumatic power available to the HydroAir PTO over a year is 1150030 kWh. Taking into account the peak efficiency of the HydroAir PTO of 75% an average turbine efficiency of 68% has been assumed for the annual yield analysis. Based on an average efficiency of 68% and an estimated 95% availability (taking into account downtime and shut downs due to weather) representing a downtime of 18.25 days the estimated annual yield is 742919 kWh which equates to a yield of 64.6%. Taking a more conservative approach and estimating a 90% availability representing a downtime of 36.5 days of the HydroAir PTO the annual yield is still estimated to be 61.2%. In both cases the HydroAir PTO exceeds the targeted annual yield of 60%.

Milestone 3.5 Complete production of manufacturing drawings

The first manufacturing drawings have been released by the Dresser-Rand Peterborough design team to Dresser-Rand Wellsville to allow the testing of the data transfer system and identify issues that may arise. Consultations with composite consultants CCG are still ongoing and detailed design of some components are likely to change with CCG's advice, thus on completion of this phase Dresser-Rand will commence production of the manufacturing drawings.

Milestone 3.6 Produce HydroAir PTO Assembly Drawing

Like milestone 3.5 Dresser-Rand is not yet in a position to produce a finalized HydroAir PTO Assembly Drawing. However the CAD model facilitating the assembly drawing is in construction and will be completed once component parts are finalized allowing the release of the assembly drawing.

Milestone 3.7 Deliver HydroAir PTO Design Report

This Milestone; Milestone 3.7 is met by the submittal of this design report.

Milestone 3.8 Deliver Draft HydroAir PTO Test Plan

The Draft Test Plan which includes factory spin test, factory spin down test, factory no load flow test, factory part load flow test and ocean trials is complete and will be accompany this Design Report and Continuation Application. The Draft Test Plan summited to the DOE summarizes the scope of testing and briefly explains each test segment. A comprehensive HydroAir PTO Test Program is being developed that will complement this high level plan.

Task 4 – Electrical Design

Milestone 4.1 Size a generator

Based on the functional requirements of the HydroAir PTO, a list of key technical requirements for the generator was created:

- Variable speed harnessing; the generator should be able to capture power from the full range of rotor speeds ranging from 350-750 rpm
- Efficiency greater than 97 % for the operating rotor speed range.
- The generator should have a low auxiliary system consumption (e.g. low heat exchange due to lower heating effects)

The Permanent Magnet Generator (PMG) is a Synchronous Machine, where the DC excitation Circuit is replaced by permanent magnets eliminating the need for brushes. Without brushes and slip rings, the Permanent Magnet Synchronous Generator has a smaller physical size, a low moment of inertia resulting in a higher reliability and a higher power density per volume ratio. By utilizing permanent magnets in the rotor circuit, the electrical winding losses (I^2R) in the rotor are eliminated. This leads to an overall increased efficiency for PMG. With this in mind, it was decided to select a PMG generator for the HydroAir turbine application.

Parameter/ Generator	**water cooled PMG**	**air cooled PMG**	**air cooled Induction Generator (IG)**	**Unit**
Rated power	500	500	500	kW
Rated voltage	690	690	690	V
Rated frequency	86.7	86.7	86.7	Hz
Rated current	480	710/480	-	A
Rated speed	650	600/650	-	Rpm
Power factor	0.95	0.95	-	
Efficiency for rated torque within speed range 350-750	**>97%**	>97%	> 93%	
Weight	**3650**	4970	8935.762	**Kg**

Table 4.1: Comparison of parameters for generator selection

A water-cooled PMG Generator will be supplied by Siemens.

1. There will be fresh water closed loop circulation by means of a pump, and a radiator (fan) arrangement will provide forced convective cooling.

