Final Technical Report

Project Title:	Reliable Lightweight Transmission for Off-Shore, Utility Scale Wind Turbines
Award Number:	DE-EE0005190
Project Period:	October 1, 2011 to March 31, 2012
Lead Organization:	Eaton Corporation Innovation Center 4201 N 27th Street Milwaukee, WI 53216
Team Members:	University of Minnesota Clipper Windpower, LLC.
Principal Investigator:	Dr. Jean-Claude Ossyra Phone: (248) 226-6203 Email: Jean-ClaudeOssyra@eaton.com
Contractual Contact:	Cindy K. Shane Phone: (414) 449-6607 Email: lucindakshane@eaton.com
Authors:	Jean-Claude Ossyra, Srinivas Patri, Venkat Durbha, Anjela Jankowski, Kim Stelson, Feng Wang, Rahul Dutta, Brad Bohlman, Jeffrey Marr, Chris Feist and Matthew Lueker

October 25, 2012

Prepared for:

U.S. Department of Energy
Golden Field Office
Gary Nowakowski, Project Officer

Acknowledgment:
"This material is based upon work supported by the Department of Energy under Award Number DE-EE0005190."

Executive Summary

This report presents key findings from the Department of Energy's Next Generation Drivetrain Project 'DE-EE0005190 - Reliable Lightweight Transmission for Off-Shore, Utility Scale Wind Turbines' executed between October 1, 2011, and March 31, 2012. The objective of this project was to reduce the technical risk for a hydrostatic transmission based drivetrain for high-power utility-size wind turbines. A theoretical study has been performed to validate the reduction of COE for the wind turbine, identify risk mitigation strategies for the drive system and critical components, namely the pump, shaft connection and HST controls and address additional benefits such as reduced deployment costs, improved torque density and improved mean time between repairs (MTBR).

For the next generation drivetrain, an advanced hydrostatic transmission (HST) drivetrain concept has been proposed which considerably reduces the weight in the nacelle and replaces the gearbox with a significantly more reliable solution. This will reduce the cost of energy (COE) considerably (>24%) through capital cost reduction (related to goal 50% improvement in torque density (Nm/m^3) while maintaining an equivalent cost/torque), reduction of Operations & Maintenance (O&M) costs, reduction of Levelized Replacement/Overhaul costs (including a >50% improvement in mean time between replacement of gearboxes and/or generators), with an increase of lifetime energy production, reduction in deployment (20% reduction) costs.

To capture the value of the advanced drivetrain concept with the most accurate data available the following approach was used for the utility size wind turbine selection:

- Study Cost of Energy impact on 2.5 MW onshore wind turbine,
- Scale and model prediction to 5 MW offshore wind turbine,
- Selected: onshore wind turbine Clipper Liberty C96, 2.5 MW, Wind Class II.

To identify the advanced drivetrain configuration with the highest potential, multiple configurations for the hydrostatic transmission (hydraulic pump, fluid lines, and hydraulic motor) and induction generator were investigated including

- drivetrain hydraulic motor and generator location in nacelle or on the ground,
- number of parallel induction generators used for drivetrain,
- size of accumulators used in hydrostatic transmission for energy storage.

For the next generation drivetrain the advanced hydrostatic transmission (HST) drivetrain concept has been analyzed for in-nacelle and on-ground solution. The highest CoE reduction compared to a gearbox solution can be obtained with an on-ground HST solution which considerably reduces the weight in the nacelle and replaces the gearbox with a significantly more reliable solution. This will reduce the cost of energy (CoE) considerably (13%) through

- capital cost reduction of 13%,
- reduction of Operations & Maintenance (O&M) costs of 56%,
- reduction of Levelized Replacement/Overhaul costs of 30%,
- a >50% improvement in Mean Time Between Replacement (MTBR) of gearboxes and/or generators,
- reduction in lifetime energy production of 11%, and
- reduction in deployment costs of 20%.

Table 1 shows the results in terms of the key metrics and improvements relative to the baseline wind turbine. The overall technical risk level of the HST drivetrain approach is low due to the extensive use of commercially available components (TRL 9) in a parallel system architecture which allows for scalability by increasing the number of components rather than their size. The only major component with a high risk is the pump (TRL 2), for which the technology and design risks have been identified.

Table 1. Overview of key metric and improvement relative to baseline (5 MW offshore, on-ground)

	Improvement vs Baseline	Source of Reduction
Capital Cost	-12.79%	Primarily through nacelle weight reduction, causing considerable material savings on wind turbine, reducing transportation and installation cost, additional potential through use of 4 pole synchronous generator without full power electronics.
O&M Cost	-56.48%	Better accessibility of generator, power electronics and hydraulic motor, longer maintenance intervals and replacement components through reduced wear in bearings, power electronics reduction with time intensive maintenance (time to detect and fix a failure).
LRC	-29.7%	Drivetrain component cost reduction, longer times between replacements, and reduction of electrical connections.
AEP	-11.43%	Variable speed transmission with lower power losses downstream of the generator.
Torque density	+20%	Space saved in the nacelle due to relocation of half of the transmission, the generator and the power electronics into the tower base at the same power (torque) transfer.
MTBR	>50%	HST repairs exceed the 7.7 years MTBR of the current gearbox due to preventive scheduled maintenance, different load conditions and higher component life.
Deployment cost	-19.85%	28% reduced transportation costs (average of material savings) and installation cost reduction of 34% (turbine material savings)
COE	-13.00%	Per CoE calculation

Table of Contents

List of Figures

List of Tables

1. System and Component Design

To convert the rotor shaft power to electric power, a continuously variable hydrostatic transmission is set between the rotor and the generator as replacement for a standard mechanical gearbox. It mainly consists of a large fixed displacement hydraulic radial piston pump, variable displacement hydraulic axis piston motors and fixed displacement hydraulic charge pumps, as shown in Figure 1. The hydrostatic transmission is designed as a closed circuit, which means the output flows of the motors are directly fed back to the pump inlet. The hydraulic reservoir is parallel. This allows operating the low pressure side at higher pressure without damaging effects to components. A charge pump is required in the closed circuit hydrostatic transmission to make up for the power losses due to pump and motor efficiencies and line transmission losses.

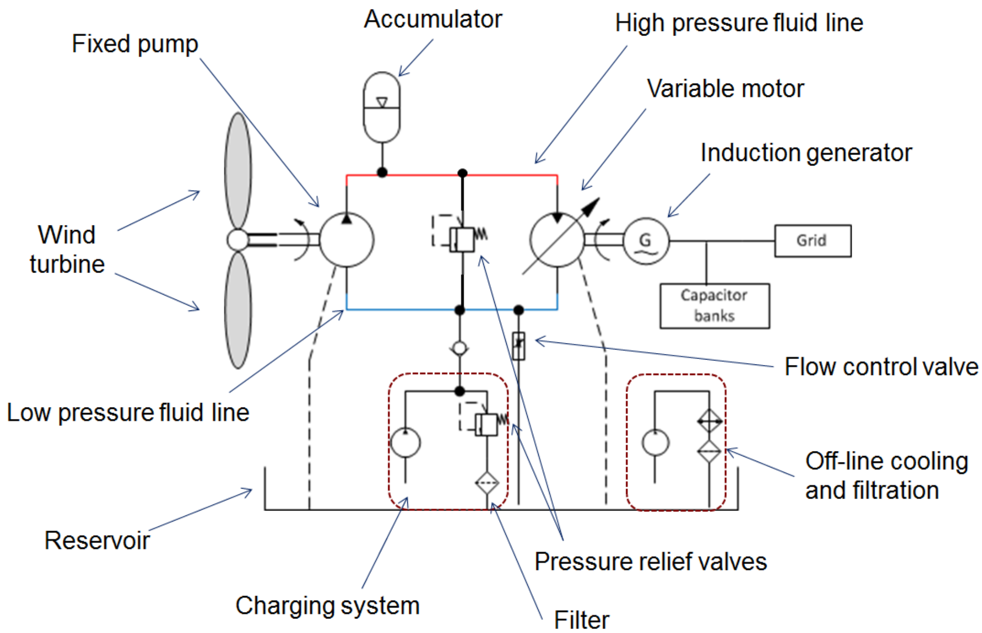


Figure 1. Simplified schematic diagram for a closed circuit hydrostatic transmission

The rotor shaft power is first converted to hydraulic power through the pump and then converted to motor shaft power through multiple motors. By using the variable displacement motors, the motor or generator shaft speed is decoupled from the rotor shaft speed so that the generator can run at near synchronous speed all the time. The ratio between speed and torque can be changed continuously within the power limits to achieve rotor speed changes with the wind speed so that the optimum Tip Speed Ratio (TSR) and the maximum rotor power coefficient (C_p) can be achieved. Figure 2 illustrates the in-nacelle configuration of our proposed drivetrain.

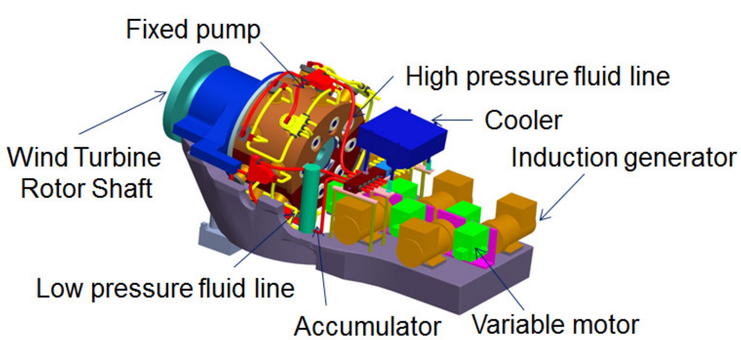


Figure 2. Solid model of hydrostatic drivetrain for in-nacelle configuration

1.1. Existing Components

Motor

Sizing/selection information

At rated conditions the required motor displacement for a 2.5 MW system is approximately 2445 cc/rev. There are no commercially available motors of this displacement. With a displacement range of 250 to 750 cc/rev, Hydrokraft motors provide the highest displacement within Eaton's portfolio. In our solution, we propose to use multiple Hydrokraft motors to achieve the total displacement at rated conditions. These motors are variable-displacement, axial piston units with a controllable swash-plate. During the turbine operation, as the rotor speed varies with the wind speed, the motor displacement is adjusted by changing the swash-plate position to achieve a constant speed at the high speed output shaft.

In our study, two specific motors were considered, the 750 cc unit and the 500 cc unit. The maximum speed of the 750 cc unit is limited to 1500 rpm, while the 500 cc unit is rated for 1800 rpm. Due to the speed range of the motor, various motor-generator configurations can be used to generate electricity. A Pugh matrix based concept comparison study was done to identify the motor-generator configuration appropriate for our drivetrain. The system operating pressure was chosen to be 5000 psi. At this pressure, the required motor displacement at rated conditions decreases to 2445 cc/rev. This net displacement is achieved by using five Hydrokraft units in parallel each one directly connected to a generator.

Scalability

The required net displacement of the motors increases in proportion to the rating of the turbine. For example, for a 5 MW turbine, the required displacement is 4890 cc/rev. This increase in required net displacement is accommodated by simply adding additional motors in parallel to the existing motors. Therefore, the proposed configuration can be easily scaled to any required power without the need for drivetrain redesign.

O&M Tasks

To determine the mean-time-between-replacement (MTBR), bearing life calculations for the motor were done. To this end, the annual pressure distribution in the high pressure lines of the motor was determined by assuming a Weibull wind distribution. The bearing life of the motor for this duty cycle was estimated to be about 3.3 years. To achieve reliable operation of the turbine, it is proposed to replace the motors once every 3 years. Associated costs have been captured in the O&M cost of the Cost of Energy model. By establishing a regular maintenance regime, any unexpected turbine downtime due to motor failure will be avoided.

Fluid Conveyance

The selection of appropriate fluid conveyance elements involves a trade-off between minimizing the fluid lines required for installation, minimizing the line losses, and availability of commercially available components. The fluid conveyance sub-system comprises the following key components: fluid lines (hoses, tubing), fittings and manifolds.

Fluid Lines

In-Nacelle Configuration

The flow from all the cylinders in the pump is collected in a manifold before being routed to the motors. Similarly, on the low pressure side, all the flow from the motors is routed to a collection manifold before being supplied to the pump. As recommended in [12], the velocity of the fluid in the high pressure lines of the transmission is limited to 4.5 m/s, and the velocity on the low pressure side is limited to 3.5 m/s. Note that at this velocity, the flow is in the turbulent regime. To identify the effect of flow losses on the drivetrain efficiency a more detailed model for fluid conveyance was studied. The section between the pump and initial collection manifold is identified to have pressure losses at a 90 degree bend fitting, the straight section of hose and the connection at the manifold. Similarly the connection from the manifold to the motor was assumed to incur losses due to a 90 degree bend fitting at the motor, the straight section of the hose and the connection at the manifold. The flow at the manifold connection was modeled as the side flow in a T-junction fitting to simulate significant pressure loss. Using the total flow rate from the pump and the number of cylinders on the pump, the flow rate from individual cylinders can be determined. The individual cylinder flow rate and the stipulated flow velocities have been used to determine the dimensions of the flow lines [14]. As per these calculations, a hose with 1.25 inch internal diameter (ID) would be sufficient between the cylinder and the collection manifold. From the packaging study, the average hose length from the collection manifold to the motors was determined. The number of lines from the collection manifold to the motor is limited by the number of motors. Therefore the velocity in these lines would be higher than the stipulated value of 4.5 m/s. To minimize the losses, hoses with 2 inch ID were selected. On the low pressure side, hoses with 2.25 inch ID are used from the motor to the collection manifold, and hoses with 1.5 inch ID are selected to run from the collection manifold to the pump cylinders.

On-Ground Configuration
It was deemed impractical to install a total of sixty pipes for fluid conveyance. A study was performed to identify the trade-off between multiple pipes of smaller diameter and a single pipe of larger diameter. The results of this trade-off are provided in a later section. But it was noted that effect on CoE improvement was minimal. Therefore, it was decided that a single pipe would be used to route the fluid from the collection manifold near the pump to a collection manifold near the motors at the tower base. A pipe with six inch ID will be required on the high pressure side, and a seven inch ID pipe will be required on the low pressure side. Such large diameter pipes, which can handle the required flow rates and pressure, are commonly used in the oil and gas industry.

Scalability
The number of fluid lines required in a 5 MW turbine is twice that of a 2.5 MW turbine. Similar scaling is observed for turbines at higher power rating. Pipes with diameter of twelve inches are commercially used in oil and gas industry. Therefore, the fluid lines can be scaled as per the constraints of the tower.

