Offshore availability for wind turbines with a hydraulic drive train

James Carroll*, Alasdair McDonald†, David McMillan‡, Julian Feuchtwang•

**University of Strathclyde, Scotland. j.carroll@strath.ac.uk, †University of Strathclyde, Scotland. alasdair.mcdonald@strath.ac.uk. ‡University of Strathclyde, Scotland. d.mcmillan@strath.ac.uk. • University of Strathclyde, Scotland. julian.feuchtwang@strath.ac.uk*

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Abstract

Hydraulic drive trains for wind turbines are under development by a number of different companies; at least one hydraulic drive train is in the final stages of development by a leading wind turbine manufacturer. Hydraulic drive trains have a number of advantages such as redundancy, modularity, compactness and track record in other industries. Currently there are few or no installed wind turbines with hydraulic drive trains onshore or offshore. As no data exists on reliability or failure rates for wind turbines with hydraulic drive trains, this paper estimates failure rates, repair time and availability for said turbines. The paper contains an availability comparison with other drive train types, in which the hydraulic drive train performs best. However, this superior performance should be validated because of the number of assumptions that have been made in this availability estimation.

1 Introduction

In modern wind turbines the slow rotational speed and high torque at the main shaft is usually converted mechanically with a gearbox into a higher speed, lower torque input to the generator. Alternatively with direct drive turbines, a much larger generator with a high torque rating is directly coupled to the turbine rotor. In hydraulic drive trains the mechanical gearbox is replaced by a hydraulic system which converts low to high speed (and from high to low torque). Hydraulic drive trains are currently not in serial production for MW scale turbines. However, hydraulic equipment is successfully used throughout the nacelle. Hydraulics is used in the pitch system, brakes, locks and lifting equipment.

Over the past decade there have been a number of research projects investigating the use of hydraulics in the wind turbine drive train to replace the gearbox and converters. These research projects have led to prototypes being produced and major wind turbine manufacturers acquiring some of the hydraulic technologies developed.

Even with the acquisition of these hydraulic drive train companies by major manufacturers there is still no failure data available in the public domain. As a result of this lack of field failure rate data this paper estimates the failure rates for a hydraulic drive train using a number of different data sources. Failure rates and repair times will be estimated through past publications [1] and the use of offshore reliability data from the oil and gas industry [2]. Recent papers [3, 4] that estimate offshore availability include a number of different offshore drive train types but due to the non-availability of data the hydraulic drive train was excluded.

Reference [3] uses a model based on a probabilistic-statistical approach to calculate the turbine access delays caused due to poor sea conditions. Along with estimated failure rates and repair times for the hydraulic drive train, this model will be used to work out the overall availability for the different drive trains.

The estimated availability from this paper could later be used to calculate an overall cost of energy (CoE) for a wind turbine with a hydraulic drive train. This CoE calculation can then be used as a means of comparing the hydraulic drive train turbine to turbines with alternative drive train types; that in turn will assist in the process of choosing the correct turbine type for a specific site.

2 Hydraulic drive train technologies

2.1 Hydraulic drive train overview

A traditional wind turbine drive train and a hydraulic wind turbine drive train can be seen in figure 1. It can be seen that the gearbox is replaced with the hydraulic system and the power converter (and possibly transformer) are no longer required. The power converter can be removed because of the ability of the hydraulic system to drive the hydraulic motors at a constant speed which in turn can drive a directly grid connected synchronous generator at a constant speed, eliminating the need for converting the frequency or voltage as carried out by the converter in a conventional variable wind turbine drive train.

Early attempts at replacing the gearbox with a hydraulic torque conversion system were unsuccessful for a number of reasons. Scaling worries and poorer efficiency, specifically part load efficiency, were some of the main reasons for the hydraulic torque conversion systems not being considered viable. New technologies that are detailed in section 2.3 may overcome the hydraulic efficiency issue.