2. Water entering the PMG will be maintained below 38^{o} C, with flowrate = 30 Lit/min

The generator has been selected after a thorough understanding of the marine conditions to be encountered in operation. Key design features are listed below:

- Rated power 500 kW; rated voltage (DC bus) 690 V; rated current 480 A; Rated speed 650 rpm; These specifications for the generator were selected based on turbine design and operating conditions to be encountered in WETS site.
- This generator will be water cooled. An inlet water temperature of 38 °C has been assumed (standard for water cooled generators for marine application).
- Mechanical design of this generator (bearings, corrosion protection, etc.) will be according to ABS Standards.
- Ingress Protection rating of IP 56 shall be provided in the build of this generator. This will protect against dust and powerful water jets.
- The generator shall have standard terminal junction box for motor leads connection
- Maximum rotation speed: 780 rpm
- Assumed maximum overload (for short durations): 1000 kW, at 650 rpm. This gives a maximum torque of 14.7kN-m
- Thermal Class rating F (155 °C)
- Noise level : less than 82 dB(A)
- Efficiency for rated torque within speed range 350-750 is greater than 97%
- Inbuilt torque sensing capability, with accuracy of +/-1.5 % of zero speed torque (Mo). The value for Mo is measured to be 11550 Nm.
- Shaft speed measurement via encoder, with an accuracy of +/- 0.001% of rated speed
- Overall generator weight : 3650 Kg
- Moment of inertia: 26.1 Kgm²

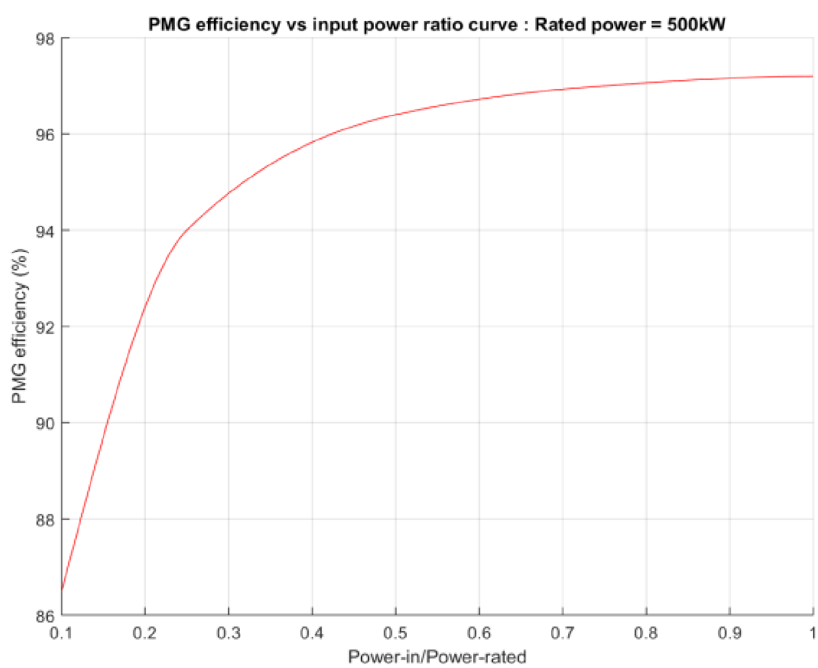


Figure 14 - PMG efficiency at various load ratio values

Integration of generator into turbine design

- Turbine will be connected to the generator by means of a direct drive shaft. This results in a simple mechanical assembly, with ease of access to turbine rotor during planned maintenance and repairs. Details of the generator support frame are under Milestone 3.1.
- Based on the speed and torque analysis for the HydroAir Turbine, the aforementioned selection of generator meets the technical requirements of power generation and also the goal of achieving a high Power to Weight ratio (greater than 60 watt/kg)

Milestone 4.2 Size a frequency converter

SINAMICS S120 Cabinet Modules are based on the SINAMICS S120 range of components. It is a modular system to configure enclosed drive line-ups with a central line infeed (rectifier) and common DC bus supplying power to multiple motor modules (inverters). A line-up comprises of cabinet modules installed side-by-side in a row. All drive components are arranged in a clear, compact layout in the individual Cabinet Modules. They offer great flexibility for the configuration of drive line-ups, and a comprehensive array of options allows the systems to be optimally adapted for a marine environment application and to meet the required drive performance. The S120 cabinet will be housed within a control chamber on the deck of the Ocean Energy buoy.

The main components of the S120 system are:

- Line Connection Modules (LCM) with line side components such as circuit breakers, fuses, contactors and line reactors.
- Active Line Modules (ALM) for four-quadrant (fully regenerative) clean power operation, with negligible line harmonics and unity or adjustable power factor depending on grid requirements.
- Motor Modules (MM), which are in individual cabinets, contain the inverters (AC to DC bridge).
- Auxiliary Control module (AUX): Control units that implement the speed, torque and power based control for power take off from the generator.
- Auxiliary Power Supply Modules to supply accessories such as blowers and for AC and DC control power.
- Central Braking Module (CBM) for Dynamic braking with external resistors.