O&M Tasks
The current recommended "Mean Time Between Replacement" (MTBR) for hoses for such an application / duty cycle is 5 years. This proposed MTBR can be improved through proactive maintenance enabled by Eaton's patented LifeSense hoses. These hoses are however currently not commercially available in the required size.

Swivel Joint
With a single pipe to run the length of the tower, a single swivel joint whose axis is aligned with the tower, can be implemented. There are no such commercially available swivel joints. However, there are many such joints in use today at a smaller scale in the mobile hydraulics industry. A common application is the joint used to accommodate the swing motion of an excavator's cab.

Hydraulic Oil
The efficiency of fluid conveyance is significantly influenced by the type of oil used in the system. The oil is required to have sufficiently high viscosity to minimize volumetric losses. Biodegradable oil is preferable, due to its limited environmental impact in case of an oil leak. Synthetic oil is preferred as it provides good operating characteristics in extreme weather conditions. Currently a synthetic, bio-degradable oil is proposed in our solution.

Charge Pump
Sizing/selection information
In the proposed drivetrain the following tasks are required of the charge pump,

a) Replenish the main HST loop to make up for leakage
b) Maintain required pressure in the return line to enable retraction of pumping cylinders
c) Provide hydraulic power for controlling the motor swash-plate

Based off of previous hydraulic system experience, a portion of flow at motor outlet should be diverted to the reservoir. This is to reduce the required capacity of the heat exchanger. The charge pumps are used to replenish this flow in the return line by pumping in the cooler oil from the reservoir. This significantly increases the required flow in the charging loop. The charge pumps were resized accordingly. A vane pump was selected to be the charge pump.

While a little more expensive than gear pumps, the vane pumps are more reliable and efficient. Another concern with gear pumps is contamination of working oil due to gear failure.

The charge pump is required to maintain a predefined pressure in the return line. This is to enable piston retraction in the cylinders of the main pump. The swash-plate control unit in the Hydrokraft motors requires a flow of 12 lpm at a pressure of 1160 psi. Since the pressure requirement for the two tasks is very different, an additional vane pump is integrated to provide the pilot flow required for the swash-plate control. The two vane pumps will be run in tandem as a double vane pump configuration. Such configurations are commercially available.

Scalability
Similar to the motor, at higher turbine power ratings, additional charge pumps can be integrated into the drivetrain without the need for a redesign.

O&M Tasks
Vane pumps will be used as charge pumps in our proposed solution. The design of these pumps provides balanced load on the driveshaft. An MTBR of 10 years is assumed for the charge pumps.

Oil Cooler
The losses incurred at various points in the drivetrain are dissipated as heat, leading to an increase in the temperature of the working oil. At higher temperatures, oil viscosity decreases and it loses its ability to maintain good lubrication properties. Consequently the system efficiency drops. Higher temperatures also lead to a potential hazard of oil combustion. To mitigate these situations, it is recommended to keep the working oil temperature below 65^{o} C. This is achieved by using oil coolers in the drivetrain. In a hydrostatic transmission, the cooler is typically connected in the return line between the motor outlet and the pump inlet.

It was proposed to divert a portion of the flow from the main return line to the tank for the purpose of filtering and cooling. The charge pumps will be used to compensate for this flow loss in the return line by supplying cooler oil from the tank. The temperature of oil in the tank is regulated by designing for sufficient dwell time and by using an off-line cooling loop. The off-line loop consists of a simple centrifugal pump driven by an electric motor, a heat exchanger and a filtration unit. The pump draws oil from the hotter section of the tank, runs it through the heat exchanger and the filtration unit before sending it back to a cooler section of the tank.

Scalability
Additional oil coolers can be added in parallel as the drivetrain size scales up.

Oil Filtration Unit
Sizing/selection information
Appropriate filtration to eliminate particulates in the oil is necessary to avoid contamination of the fluid and critical machined surfaces, and thus achieve reliable system operation. The recommended ISO filtration code for a hydrostatic transmission at the design pressure of 5000 psi is 16/14/11. This code stipulates that a milliliter of working fluid is limited to contain 320 to 640 particles of 2 μm size, 80 to 100 particles of 5 μm, and 10 to 20 particles of 15 μm size. Due to the operating pressure and the required flow rate, it is impractical to use filtration units in the high pressure line. In our solution we propose to achieve filtration in the off-line loop.

Scalability
There are currently no commercially available filters that can handle the entire system flow. Therefore the flow in the off-line loop is distributed through multiple commercially available filters. At higher turbine ratings, more such filters can be integrated in the drive train to achieve the required filtration. Packaging studies will be required at wind turbine power ratings of 15-20 MW to understand any potential space constraints in the nacelle.

Generator

Sizing/selection information

The concept identified as the best solution from the Pugh matrix analysis stipulates a generator coupled to each motor. In our study, we assume that the wind power at the rotor of our proposed drivetrain is the same as the baseline. This available wind power is multiplied by the drivetrain efficiency to obtain an individual generator power rating of about 450 kW. For a 2.5 MW system, five such generators are required. Note that the net power capacity of the generators is less than 2.5 MW. This is due to higher inefficiency of the hydrostatic transmission. The negative effect of this efficiency loss on CoE is offset to some extent by the lower cost incurred on the generators. In our solution, we propose to use squirrel cage induction generators (SCIG) for power generation. SCIG's have been commonly used in fixed speed wind turbines. This generator topology is simple to design, reliable and robust. In addition, as the hydrostatic transmission is continuously variable, both the turbine rotor and the generator can be operated at their most efficient points. Previous studies [39] have shown that by designing appropriate controllers, the operational efficiency of multi-induction generator configuration can be improved. For our CoE model, the cost of these generators as reported in [39] is used. A solid model for a similar sized commercially available generator was created form generator volume and weight specifications. This information has been used to calculate the cost reduction in structural material due to potential weight savings of our proposed drivetrain.

Scalability

As the turbine power ratings vary, additional motors and generators can be added in parallel to match the rated power requirements. Therefore the proposed solution can be scaled to any desired power without requiring extensive redesign of existing drivetrain.

Power Electronics

Sizing/selection information

When connecting to the grid, squirrel cage generators can exhibit reactive "inrush" current, resulting in voltage fluctuations in the grid. To mitigate inrush current magnitude, reduced-voltage solid-state starters are used in conjunction with the induction generators. Each generator is provided with its own starter. Induction generators consume reactive power and cannot provide reactive power compensation to the grid. Therefore capacitor banks are used to provide reactive power to the generators, and if required to the grid. For component sizing calculations a Power factor of 87% at an efficiency of 96.6% at full-load has been assumed (from NREL report – low vs. industry estimates) with a Terminal output voltage of 690V.

Scalability

As the required number of generators increases with the power rating of the turbine, it is fairly simple to added more soft starters and capacitors for feasible operation.

Accumulator

Sizing

The accumulator has been selected to represent enough volume for flow ripple reduction. The final volume has to be optimized with the finalization of the pump design which is the source of these flow ripples.

Torque Coupler

Sizing/Selection

The torque transferred from the motor to the generator at rated conditions is about 2.25 kN-m. Due to the modularity of our design, this number is independent of the wind turbine rating. In addition to torque transfer, the torque coupler is selected to provide insulation for preventing stray electricity leakage from the generator to the motor.

Sensors

Sizing/Selection

Pressure sensors are required in the system at multiple points for feedback control and system monitoring. In addition, oil temperature sensors and oil level sensors will also be integrated for monitoring the system operating conditions. This will enable safe and efficient operation, and also facilitate quick location of any major faults in the drivetrain. Other sensors to monitor the turbine structure will be same as the baseline.

1.2. New Major Sub-systems and Components

1.2.1 Pump Design

The primary critical component for the drivetrain design is a reliable lightweight pump which is scalable for utility wind turbines ranging from 2.5 MW to 10 MW. There is a vast selection of smaller hydrostatic pump types available today. The available designs include gear pump, gerotor pump, vane pump and piston pump. Among them, the piston pump provides the most robust and reliable design with highest hydraulic power density. Due to these characteristics, these pump types are the most common transmission pumps in mobile drivetrain applications because they allow the use of high operating pressures, which allows for high power densities yet are robust and reliable. An example, and one of the largest commercially available HST pumps, is the Eaton 750 cc/rev, 787 kW, axial piston swash plate design with a power density of 1.7kg/kW.

Piston pumps are available in two specific designs, an axial piston configuration and a radial piston configuration. Commercially available axial piston pumps will provide a reliable solution in the selected pressure range at acceptable high efficiency, but these units are usually designed for shaft speeds of about two orders of magnitude higher than the wind turbine rotor speed. Also, current axial piston pump sizes are designed for output powers an order of magnitude lower than the turbine requirements.

Radial piston pumps will provide a reliable solution in the selected pressure range at high efficiency. These units are usually designed for shaft speeds within the range of the wind turbine rotor speed. Here again, commercially available radial pumps are designed for output power which is an order of magnitude lower than required. The weight of these commercial pumps is currently very high (e.g. Haegglunds MB 4000, 11tons for 800kW), and will require a redesign to enable targeted weight savings and the associated reduction in cost of energy.

1.2.1.1 Design Concept Review and Selection

Multiple designs for a lightweight, large capacity, robust piston pump, which utilizes the existing low speed shaft, have been reviewed. Critical characteristics include a low weight of the pump, ease of integration (initial build), ease of maintenance with respect to accessibility, repair without an additional onsite crane (other than an internal hoist), utilization of standard cylinders and valve sizes, utilization of standard manufacturing processes, utilization of standard materials and low overall cost.

As a baseline for the weight reduction published information for a wind turbine has been used. This information has been translated into a simplified solid model for the wind turbine representing the weight, size and location of the main components rotor, low speed drive shaft, bearings and bearing support, gearbox, generator, the bed plate (main frame for component support, also represents weight of power electronics) and the nacelle cover. For the weight savings calculations solid models have been used for different pump concepts and integrated into the simplified baseline model to determine weight and space savings. As this model was intended for mass (weight, volume) location only, the model will need to be validated through load calculations of the load carrying structure.

Information from the Clipper Liberty 2.5 MW wind turbine has been used for the baseline wind turbine. The resulting weight savings have been linearly scaled to predict the weight savings for a 5 MW pump concept including the systems weight saving. Solid models for the main frame, the rotor shaft, the main bearings and the bearing housing have been used for representing the turbine weight, size and location of the main components. The weight and location of other components located in the nacelle (e.g. power electronics) and the nacelle cover has not been modeled but rather taken into consideration as total value in the cost of energy calculation. The objective for this study has been expanded from weight (lowest weight possible) and volume only, to also include ease of integration (initial build), ease of maintenance with respect to accessibility, repair without an additional onsite crane (other than an internal hoist), utilization of standard cylinders and valve sizes, utilization of standard manufacturing processes, utilization of standard materials and low overall cost.

Axial Piston Design
An axial piston design was investigated where the actuation cylinder axes are located in parallel with the rotor shaft axis around the rotor shaft. Standard cylinders with integrated valves have been chosen as required. The actuation of the cylinder rod and piston is to be realized with cam roller/follower design. The cam surface is located perpendicular to the rotor shaft axis. The main disadvantages of the axial piston design regarding higher manufacturing costs, reliability issues and load complexity outweigh the main benefit of a slightly more compact design.

Radial Piston Design
A radial piston design has been investigated, where the actuation cylinders have been located perpendicular to the rotor shaft axis around the rotor shaft. Standard cylinders with integrated valves have been chosen as required. The actuation of the cylinder rod and piston is to be realized with cam roller/follower design. The cam surface is located in parallel with the rotor shaft axis. This arrangement will only cause compression of the cam material because the high cylinder forces (translating the high torque of the rotor shaft). The material thickness of the cam has to be designed appropriately. Opposing cylinders have been chosen for this design concept but force imbalance is possible and needs to be investigated. The cam connection to the rotor shaft does not need to be designed for compensating thrust loads. Multiple cam lobes have been used to allow for multiple actuation of every cylinder per revolution (compact design).

The radial pump utilizes standard cylinders, valves and materials. The manufacturing processes for the housing will be casting, all other components like the roller and follower including the cam will be manufactured using standard materials and processes. The pump design will allow for cam misalignment and varying loads without reduction of 20 years life time. Scalability in power is achieved through increase number of cams on the shaft without increasing the diameter, but pump length only. The pump housing is connected to a fixture of the turbine base (main frame). The cylinders are mounted to the pump housing. Each cylinder rod (with piston on the outer end inside of cylinder) is connected with a follower. The roller, which is moving along the cam, is connected to the follower via a set of bearings alongside the cam to translate the roller rotation to a linear motion of the cylinder piston. Each cylinder is connected with a low and a high pressure line. The lines of each pressure level are interconnected in manifold blocks to route the fluid in fewer lines between the pump and the hydraulic motors of the HST drivetrain.

1.2.1.2 Cylinder and Valves
In this pump, commercial, off-the-shelf (COTS) cylinders are used as the pumping elements. The current commercial technology in piston seals enables very high volumetric efficiency (low leakage) in the cylinder while providing sufficient lubrication against friction. Therefore the pump is projected to have a very high overall efficiency. The cylinder pistons will be driven by an internal cam mounted on the low speed shaft of the turbine. The flow to and from the pump is controlled by check valves. The cylinder pistons are not designed to be self-retracting – an initial low pressure is required at the pump inlet for piston retraction.

The required displacement for a pump operating at rotor shaft speed is determined by the rated power of the wind turbine, the rotor rotational speed and the operating pressure at the rated conditions. In the initial iteration for the 2.5 MW system, the operating pressure was increased to 5000 psi. The required pump displacement for a 2.5 MW system was calculated to be 299,337 cc/rev.

In the compression chamber of a piston pump, a fluid will be inserted at low pressure, cut-off from the low pressure supply, compressed to high pressure and then delivered to a high pressure line. The fluid management in and out of the chamber will be realized through valves on the low pressure side and high pressure side, with special opening characteristics (both off-the shelf components). The compression of the fluid will be realized through the motion of a piston. The piston is connected to a driving rod, which in our design is connected to the follower assembly. The piston is contained in the cylinder (compression chamber). The approach is to use a standard cylinder assembly for the fluid compression together with integrated standard valves.

The valves available for fluid flow control in and out of the cylinder are available in flow rates up to 300 lpm and pressures up to 5,000 psi (e.g. 3CA300 - Check Valve). The selection of the valve size will depend on the pressure drop across the valve spool if the valve is opened as well as the opening and closing characteristic during transition. The selection of these components has an impact on critical pump characteristics (weight, packaging, costs). Finding the optimum combination of components is important to optimize the design.