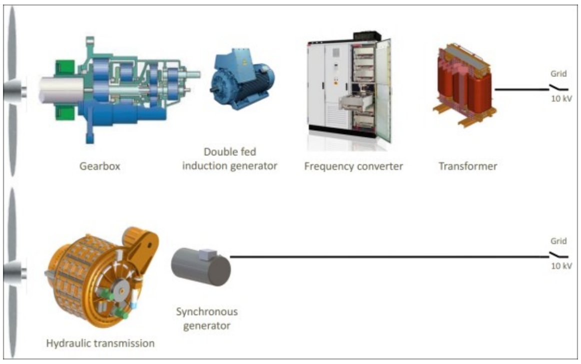


Figure 1: Traditional vs. hydraulic drive trains [7]

2.2 Hydraulic drive train advantages and disadvantages

Manufacturers originally investigated hydraulic drive trains due to a number of possible advantages offered by a system that could potentially remove the gearbox, converter and transformer. Both the gearbox and converters suffer from high failure rates and high downtimes. As seen in [1], the gearbox failure rate is lower than the converter failure rate but the gearbox downtime is higher than the converter downtime. The opportunity to remove these high failure rate and high downtime components in a hydraulic drive train system may provide it with a competitive edge when it comes to reliability.

Manufacturers are continuously trying to save weight in their nacelle designs and the hydraulic drive train also offers this opportunity. The removal of weight bearing components can be seen in figure 1. Loading fluctuations that traditionally can cause problems for mechanical components are also removed through the use of accumulators in the hydraulic system which smooth loading fluctuations from sudden short gusts of wind and turbulence.

One of the hydraulic drive train system reviewed used an electrically excited wound rotor brushless synchronous generators. This removes the need for rare earth materials such as permanent magnet materials as seen in medium speed, fully rated converter, permanent magnet generator configurations. Being brushless removes that failure mode which is one of the highest failure modes for generators [5].

The hydraulic system consists of a pumping unit that contains a number of cylinder and piston modules; this type of system lends itself to modular replacement, something which is generally speaking not possible with a gearbox. Weight reasons mean individual piston and cylinder modules could be replaced with far greater ease than a full gearbox. This in turn would eliminate the need for external lifting equipment. Offshore, this modularity and elimination of the requirement for external lifting equipment is an even greater advantage due to access constraints and the costs associated with jack up vessels. Other advantages stated by hydraulic drive train proponents are the ability to place motors and generators in the tower or at ground level (even though this may lead to lower efficiency) and the removal of mechanical alignment issues associated with gearboxes, bearings and generators in traditional systems.

2.3 Hydraulic drive train operation

Traditional hydraulic systems that cannot vary the amount of hydraulic fluid displaced use pumps consisting of camshafts, pistons and valves. They regulate the hydraulic fluid drawn into the cylinder chamber on the back stroke and allow high pressure fluid out on the forward stroke. However, in a conventional variable displacement hydraulic system a swash plate is used to control the amount of hydraulic fluid displaced. This swash plate control mechanism has traditionally led to poor efficiency.

The research projects mentioned earlier in section 2.1 claim to have overcome the major efficiency and part load efficiency issues through the use of fast acting microprocessor controlled solenoid valves to deal with controlling the amount of hydraulic fluid displaced. This method of controlling the displacement of the hydraulic fluid has been called “Digital Displacement” by one company that was created out of a research project and later acquired by a wind turbine manufacturer [6]. Another company refers to it as “digital valve technology” and “digital hydraulic motor” [7].

Unlike the conventional variable displacement technology that uses the swash plate, digital displacement technology has low and high pressure valves that can be opened and closed independently with each stroke. This is achievable through the use of a small electromagnetic latch. An embedded controller and power FET controls the solenoid valve in each latch. The power FET is a semi-conductor device that can control current and act as a switch. [6]

2.4 Overcoming efficiency issues

It is this accurate low pressure valve control in the pump, high pressure valve control in the motor and natural release of pressure through the passive high pressure valves in the pump that eliminates the losses associated with swash plate pumps operating at partial load.

It is claimed that this improved valve control offers nearly uniform efficiency throughout the partial load stages of operations, partial load efficiencies of over 90% have been reported by the companies that are using this digital valve technology [6, 7]. It is this improvement in partial load efficiency that has made hydraulic drive trains viable in wind turbines.

3. Hydraulic drive train failure rate calculation:

3.1 Method

Unlike traditional drive train types no reliability or failure rate data currently exists for hydraulic drive trains making it impossible to calculate failure rate or availability figures based on past data as in references [1, 3]. Failure rate and availability figures have to be estimated and calculated using an alternative estimation method.