Standardized power and control interfaces facilitate easy configuration, installation and integration with the plant. Pre-manufactured connections are provided for interconnecting the DC bus and auxiliary power between cabinet modules. Communication between power modules and control units is done via DRIVE-CLiQ, the flexible backplane bus.

DRIVE-CLiQ (Drive Component Link with IQ) is a communication system for connecting the various SINAMICS components (e.g. Control Unit, Line Module, Motor Modules, generators, and encoders). DRIVE-CLiQ forms the backplane for the complete drive system. The standardized cables and connectors reduce the variations across different parts and cut storage costs. Converter boards (Sensor Modules) for converting standard encoder signals to DRIVE-CLiQ are available for third-party motors or retrofit applications.

DRIVE-CLiQ supports the following functions:

- Automatic detection of components by the Control Unit
- Standard interfaces to all components
- Standardized diagnostics down to component level
- Standardized service down to component level
- Electronic rating plate

The electronic rating plate contains the following data:

- Component type (e.g. SMC20)
- Order number (e.g. 6SL3055-0AA0-5BA0)
- Manufacturer (e.g. SIEMENS)
- Hardware version (e.g. A)
- Serial number (e.g. "T-PD3005049”)
- Technical data (e.g. rated current)

Technical specifications for S120:

Module ID	**Module description**
AUX1	Auxiliary control cabinet
LCM1	770 Amp Line connection module
ALM1	800 kW Active line module
MM1	735 Amp Motor module
MM2	630 kW Braking module

Table 4.2 Module list

Note: Refer to figures 15 and 16 for relative location of above modules in S120

Physical Dimensions: 4400 mm (wide) X 600mm (deep) X 2700 mm (high)
Weight: 2600 kg