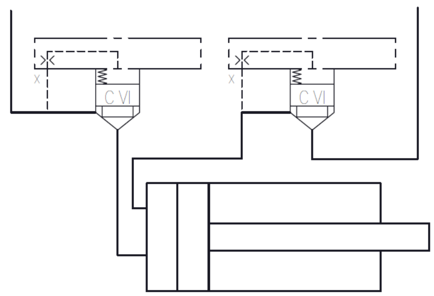


Figure 3. Hydraulic schematic for valves and cylinder

1.2.1.3 Shaft Connection Mechanism

The most critical component for the selected pump concept is the shaft connection mechanism. The use of the wind turbine low speed shaft for this pump will require a shaft connection mechanism which will translate the shaft rotation into a hydraulic power generation motion, but at the same time be robust against bending forces and torques applied to the driveshaft. These loads are caused by the weight of the hub and the blades acting on a lever and by cyclic drive shaft loads. Two principal connection types have been identified for the radial piston pump. A crank shaft type mechanism has commonly been used for early radial piston aircraft motors or for internal combustion engines. The second mechanism uses a roller lifter connected to the piston rod which travels on a cam. Both concepts have advantages and disadvantages. The ability of these mechanical connections to cope with the operating condition of the rotor shaft will be a key criterion to design a robust pump. The connection will have to be designed to allow for shaft bending and additional loads other than the driving torque from the rotor.

A cam shaft design has been chosen for this pump concept. To build a compact design with short cylinder strokes (cylinder length, outer radial pump dimension), multiple lifts of a cylinder piston per revolution are required. Multiple lifts per revolution cannot be obtained with a single crank shaft design. The pump will use a cam design with multiple lobes, which will allow actuating the cylinders multiple times per revolution and therefore allows for a more compact design. To minimize the diameter of the cam profile for manufacturing cost limitation purposes and to allow an easy accessibility of the cylinders, an external cam shaft design has been chosen.

The design of a cam-follower system for mega-watt wind turbines that can have a service life of 20 years represents a technical challenge given the contact pressures associated, the large mass of the components in contact and the unknown dynamic effects. The follower needs to be kept in rolling contact with the cam at all times and operating conditions to avoid premature surface wear issues.

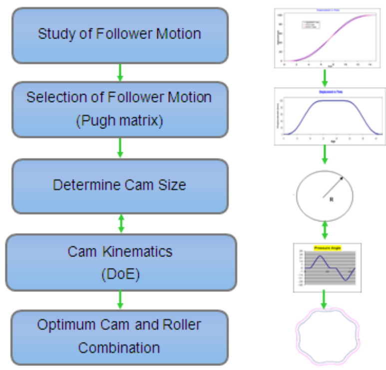


Figure 4. Design process for cam design

For identifying the cam layout, the follower motion has been studied for different cam profiles. Each of the main concepts has been modeled to understand the best profile to choose. After comparison the best cam profile has been selected based on the sum of weighted (priority) characteristics. After the profile has been known, the cam size has been selected based on the required inner radius of the cam: The lower limit of the inner radius of the cam is a function of the outer rotor shaft radius and the required material thickness of the selected cam material for carrying the torque and loads. The upper limit of the inner radius of the cam is given through a compact design. With the selected cam radius and the cam profile, the cam kinematics has been derived. With the specified roller radius for the follower assembly the performance characteristics has been validated.

The identified initial cam design has to be modified to withstand misalignments between the cam and the roller/follower assembly. To characterize the dynamic component interactions within the pump and between the pump and the wind turbine system, a multi-body dynamic model was generated, which used the cam profile and the dimensions of the roller elements that form the followers, in order to evaluate transient and steady state responses of the cam-follower mechanism.

Load bearing calculations have been performed to determine the size of bearings required to take the cylinder loads and survive for a design life of 20 years. Bearings have been identified and selected. The calculated basic rating life according to ISO 281:1990 with simplified mean load calculation for the bearing loads exceeds the wind turbine operating lifetime (20 years) under the following assumptions: assumed life time operating load conditions (30% rated speed/full load, 30% reduced speed/half load, 30% low speed/low load, 10% no load). These assumptions need to be challenged due to the new level of knowledge gained through the load study (severity of loads under which conditions) and the performance study, which gives more information regarding the assumed operating conditions (time under load conditions). A bearing sizing study needs to be performed to validate the assumptions.

Scalability
At higher turbine ratings, multiple cams can be mounted on the low speed shaft, engaging multiple banks of cylinders as pumping elements. This design provides sufficient modularity to enable scaling of pump displacement to the required power rating.

O&M Tasks
The cam and the roller bearings in the pump are sized for a design life of twenty years. The other elements in the pump, such as the cylinders, valves, hoses and fittings will require regular maintenance. The pump is designed to enable quick replacement of these components without the need for an external on-site crane.

1.2.1.4 Load Analysis
The load profile for the low speed shaft needs to be characterized to design the connection mechanism appropriately to increase the reliability of the pump. A shaft load study has been performed to characterize the conditions for the mechanism appropriately (see analysis section). Torque spikes of the rotor shaft caused by wind gusts will create pressure spikes within the hydraulic system. These spikes will be limited through pressure limiting valves and dampened through the high inertia mass of the hydraulic fluid in the long transmission lines (length of tower height) and the accumulator in the system. This will reduce damaging pressure levels on the pump side while separating these loads from the hydraulic motors and the generator. A simulation model has been set up to instigate the pressure spike reduction and has been connected for co-simulation with the Wind Turbine Generator dynamic model (see section analysis).

1.2.2 Controller Design

There are usually three regions in a typical wind turbine, which are region 1, 2 and 3 respectively (shown in Figure 5.). In Region 1, the wind speed is lower than the cut-in wind speed; the wind is not powerful enough so the turbine is not running in this region but ready to run. Region 2 is the region where the wind speed is between the cut-in wind speed and the rated wind speed. In this region, the turbine is controlled in the way so that it runs at the optimum tip speed ratio all the time and the maximum rotor power coefficient can be achieved. The rotor pitch angle is at the fine pitch angle in this region. At the rated wind speed, the turbine reaches the rated power. When the wind speed is between the rated wind speed and the cut-out wind speed, the turbine enters into region 3. In this region, the pitch controller takes effect to maintain the turbine output at the rated power level when the wind speed is above the rated wind speed. When the wind speed is above the cut-out wind speed, the wind power is so powerful that the whole turbine needs to be shut down for safety reasons.

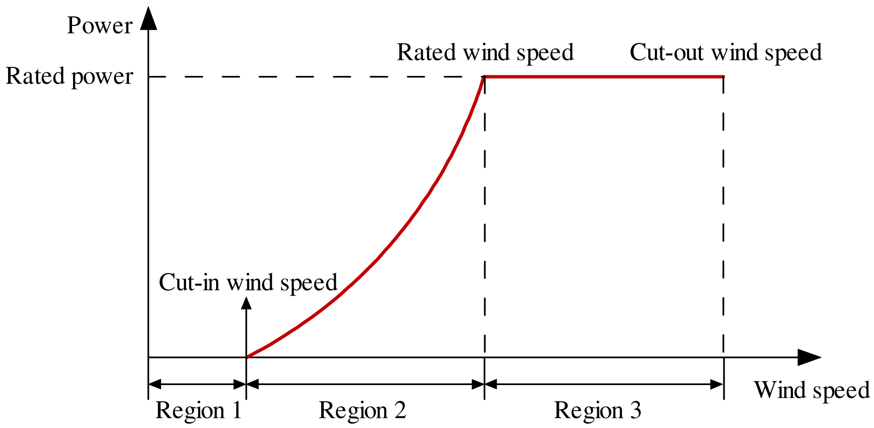


Figure 5. Different regions of a typical wind turbine

In the actual condition, the wind speed changes temporally and spatially all the time. Moreover, the speed of the wind that hits the turbine can vary significantly across the rotor plane. Since the wind speed measurement is usually not accurate enough, it is not practical to use the wind speed information in the turbine control. The rotor speed measurement is usually the only measurement used in the actual turbine control. The control block diagram of a typical wind turbine (with gearbox transmission) is shown in Figure 6.

There are two controllers in the turbine: pitch controller and torque controller. The pitch controller takes the rotor speed error (difference between desired rotor speed and actual rotor speed) as input and generates the pitch angle command to the pitch system. The pitch system usually consists of pitch motors which actuate the blades. The pitch system takes the pitch angle command and turns the blades to the desired angle. In region 2, the pitch angle command is the fine pitch angle. In region 3, the pitch angle command changes with the rotor speed error.

The torque controller takes the actual rotor speed and generates the torque command to the power converter. The power converter takes the torque command and controls the generator torque through the power electronics. This generator torque gives the rotor reaction torque through the gearbox. In region 2, the torque command is proportional to the square of the rotor speed (known as "kw^2" control). This control makes the rotor work at the optimum tip speed ratio all the time and therefore maximizes the power capture in region 2. In region 3, the torque command is set at the rated generator torque. This makes the generator run at the rated generator power.

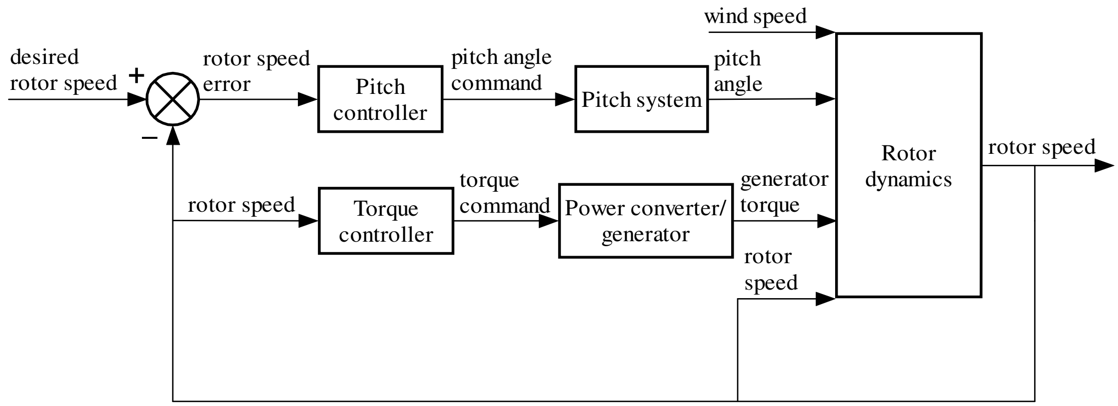


Figure 6. Control block diagram of a typical wind turbine (with gearbox transmission)

The turbine control changes when the hydrostatic transmission is applied to the wind turbine. The turbine control with a hydrostatic transmission is almost the same as the turbine control with gearbox transmission except for the "Power converter/generator" block in Figure 6. In the hydrostatic wind turbine, instead of controlling the generator torque through a power converter, the rotor reaction torque (also pump torque) is controlled by the line pressure. By changing the motor displacement, the line pressure as well as the pump torque can be adjusted. The block diagram of the pump torque control is shown in Figure 7.

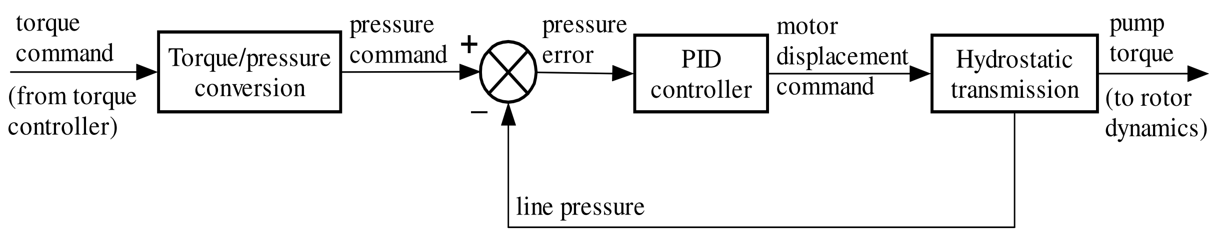


Figure 7. Block diagram of the pump torque control in the hydrostatic wind turbine

The relationship between the torque command and the pressure command is:

$$p_c = T_c \frac{\eta_{pm}}{D_p} \tag{1}$$

where T_c is the torque command, p_c is the pressure command, D_p is the pump displacement, η_{pm} is the pump mechanical efficiency. The accuracy of converting the torque command to the pressure command is determined by the pump mechanical efficiency. As it is discussed in the pump model, the pump mechanical efficiency is a function of the pressure and the speed. In the simulation model, the rotor speed and the line pressure are used to give a more accurate pump mechanical efficiency.

1.3 Key Technical and Performance Metrics

This section discusses the most important metrics by which each component or sub-system's performance has been measured. All of the values of these metrics have been used to size the HST components. The following table shows the key performance metrics for the HST. Brief explanations of each metric follow.

Table 2. List of key performance metrics

Sub-system	Key Performance Metrics
Pump	Efficiency, Speed, Pressure Rating, Flow & Reliability
Motor	Efficiency, Output Speed, Pressure Rating & Reliability
Fluid Conveyance	Flow Velocity (Reynolds Number), Pressure Drop
Charge Pump	Efficiency
Cooler	Cooling Capacity (Power Dissipated)
Filter	Particle Count, Flow Capacity
Generator	Efficiency, Speed, Power, LVRT, Flicker & Harmonics
Accumulator	% Flow Ripple Reduction

Pump: The pump converts the rotary motion of the low speed shaft into fluid flow. In fulfilling this function, the pump has to be able to operate at high efficiency (to minimize parasitic losses), handle the input speeds from the shaft, and generate the flow required to drive the motors (and generators) at the required speed. As such, the key performance metrics for the pump are: *efficiency, input speed, pressure capability and flow.* A related performance metric is the *reliability* of the pump – this is addressed in the "3.1 Pump Design Risk" section of this report. The pump efficiency depends on the leakage and frictional losses. In general, the leakage losses in the commercial cylinders we have selected are extremely low (of the order of 20 – 60 cc/h). By optimizing the seal material and cylinder surface properties, the frictional losses can be minimized.

Motor: The function of the motor is just the opposite of the pump – it converts fluid flow into mechanical energy that drives the generator. Therefore, the same performance metrics apply to the motor as the pump: *efficiency, output speed, pressure capability and reliability.* The motors we have chosen are COTS – Commercial, Off The Shelf. Therefore, catalog data exists relating to all of these metrics. The only life-limiting components of the motors are the shaft bearings. The rest of the motor is designed for infinite life, assuming proper maintenance of fluid quality. Replacement of the bearings is accounted for in the O&M costs of the system.