The estimation method relies on past publications detailing failure rates and downtimes for traditional drive train turbine types and offshore failure rate data from the oil and gas industry. The following steps are taken to estimate the offshore availability for a wind turbine that has a hydraulic drive train:

1. The hydraulic drive train is broken down into its individual components.
2. These individual components are further broken down to obtain a parts list for each component detailing what each component consists of e.g. piston, valve, seal etc.
3. This parts list is then used to obtain offshore failure rate data from OREDA (**O**ffshore **RE**liability **DA**ta) for each part [2]
4. Downtimes and failure rates for the turbines sub-assemblies outside of the drive train are obtained from past publications which detail downtime and failure rates from traditional turbines. [1,3]
5. Offshore delays due to inaccessibility from poor sea conditions are calculated using the offshore delay model described in section 5.1.
6. These offshore delays, downtime data and failure rates are then used to calculate the overall offshore availability for the hydraulic drive train turbine.
7. The hydraulic drive train turbine is compared to turbines with alternative drive train types from similar studies [3]

3.2 OREDA data

As mentioned in step three of section 3.1, OREDA data is used to obtain offshore failure rate data for the parts and components used in the hydraulic drive train. OREDA is a co-operation agreement between eight global oil and gas companies. The eight companies have agreed to create a shared reliability, safety and maintenance database for their exploration and production equipment. This data comes from offshore sites throughout the globe and covers a large range of equipment types and operating conditions. Both offshore subsea and topside equipment are included in the database; however it is the topside data that is relevant for this analysis. OREDA also publishes books detailing their failure data, the fourth edition of this book was used to obtain data for this analysis. [2]

The OREDA failure rates in step 3 could be slightly different than wind turbine failure rates because of differences in stop-start patterns and loading and partial loading issues. A brief comparison of an offshore generator of a certain power rating from a wind turbine with an offshore generator with a similar power rating from the OREDA data showed a difference in failure rates of 5 to 10%. The wind turbine's generator had the higher failure rate and this could possibly be explained by the stop start and partial loading aspects of wind turbines.

4. Break down of hydraulic drive train

4.1 Components of the hydraulic drive train

As mentioned in step one of section 3 the hydraulic drive train is broken down into its components. For the purpose of this analysis, the components in a hydraulic drive train have been identified using the schematic shown in figure 2 which was taken from one of the manufacturer's hydraulic drive train documentation [6].

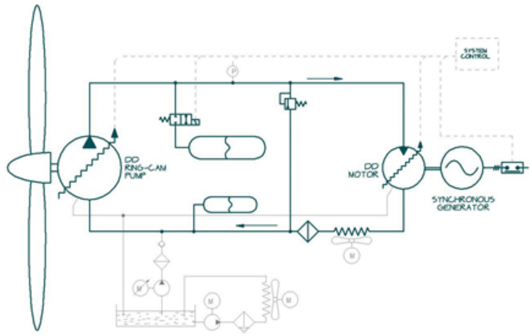


Figure 2: Hydraulic drive train schematic [6]

From the above schematic it can be seen that the hydraulic drive train consists of the digital displacement pumping system, 2 hydraulic accumulators, a hydraulic motor and a synchronous generator. As mentioned in the method in section 3.1 each of these components is broken down into a parts list. This part lists can be seen in the following sections.

4.2 The hydraulic pumping system

A scaled down version of the digital displacement pumping system from the promotional literature of one of the hydraulic drive train companies is shown in figure 3. For illustration purposes this version shows 6 pistons and 6 cylinders in the pumping system. However from the manufacturer's material [6] it is assumed that the 1.5MW prototype that has been successfully tested consists of a parts list of a radial pump with 68 pistons, 68 valves, 68 microcontrollers and 68 power FETs.

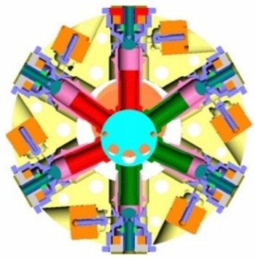


Figure 3: Digital displacement pumping system [6]

4.3 Hydraulic accumulators

A number of different hydraulic accumulators exist. A bladder accumulator in which the charge is accumulated through the compression of a gas filled bladder is one popular type of accumulator. Another type is a spring accumulator in which a spring is used instead of the gas filled bladder.