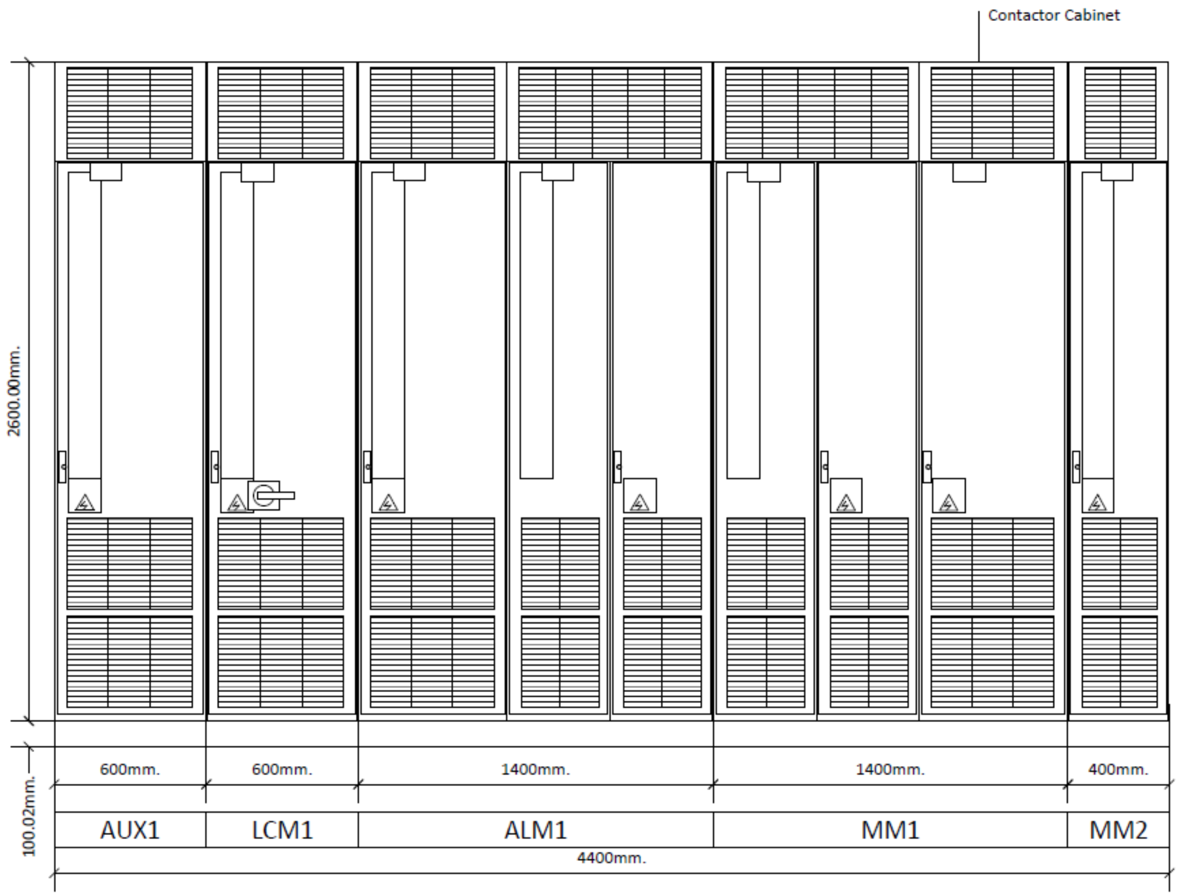


Figure 15 – VFD Module dimensions

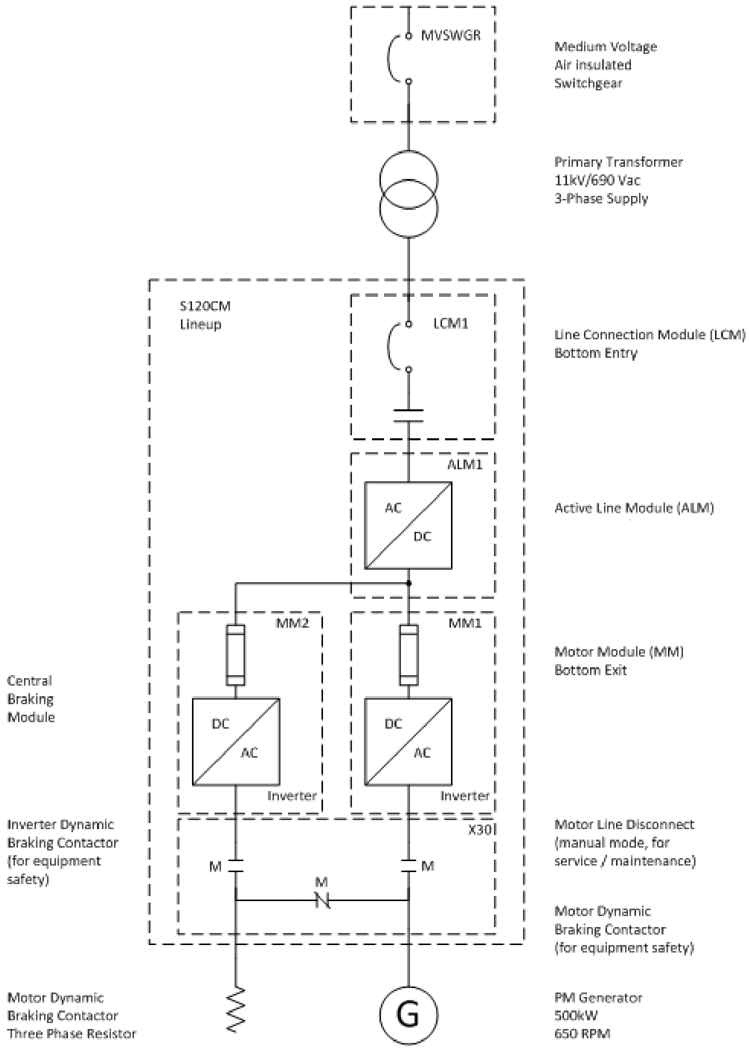


Figure 16 - Single Line Diagram for Electrical System

<u>Compliance Standards for S120</u>

- EMC-compliant configuration that is safe functionally and in operation
- Applicable North American electrical standards
    - NEMA ICS 7: Industrial Control and Systems : Adjustable-Speed drives
    - NEMA ICS 7.1: Safety Standards for Construction and Guide for Selection , Installation and Operation of Adjustable-Speed Drive Systems

- NEMA 250: Enclosures for Electrical Equipment (up to 1000V)
- UL508C: Power Conversion Equipment

Converter Control and Interface (CCI)

The CCI is a PLC based system to perform the visualization, operator interface, alarm and fault diagnostics, as well as technological control of the system and interface to higher-level control system. The CCI can also be used to interface with the condition monitoring instruments on the turbine and buoy.