Fluid Conveyance: This sub-system, comprising several elements such as hoses, fittings and tubing, transports the working fluid between the pump and motor. Losses in this sub-system are comprised of pressure drops and frictional losses. The performance metrics that determine the magnitude of these losses are: *flow velocity (Reynolds number) and pressure drops.* The number of "discontinuities" in the fluid conveyance, such as bends, valves and fittings has been minimized in order to keep the losses to a minimum. Fluid lines have been sized to keep the Reynolds number in the laminar range to avoid excessive losses.

Charge Pump: The charge pump supplies flow to make up for leakage losses in the HST, as well as pressurized fluid to power the displacement control mechanism of the motors and the suction stroke of the pumping cylinders. The power required to perform this function counts as a parasitic loss on the overall system. It is critical to keep the efficiency of the charge pump as high as possible. The performance metric of this component is thus *overall efficiency.* This has been achieved in the proposed HST by specifying a vane pump (as opposed to a gear pump) for higher efficiency.

Cooler: Heat generated in the HST system as a result of losses is dissipated via a cooler. The working temperature of the fluid is thus controlled to extend the life of the fluid and keep its viscosity from fluctuating out of the optimum range for highest efficiency of the pump and motors. The performance metric of the cooler is thus *cooling capacity (or power dissipated).* The HST system has been designed to take advantage of the heat dissipation occurring in the fluid conveyance system.

Filtration: Fluid cleanliness is extremely important for an HST, as contaminants in the working fluid can lead to premature wear and failure. The filtration level is determined by an ISO code that specifies the number of particles of a certain size, for example, 5 microns, that would be allowed through the filter. It is also important to select filters that have sufficient flow capacity in order to avoid an excessive pressure drop. The performance metrics of the filtration system therefore, are the *particle count* per its ISO filtration code *and flow capacity.*

Generator: The key performance metrics of the generator are: *speed, efficiency, power, LVRT, flicker and harmonics.* Speed, efficiency and power relate to the normal operating condition where electrical power is being generated at the required frequency with minimum losses. LVRT, flicker and harmonics are metrics related to abnormal conditions that have a detrimental effect on the quality of the power.

Accumulator: The pump in our HST has discrete pumping chambers, therefore, the output flow will have a waveform commonly called a “ripple”. The ripple causes a time-varying increase and decrease in pressure (and losses), and can have other effects such as noise. It is desirable to minimize this flow ripple to achieve a smooth flow. This is typically done through the use of an accumulator. The key performance metrics of such an accumulator would be *% flow ripple reduction*, as measured by the flow ripple before and after flow passes through it.

2. Analysis Results

2.1 Technology Readiness Level (TRL) Analysis for Major Sub-systems

The following table lists the TRL of major sub-systems in our proposed HST. Brief explanations of the assigned TRL’s follow.

Table 3. TRL of Major Sub-systems

Sub-system	TRL
Pump	2
Check Valves	9
Motor	9
Fluid Conveyance	9
Swivel Joint	2
Hydraulic Fluid	9
Charge Pump	9
Cooling System	9
Oil Filters	9
Generator	9
Power Electronics	9
Accumulator	9
Torque Couplers	9
Sensors	9

Pump: To mitigate potential risks, the pump is designed from COTS components where ever possible. Preliminary modeling studies and calculations demonstrate feasibility of the proposed design. The cam-based actuation mechanism requires high-precision machining to control surface quality. The contact area between the cam surface and cylinder follower must be well-lubricated in order to keep Hertzian stress low. A TRL of 2 is assigned to the pump.

Check Valves: Commercially available slip-in cartridge check valves with the required flow capacity were identified within Eaton’s portfolio. No design modifications or technology validation is necessary to use these components in our proposed HST system. Therefore, these valves have a TRL of 9.

Motor: Hydrokraft motors are currently used in many commercial heavy duty applications, at the same pressure and flow conditions required in our proposed drivetrain solution. No design modifications or technology validation is necessary to use these components in our proposed HST system. Therefore, a TRL of 9 is assigned to the Hydrokraft motors.

Fluid Conveyance: The proposed solution uses only commercial of-the-shelf components. The selected hoses are regularly used in various applications requiring similar flow and pressure requirements. The large diameter tubing proposed for the on-ground configuration is commonly used in the oil and gas industry. No design modifications or technology validation is necessary to use these components in our proposed HST system. Therefore, the TRL level of the hoses and the piping is 9.

Swivel Joint: The current design of the fluid lines for on-ground configuration will require development of a new swivel joint. However, this is not anticipated to be a major technology development effort. The existing swivel joints used in the mobile, off-highway construction machinery market can be scaled up to enable connection to large diameter pipes. Therefore, a TRL value of 2 is assigned to the swivel joint.

Hydraulic Fluid: There is commercially available oil that can be used in our system. Therefore a TRL of 9 is currently assigned. However, the oil tends to degrade over time under operational stress, and will need to be replaced every 3 to 6 years. Development of oil that can last the entire lifetime of the turbine is required.

Charge Pump: The selected charge pumps are currently being used in commercial applications at the required specifications. Therefore, a TRL of 9 is assigned.

Cooling System: Commercially available air-oil coolers of the required capacity were identified. No design modifications or technology validation is necessary to use these components in our proposed HST system. The oil coolers are therefore assigned a TRL of 9.

Oil Filters: The filters selected are currently being used in situations that are anticipated in our proposed solution. No design modifications or technology validation is necessary to use these components in our proposed HST system. We have therefore assigned a TRL of 9 to the filters.

Generator: The squirrel cage induction generator has been successfully used in wind turbines built using the 'Danish Concept'. No design modifications or technology validation is necessary to use these components in our proposed HST system. Therefore, a TRL of 9 has been confidently assigned to them.

Power Electronics: Like the squirrel cage induction generator, the soft starters and the capacitor banks that comprise the power electronics in our proposed system are mature technologies, and hence are assigned a TRL of 9.

Accumulator: The accumulator that we propose to use in order to control the flow ripple from the pump is a mature technology that is widely used in the commercial hydraulics industry. No design modifications or technology validation is necessary to use these components in our proposed HST system. Therefore, a TRL of 9 is assigned to the accumulator.

Torque Coupler: A Rexnord torque coupler with an elastomeric element for insulation is selected in our system. The coupler is a commercially available product and is therefore assigned a TRL of 9.

Sensors: The system conditions that are required to be monitored are within the realm of sensors required in many industrial and mobile applications of hydraulic systems. No design modifications or technology validation is necessary to use these components in our proposed HST system. Therefore, a TRL of 9 is assigned to the sensors.

2.2 System Performance Modeling Results

The University of Minnesota Eolos team was tasked with determining loads that would be present on a 2.5 MW turbine. This information has been used to help guide the initial conceptual design of the hydrostatic drivetrain and supporting structure.

From the literature review, the team found unsteady loading conditions for any given WTG is difficult to calculate due to the following issues.

- Highly variable incoming wind conditions over time and space of the rotor
- Variable control algorithms
- Variable WTG geometries

For these reason, the literature mainly utilized tools available for calculating these loads. These numerical tools includes FAST (Fatigue, Aerodynamic, Structures, and Turbulence) and other software developed by the National Renewable Energy Laboratory (NREL). These tools model incoming wind, the wind turbine structure, including blade lift/drag, and allow easy manipulation of control strategies for generator torque, blade pitch and yaw.

In the publicly available literature, there are three turbine sizes for which loading was available: 750kW [Oyague, 2011], 1.5 MW [Poore, 2000], and a 5 MW [26]. These reports use FAST and the commercial code ADAMS (Automatic Dynamic Analysis of Mechanical Systems) coupled with AeroDyn.

For the current project, FAST modeling was chosen to provide the rotor shaft loading for a 2.5 MW rated wind turbine. A geared generic baseline turbine was first modeled, followed by a model implementing the HST. This approach led to a non-proprietary model with control laws that were easily modified. This was critical when modifying the models to contain an HST.

Loading cases were determined from [40] and slightly modified. The loading results were compared to measurement results and determined to be generally in good agreement. The loading results included ultimate, fatigue, and torque histogram loads.

After performing the baseline geared modeling result, the team investigated applying an HST in the FAST model. After determining the necessary modification to the control and correct method of implementation, the same loading results were found for the HST based wind turbine. The control algorithms for both the geared and HST equipped wind turbine used the same control laws with regard to the low speed shaft speed and blade pitch.

There was no appreciable difference between the geared and HST equipped turbine. More in depth modeling is required after initial design to ensure maximum loads are determined. The generalized 2.5 MW model were also used in the annual energy production (AEP) calculation in other sections of the report.

2.2.1 Current State of Hydrostatic and Other CVT Technologies for WTGs

As of spring 2012, there are few operating grid-connected hydrostatic drivetrains operating. This is likely due to the advancement of power electronics during the 1980s and 1990s [8]. Power electronics allow variable hub speeds used to maximize aerodynamic efficiency at a variety of wind speeds. The multi megawatt turbines are dominated by variable speed machines. This removed the need for a continuously variable transmission to achieve maximum aerodynamic efficiency while maintain AC grid quality output power. Chapdrive and Mitsubishi/Artemis are the major active companies with Mitsubishi, using Artemis hydrostatic technology, appearing to make a major breakthrough with an onshore 7MW prototype deployment in the UK in 2013. Chapdrive currently operates a 900 kW wind turbine with a hydrostatic transmission. The Schacle-Bendix Turbine in southern California was the largest turbine of its time and used a hydrostatic system to achieve variable speeds to maximize output at a variety of wind speeds, but was a failure. Voith Windrive® uses a hydrodynamic device similar to a torque converter to achieve variable speeds; however, this should not be confused with hydrostatic transmissions. Mechanical CVT's are also discussed as they provide variable rotor speed with a fixed generator speed, which a shared major benefit with hydrostatic transmissions.

2.2.2 Loading Literature Review Conclusion

From the loading literature review, the team found that the best approach to determine wind turbine loading is with the use of FAST model. This publicly available model offered to expedite the process of determining loads. Design load conditions in both GL and IEC 61400-1 would be used for simulation, analysis, and reporting. IEC 61400-4 would be used to provide more transmission based reporting. This approach leads to a loads analysis aimed towards determining the anticipated loads seen in a HST transmission in a wind turbine. The approach is similar to that of Poore, 2000, where a baseline geared wind turbine was used to determine preliminary loading on other drivetrain possibilities.

2.2.3 Conclusions from Loading Results

The design loads in the previous section provide a guide for the initial conceptual design of the drivetrain and supporting structures for a wind turbine at the 2.5 MW rating. Design load cases outlined in IEC 61400-1 provided wind models to be used in the loads analysis. For the first phase, load cases outlined for normal operation of the wind turbine were chosen. Transient and fault occurrence loads were not modeled. The model developed uses one DOF enabled, generator speed. With additional engineering in the development of the turbine controls, additional DOFs should not alter the ultimate loads significantly.

The load results performed using FAST were compared with the OEM production loads. The design loads were in good agreement and validated the baseline model as a tool for estimating drivetrain structural forces and moments. A loads analysis was also performed on an equivalent wind turbine with a HST. When comparing the results of the HST version to that of the baseline, geared wind turbine, the differences were minimal. Further, when observing the time series output of the moments about the *y* and *z* axes at the time at which the maximum values occurred, it was clear the loads where not in phase. Because of this, differences in loads are likely do to the differences in rotor position between the two models. A blade may be in an unfavorable position when a gust of wind occurs, driving a larger ultimate load.

Additional load cases are required for the mechanical design of the drivetrain. A more exhaustive loads analysis is needed to fully comply with industry standards. Due to the iterative process in control design and the resulting changes in ultimate loads, this step should be completed when the drivetrain design is at a more mature state.

2.2.4 Overall Conclusion and Next Steps

The loading results obtained thus far show there are no major detrimental loads added to a 2.5 MW WTG by adding a HST with similar control strategies to the baseline. Adequate loads for the LSS have been given for initial conceptual design. Modifying the control strategies for the HST to maximize the unique continuously variable nature is an important step prior to finalizing these loads. Once new control strategies have been finalized, loading simulations should be completed with the goal of balancing structural forces and moments and maximizing AEP. The FAST simulations outlined here create an excellent starting point for this analysis. Though, higher fidelity models for the HST system and the wind turbine, by adding DOFs, are needed to fully understand the interaction of the two systems.

2.2.5 Simulation Model of Hydrostatic Wind Turbine

This section describes the development of a dynamic simulation model for off-shore, utility scale hydrostatic wind turbine in MATLAB/Simulink. The model is a physical equation based dynamic simulation model which intends to simulate both the quasi-static and the dynamic conditions. The input and output causality of each component is verified by the Bond graph method. To apply the hydrostatic transmission to a wind turbine, a torque/pressure controller is specifically designed to control the hydrostatic transmission. To maximize the energy capture, some parameter sensitivity studies are also conducted. To compare hydrostatic wind turbine with commercial gearbox wind turbine, the Clipper C96 2.5MW turbine is selected as the baseline.

In the hydrostatic wind turbine model, the rotor uses the same rotor torque and speed data as the Clipper C96 2.5MW turbine. The hydrostatic transmission consists of a large fixed displacement radial piston pump, six identical variable displacement piston motors, six fixed displacement charge pumps with a charge pump attached to each variable motor, and two long pipelines connecting the pump in the nacelle and six variable motors on the ground. There are six identical induction generators which are also on the ground. Each generator is driven by one variable motor independently. The schematic diagram of the hydrostatic wind turbine is shown in Figure 8. Please note that the final configuration as reported in earlier sections has five motors and five generators. This is the result of a design iteration performed after this simulation study was done. The AEP as reported in this section is not anticipated to change significantly for this new configuration, as the operating pressure at rated conditions increases by only 500 psi.

