PROcedures for TESTing and measuring wind turbine components; results for yaw and pitch systems and drive train

J.G. Holierhoek

D.J. Lekou[1]

T. Hecquet[2]

H. Söker[3]

B. Ehlers [4]

F.J. Savenije

W.P. Engels

R.P. van de Pieterman

M. Ristow[5]

M. Kochmann[5]

K. Smolders[6]

J. Peeters[6]

[1]Wind Energy Section, Centre for Renewable Energy Sources and Saving, Greece

[2]SWE, Universität Stuttgart

[3]DEWI, Wilhelmshaven

[4]Suzlon Energy GmbH, Rostock

[5]Load Assumptions, Germanischer Lloyd Industral Services GmbH, Hamburg

[6]R&D technology, Hansen Transmissions International, Lommel

Published online in Wiley Online Library (wileyonlinelibrary.com). DOI: 10.1002/we.1511, Wind Energ. (2012)

ECN-W--12-030 JULY 2012

WIND ENERGY
Wind Energ. (2012)
Published online in Wiley Online Library (wileyonlinelibrary.com). DOI: 10.1002/we.1511

RESEARCH ARTICLE

PROcedures for TESTing and measuring wind turbine components; results for yaw and pitch system and drive train

J. G. Holierhoek[1], D. J. Lekou[2], T. Hecquet[3], H. Söker[4], B. Ehlers[5], F. J. Savenije[1], W. P. Engels[1], R. P. van de Pieterman[1], M. Ristow[6], M. Kochmann[6], K. Smolders[7] and J. Peeters[7]

[1] Unit Wind Energy, Energy research Centre of the Netherlands, P.O. Box 1, 1755 ZG Petten, The Netherlands
[2] Wind Energy Section, Centre for Renewable Energy Sources and Saving, Greece
[3] SWE, Universität Stuttgart, Stuttgart, Germany
[4] DEWI, Wilhelmshaven, Germany
[5] Suzlon Energy GmbH, Suzlon Energy GmbH, Rostock, Germany
[6] Load Assumptions, Germanischer Lloyd Industrial Services GmbH, Hamburg, Germany
[7] R&D technology, Hansen Transmissions International, Lommel, Belgium

ABSTRACT

PROcedures for TESTing (PROTEST) and measuring wind energy systems) was a pre-normative project that ran from 2008 to 2010 in order to improve the reliability of mechanical components of wind turbines. Initiating the project, it was concluded that the procedures concerning these components should be further improved. Within the PROTEST project, complementary procedures have been developed to improve the specification of the design loads at the interfaces where the mechanical components (pitch and yaw system, as well as the drive train) are attached to the wind turbine. This is required, since in optimizing wind turbine operation and improving reliability, focus should be given to the design, not only to safety related components but also to the rest of the components affecting the overall behaviour of the wind turbine as a system. The project has resulted in a proposal for new design load cases, specifically for the drive train, a description of the loads to be defined at the interfaces of each mechanical system, as well as a method to set up and use the prototype measurements to validate or improve the load calculations concerning the mechanical components. Following this method would improve the reliability of wind turbines, although more experience is needed to efficiently use the method. Examples are given for the analysis of the drive train, pitch system and yaw system. Copyright © 2012 John Wiley & Sons, Ltd.

KEYWORDS
reliability; mechanical systems; interfaces; drive train; prototype testing

Correspondence
J. G. Holierhoek, Unit Wind Energy, Energy Research Centre of the Netherlands, P.O. Box 1, 1755 ZG Petten, The Netherlands.
E-mail: holierhoek@ecn.nl

Received 9 March 2011; Revised 13 October 2011; Accepted 24 January 2012

1. INTRODUCTION

High reliability of wind turbines and their components is one of the prerequisites for an economically viable exploitation of wind farms, especially for offshore power plants. The current level of reliability needs to be increased in order for offshore wind energy to become economically competitive. In this section, first, an introduction into the current status of reliability of the mechanical components of wind turbines will be given, followed by a description of the PROcedures for TESTing (PROTEST) project.

The reliability of wind turbines and their components is, at present, still at a level that leaves room for improvement. Wind turbines still show failure rates of between 2 to 5 failures per year that need visits from technicians (derived from i.e.[1–3]). Although electrical components and control systems fail more often, the costs related to repair of failed mechanical systems (drive train, pitch and yaw systems) are dominating the operation and maintenance costs and the downtime. For this reason, ways to improve the reliability of the three aforementioned systems need to be addressed.