For the purpose of this analysis it has been assumed that the type of accumulator used is a piston accumulator as seen in figure 4. In a piston accumulator a cylinder contains a piston that is pushed to the bottom of the cylinder using a charging gas, the pressurized fluid from the hydraulic pump enters the accumulator and pushes back the piston compressing the gas which will later be used to release the hydraulic fluid at a constant speed. As seen in figure 4 the hydraulic accumulator parts list consists of valves, piston, and the casting. The system seen in figure 2 contains two accumulators.

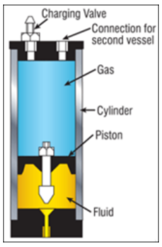


Figure 4: Piston hydraulic accumulator [17]

4.4 Hydraulic motors

The internal workings of a hydraulic motor is very similar to that of a hydraulic pump run backwards [8]. As described in section two, in new hydraulic systems the motor also includes a microcontroller for the high pressure valve. As a result, the parts list for the digital displacement motor consists of a pump, microcontroller and power FET.

4.5 Synchronous generator

The generator type used is a fixed speed brushless wound rotor synchronous generator [6]. This generator type does not require brushes or rare earth materials and its fixed speed allows for the removal of the converter and transformer.

5 Offshore delays

5.1 Overview of model

This section provides an overview of the model used to calculate offshore delays for different drive train types from paper [3]. For offshore availability it is not sufficient to look at onshore lead time and repair time. Delays due to sea conditions and the travel and positioning times of the vessels must also be included. The model used to estimate offshore availability is based on the probabilistic-statistical approach detailed in [9] and implemented in [10]. Given a number of statistical parameters related to the wave regime at the wind farm site and data on reliabilities and repair times for different components, delays are calculated. This avoids the need to run multiple lengthy simulations and makes it simple to explore the effect of changes in parameters, such as failure rates.

The model takes into account delay time predicted from sea conditions, travel time from the position of the site and average positioning time depending on the vessel type required to repair the failure. The onshore repair time is then added to the delay times calculated from the model to determine the overall downtime. Full details on the operation of the model can be found in [9] and an overview is provided in the following paragraphs.

Three different vessel types are used in the model and each turbine failure is allocated to the vessel type required to repair that failure. Each vessel type has a sea condition threshold above which it cannot operate, and is then used, along with the past sea condition data, to calculate an expected delay time using the probabilistic model developed in reference [9]. The model is based on a number of simplifying assumptions given below:

- A failure will occur independently and unsystematically. In reality a failure will not be independent; it will be influenced by factors like wind speed and wave conditions. Higher wind speeds and rougher sea states would in reality lead to higher failure rates and reduced access, which in turn would lead to reduced availability [11].
- The repair will occur in a single trip and not be broken into multiple trips;
- Sea condition forecasts will always be available for the length of time required to complete the repair [9].

From the event tree in figure 5, and a more detailed one that can be developed from it, probabilities and expected delay times are allocated to each branch of the tree. These probabilities and times are calculated directly from parameters of the wave height probability distribution and wave height duration probability distributions, which in turn are calculated from significant wave height records from the site in question (see [9]). Data are also required for each vessel's positioning time and a speed which can be used to calculate travel time.

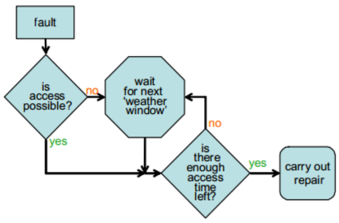


Figure 5: Repair event tree [9]

The analysis for this paper was based on a site that is 16km from shore. The wave height duration distribution for this site was derived using the method in [12] and the wave height distribution figures from [13]. The sites wave and wind characteristics can be seen in table 2. The modelled offshore availability figures depend on the wind and wave characteristics, and would vary as these inputs vary, further work could look at the sensitivities of variance to these inputs.