The Human Machine Interface (HMI) is an integral component of the CCI. It will consist of a TP1200 (12 inch touch panel interface) mounted display, programed with 2 screen views to display operational parameters and alarm/fault messages. The screen can display system variables such as operational status, power, current, voltage, speed, torque, component temperatures etc.

Servo Control Algorithm

Servo control is employed for cyclic processes with precise, highly dynamic position control and servo motors, e.g. in textile, packaging, printing machines and machine tools. Servo control is a form of vector control adapted to control permanent magnet AC generators. Servo control compares the rotor angle with respect to the stator angle to determine the load on the system. This information is then used to adjust the current amplitude and angle in the stator to precisely control the torque of the generator. In this design, the use of servo control is dictated by the generator selected with the added benefit of having very fast and precise control loops.

The proposed control concept is to program the controls to apply a load to the Generator based on the speed of the generator shaft.

1. As the speed increases from zero speed to 350 rpm the load will be applied according to the square of the speed minus a factor, to allow the system to accelerate.
2. When the system reaches 350 rpm, the load applied will be full rated load for the speed of the shaft, allowing for maximum power recovery.
3. If the torque applied is greater than the load torque, the shaft will continue to accelerate until the torque applied and the load torque is balanced.
4. If the speed reaches maximum rated speed, the load torque will be increased by the controller to try to limit the speed. This increased load can only be sustained for a short period of time due to the I^2R losses in the inverter and generator. The permissible duration is dependent on the magnitude of the load. The system will continually monitor the temperature of the generator and inverter, and will shut down in the event of overheating.

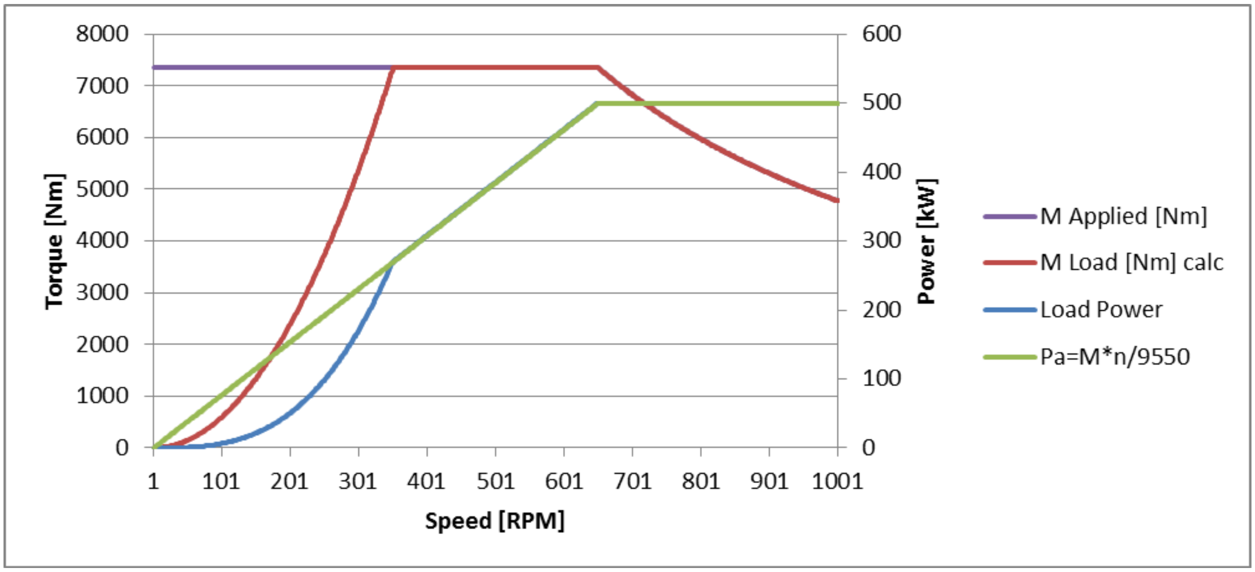


Figure 17 – Torque-speed and Power –speed curves

5. The Active Line Module of the inverter system monitors the DC bus of the system and draws power in from the line or regenerates back to the line based on the voltage of the DC bus. As the Inverter loads the generator, a current flow from the generator to the inverter is produced. This current acts to increase the voltage on the DC bus. This voltage increase is sensed by the Active line module which then regenerates this energy to the line in an effort to maintain the DC bus voltage at a constant value.