In depth studies, e.g.,[4] and discussions with turbine manufacturers, component suppliers, and certification bodies during the Dutch Wind Workshops in Petten, the Netherlands in 2006, revealed that one of the major causes of failures of mechanical systems was insufficient knowledge of the loads acting on these components. This lack is a result of the shortcomings in load simulation models and in load measurement procedures at the level of these components.

It was also concluded that, at present, the procedures, tools and guidelines in designing rotor blades and towers of wind turbines are much more specific than the procedures in designing other mechanical components such as drive trains, pitch and yaw systems. The lack of clear wind energy specific procedures in designing mechanical components and specifying the loads on these components should be resolved, thereby, preventing early failures. These wind turbine components are complex electromechanical systems themselves. Although they are indispensable in the operation of the wind turbine, these systems are, in the process of wind turbine design, usually treated as off-the-shelf systems and are purchased as such from second companies producing them for use in wind turbines. Given the increase in size of the wind turbines, the design for these components can no longer be supported through findings from other sectors, e.g. heavy industrial crane manufacturers or naval applications. Therefore, refined design guidelines should be developed for these components. An additional problem comes from the aeroelastic simulations for the estimation of loads acting on the wind turbine components. These provide only limited information to the manufacturer of the system, since oversimplified models for the pitch, the yaw and the drive train systems are used.

PROTEST, the acronym of the full title 'PROcedures for TESTing and measuring wind energy systems' was a pre-normative project which intended to provide complementary procedures to better specify and verify the local component loads acting on mechanical systems in wind turbines. PROTEST, funded by the European Commission within the Seventh Framework Programme, ran from March 2008 to August 2010. Seven different partners were involved: University of Stuttgart, Greece's Centre for Renewable Energy Sources and Saving, Suzlon Energy GmbH, Germanischer Lloyd (GL), Hansen Transmissions International and the German Wind Energy Institute (DEWI), with the Energy research Centre of the Netherlands (ECN) as the coordinator of the project. During the project, intermediate results were also distributed to the GRC[5] and the IEC-4[6], whereby further input was gained from these large groups of experts.

This paper will give an overview of the main results obtained in the PROTEST project. First, the current state of the art will be shortly discussed, followed by a description of the six-step approach, which was developed during the project. Then, the results for each of the mechanical components that were treated, the drive train, the pitch system and the yaw system, are given. For each system, the required additional design load cases (DLC) will be discussed (when applicable), followed by the definition of the interfaces and the description of the load definitions at these interfaces. Also, a small selection of the results obtained from the analysis and measurements are shown for each component. These results show the application of the PROTEST approach in the use of measurements to obtain required input parameters for simulation tools (drive train example), to validate and/or tune the simulations (pitch example) and to validate the design loads (yaw example). Finally, the main conclusions of the project are described.

2. STATE OF THE ART

Before trying to improve the reliability of the components by suggesting a new approach, it is of course necessary to first look at the current situation. In the PROTEST project, the state of the art at the start of the project in 2008 was therefore studied, for which the main results are described in this section.

The design process of wind turbine components is based on aeroelastic calculations of various DLCs, which are described in standards e.g. the IEC 61400-1[7] or guidelines such as GL guidelines for the certification of wind turbines.[8] These wind turbine standards have been developed to ensure the engineering integrity of wind turbines. In increasing reliability, it was questioned whether these DLCs are sufficient for the design of machinery components, especially the drive train, and also for the pitch system and yaw system. For the drive train, an additional standard (IEC 61400-4) is currently under development. IEC 61400-4 is a good starting point in defining specific DLCs, but no detailed list of DLCs is provided. Therefore, the PROTEST project has evaluated whether the currently considered load cases are sufficient for the drive train, pitch and yaw system or which additional DLCs have to be considered. The results of the evaluation can be found in the next sections.

In general, the state-of-the-art aeroelastic simulation codes use a simplified representation of the machinery components, especially the drive train, which result in neglecting the interactions of the components. The GL guidelines from 2004[8] required the consideration of internal component dynamics through drive train resonance analyses to identify possible resonances, but the results were not linked to wind turbine loading. To overcome the shortcomings of this simulation approach, advanced wind turbine simulation codes should consider more detailed models of the drive train components also for aeroelastic simulations in the time domain.