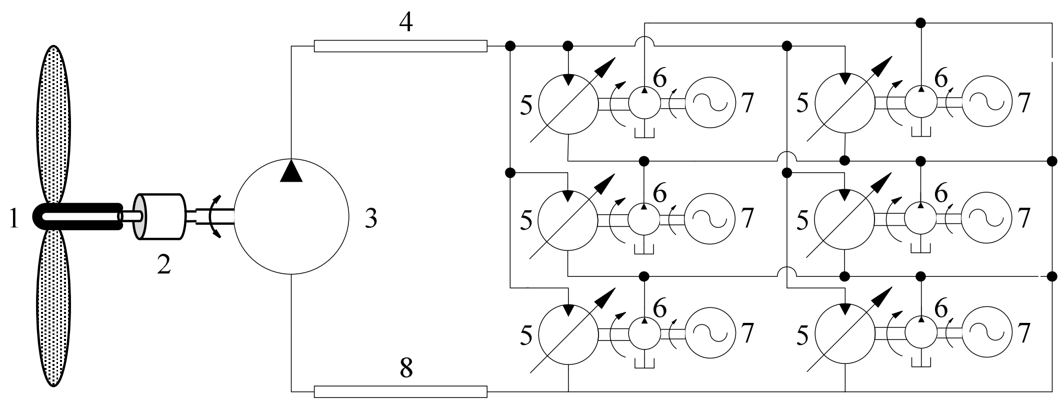


Figure 8. Schematic diagram of the hydrostatic wind turbine
(1. rotor 2. rotor shaft 3. fixed displacement pump 4. high pressure line
5. variable displacement motor 6. charge pump 7. induction generator 8. low pressure line)

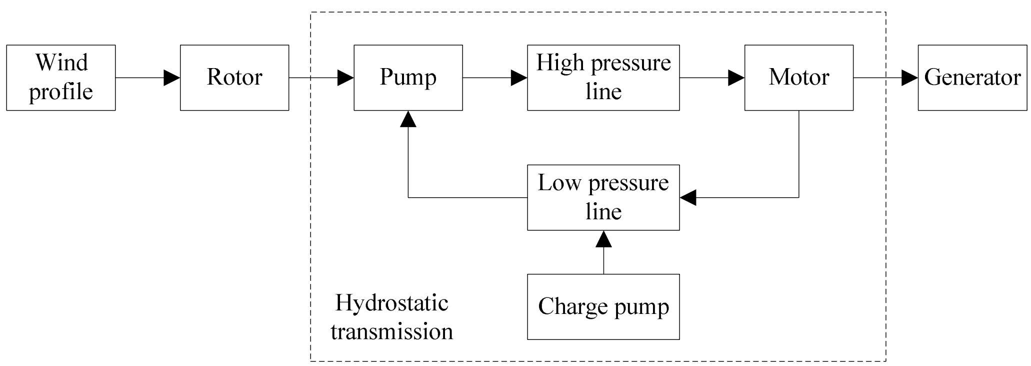


Figure 9. Power flow of the hydrostatic wind turbine

The power flow in the hydrostatic wind turbine is shown in Figure 9. The rotor captures the wind power and turns it into rotor shaft power. The pump takes the rotor shaft power and turns it into hydraulic power. The high pressure line transmits this hydraulic power to hydraulic motor. The hydraulic motor converts majority of this hydraulic power into motor shaft power. The generator turns the motor shaft power into electric power. The charge pump supplies additional hydraulic power to make up for the power losses due to components efficiency and line transmission losses. Each component model will be explained in detail in the following sections.

2.2.5.1 Wind Profile

Wind speed varies all the time and so is the wind power, which varies with the cube of wind speed. Not only is the wind speed changing all the time, its direction also changes with time. Wind speed also varies spatially. An accurate representation of wind speed across the wind turbine blades therefore involves specifying both spatial and temporal variation of wind speed magnitude and direction.

The AEP calculation, described in detail in later chapter, is based on a statistical assumption where a wind regime is divided into many steps or intervals and a histogram of hourly mean speed distribution is generated for a full year. For any given wind speed, it is assumed that during any particular hour of its occurrence, the speed remained constant. Although in reality it will vary about that value and the variance is dependent on the level of turbulence intensity at the point of measurement. Thus for the AEP calculation, wind speed is assumed to be constant during turbine simulations.

2.2.5.2 Rotor

The turbine blade converts the wind power into rotor shaft power. This wind induced rotor torque is dependent on the incident wind speed seen by the blade and the angle of attack, which in turn are dependent on the wind speed (including direction) hitting the rotor plane, rotor speed and the blade pitch angle. Thus the rotor torque is a function of wind speed v, rotor speed ω and pitch angle α and this can be expressed as:

$$\tau_{wind} = \frac{\rho A v^3 C_p(v, \omega, \alpha)}{2\omega} \tag{2}$$

where ρ, A, C_p represents air density, rotor swept area and rotor power coefficient respectively. In the rotor model, the rotor torque is represented as a 3-D lookup table which takes the wind speed, rotor speed and pitch angle as input indices and calculates the rotor torque. Since the input index only has limited discrete values, interpolation and extrapolation are used to find the intermediate data points.

2.2.5.3 Hydrostatic Transmission

To convert the rotor shaft power to electric power, a continuously variable hydrostatic transmission is set between the rotor and the generator. It mainly consists of a large fixed displacement radial piston pump, six variable displacement bent axis piston motors and six fixed displacement charge pumps, which is shown in Figure 10. The hydrostatic transmission is designed as a closed circuit, which means the output flows of the motors are directly feed back to the pump inlet and there is no need of reservoir. This makes the whole transmission more compact. Charge pump is required in the closed circuit hydrostatic transmission to make up for the power losses due to pump and motor efficiencies and line transmission losses. The schematic diagram of the hydrostatic transmission is shown in Figure 10, only one motor and one charge pump are shown for simplicity.

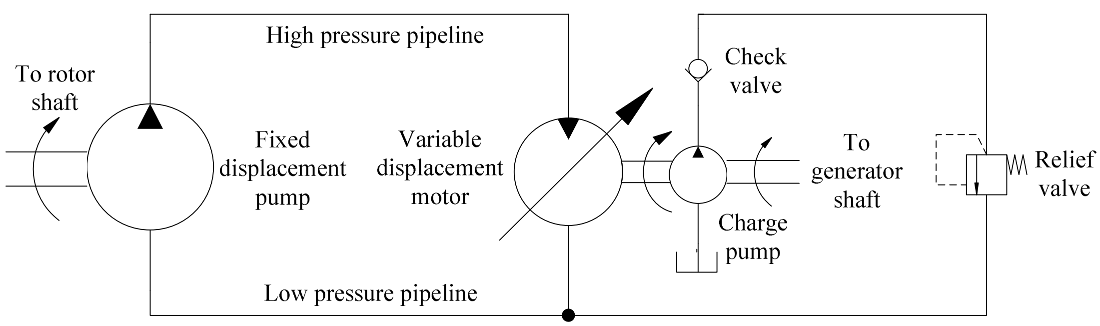


Figure 10. Schematic diagram of the closed circuit hydrostatic transmission (only one motor and one charge pump are shown for simplicity)

The rotor shaft power is first converted to hydraulic power through the pump and then converted to motor shaft power through multiple motors. By using the variable displacement motors, the motor or generator shaft speed is decoupled from the rotor shaft speed so that the generator can run at near synchronous speed all the time. In turbine control region 2, the rotor speed changes with the wind speed so that the optimum Tip Speed Ratio (TSR) and the maximum rotor power coefficient (C_p) can be achieved. The motor displacement is continuously adjusted to adapt the time changing wind speed and the rotor speed.

The hydrostatic transmission consists of fixed displacement pump, high/low pressure pipelines, variable displacement motors and charge pumps. The flow chart of the closed circuit hydrostatic transmission is shown in Figure 11. The components input and output causalities are verified by Bond graph method. Each component model will be explained in the following sections respectively.

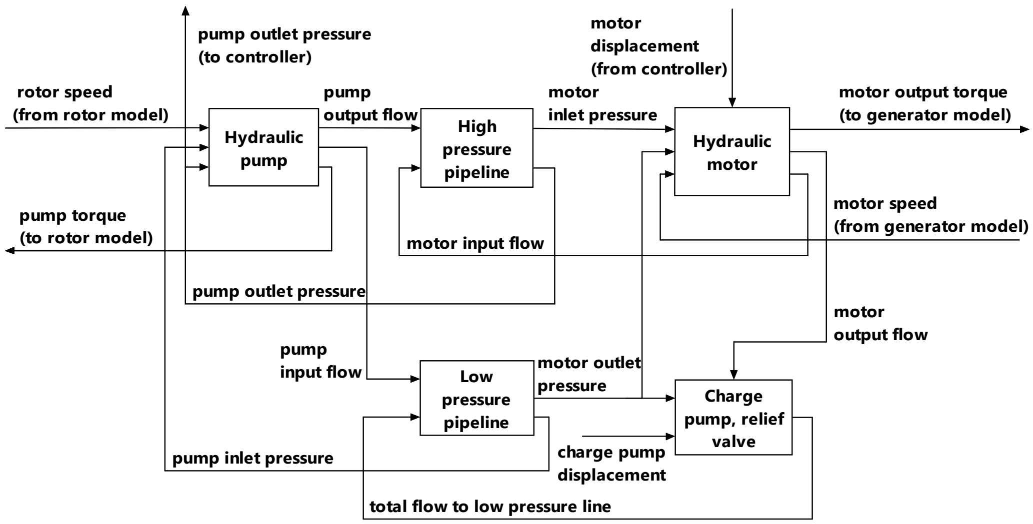


Figure 11. Flow chart of the closed circuit hydrostatic transmission

2.2.5.4 Annual Energy Production Calculation

The annual energy production is a statistical estimate of the expected annual energy production of a turbine for a given a wind speed distribution. To calculate the AEP at a location, the power output *versus* wind speed characteristic of the turbine and the wind speed distribution at that particular site must be known.

With a hydrostatic transmission applied to the wind turbine, the generator speed is decoupled from the rotor speed so that the generator can run at near synchronous speed at different wind speeds. This eliminates the use of power electronics while it is necessary in a wind turbine with gearbox transmission. In a hydrostatic wind turbine, the rotor speed changes with the wind speed so that the optimum Tip Speed Ratio (TSR) and maximum rotor power coefficient (C_p) can be achieved. The generator speed maintains at near synchronous speed while the rotor speed changes. This is accomplished by changing the motor displacement to adapt the different rotor speeds.

To conduct a fair comparison between hydrostatic wind turbine and the Clipper C96 2.5MW wind turbine, some assumptions should be made. It is assumed that both turbines are based on the same rotor size. This means that the rotor torque capability and maximum rotor speed of the hydrostatic wind turbine are the same as the Clipper C96 turbine. In the current study, the hydrostatic transmission is designed such that it can generate as higher AEP as it can, given the same rotor torque capability and same maximum rotor speed. Designing a hydrostatic wind turbine to achieve lower cost of energy (CoE) than commercial wind turbine is a complex work, since the CoE is not just influenced by the AEP, it is also influenced by costs. The cost of energy analysis is out of scope of the current simulation study. The system configuration shown in Figure 8 has already taken some preliminary CoE analysis into considerations. Given that system configuration, the objective of the current simulation study is to maximum the AEP.

Turbine operation points

The AEP simulation results with and without **2A_optimized** are shown in Table 4.

Table 4. AEP simulation results with and without 2A_optimized

	Clipper C96	**HST (Axial piston motors)**	
		Region 2A not optimized	Region 2A optimized
Gross AEP (MWh)	10,251	9219	9264
Net AEP (MWh)	8970	8067	8106
Net CF (%)	41.0%	36.8%	37.0%

Motor efficiency

Bent axis motors are considered to have a higher efficiency than axial piston motors at low displacement fraction. The AEP simulation results with bent axis piston motors and axis piston motors are compared in Table 5. It is shown that by using high efficient bent axis motors, the output power of the hydrostatic turbine increases and therefore the AEP can be improved.

Table 5. AEP simulation results with bent axis and axial piston motors

	Clipper C96	HST WT (Region 2A optimized)	
		With axial piston motors	With bent axis motors
Gross AEP (MWh)	10,251	9264	9406
Net AEP (MWh)	8970	8106	8231
Net CF (%)	41.0%	37.0%	37.6%

In-nacelle and on-ground solution

With motors and generators on the ground, the weight in the nacelle can be reduced. Also it can reduce maintenance cost as it is much easier to do the repair or replacement on the ground. However, due to the long pipelines between pump and motors, the pipeline loss is higher than the in-nacelle solution. The AEP simulation results with in-nacelle and on-ground solution are compared in Table 6. It is shown that the AEP results with two solutions are very close. By studying the pressure drop across the pipeline, it is found that the pressure drops across the pipeline in two solutions are much lower than system pressure, which means it does not influence the system efficiency a lot.

Table 6. AEP simulation results with in-nacelle and on-ground solution

	HST WT (Region 2A optimized, bent axis motors)	
	In-nacelle solution	On-ground solution
Gross AEP (MWh)	9,431	9406
Net AEP (MWh)	8252	8231
Net CF (%)	37.7%	37.6%

Components sizing

With reduced pump and motor size, the line pressure is increased, which improves the efficiencies of both the pump and motors. The AEP simulation results with different pump and motor size are compared in Table 7. It is shown that the AEP is improved considerably by reducing the pump and motor size.

Table 7. AEP simulation results with different pump and motor sizes

	HST WT (Region 2A optimized, bent axis motors, on-ground solution)	
	pump size: 333297 cc/rev motor size: 500 cc/rev (number: 6)	pump size: 422300 cc/rev motor size: 750 cc/rev (number: 6)
Gross AEP (MWh)	9,607	9406
Net AEP (MWh)	8406	8231
Net CF (%)	38.4%	37.6%

Motor operation strategy

When the motor displacement command is given by the turbine controller, there are two different strategies to operate multiple motors. One strategy is to switch the motor one by one, all the other motors run at the full displacement except for the last motor. Another strategy is first to determine the numbers of motors which should be switched on and then distribute the total displacement evenly to these motors. The AEP simulation results with different motor operation strategies are compared in Table 8. It is shown that the difference between these two motor operation strategies is very small.

Table 8. AEP simulation results with different motor operation strategies

	HST WT (Region 2A optimized, bent axis motors, on-ground solution) pump size: 333297 cc/rev, motor size: 500 cc/rev (number: 6)	
	Motor switched one by one	Motor displacement evenly distributed
Gross AEP (MWh)	9,607	9,660
Net AEP (MWh)	8,406	8,452
Net CF (%)	38.4%	38.6%

2.2.5.5 Energy Storage Analysis

Use of energy storage with renewable energy resources is an active field of study. Many different storage technologies have been investigated for use with renewable resources. Energy storage capabilities can alleviate the problems arising out of the mismatch between the demand and production in electric grids with a high renewable energy penetration. Depending on the type of storage technology, the application objective and needs differs from one another. Storage technology having lower energy density but high power density and cycling rates like flywheel, supercapacitor etc have been studied and used for applications like power smoothening, power ramping and decentralized applications. Technologies like pumped hydro and Compressed Air Energy Storage (CAES) have been used for load leveling applications.