Wave location parameter	0.36	m
Wave shape parameter	1.36	
Wave scale parameter	1.031	m
Wind location parameter	1.53	m/s
Wind shape parameter	2.12	
Wind scale parameter	9.16	m/s

Table 1: Wind and wave parameters

5.2 Modularity in the delay model

As mentioned in section 5.1, failures are allocated a vessel type required for repair, e.g. it may be stated that for gearbox repair a crew transfer vessel (CTV) is required for 60% of repairs, a fast rescue craft (FRC) for 20% of repairs and a jack-up vessel for 20% of the repairs. The hydraulic system is modular with no component weighting over 25kg [6]. The model captures this modularity advantage in the repair vessel allocation e.g. instead of requiring a Jack-up vessel and FRC for 40% of the failures as in the gearbox the hydraulic system requires a CTV for 100% of the failures. The result is a lower cost and shorter delay time through the use of the CTV for all failures and this advantage is captured in the model.

6 Results

6.1: Overall hydraulic system offshore failure rates

The overall failure rate for the hydraulic system that replaces the gearbox, converter and transformer is 0.1029 failures per turbine per year. Figure 6 shows this broken down into subsystems.

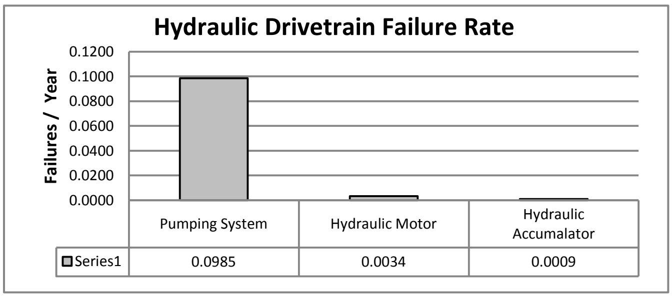


Figure 6: Hydraulic system failure rate

6.2: Pumping system offshore failure rates

The pumping system is the largest contributor to the overall failure rate with 0.0985 failures per turbine per year. The breakdown of the pumping system failure rate can be seen in figure 7. Based on section 4.2, and for the purpose of obtaining failure rates, the pumping system is assumed to consist of a standard pump with 68 valves, 68 pistons, 68 micro controllers and 68 FETs. Each parts failure rates can be seen in Figure 7. These failures rates were obtained from [2] page 175. The microcontroller failure rate was obtained from [14] and the failure rate of 11 failures in time (FIT) for the power FETs was obtained from [15].

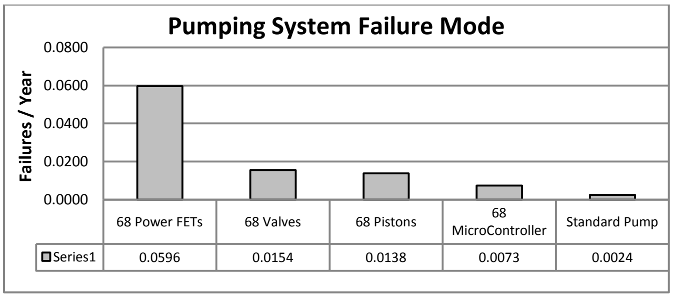


Figure 7: Pumping system failure mode and rate

6.3: Hydraulic motor offshore failure rates

Figure 8 shows the second largest contibuter to the hydraulic system failure rate is the hydraulic motor with a failure rate of 0.0034. As detailed in section 4.4 the hydraulic motor has been assumed to consist of a standard motor/pump, a micro processor and a power FET.

The failure rates for each of these items can be seen in figure 8. As with the pumping system, the failure rates from the overall pump/motor come from [2], the microprocessor from [14] and the power FET from [15].

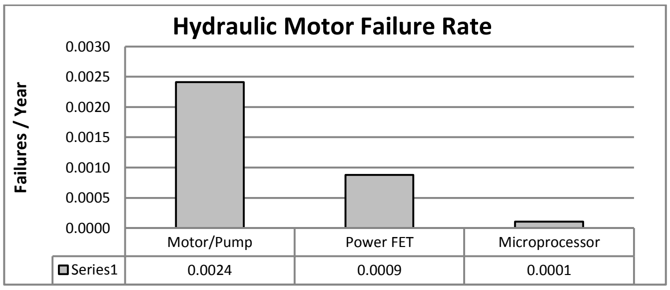


Figure 8: Hydraulic motor failure rate and mode

6.4: Hydraulic accumulator offshore failure rates

The final contributor to the hydraulic system failure rate is the hydraulic accumulator. As stated in section 4.3, the hydraulic accumulator has been assumed to be a piston accumulator consisting of the parts detailed in figure 9. The failure rate for each of these parts were obtained from [2].