For load measurements, performed to support the design process and for certification, various measurement load cases are defined in guidelines and standards, especially IEC/TS 61400-13.[9] At present, measurement campaigns are used to validate the global design loads with measurement loads. It is not yet clear whether the provided measurement load cases are sufficient to validate the design loads of all the wind turbine components. Moreover, the guidelines and standards define the measurement campaign in detail, but no procedure is given on how to validate the global design loads with the measured loads or how to validate the simulation performed. Furthermore, at present, there are no guidelines or standards in defining a measurement campaign for wind turbine components. Since no procedure is given for the validation of the global loads, this lack of information is even more relevant for the validation of component loads.

As compensation, the PROTEST project has developed procedures for performing such a measurement campaign and for validating the loads, used for the component design, with the component measurement data. This approach is described in Section 3.

When looking at the future prospects for the design and development procedures for mechanical components, it is the expectation of the authors that it will follow a similar route as the design and development of rotor blades over the last 20 years. The design process for rotor blades (and also for the tower) is critical for safety: failures will lead to unsafe situations. Therefore, in the past, safety standards have been developed for wind turbines,[10] together with technical specifications on how to carry out full scale blade testing[11] and prototype measurements[9] in order to prevent critical failures. Failures of other mechanical systems, however, are mainly critical for reliability: failures will lead to standstill and economic losses only.

In short, improvements to the design and development procedures for mechanical components in the following areas are expected:[12]

- Design approach
  The common current design approach for blades and towers is much more extensive than the approach used for mechanical components. Although this is not included in the standards, it is illustrated by the different guidelines, e.g..[8,13] It should be assessed if the current DLCs cover everything for the three discussed mechanical components, where not only fatigue and ultimate strength should be considered. Also, the dynamic properties should be modelled and tested in more detail, as is currently the case for blades but not for the other components.
- Measurement procedures
  Similar to the standards for blade measurements (IEC 61400-23[11] and IEC 61400-13[9]), guidelines should be set up for defining an appropriate measurement procedure and for analysing and reporting the measurement data for the components, but with more emphasis on using measured data in the design process, for validating the input used either in the process or the design process per se.
- Standards and certification
  New standards and guidelines will differentiate on the basis of the size of the wind turbine and onshore or offshore turbines. Redundancy or maintainability will become important. The certification process will focus more on supervising the quality of the manufacturing process.
- Data exchange
  After validation of the prototype turbine, it is necessary that a certain minimum of data flows from the wind farm owner to wind turbine manufacturer to component manufacturer.

A more elaborate discussion on this subject can be found in the PROTEST report dealing with the state of the art[12].

3. SIX-STEP APPROACH

Two different goals should be distinguished when setting up a prototype measurement campaign. First, the campaign can be set up to verify the initial design loads. Second, and the main focus of PROTEST, the campaign can be used *to tune and validate the models* that have been used for the design of the mechanical systems. However, because of the large differences between the different wind turbine concepts for each of these mechanical systems, as well as the differences in the corresponding models that are used, it appears to be impossible to set strict standards for any of these systems. The model that is used determines the measurements needed. Therefore, a completely new and more flexible approach is suggested, a six-step approach, letting go of the current less flexible approach in the guidelines and standards.

The six steps that are to be followed in setting up a measurement campaign for a mechanical system are (see also Figure 1):

- Step 1: Identify the critical failure modes or phenomena for the component.
- Step 2: Design the model.
- Step 3: Run the model for (various) critical DLCs.
- Step 4: Determine the input and output parameters of the model, estimate their accuracy, and whether they need to be verified/measured.
- Step 5: Design the measurement campaign used to validate the model and verify the input parameters used in the model, e.g. stiffness values.
- Step 6: Process the measurement data and check or improve the model or the model parameters.

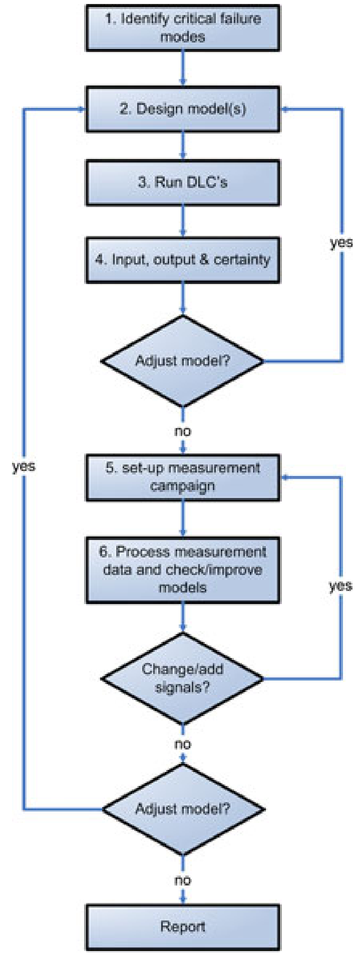

Figure 1. The six-step approach.