Milestone 4.3 Specify the instrumentation used to collect data for the selected test platform

The generator shall be mounted with following inbuilt instrumentation.

Parameter	Quantity	Measurement Range	Accuracy	comments
Torque measurement	1	+- 11500 Nm	+- 1.5 %	4 kHz sampling; measured value averaged over 3 sec
Speed measurement	1	+- 650 rpm	+- 0.001%	4kHz sampling
Winding Temperature	6	+-200 °C	Class B	PT 100 Resistance Thermometer
Bearing Vibration	2 (X,Y axes)	Acceleration: frequency response 3 - 10000 Hz	2%	4-20 mA output

Additional sensors will be mounted on the turbine to measure various physical properties. These include:

Parameter	Sensor Instrument	quantity
Static pressure (gauge) [Guide vane + rotor]	pressure transducer	36
Dynamic pressure (gauge) (differential Pressure)	pressure transducer (flow-rate measurement purpose)	12

Air flow rate	pitot tube	12
Temperature for turbine and duct	PT 100	2
Solenoid for Shut Off Valve (SOV)	6 port solenoid manifold	1
Position switch for piston inside cylinder	2-color indication type REED switch	48

Milestone 4.4 Ensure power quality is grid compliant for selected test platform.

The inverters on the PMG side are coupled through a common DC bus which allows energy exchange to the Active Line Module (ALM). This ALM is configured for 4-quadrant (fully regenerative) clean power operation. This ensures negligible line harmonics and also permits adjustable power factor, depending on the grid requirement. Additional data will be collected from the WETS electric grid provider to ensure these requirements are comprehensively documented.

Key power quality features:

- VFD drive will comply with IEEE 1547 standard and IEEE 519 (harmonics)
- Total harmonic distortion level (THD) will be limited to less than 5 %

Milestone 4.5 Produce instrumentation diagrams for selected test platform

Dresser-Rand has currently not completed the instrumentation diagram for the selected test and will complete the instrumentation diagram in budget period II. Instrumentation has been identified including the location of this instrumentation on the turbine.

Task 5 – Structural Design

Milestone 5.1 Integrate the HydroAir PTO design with the test site

The integration of the HydroAir PTO with the test platform is ongoing. Dresser-Rand is in discussion with OE to finalize details. Interfaces have been defined and selected in Milestone 2.6 – define all mechanical interfaces.

Milestone 5.2 Estimate Center of Gravity (CG) and mass distribution of the overall HydroAir PTO

The CG will be estimated through the use of inbuilt CAD tools providing an estimation of the PTO CG. This cannot be calculated until all component parts are finalized.

Milestone 5.3 Verify HydroAir PTO components meet availability, OpEx requirements

Dresser-Rand has undertaken a FMEA to reduce downtime and OpEx (Operational Expenditure). Through compensating measures the availability of the HydroAir PTO has been increased.

Milestone 5.4 Produce interface components manufacturing drawings

Turbine support, generator support and chamber interface has been designed from Dresser-Rand perspective. The interface requirements have been communicated to OE, and are dependent on OE securing the contract funding with DOE.

Milestone 5.5 Deliver report on results from validation of structural requirements of the design

Structural validation has been completed on mechanical and frame design. The validation on composite components will progress upon completion of the composite design report from CCG.

Subtask 5.1: Structural Design Risks and Mitigation Measures

Extensive FEA and FMEA have been used with worst case loading to ensure structural reliability and avoid structural failure in the event of overload. All loading scenarios have been identified; these include pressures from the plenum chamber and loading due to acceleration of the buoy. Materials specified are suitable for a marine environment with the main ducting, guide vanes, and rotor blades composite; CCG will perform the relevant structural analysis to ensure structural reliability. The number of webs and their profile between the inner and outer interface frame have been optimized with FEA to deal with thrust loads on the SOV resulting from high chamber pressures;