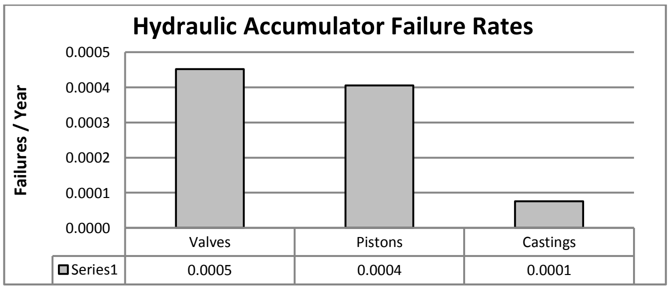
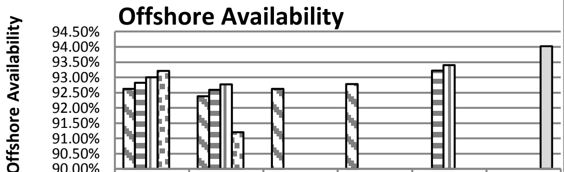


Figure 9: Hydraulic accumulator failure rate

6.5: Availability calculation and comparison

For the purpose of this analysis, the failure rate and downtime for the fixed speed wound rotor synchronous generator is assumed to be the same as in reference [3]. In this analysis the failure rates and downtimes for the turbine components outside of the drivetrain have been assumed to be the same as the failure rates and downtimes in reference [1]. Within the drive train, failure rates are calculated as detailed in the earlier sections of this paper. As a conservative estimate of the downtime for the hydraulic torque conversion system the gearbox downtime has been used. In reality, the hydraulic torque conversion downtime is likely to be lower than that of the gearbox because with parts weighing no more than 25kg external cranes or lifting equipment will never be required for the hydraulic system. As no gearbox, converter or transformer is included in the hydraulic drivetrain, no failure rates or downtimes are included for these components. The offshore delay was worked out using the model described in section 5.1. Based on the above data and estimates, the overall offshore availability for a hydraulic drive train turbine was then estimated to be 94.02%. A comparison with availability from other drive train types that were estimated in [3] can be seen below:



	FRC PMSG	FRC WRSG	FRC SCIG	DFIG WR	Brushless DFIG	Brushless WR
3 Stage Gearbox	92.62%	92.38%	92.62%	92.78%		
2 Stage gearbox	92.83%	92.59%			93.22%	
Single Stage	93.00%	92.76%			93.40%	
No Gearbox Direct Drive	93.22%	91.21%				
Hydraulic						94.02%

Figure 10: Availability Comparison

7. Discussion and conclusion

The previous section shows the estimated availability for a hydraulic drive train offshore turbine is greater than similar estimates for other drive train types in [3]. The estimated availability of 94.02% is 0.66% greater than the second highest performing drive train. The main drivers for this superior availability are the removal of the power converter failure mode, the removal of the gearbox failure mode, the lower generator failure rate (due to it being brushless) and the decrease in downtime due to the modularity of the hydraulic system. The newly introduced failure modes from the hydraulic systems seem to be overcome by the reliability improvements previously mentioned. The estimated offshore failure rate of 94.02% is in the average European offshore availability range of 90% - 95% [16]

Due to the large number of assumptions made in this analysis, it is recommended that further work be completed to verify the hydraulic drive train's superior availability. As no failure rate or reliability data currently exists in the public domain for hydraulic drive trains, it is felt that even though this paper includes a number of assumptions and estimates it can be a starting point for further hydraulic drive train reliability analyses.

In this study the components used in the hydraulic drive train have been determined through manufacturers' websites and promotional material. Further work could include, working with the manufacturers to verify the components used and sourcing failure rate data for any of the components not already included or improved failure rate data for the components already included. Further work could also include correcting the OREDA failure rate data to take into consideration the impact of the stop start factor and partial loading experienced by the wind turbine detailed in section 3.2. Due to the modularity of hydraulic drivetrains redundancy may be another advantage; further work could try to capture this redundancy in the availability modelling.

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