It is clear that using this approach, the measurement campaign of a certain component will not be the same for every type of wind turbine; it depends on the configuration, as well as on the model that is used.

Note that these steps will not always be performed sequentially, as illustrated in Figure 1.

4. DRIVE TRAIN

One of the mechanical components that were investigated within the PROTEST project was the drive train. For the drive train, three new DLCs were identified, as described in Section 4.1. The interfaces and the loads transferred across these interfaces are discussed in Section 4.2. A small selection of the results obtained by using the six-step approach for the drive train are discussed in Section 4.3.

4.1. New design load cases

Failures of drive trains are, by now, well-known problems in wind turbines.[5,14] One of the identified causes of these failures is the difference between the analysed loads on the gearbox and the actual loadings. This is often because of the oversimplification of the drive train model in complete wind turbine simulations, whereby the input to the detailed drive train models is also oversimplified. Therefore, three shortcomings in the current procedure used to validate the design of wind turbine drive trains have been identified.[15] These three shortcomings can result in significant differences between the results of the analysis of the wind turbine model and the real turbine. Therefore, for the design of the gearbox and the complete drive train, these three short comings should be taken into account in the requirements prescribed by IEC-61400-1.[7] During the project, they were included in the latest version of the GL guidelines.[16]

DOI: 10.1002/we

It should be noted that, strictly speaking, the first two cases are not DLCs. A DLC is defined as a set of external conditions such as wind conditions, combined with a design situation, additional conditions, the type of analysis and the partial safety factors. This is not specified as such in the newly proposed DLCs. Next to this, a violation of the tolerance criterion cannot be accounted for in the DLCs; it must be assumed that a turbine is constructed according to the requirement specification.

- DLC - Misalignment
  Misalignment of the drive train may cause constraining forces in the gearbox. Misalignment can originate from the interface of the main shaft assembly and gearbox and from the connection between the gearbox and generator. To take these into account and analyse these constraining forces, it should be specified to what extent of misalignment in the drive train shall be considered. The specified tolerances shall apply for the operating condition of the wind turbine and the deflection caused by the flexible mounting of the drive train components on the supporting structure shall be taken into account. Besides the flexible mounting of the drive train, the deformation of the supporting structures, themselves, (main bearing housings, main frame and generator carrier) during an operation will apply reactive forces to the drive train. Also, these forces need to be considered in the determination of design loads for the drive train components.[15]

  In the first step, it can be assumed that the additional loads originating from misalignment of the drive train will only marginally affect the overall system response of the wind turbine. This means that the global loads at the current interconnection points will not shift significantly, whereas the local component loads will be notably affected. From this assumption, it appears acceptable to analyse the drive train and the supporting structure as separated from the remaining wind turbine. For this purpose, a detailed time domain calculation model of the drive train and adjacent components needs to be implemented and analysed. Certain transient events, as well as DLCs of normal operating conditions, should be analysed by this means.
- DLC - Resonance
  The drive train consists of a number of sub-assemblies which, together, form a dynamic system. The intersection of the systems natural frequencies and excitation frequencies may lead to load increasing resonances that will affect the main drive train components. In order to identify and investigate resonances, a resonance analysis has to be performed. Depending on the phenomena to be analysed and the frequency ranges, different models and tools with varying levels of complexity can be used. Depending on the excitation mechanisms, different frequency ranges need to be analysed, e.g. [0–5 Hz], [5–50 Hz], [50–200 Hz], [200–500 Hz] and [500-2000 Hz].[15]

  The dynamic behaviour of the drive train depends mainly on the mass, inertia and stiffness properties of the components in the drive train. Varying drive train configurations might cause variations of these properties. Hence, a new analysis of the drive train dynamics is necessary if different types of a component are installed in the same type of wind turbine, e.g. rotor blades, main shaft or gearbox.[15]
- DLC - LVRT
  Fault or loss of the electrical network connection is included in DLCs 2.3 and 2.4[7]. However, in practise, the tools are not yet good enough to completely analyse these DLCs. The low voltage ride through (LVRT) should be described in more detail; many different shapes of the low voltage can be specified and have different effects on the turbine. The different grid codes that exist in different countries further complicate this DLC. This combination deems it impossible to prescribe the LVRT DLCs in detail. It is clear that it will also be very hard to find the most critical cases for a specific turbine. Combined with wind speed, a detailed approach of the LVRT can result in a large number of DLCs to be analysed. The details of this process, therefore, cannot be specified during this project. The LVRT DLCs are however of significant importance for both fatigue and ultimate strength.
  Wind turbine tools should be coupled to power system simulation tools, as performed in[17]. But compared to the analysis there, more detail will be needed in the drive train model, and higher frequencies need to be addressed.