The purpose of this storage study is to investigate the use of commercially available hydraulic accumulators to capture more energy than a hydrostatic turbine without energy storage systems. Hydrostatic drives can easily incorporate hydraulic accumulators and also provide convenient energy management capability due to the flexible nature of fluid power transmissions.

To investigate the feasibility of wind energy storage it is necessary to understand the conditions for storage of wind power. The storage opportunities in a wind turbine can be illustrated by the following wind power production profile.

It is seen here that with storage the loss in power due turbulence is made up by the storage system in the wind speed range of 9.5 m/s to 15 m/s. Table 9 compares the AEP of the turbine with and without storage and the sizes of pump/motor and accumulator required to achieve this.

Table 9. AEP comparison with and without storage and the sizes of pump/motor and accumulator

	Turbine without storage	Turbine with storage
Max rotor power (MW)	**2.781**	**3.337 (20% more)**
Generator rating (MW)	2.325	2.325
Accumulator size (liter)	NA	2500
Pump/Motor size for each motor (cc)	NA	250 (1500 in total)
AEP (MWh)	9,429	9,644
% Increase in AEP	**0**	**2.28**

The AEP in this case increases 2.28% due to storage. If we consider only one motor M1 and pump/motor M2 then the required displacement of M2 is 1500 cc and the accumulator size is 2500 liter. This is the optimum accumulator size calculated by a sensitivity analysis.

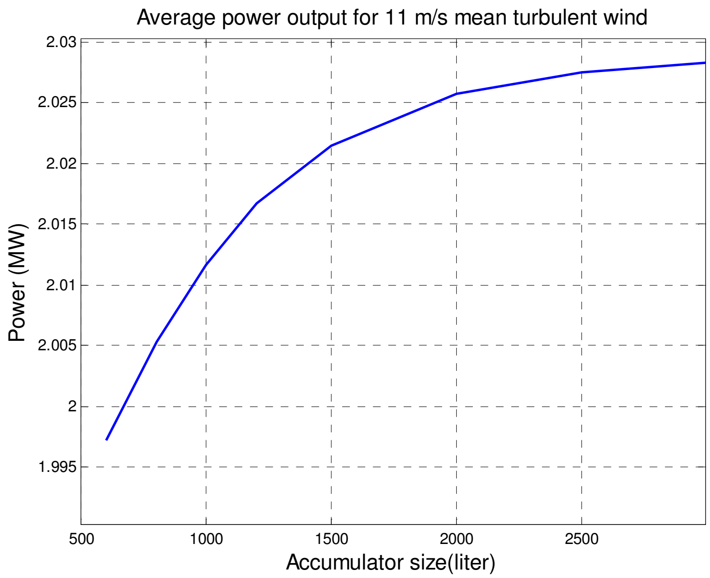


Figure 12. Accumulator size study

To evaluate the optimum accumulator size multiple simulations were run using turbulent wind conditions with a mean of 11m/s and the average power was recorded. The accumulator size was progressively increased and it reached a point of diminishing return. Figure 12 shows the results of this analysis. It can be observed that beyond an accumulator size of 2500 liter, the increase in average power is minimal. Hence an accumulator size of 2500 liter can be considered the optimum accumulator size. This method of energy storage assumes that the turbine can handle 20% more power than the generator and gearbox.

This is equivalent to saying that the generator is downsized. It can be argued that if a larger generator and gearbox that can handle 20% more power then it may be able to increase the AEP without using storage systems. Ultimately the feasibility of this storage system can be determined only by an economic analysis. To handle higher power level during storage, the turbine blades, hub, main shaft and the tower needs to be redesigned. It was difficult to estimate the increase in cost of these upgraded components. To simplify the cost analysis it was decided to keep the maximum power level of the turbine at the rated level while using storage systems and downsize the generator. This way the cost of the turbine components remains unchanged and the only cost of the downsized generator needs to be assessed. The rated generator power level for the hydrostatic turbine is 2.325MW. The generator was downsized to 2 and 1.8 MW and the controller was redesigned to accommodate these changes. The AEP analysis was done for these cases and the results of these studies are presented in Table 10.

Table 10. Generator downsizing study

	Baseline Turbine (Clipper C96)	Turbine without storage	Turbine with storage (13% downsized Generator)	Turbine with storage (22.8% downsized Generator)
Generator rating (MW)	2.5	2.325	2.02	1.794
Accumulator size (liter)	NA	NA	2500	2500
Pump/Motor size for each motor (cc)	NA	NA	166.7 (1000 in total)	250 (1500 in total)
Gross AEP (MWh)	10,251	9,429	8,974	8,423

2.3 CoE Analysis Results

The CoE analysis result is structured into the main contributors of the calculation: Capital Cost, O&M cost, LRC and AEP.

2.3.1 Initial Capital Cost

Component Cost

The initial capital cost (ICC) of the baseline was determined from the latest NREL cost model [14]. The ICC for 5 MW offshore wind turbine was scaled by a factor of 1.34 as recommended in the OSW strategic document [5]. The cost information for a 2.5 MW system is scaled by a factor of 1.3 to adjust for inflation. A study was commissioned by EATON to identify the potential cost saving due to lower weight of the hydrostatic drivetrain. The study, conducted by Frost and Sullivan [4], had the following recommendations,

- Every 1 ton reduction in the tower is a 2-ton reduction in the foundation weight,
- Every 1 ton reduction in the top tower is a 1.6-ton reduction in the tower weight.

In our study we identified the weight of major components in the drivetrain to calculate reduction in cost as recommended by the Frost and Sullivan study. As most components in our drivetrain are commercially available, their weight was identified from the catalogue. For components such as the cam, which are not yet commercially available, the weight was estimated from the solid model of the pump design. Other assumptions in identifying the system weight are,

- The weight of the rotor shaft is based off the Clipper Liberty 96 rotor shaft with material added, which results in a weight slightly lower than the baseline for 2.5 MW. The weight reduction was appropriately scaled for the 5 MW turbine.
- The mainframe weight savings were scaled with respect to the total weight savings in the nacelle not including the mainframe.
- The nacelle cover for in-nacelle configurations is the same size as the baseline configurations. The on-ground solution assumes an 18% material reduction in the nacelle cover as recommended in the 5 MW weight study. This number is appropriately scaled to the 2.5 MW wind turbine.
- The top tower weight is defined as the sum of the nacelle weight and weight of the rotor assembly.
- The foundation weight for the 2.5 MW baseline wind turbine is obtained by assuming concrete density of 2400 kg/cubic meter and a volume of 400 m^3. The foundation weight for the 5 MW baseline is obtained from [4].

Some key highlights for the capital cost comparison between the baseline and Eaton solutions are,

- The cost of low speed shaft is lower for Eaton solution. This is due to material savings.
- The driveline cost of the HST is lower than for the baseline based upon hydrostatic transmission component costs determined from quotes provided by internal and external vendors. The cost of induction generators, capacitor banks for reactive power compensation and cables were obtained from the WindPact study [40] and scaled to the appropriate power level.
- The cable cost for on-ground solutions is lower. This is due to proximity of generators to the transformer. This cost is obtained by considering the length of cable required to from transformer to the tower base.
- The reduction in mainframe cost is due to material savings in our proposed solution.
- It is assumed that the hydraulic cooling system is required in the baseline solution to regulate the temperature of full power convertor electronics. Since our proposed solution does not require full power conversion, this cost is not included in our CoE calculation.
- A smaller nacelle is sufficient for on-ground solution. The corresponding savings in the material required are translated to lower cost.
- As per the recommendations in [4], reduction in weight of tower and foundation is calculated for Eaton solution. The cost associated with tower and foundation is proportionally lowered.

Transportation Costs

The transportation cost for 5 MW offshore wind turbine identified in [4] have been used in our study. The inland transportation distance is assumed to be 30 miles and transport over water is assumed to be 15 miles. It is assumed that it takes an average of nine hours to transport the turbine components from the loading station at the port to the offshore installation point. The total cargo transported in a single trip is determined by the weight of the cargo. Due to the reduced nacelle volume (and weight) for on-ground solution, about six wind turbines of this configuration can be transported at a time, as opposed to five turbines for in-nacelle/baseline configuration. Similarly, about seven foundations for on-ground can be transported in one trip, while about six foundations required for Eaton's in-nacelle configuration can be transported in a trip. Due to the larger volume, about five foundations required for baseline can be transported in one trip.

Transportation calculations for the 2.5 MW On-Ground and In-Nacelle solutions have been based upon the Frost and Sullivan Transportation and Installation study [4]. While the study was made for a 5 MW offshore wind turbine, transportation numbers have been calculated for inland transport covering 30 miles distance. The baseline transportation costs have been derived from the NREL document, reduction due to weight savings assumed to be the same as for the 5 MW calculations.

Installation Costs

For the cost of installing the tower per installation study [4] a 25% downtime and a 45-hour tower installation time have been assumed for the baseline turbine. The baseline costs were adjusted accordingly to accommodate for the difference in our baseline weight and the weight of baseline tower in the Frost and Sullivan study. The cost of installing a tower with Eaton's drivetrain is calculated by adjusting the baseline cost proportional to percentage change in weight of the tower.

For foundation installation costs per installation study [4] it has been assumed that it takes about 30 hours to install the foundation. The assumed timeline includes 25% downtime due to weather conditions. The baseline costs were adjusted to accommodate the higher baseline weight according to the Frost and Sullivan study. Again, the cost for Eaton's solution is calculated by adjusting the baseline cost proportional to change in foundation weight. Wind turbine assembly and installation remains the same between the NREL baseline and the Eaton HST solutions for the 2.5 MW applications.

2.3.2 Operating and Maintenance Costs

The Operation and Maintenance Costs (O&M) are separated into two models, scheduled and unscheduled maintenance based off the WindPACT Advanced Wind Turbine Drive Train Study [40]. Since the WindPACT study was based on a 1.5 MW system, the cost of equipment required for major replacements for this study has been proportionally scaled to reflect the size of turbine in our study. The maintenance labor cost has been assumed to be $65/hour based on the WindPACT study recommendation. It is assumed that both the baseline and Eaton's solution for the 2.5 MW wind turbine incur an operation cost of $10,000. The operation cost for the 5 MW drivetrain is assumed to be $20,000. The maintenance tasks are categorized under scheduled and unscheduled maintenance and their associated cost is appropriately calculated. In the following some of the key assumptions in our calculations are highlighted.

Baseline – Onshore

For the unscheduled maintenance model all assumptions regarding the component costs (linked to capital costs for components), the spare costs (e.g. shelving or expedition costs), spare Weibull average (type of failure), average failure/year, MTBE, MTTR have been kept. The equipment cost includes an onsite crane for heavy lifting at assumed $50,000 per event and an onsite crane for medium weight lifting at assumed $25,000 per event. The WindPACT study assumed gearbox failures to occur every 20 years. Newer studies suggest that gearboxes fail every 7.7 years as identified in the Windpower monthly article 'Breaking down the cost of wind turbine maintenance' [30]. Based on these results, the baseline unscheduled maintenance model has been updated to reflect the higher gearbox failure rate. The generator failure rate has been updated with the gearbox failure rate to be consistent with the WindPACT model and the underlying assumptions for it. This is the reason the baseline O&M costs in the Eaton CoE model are twice as high as the O&M costs in the NREL baseline model.

For the scheduled maintenance model all assumptions regarding events/year, intervals, duration/event, number of crew and material/event have been kept. No special equipment is assumed to be necessary.

Baseline – Offshore

For the unscheduled maintenance model for offshore applications, assumptions have been updated to reflect the change in drivetrain size (5 MW instead of 2.5 MW) and the location on the sea. The location on the sea requires additional time to travel and higher equipment costs due to boat rental. The assumptions for these costs have been based on the Frost and Sullivan study [4] where boat rental costs and travel time to/back from the wind turbine installation site have been identified. The updated equipment costs reflect the rental of a ship for the identified time to repair the component. The mean time to repair (MTTR) reflects the time it takes to get to and return from the wind turbine site (30 miles distance).

For the scheduled maintenance model the assumptions for the onshore model have been updated regarding the duration/event to reflect the larger power scale (size of drivetrain) and the time to travel from a harbor to the wind turbine and back to the harbor (portion of overall travel time for maintenance items that can be done in parallel). The material costs reflect the larger power scale and the special equipment costs reflect the rental of a ship to get to the wind turbine site (30 miles distance).

HST Drivetrain

The cost incurred for maintenance of the turbine components, excluding the drivetrain, is assumed to the same as baseline. For evaluating the maintenance cost of the drivetrain, the required maintenance tasks and their expected duration was identified. The labor costs are assumed to be same as the baseline. The maintenance strategy for our drivetrain is preventative. By scheduling more maintenance tasks we anticipate fewer unexpected failures. While this would increase the cost of scheduled maintenance, the unscheduled maintenance and the levelized replacement costs of the drive train are lowered. This proactive maintenance strategy is facilitated by the modular design of our drivetrain, whereby most components of the system can be serviced using an onboard crane. Subsystems with an anticipated MTBR of ten years or more are assumed to require unscheduled maintenance. Maintenance events for all other subsystems are scheduled as per the identified requirement. Some key highlights of maintenance tasks are,

- Pump: cylinder seals and bearings are included in the unscheduled maintenance of the pump. The pump is using bearings on in the cam/follower assembly designed to meet 20 years of operation. The failure rate of pump is assumed to be as high as the failure rate of the rotor shaft (LSS) wind turbine main bearing.
- Failure rates for the charge pump, swivel fittings, valves, sensors, charging station electric drives and coupling between motor and generator are assumed based on Eaton's component knowledge in similar hydraulic system applications.

Typical assumptions for hydraulic system maintenance regarding oil change, hose replacement and motor life (bearings).

- The load calculation on the motor shaft bearings for the simulated duty cycle suggests a bearing life of 3.3 years. Therefore, it is proposed to exchange the motors once every 3 years. This task can be executed using an on-board crane. The bearings and the shaft seals are replaced off-site at a repair facility. These refurbished motors will be used in subsequent replacement cycles.
- It is recommended that the hydraulic oil be replaced once every 3 years. This can be doubled by maintaining an inert atmosphere in the reservoir, thus preventing oxidation and the associated degeneration of oil. The inert atmosphere is maintained by filling the reservoir with nitrogen.
- The hydraulic hoses are scheduled to be replaced once every five years. It is assumed that filters will require replacement once every three years.

The cost for the replacing components is obtained from the bill of materials.

HST Drivetrain – In-Nacelle vs On-Ground

The major differences of the maintenance costs for different drivetrain motor location are the cost of oil exchange (Costs: longer lines, much more fluid required) and the time to exchange the motors in the nacelle (Labor time: onboard hoist, but added time for getting components to tower top and down).