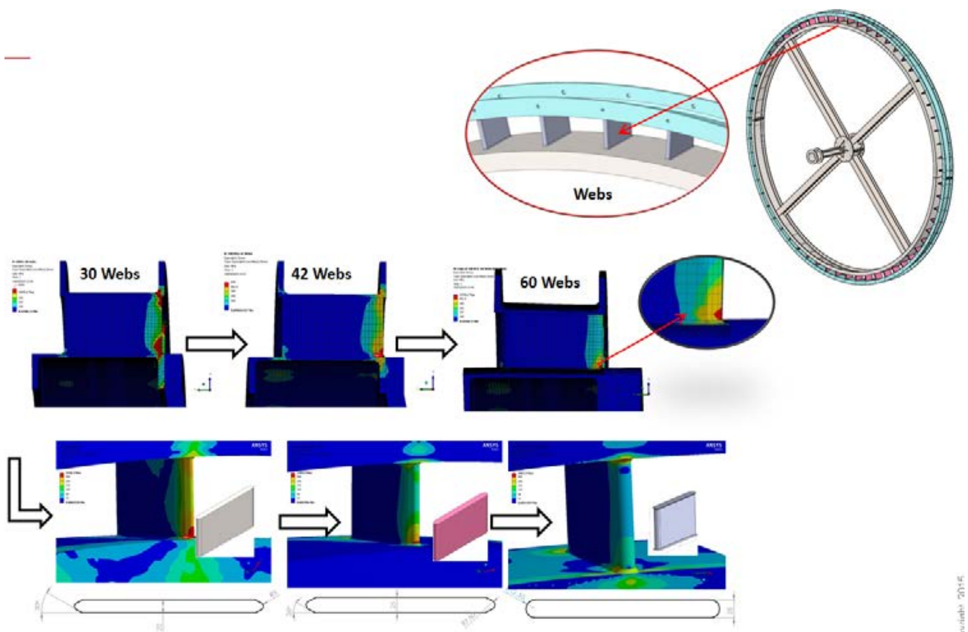


Figure 18 – examples of FEA results on Spider frame web structure

The number of SOV blades has also been reduced to avoid excessive deflection from high chamber pressures when closed. FEA was used to optimize the number and thickness of blades ensuring reliable operation.

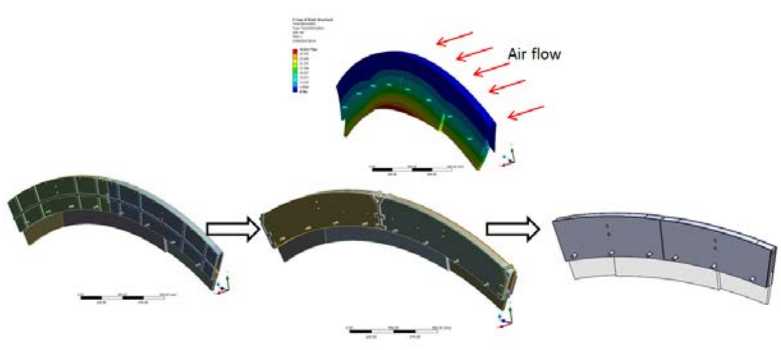


Figure 19 – SOV design optimization based on FEA results

Extensive FEA has been carried out on the generator support stand to ensure that deflection does not impact on the operation of the turbine. Acceleration and movement of the buoy has been taken into account to simulate the changing of distribution of mass. Fatigue has been taken into account for components subjected to cyclic loading to ensure life cycle is acceptable.

Task 6 – Shut Off Valve Design

Milestone 6.1 Complete design for a custom or project specific Shut Off Valve

The design of a custom SOV has been completed by Dresser-Rand. The design is predominantly based on the SOV used on the OceanLinx turbine but has been sized to meet the operational conditions of the 1.65m Radial HAT and the WETS.

The SOV consists of 24 horizontally mounted blades that protrude into the annulus blocking the flow of air from the plenum chamber. Each blade is controlled by a single, double acting pneumatic actuator. The SOV has been designed to withstand chamber pressures up to 100kPa. FEA was used extensively to produce an optimized blade design that could stand up to the loads imposed by the chamber pressure. A larger amount of blades has been used compared with the prototype built for OceanLinx to reduce the deformation of the blades and resultant bending stresses. An advantage of using a large number of blades is the reduction in blade size; in the unlikely event of a blade not operating the annulus exposed is reduced limiting the airflow through the turbine. Conversely if one blade fails to operate when the SOV is opening a smaller area of the annulus is obstructed reducing the effect on turbine performance until maintenance can be performed.