By including detailed analysis of the aforementioned cases, the oversimplification of the drive train model in the complete wind turbine simulations can be compensated for. Only once, these new DLCs have been used by manufacturers for different designs; the validity of the assumption that this would increase the reliability of the designed drive trains can be evaluated.

4.2. Drive train interfaces and definition of transferred loads

When determining the procedure to be used in describing the method defining the loads at the interconnection points, the specification of the interfaces of the gearbox and the drive train and its subcomponents is required. This includes isolation of each system or subcomponent from the overall wind turbine structure and creating an adequate description of the sectional loads at the interconnection points (interfaces).

For the *gearbox*, IEC 61400-4[6] identifies the interconnection points (interfaces) commonly applied in modern wind turbine designs. The four identified interfaces are[18] the low speed shaft to the gearbox (specifically, the gearbox entrance stage), the high speed shaft to the gearbox (specifically, the gearbox output stage), the nacelle main frame through the supporting positions of the gearbox to the gearbox (specifically, the gearbox housing) and the mounting positions of the gearbox on the nacelle main frame via torque arms to the gearbox (specifically, the gearbox housing).

Loads transferred across the gearbox system depend on the configuration of the wind turbine. Therefore, detailed analysis would have to be based on detailed configurations. In an ideal situation, the purpose of the gearbox would be to transmit the torque and the rotation (revolutions) of the rotor to the generator through the high speed shaft, counteracting all other loads arriving at the gearbox from the rotor through the low speed part of the drive train. To this end, the torque arms of the gearbox are used to counteract the torque reaction of the gearbox from the rotor. The forces and bending moments are either counteracted through the main bearing(s) of the main shaft or (depending on the configuration) through bearings of the gearbox.

Bending moments and torsion (torque) are usually measured on the main shaft during conventional load measurement campaigns (as specified in IEC/TS 61400-13[9]). The force measurements, however, are not required, and usually these measurements are not performed. The forces (and moments) on the main shaft can be estimated through aeroelastic simulations, but to obtain the forces and moments on the high speed shaft or the forces on the torque arms through aeroelastic simulation, detailed information on the gearbox and the drive train is necessary.

A summary of the recommended measurements during an experimental campaign specifically designated on the gearbox is presented in Table I. Synchronization of the general loading conditions is required with the wind turbine operation data (WTOD). The WTOD consists of the status, hub wind speed and direction, rotor angular speed and azimuth angle, pitch angle, yaw angle and generator power.

For the *drive train*, IEC 61400-4[6] identifies the interconnection points (interfaces), depending on the wind turbine configuration, similar to the gearbox. For example, the following interfaces can be defined for a modular drive train with a three-point suspension, as illustrated in figure 2:[18]

1. The rotor hub to the drive train (on the low speed, main shaft);
2. The main bearing of the drive train (on the low speed shaft) to the nacelle main frame;
3. The torque arm on the gearbox to the nacelle main frame;
4. The nacelle main frame to the support points of the gearbox of the drive train;
5. The nacelle main frame to the support points of the generator of the drive train;
6. The generator (on the high speed shaft) to the drive train, internal interface of the drive train;
7. The mechanical brake to the drive train (on the high speed shaft), internal interface of the drive train;
8. The coupling on the high speed shaft of the drive train, internal interface of the drive train; and
9. Others (e.g. interfaces for lubrication systems, sensors).

Loads transferred across the drive train on specific interface points depend on the configuration of the wind turbine. Similar to the case of the gearbox, detailed analysis of the loads transferred through each component of the drive train would have to be based on the specific configuration of the wind turbine. In an ideal situation, the purpose of the drive train would be to transmit the torque and the rotation (revolutions) of the rotor to the generator, counteracting all other loads of the rotor through the interfaces with the nacelle main frame.

Table I. Definition of the loads at the interfaces of the gearbox.