HST Drivetrain – Offshore
For the unscheduled maintenance model offshore, assumptions have been updated to reflect the change in drivetrain size (5 MW instead of 2.5 MW) and the location on the sea. The assumptions which reflect the travel time and equipment required (boat rental) at sea due to the wind turbine location on sea are the same as for the baseline wind turbine.

For the scheduled maintenance model the assumptions for the onshore model have been updated regarding the duration/event to reflect the larger power scale (size of drivetrain) and the time to travel from a harbor to the wind turbine and back to the harbor (portion of overall travel time for maintenance items that can be done in parallel) consistent with the assumptions used for the baseline wind turbine model.

Result
The overall O&M costs for the HST drivetrain wind turbine are much lower than those of the baseline wind turbine. The higher scheduled maintenance costs of the HST system due to hydraulic components maintenance requirements have been offset by lower unscheduled maintenance costs. The high difference between the HST and the baseline wind turbine on unscheduled maintenance cost is caused by high baseline maintenance costs if the rate of gearbox failures is considered.

2.3.3 Levelized Replacement Costs

The Levelized Replacement Cost (LRC) for the turbine includes major overhaul of high cost components of the wind turbine. It is calculated as recommended in Appendix D of FOA [13] as per the following table.

	PV(n)	=	$PVF(n_{mp}) \times RC(2010) \times (1.03)^n$
where:	PV(n)	≡	Present Value of annual stream of reserve fund for event occurring in year (n)
	$PVF(n_{mp})$	≡	Present Value Factor for mid-point year of reserve fund payment stream
		=	$(1 + i)^{-n_{mp}}$
	I	≡	Nominal discount rate = (0.0925)
	RC(2010)	≡	Replacement/Overhaul Cost in year 2010
	Note: in the formula above, 1.03^n is an inflation factor		

	LRC	=	$0.8 \times CRF \times \sum PV(n)$
where:	CRF	≡	Capital Recovery Factor
		≡	$i_{const} / (1-(1+ i_{const})^{-30}) = 0.073$
and where:	i_{const}	≡	Constant dollar discount rate = 0.0607
	Note: the factor 0.80 accounts for depreciation of each replacement (this factor was derived from a utility-scale finance model1.)		

[1] George, K.; Schweitzer, T. (2006). Primer: The DOE Wind Energy Program's Approach to Calculating Cost of Energy. ; NREL/SR-500-37653.

In the HST drivetrain, only the elements of the pump, namely the cylinders, the side load bearings and the cam are included in calculating the material cost for replacement. As mentioned in [40], the material cost for replacement is assumed to be 20% of the initial capital cost. Due to the modularity of our drivetrain, equipment such as an external crane is not required. Therefore this cost is excluded in calculating the LRC of our drivetrain.

2.3.4 Annual Energy Production

Annual energy produced by a 2.5 MW wind turbine with a hydrostatic transmission is calculated as explained in the analysis section of this report. To predict an appropriate AEP, a regression model between the assumed drivetrain efficiency at rated power and the AEP has been used. The drivetrain efficiency at the rated power is recalculated by considering the Hydrokraft motors in the drivetrain and including the parasitic losses. Through the regression model the AEP prediction has been adjusted for drivetrain efficiency. Due to the lower efficiency of Hydrokraft and inclusion of parasitic losses, the net AEP for our drivetrain has been lowered to 7907 MWh. This value for projected AEP is used in calculating the CoE of the hydrostatic drivetrain. It has been assumed that the availability of both baseline and Eaton's solution are the same at 95%.

The AEP for 5 MW baseline wind turbine has been obtained from OSW strategic plan document [5]. The energy produced by the Eaton drivetrain has been obtained by scaling the baseline AEP. The scaling factor is defined as the percentage difference between the baseline and Eaton's solution for the 2.5 MW wind turbine.

2.3.5 Cost of Energy Results

As recommended in the FOA document [45], a discount rate of 7% for a 20 year project life is assumed. The insurance, warranty and fees have been estimated at 1% of the initial capital cost. A tax break of 40% is assumed on the O&M costs. The CoE for the baseline and for the Eaton's solutions is listed in the following table. The CoE factors ($/kWh) below the AEP have been calculated dividing the cost items (in $k) above AEP by the AEP (MWh).

Table 11. CoE comparison between the baseline and Eaton drivetrains.

Representative Categories	Offshore 5 MW Baseline	Offshore 5 MW HST On-Ground	Improvement from baseline	Offshore 5 MW HST In-Nacelle	Improvement from baseline	Onshore 2.5 MW Baseline	Onshore 2.5 MW HST On-Ground	Improvement from baseline	Onshore 2.5 MW HST In-Nacelle	Improvement from baseline
Capital Cost [$k]	18751	16353	13%	16906	10%	4625	4457	4%	4390	5%
Operations & Maintenance Cost [$k/year]	438	191	56%	187	57%	111	50	55%	42	62%
Levelized Replacement Cost [$k/year]	82	58	30%	58	30%	22	17	23%	17	23%
AEP net [MWh]	17905	15858	-11%	15999	-11%	8970	7907	-12%	7974	-11%
Capital Cost COE Factor [$/kWh]	0.0838	0.0825	2%	0.0845	-1%	0.0412	0.0451	-9%	0.044	-7%
O&M COE Factor [$/kWh]	0.0245	0.012	51%	0.0117	52%	0.0124	0.0064	49%	0.0053	57%
LRC COE Factor [$/kWh]	0.0046	0.0036	21%	0.0036	21%	0.0025	0.0022	12%	0.0022	13%
Total System CoE [$/kWh]	0.1128	0.0981	13%	0.0998	12%	0.0562	0.0537	4%	0.0515	8%
CoE Improvement		13%		12%			4%		8%	

The CoE for a wind turbine with the new advanced HST drivetrain can be up to 13% lower than the standard wind turbine with a geared drivetrain solution. The impact of capital cost savings for the HST drivetrain (up to 13%) compared to the baseline has been reduced by the lower AEP due to lower system efficiencies for the proposed low risk off-the-shelf-component solution (11% less efficient), and ranges from 2% cost factor reduction for offshore to 10% cost factor increase. The highest impact on the CoE improvement is caused by the O&M cost savings (improvement 55% - 62%) and LRC savings (improvement 23% - 30%). The savings are based on the higher reliability of the drivetrain compared to the standard gearbox drivetrain and the use of modular lightweight drivetrain components which can be handled for maintenance/repair with the turbine's internal hoist without the need for an expensive onsite crane.

The CoE results for a larger 5 MW offshore wind turbine show a higher potential than the 2.5 MW onshore wind turbine. This is caused by higher capital cost savings of the drivetrain and wind turbine components. For the 2.5 MW onshore wind turbine, the in-nacelle HST drivetrain solution, with an 8% CoE improvement over the baseline, is superior to the on-ground HST solution with 4% CoE improvement over the baseline, caused by higher O&M cost savings. The on-ground HST drivetrain solution requires higher effort and more material to be maintained due to its size and required real estate. For the 5 MW offshore wind turbine, the in-nacelle HST drivetrain solution shows a 12% CoE improvement, and the on-ground solution shows a 13% CoE improvement over the baseline. The higher O&M cost savings of the in-nacelle solution are roughly balanced by the higher capital cost savings of the on-ground solution due to the higher material cost savings on the tower and foundation.

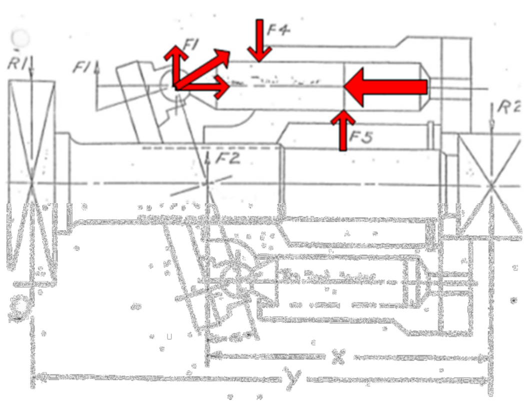
The results obtained by this study are derived with high confidence in its prediction. The obtained improvements in CoE for different sizes of on-shore and off-shore turbines are considerably better than the baseline, especially if considered together with the low technical risk for the alternative drivetrain solution for utility scale wind turbines.

2.4 Performance and Cost Tradeoffs Analysis

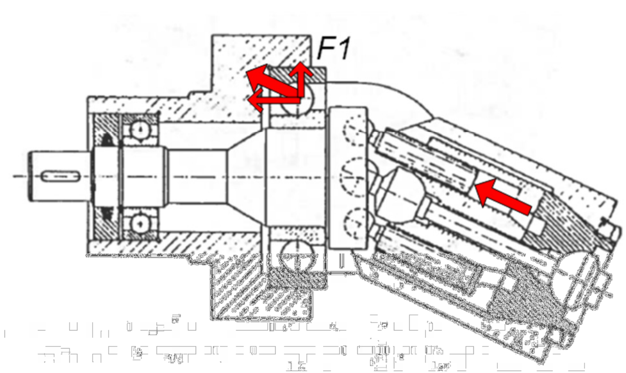
Several trade-off studies were performed during this study to investigate methods of lowering the CoE, either by reducing the cost of the system or increasing the AEP. The following are the major trade-off studies conducted.

On-ground vs. in-nacelle HST configuration: As mentioned earlier in the AEP calculation section, the nacelle weight can be significantly reduced by placing the motors and generators on the ground. Not only the tower and foundation weight can be reduced, but also maintenance costs can potentially be reduced significantly for components located on the ground, since it is much easier to service those components. The AEP analysis showed that the difference between these two configurations is negligible. However, cost analysis showed that in both 2.5 and 5 MW turbines, the annual O&M costs are lower for the in-nacelle configurations. The difference is mainly the significantly higher volume of hydraulic fluid required for the on-ground configuration which has to be periodically replaced. Therefore, from an O&M cost perspective the in-nacelle configuration is clearly better.

Axial vs. bent-axis design motor architecture: The AEP analysis showed a comparison between "axial" and "bent-axis" motors with regard to AEP. These terms refer to the design architecture of the motors as shown in Figure 13. The main difference lies in the resolution of the internal forces on the pistons. In the axial design, there is a significant side component on the piston; in the bent-axis, the side component is taken up by the shaft bearing. These side components are identified by "F1" in both pictures below. This causes a large difference in the mechanical losses, and thus overall efficiency. On average, the bent-axis design is more efficient than the axial design by 5 to 10 percentage points, depending on the operating conditions. However, the construction of the bent-axis unit is much larger, requiring expensive bearing packs for the loads.



Axial Architecture



Bent-axis Architecture

Figure 13. Comparison of the different motor architectures considered in this study

Since no bent-axis design units exist in the displacement required for such a system, the trade-off analysis was limited to AEP prediction based on efficiency data that exists for smaller units in both designs. The development of large, bent-axis motors is one possible technical solution to increasing the AEP; this benefit, being fairly small according to our prediction, would have to be balanced against the cost of development.

Motor sizing: This is a conceptual trade-off performed on two motor sizes, 500 cc and 750 cc. The biggest motor sizes are selected to minimize the total number of required motors. The metrics used in this trade-off analysis are shown in Table 14. As seen in the table, cost of these motors was given the highest importance metric. Since these motors are COTS components in Eaton's portfolio, there is precise cost information available. The second biggest differentiating factor was the speed of operation. This has a large influence on efficiency as well as component durability. Even though the 750 cc unit could provide approximately one year of additional operational life when run at 1240 rpm versus the 500 cc unit at 1860 rpm, the cost difference between the two proved to be decisive in tipping the balance in favor of the 500 cc unit.

Single vs. multiple fluid lines (on-ground configuration): Pressure drop in the fluid conveyance constitute a power loss. This power loss can be lowered by keeping the flow Reynolds number lower. One way to achieve this is to use large diameter pipes. The other possible solution is to divide the flow among multiple lines of smaller diameter. An added advantage of multiple lines the large convective area available for heat transfer. But the large number of connection points was deemed impractical for installation and service. The choice was to use a large diameter pipe to transport fluid between nacelle and ground – especially since a COTS component was available for this function.

Table 12. Cost trade-off study to differentiate between single large diameter line and multiple smaller diameter lines

	CoE improvement for 5 MW on-ground	CoE improvement for 2.5 MW on-ground
Multiple 2.36"ID pipes	11.6 %	4.9%
Multiple 3.35"ID pipes	11.7%	5.0%
Single 6" ID pipe	11.8%	4.5%

Various combinations of motor-generator: In our study, six feasible concepts were identified and they are listed in Table 13. Using the Pugh matrix analysis shown in Table 14, concept 5 was identified as the best solution. The Critical-To-Quality (CTQ) parameters and their relative importance were identified after consultation with Clipper.

Table 13. List of all the feasible motor-generator combinations identified for a 2.5 MW wind turbine

Concept	Motor			Gear drive	Generator	
	No: of units	Disp. (cc/rev)	Speed (rpm)		No: of units	Power rating (kW)
1	5	750	1500	2	2	870
					1	430
2	4	750	1860	0	4	543
3	6	750	1240	0	6	362
4	6	750	1240	3	3	724
5	6	500	1860	0	6	362
6	6	500	1860	3	3	724

Table 14. Pugh matrix analysis to identify the appropriate motor-generator combination

Pugh Matrix		HST for Wind Turbine					
Weighted Sum		48	92	96	44	118	62
CWNs or CTQs	Importance	Concept 1	Concept 2	Concept 3	Concept 4	Concept 5	Concept 6
Initial Capital Cost	5	5	9	3	1	9	7
Reliability	4	3	1	9	5	7	3
O & M Cost	3	1	9	7	1	7	1
Efficiency	2	3	5	9	7	3	1
LRC	2	1	3	3	1	9	5
Scalability	1	9	5	1	9	1	9

On-motor vs. stand-alone charge pump: This trade-off refers to the choice of having individual charge pumps on each motor, typically driven through a coupling on the back of the main motor shaft, versus a stand-alone charge pump system. For the on-motor configuration, the displacement of each charge pump is small since there are many pumps. At this displacement, COTS pumps are available. The cost is significantly lower, not only because they are commercially available at high volumes, but they do not require a separate prime mover. On the other hand, for the stand-alone configuration, the costs are significantly higher since the only pumps available at the required displacement are very expensive.