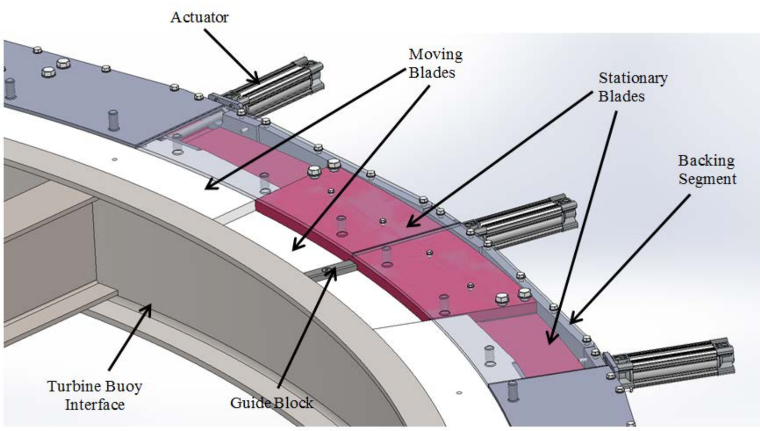


Figure 20 - SOV Major Components

To totally close off the annulus without any gaps moving blades are overlapped, a stationary blade is used to offset each moving blade. A guide block ensures the positioning of moving blades to maintain an adequate overlap.

Top and bottom plates seal the SOV to the atmosphere and interface with the turbine buoy interface and outer ducting. To aid maintenance, inspection holes will be included allowing a threaded plug to be removed prior to the insertion of a boroscope. Blades can be replaced by the removal of the actuator and backing segment allowing replacement in situ.

The 24 actuators are double acting pneumatic cylinders operating at 5 – 6 Bar. The air reservoir is sized to provide enough redundant operations so that in the event of a power loss the SOV can be operated and the turbine shut down. The materials chosen for the SOV have been carefully considered to give optimum performance. This includes materials for sliding parts with a low coefficient of friction, and materials exposed to the elements with adequate corrosion resistance.

Figure 20 - Plan view of a section of the SOV

Milestone 6.2 Produce manufacturing drawings

The first drawings of the SOV have been released by Dresser-Rand Peterborough to allow Dresser-Rand Wellsville to familiarize them and resolve any potential issues with data transfers. Current consultations with CCG are ongoing, as some composite components interface with the SOV some manufacturing drawings cannot be completed until such time as the composite components are finalized.

Milestone 6.3 Produce instrumentation diagram

The instrumentation diagram for the SOV is not currently complete; however the majority of instrumentation components have been specified. A 5 port single direct solenoid valve, with spring return mechanism is proposed for actuating the cylindrical pistons to operate the SOV. Each solenoid valve shall regulate the pneumatic flow to 6 cylinders. Hence, a total of 4 such solenoids will be used. The circuit will also have a particulate filter with coalescer before the air enters the valve assembly. The compressed air will be supplied by a pump, and an accumulator of 100 liters has been sized, to meet the air flow requirements. The coefficient of flow (C_v) is under design, based on the response time needed to shut-off the webs of the turbine.

Milestone 6.4 Produce an assembly drawing

The SOV assembly drawing is currently underway; however the CAD model facilitating this drawing is complete. This represents a significant amount of the work involved in the creation of an assembly drawing.

Subtask 6.1: Shut Off Valve (SOV) Risks and Mitigation Measures

Dresser-Rand undertook a FMEA as part of Milestone 1.4. In the event of a failure or partial failure where the annulus is not closed off airflow will continue to pass through the turbine. This presents a problem in severe operating conditions or when carrying out maintenance.

The FMEA highlighted areas where compensating provisions could reduce the probability of a failure. The number of blades has been doubled from 12 to 24 to mitigate the failure of a blade breaking due to overpressure from the plenum chamber. Webs have been incorporated into the annulus before the SOV to mitigate the effects of debris impacting the blades or stopping the blades from closing fully. A periodic operational check will be implemented to ensure functionality of the SOV.

Task 7 – Value Analysis / Value Engineering

This task is currently ongoing and is to be completed as shown on the updated IMS. CCG has been contracted to support Dresser-Rand with this task. CCG will produce a Material Study and Manufacturing Methods Report that includes various manufacturing techniques along with a costing analysis of the components assessed. Report will document VA/VE analysis taken with the Dresser-Rand HAT design team and identification of key manufacturers who can produce at volume. The report will also include potential changes in manufacturing methods due to increased volume of piece parts and cost impact.