Interface	Loading	Synchronicity	Analysis
Main shaft and gearbox	Loads: main shaft axial and shear forces, bending moments and torsion (torque) Kinematics: main shaft angle and speed, and axial displacement	WTOD	Mean loads fatigue loads (RFC, LDD)
High speed shaft and gearbox	Loads: axial and shear forces, bending moments and torsion (torque) Kinematics: high speed shaft angle and speed, and axial displacement	WTOD	Mean loads fatigue loads (RFC, LDD)
Torque arm and gearbox	Kinematics: axial, vertical and tangential displacement[a]	WTOD	
Gearbox housing	Frequency response of gearbox housing at suitable frequency ranges	WTOD	
Additional measurements (internal to the gearbox system)	Lubrication temperature on gearbox bearings, gear meshes or overall volume temperature	WTOD	

[a]The overall three-dimensional rotation of the gearbox can be estimated/measured through the combination of the three-dimensional displacements measured on each torque arm.

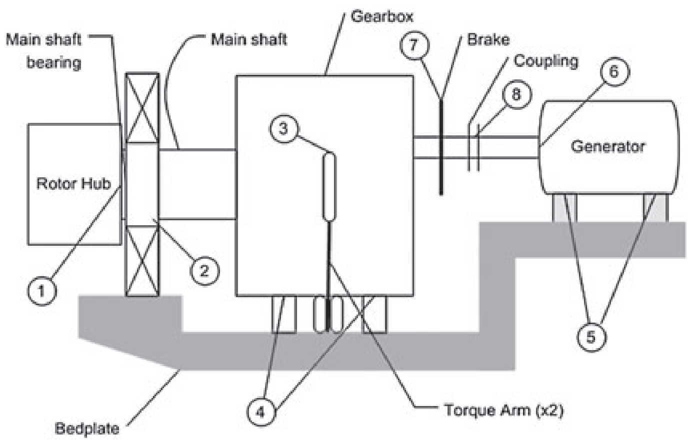

Figure 2. Schematic of the nacelle layout, with a triple-point suspension, showing the interfaces.[18]

Classifying the general loads transferred across the interfaces of the drive train as loads, kinematics and dynamics and the following parameters should be defined for the drive train.

Loads:

- Axial and shear loads, bending moments and torsion of the low speed shaft (at the rotor interface);
- Axial and shear loads, bending moments and torsion of the high speed shaft (at the generator interface);
- Forces and moments of the torque arms; and
- Forces at the main bearing(s) on their interfaces on the nacelle main frame (if applicable).

Kinematics:

- Displacements at the supports;
- Positions (angle, speed of rotation and axial displacement) of moving (rotating) elements (e.g. shafts); and
- Rotation and axial displacement of the torque arms.

Dynamics:

- Accelerations of moving (rotating) elements (e.g. shafts).

Identical to the gearbox discussion, synchronization of the general loading conditions is required with the WTOD.

If any other drive train component is analysed as a stand alone, e.g. main bearing, interfaces should be identified for this component as well.[18]

On the basis of the results for the loads transferred across the interfaces of the *gearbox*, Table I presents a summary of the recommended quantities to be measured during an experimental campaign focusing on the gearbox. The same table can be used as a starting point for the definition of loads transferred across the interfaces of the drive train, properly adjusted for the specific wind turbine configuration.

IEC/TS 61400-13[9] should be followed wherever possible. However, in order to better illuminate the load cases that affect the components/systems under study, the following presentation/analysis should be added for the load measurements regarding the drive train and the gearbox of the wind turbine.

- A selection of measurement cases, which can be used for the validation of wind turbine design models should be made, assuring the atmospheric conditions and the specific turbine characteristics, as described in IEC 61400-4.[6] This is necessary in enabling the accurate reproduction of the as-measured response, using data from the field tests.
- Analysis specifically intended for the verification of design assumptions for the gearbox, including torsional vibration, combined structural response and reaction at the gearbox supports and interfaces, as described in IEC 61400-4.
- Analysis regarding the drive train resonances including vibration levels at representative locations (possible corresponding to work shop testing locations), following IEC 61400-4.
- Measurements and analysis regarding the lubrication delivery/cooling system effectiveness including temperatures, as described in IEC 61400-4.

According to IEC 61400-4, in addition to load measurements prescribed in the IEC/TS 61400-13, the torque on the low and the high speed shaft should be measured in experimental campaigns requiring the verification of the gearbox and the drive train. Additionally, the shaft speed should also be measured. According to IEC 61400-4, additional load measurements for forces and bending moments may be required for the evaluation of the gearbox interface loads and design assumptions.