Generator selection: The continuously variable nature of our proposed drivetrain enables operation of the generator shaft at a fixed speed. This eliminates the need for power convertors to match the grid frequency. A cost comparison study was performed to understand the benefits of using doubly fed-induction generators (DFIG) with power electronics versus squirrel cage induction generators (SCIG) with soft starters and capacitor banks. The cost information for this study was obtained from an earlier study [40]. As shown in Table 15 the cost of using SCIG is significantly lower than DFIG. In addition, the wound rotor design of DFIG requires regular maintenance of slip rings. Therefore SCIG is preferred over DFIG in our drivetrain.

Table 15. Cost per kW comparison between DFIG and SCIG. Data obtained from WindPACT study [40]

	DFIG	SCIG
Generator + PE cost [$/kW]	86	47

The constant speed at the generator shaft also facilitates use of synchronous generators. Due to their projected high cost, permanent magnets are not considered in our study. Field excited synchronous generators (FESG) are a feasible alternative to SCIG. While we currently do not have specific cost information for these generators, they generally tend to be more expensive than SCIG. They also require additional maintenance of the field coils. Synchronous generators enable voltage regulation, and can therefore have better response to grid faults. The SCIG will require additional voltage regulation devices like the STATCOM to provide an appropriate response to grid faults. The required design and control strategy for this operation will be investigated in the next phase of the project.

Fluid (cost vs. type): The choice of fluid for the HST is an important one as it has an influence on efficiency, environmental impact and operating costs. To mitigate potential environmental impact of any fluid leaks we selected a biodegradable fluid. However, this fluid needs to be replaced every 3 years, which leads to a high recurring O&M cost. We have performed a preliminary analysis of alternative fluids and determined that a polyalphaolefin (PAO) based, anti-wear hydraulic fluid would be a good choice for a design life of 20 years (no replacement). Preliminary indications are that the cost would be approximately the same, but it is not biodegradable. Therefore, a final analysis is still necessary to understand the trade-off between significantly lower O&M costs and system changes that are necessary to address PAO based oil.

Accumulator for energy storage (cost vs. AEP benefit): As stated in the earlier section on energy storage analysis, the trade-off in our study was limited to the cost benefit of including an accumulator with reduced generator size as opposed to generators with higher power rating. As evident from Table 7, there is a net reduction in AEP with accumulators and downsized generators. Therefore the cost of generator downsizing must significantly outweigh the cost of accumulators in order to realize a net benefit in overall cost. The recommended accumulator size of 2500 liters is not commercially available. Therefore it was proposed to use multiple smaller accumulators. About six of these accumulators are required to achieve the optimum storage capacity. The combined cost for six accumulators is higher than the total cost incurred on the generators. Any downsizing of the generator will provide a cost benefit that is a fraction of the net current cost of the generators. Therefore the use of accumulators for energy storage was deemed commercially unviable at this time. Further improvements in compressed air energy storage technology will be required to re-evaluate this mode of energy storage.

3. Design Risks

3.1 Pump Design Risks

High risk: Simplified load prediction for initial pump design target

The loading results obtained thus far show that there are no major detrimental loads added to a 2.5 MW scale wind turbine generator by a hydrostatic transmission drivetrain with similar control strategies as the baseline. Adequate loads from the hub to the rotor shaft (LSS) have been assigned for the initial design. The approach of using a reduced model in FAST with interface defined for HST co-simulation has been useful with an assumed simplified controller. The details for the controller dynamics and the actuator dynamics for the hydraulic motors have not been modeled yet. Modifying control strategies for the HST is an important step prior to finalizing these loads. Once these strategies have been finalized, loading simulations should be started again with the goal of balancing loading and AEP to minimize COE. With this said, the NREL simulations and models created in this project is an excellent starting point for this analysis. Dynamic WTG load situations have been neglected so far, and need to be considered if special load conditions for the pump exist that have to be known to achieve a robust design.

High risk: Contact stresses on pump cam/roller interface

The compact design of the cam and follower system requires operation under very high loads. These loads create high contact stresses in the cam surface and the roller surface. Principally it would be a low risk to design for it if the loads are understood sufficiently. The challenge is to understand the system interaction sufficiently. The issue is that due to the high loads, and components which are not stiff to achieve a high compactness, the relative position between the parts is constantly changing, which in turn changes the load between the components again. To predict the interaction sufficiently in order to design the cam and roller surface areas to be robust against all changes and load conditions, a detailed interactive model for force prediction is required. This model will be extended by kinematic models and FEA models to determine the real surface conditions.

High risk: Robust design of roller/follower assembly

The roller is carried by the follower. A bearing sizing study needs to be performed to confirm the bearing specification from this project work.

Cylinder piston bearing/seal design for 20 year life

The standard cylinder rods including the pistons are designed for different loads and shorter number of cycles in its lifetime. It is low risk to design for the new loads and number of cycles but the forces need to be understood. Seal/bearing material other than current standard may be necessary.

MTBF/reliability prediction

The reliability of the pump is assumed to be very high based on the assumption of infinite design life of critical components and well understood loads and wear characteristics of moving parts. These assumptions need to be validated with a development plan. The critical components which require infinite life are the housing (structure), the cylinders, housing and cam connections to wind turbine components, the cam and roller/follower assembly. Due to the lack of historic data on similar components in similar applications, a reliability prediction model is required. This model will need to connect derived reliability data of its components to the pump level and also identifies the operating conditions for the pump component to test for failure. Tests on critical technology components will help to establish an early reliability prediction.

3.2 HST System Design Risks

Environmental impact of fluid leaks

Another risk is the environmental impact of fluid leaks. Fluid lines and connections are designed to contain pressurized fluid even under extreme conditions without leakage. Two potential root causes of leakage are possible: leakage due to wear and maintenance, and due to line rupture. Relative motions of parts over time, maintenance actions with breakage into the hydraulic lines or replacement of hydraulic components require opening of the hydraulic system and inherently a potential for fluid leakage. The fluid is supposed to be drained prior to these measures so the fluid loss to the environment is very limited. If experienced personnel work on the hydraulic system, a proper reassembly and tightening of hydraulic connections is guaranteed to reduce leakage. If a line ruptures either through unexpected external forces, unexpected wear or pressure spikes, a high amount of fluid can be released. Typical counter measures are protective sleeves for the lines, emergency flow fuses (cut off lines to contain fluid) and fluid capturing vessels are of choice as well as the use of environmental fluids (not mineral oil based, e.g. water based or biodegradable).

Resonance from long fluid conveyance lines

In the on-ground solution, the long fluid conveyance between the pump and the motors may induce resonance in the turbine structure. This resonance frequency needs to be investigated so as to protect the turbine from this resonance.

4. Final Report References

1. *Aerodrive, Inc. completes design of 2 MW Continuously Variable Transmission for Utility-scale Wind Turbines.* http://cleanenergynews.blogspot.com/2011/02/aerodrive-inc-completes-design-of-2mw.html, Accessed February 2, 2012.

2. Appleyard, D. *Wind Turbine Gearbox Study Raises Reliability Hopes.* Renewable Energy World, http://www.renewableenergyworld.com/rea/news/article/2011/11/wind-turbine-gearbox-study-raises-reliability-hopes.

3. Ariduru, S. (2004), *Fatigue Life Calculation by Rainflow Cycle Counting Method*, Thesis, Middle East Technical University.

4. Internal Report - *Assessment of Transportation and Installation Cost Savings of Hydrostatic Transmissions in Offshore Wind Turbines*, Frost & Sullivan for Eaton Corporation, November, 2011.

5. Beaudry-Losique, Jacques et al., *Creating and Offshore Wind Industry in the United States: A Strategic Work Plan for the United States Department of Energy, Fiscal Years 2011-2015,* U.S. Department of Energy, Office of Energy Efficiency and Renewable Energy, Wind & Water Program, September 2, 2010.

6. Boukhezzar, Boubekeur, and H. Siguerdidjane, *Nonlinear Control of a Variable-Speed Wind Turbine Using ad Two-Mass Model.* IEEE Transactions on Energy Conversion. Vol. 26, No. 1, March 2011.

7. Camp, T.R., et. al., *Design Methods for Offshore Wind Turbines at Exposed Sites: Final Report*, EU Joule III JOR3-CT95-0284, Nov 2003, Funded by the European Commission.

8. Carlin, P.W., et al., *The History and State of the Art of Variable-Speed Wind Turbine Technology*, Technical Report NREL/ TP-500-28607, February, 2001.

9. Cotrell, J., *Assessing the Potential of a Mechanical Continuously Variable Transmission for Wind Turbines,* Conference Paper NREL/CP-500-38212, August, 2005.

10. Cotrell, J., *Motion Technologies CRADA CRD-03-130: Assessing the Potential of Mechanical Continuously Variable Transmission,* Technical Report NREL/CP-500-36371, September 2004.

11. DeWind D8.2 Brochure. http://www.dewindco.com/eng/product/summary.asp, Accessed January 26, 2012.

12. Faulstich S. et al. *Windenergy Report Germany,* Institut für Solare Energieversorgungstechnik (ISET), 2008.

13. Fawcett, S.W.J, et. at., *Hydrostatic Transmission for a 1 MW Wind Turbine Design Study,* Great Britain Department of Energy, 1990.

14. Fingersh, L., *Wind Turbine Design Costs and Scaling Model*, Technical Report NREL/TP-500-40566, December 2006.

15. *Fluid Conveyance Products, Master Catalog,* Aeroquip.

16. Germanischer Lloyd Industrial Service Gmbh, *Guideline for the Certification of Wind Turbines Edition 2010.*

17. Hjem – Chapdrive, www.chapdrive.com, Accessed January 26, 2012.

18. IEC 61400-1:2005(E), *Wind turbines- Part 1: Design requirements.*

19. IEC 61400-13:2001(E), *Wind turbine generator systems – Part 13: Measurement of mechanical loads.*

20. ISO 61400-4:2005(E), *Wind turbines – Part 4: Design and specification of gearboxes.*

21. Jamieson, Peter, *Innovation in Wind Turbine Design,* John Wiley and Sons, Ltd, 2011.

22. Johnson, K.E., et. al., *Methods for Increasing Region 2 Power Capture on a Variable Speed HAWT,* Conference Paper NREL/CP-500-34979, November, 2003.

23. Jonkman, B.J., *Turbsim User's Guide: Version 1.50*, Technical Report NREL/TP-500-46198, September, 2009A.

24. Jonkman, J. et. al., *Definition of a 5- MW Reference Wind Turbine for Offshore System Development.* Technical Report NREL/TP-5000-38060, February, 2009B.

25. Jonkman, J., et. al., *FAST User's Guide,* Technical Report NREL\EL-500-38230, August 2005.

26. Jonkman, J.M., *Dynamics Modeling and Loads Analysis of an Offshore Floating Wind Turbine,* Technical Report NREL/TP-500-41958, November 2007.

27. Kelley, N.D., and B.J. Jonkman, *Overview of the TurbSim Stochastic Inflow Turbulence Simulator Version 1.21*, Technical Report NREL/TP-500-41137, April 2007.

28. Link, H., et. al., *Gearbox Reliability Collaborative Project Report: Findings form Phase 1 and Phase 2 Testing*, Technical Report NREL/TP-5000-51885, June 2011.

29. Madsen, P.H., et. al., *Predicting Ultimate Loads for Wind Turbine Design.* Conference Paper NREL/CP-500-25787, January, 1990.

30. Milborrow, D. *Breaking down the cost of wind turbine maintenance,* Windpower Monthly, Source: http://www.windpowermonthly.com/news/1010136/Breaking-down-cost-wind-turbine-maintenance/, 15 June 2010.

31. Musial, W., Butterfield, S., and McBiff, B., *Improving Wind Turbine Gearbox Reliability: Preprint,* Conference Paper NREL/CP-500-41548, May 2007.

32. *NWTC Design Codes (AeroDyn by Dr. David J. Laino)*. http://wind.nrel.gov/designcodes/simulators/aerodyn/. Last modified 21-February-2012; accessed 21-February-2012.

33. *NWTC Design Codes (FAST by Jason Jonkman, Ph.D.)*. http://wind.nrel.gov/designcodes/simulators/fast/. Last modified 08-March-2012; accessed 08-March-2012.

34. *NWTC Design Codes (IECWind by Dr. David J. Laino)*. http://wind.nrel.gov/designcodes/preprocessors/iecwind/. Last modified 18-March-2010; accessed 18-March-2010.

35. *NWTC Design Codes (MCrunch by Marshall Buhl)*. http://wind.nrel.gov/designcodes/postprocessors/mcrunch/. Last modified 05-December-2011; accessed 05-December-2011.

36. *NWTC Design Codes (TurbSim by Neil Kelley, Bonnie Jonkman)*. http://wind.nrel.gov/designcodes/preprocessors/turbsim/. Last modified 03-February-2011; accessed 03-February-2011.

37. Oyage, F., *Gearbox Reliability Collaborative (GRC) Description and Loading*, Technical Report NREL/TP-5000-47773, November 2011.

38. Oyague, F., *Gearbox Modeling and Load Simulation of a Baseline 750-kW Wind Turbine Using State of the Art Simulation Codes,* NREL/TP-500-41160, Feb 2009.

39. Oyague, F., *Gearbox Reliability Collaborative (GRC) Description and Loading, Technical Report* NREL/TP-5000-47773, November 2011.

40. Poore, R. and T. Lettenmaier, *Alternative Design Study Report: WindPACT Advanced Wind Turbine Drive Train Designs Study*, Subcontract Report NREL/SR-500-33196, February, 2002.

41. Puigcorbe, J. and de-Beaumont, A. *Wind Turbine Gearbox Reliability.* Renewable Energy World, http://www.renewableenergyworld.com/rea/news/article/2010/06/wind-turbine-gearbox-reliability.

42. Ragheb, A. and Ragheb, M. *Wind Turbine Gearbox Technologies.* Proceedings of the 1st International Nuclear and Renewable Energy Conference (INREC10), Amman, Jordan, March 2010.

43. Rex, A., and K. Johnson, *Methods for Controlling a Wind Turbine System with a Continuously Variable Transmission in Region 2,* Journal of Solar Energy Engineering, Vol. 131, August 2009.

44. Schmitz, J., and N. Vatheuer. *Hydrostatic Drivetrain in Wind Energy Plants.* RWTH Aachen University, Germany. Institute for Fluid Power Drives and Controls. EWEA 2011 Proceedings abstract.

45. Wind Transmission, Artemis Intelligent Power LTD, http://www.artemisip.com/appli_renewable.htm, Accessed January 26, 2012.

46. Wright, A.D. and L. J. Fingersh. *Advanced Control Design for Wind Turbines: Part 1: Control Design, Implementation and Initial Tests.* Technical Report NREL/TP-500-42437, March 2008.