Following IEC 61400-4, the sampling rate should be adequately selected (in cooperation with the gearbox manufacturer) for each application, higher than three to five times the relevant vibration frequency.

Additionally, following IEC 61400-4, a Campbell diagram (plot of the system forcing and response frequencies) should be provided through the complete operating speed range to evaluate the resonance risk.

Finally, measured temperatures at specified locations on the gearbox and lubrication system should be reported, with emphasis on maximum temperatures and maximum temperature durations. If applicable, during the measurement campaign, lubricant analysis shall also be performed and reported.

4.3. Results

The six-step approach (Section 3) is tested in a case study on the drive train of the S82 1500kW wind turbine.[15,19] The analysis was not performed for one specific failure mode (Step 1) but with different targets, e.g. validating the drive train stiffness.

Different models were designed (Step 2) to perform the analysis: a FLEX5 (DTU, Lyngby, Denmark) model, a model in SIMPACK (SIMPACK AG, Gilching, Germany) with a topology similar to the FLEX5 model and a wind turbine model with a more sophisticated torsional drive train model, also in SIMPACK. Different load cases were calculated using these three models (Step 3).

In Step 4, the uncertainties in the input and output parameters of the models have been assessed. For the masses and inertias of the drive train components, the uncertainty is small, but for many other input parameters, the uncertainty becomes larger. The most difficult parameters to precisely determine are the damping values. Approximated values that have been determined empirically depend not only on the material properties (material damping) but also greatly on the component geometry (structural damping). Furthermore, damping is described by the medium in which the mechanical parts are moving (viscous damping). In addition, the complexity of the equations of motion behind the multi-body system makes it difficult to analytically derive the uncertainty of the output result from the uncertainties of the different inputs. An attempt in getting an impression of model reliability is to carry out a sensitivity analysis on the different input parameters. It gives a rough approximation of the influence of input uncertainties on the simulation results.

In Step 5, the measurement campaign was designed. This campaign should enable verification of the models. The measurements included a number of signals with special relevance for drive train load assessment:

- torque and bending loads on the main shaft
- torque on the high speed shaft
- rotational speed high speed shaft
- rotational speed intermediate speed shaft
- rotational speed main shaft
- rotor position main shaft
- axial displacement of high speed shaft
- axial displacement of intermediate speed shaft
- axial displacement of low speed shaft
- displacement of the gearbox in the nacelle
- outdoor temperatures
- ambient temperatures
- air flow (cooler input and output) temperatures
- bearing temperatures high speed shaft
- bearing temperatures intermediate speed shaft
- oil sump temperatures
- oil in cooler temperatures
- oil pressures

These measurements have been realized on a SUZLON (Suzlon, Pune, India) S82 1500kW wind turbine situated nearby Sankaneri, Tamil Nadu in India. At least, measurements of rotational speeds and torques at input and output shaft of the gearbox were considered necessary. In the context of drive train model validation and in the attempt to quantify the relevant parameter stiffness, damping and inertia of the drive train, the focus has been placed on the rotational speed, angle and torque measurements at the main shaft (low speed shaft) and the generator shaft (high speed shaft).

Finally, in Step 6, the measurement results were processed. For example, a statistic approach was used to determine the overall stiffness of the drive train. In stationary operation near the first drive train resonance frequency, it is assumed that the effects of inertia and damping are small and can be neglected when looking at consecutive periods of steady state operation. This means that the quasi-steady state variation of the torque will be determined through the drive train stiffness

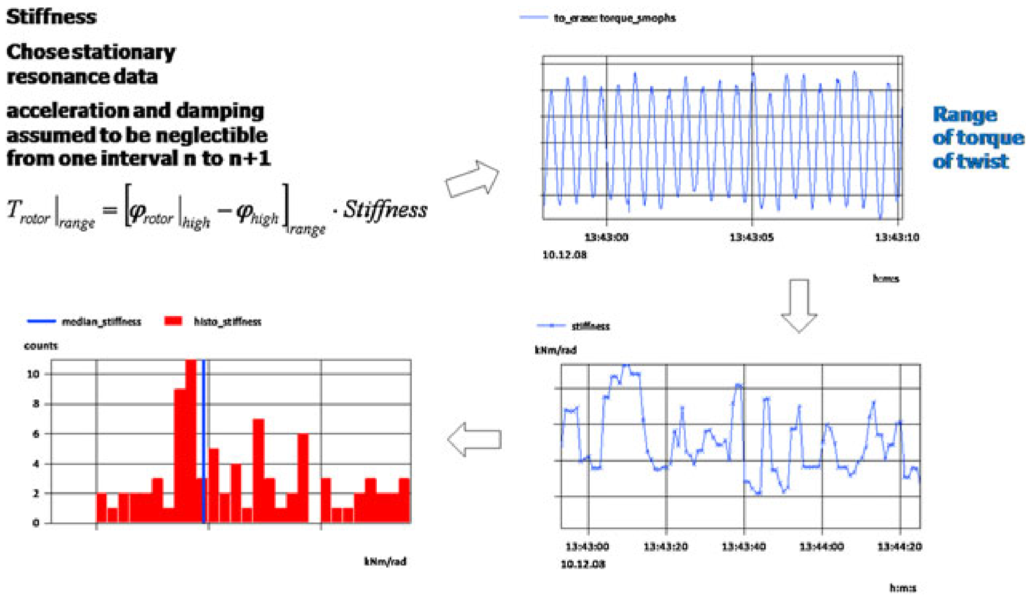

Figure 3. The median value and frequency distributions for the stiffness solutions.

and the differential angle of both shaft ends derived from the rotational speed sensors. This results in a simple equation of motion

$$T_{rotor}|_{range} = [\phi_{rotor}|_{high} - \phi_{high}]_{range} \cdot Stiffness \tag{1}$$

where $T_{rotor}|_{range}$ is the mechanical torque in the rotor at the low speed shaft converted to the high speed side using the gear ratio and by making assumptions for gearbox and generator losses, $\phi_{rotor}|_{high}$ is the angle of the low speed shaft converted to the high speed side using the gear ratio, ϕ_{high} angle of the high speed shaft and $stiffness$ the overall stiffness of the system.

In the campaign, the angles of the shafts have been derived on the basis of the measured speeds (relative to the gearbox). The first to perform is to obtain the torque variation range ($T_{rotor}|_{range}$) through a given period of time. This time must be long enough to allow the measured torque signal to include at least one full swing from the minimum to the maximum torque of that quasi-stationary oscillation. The time period has been chosen to be 5 s, which approximately is 1.3 times longer than the period of one low speed shaft revolution and is several times longer than the drive train natural frequency.

The rotational angle of the low speed shaft has been translated to the equivalent angle increment at the high speed shaft, i.e. has been multiplied by the ratio of the gearbox. As previously carried out for the torque, the variation of the angular difference between high speed shaft angle and translated rotor shaft angle has been determined.

Solving equation 1 provides a solution for the overall stiffness every 5 s or 120 solutions in a 10 min time series in which the drive train natural frequency is excited according to the initial assumption for equation 1 to hold true. Figure 3 shows the median value and frequency distribution for stiffness solutions.

5. PITCH SYSTEM

As stated before, the PROTEST project focused on the drive train, pitch system and yaw system. For the pitch system, no new DLCs were identified. The critical load sets are already included in the current sets of DLCs.[20] In Section 5.1, the interfaces and the loads transferred across these interfaces are discussed. The new six-step approach has been utilized to set up and use a measurement campaign in order to validate a friction model for the blade bearing. The method and results for this approach are discussed in Section 5.2.

5.1. Pitch system interfaces and definition of transferred loads

When determining the procedure used to describe the method to define loads at the interconnection points, the specification of the interfaces of the pitch system and its subcomponents is required. This includes isolation of each system or subcomponent from the overall wind turbine structure and creating an adequate description of the sectional loads at the interconnection points (interfaces).

For the pitch system, the working draft IEC 61400-4,[6] where the relevant issues of the wind turbine gearbox are discussed, was used as a starting point to determine the interfaces and what kind of information are necessary at these interfaces used to create a reliable design for the mechanical components of the pitch system. Four interfaces for the pitch system can be defined (see figure 4): the interface between the blade and the pitch system (bearing); the interface between the hub and the pitch system (bearing); the interface between the hub and the pitch system (transmission and drive); and the interface between the controller and the pitch system (drive).[21]

Specifically, for the pitch system, there are two distinct cases: the pitch system is used to keep the blade at a pre-defined position (as defined by the controller), or the pitch system is used to bring the blade into the required position (pitching). These two cases should be clearly discerned and connected with wind flow conditions and operating states of the wind turbine, as the intermittent/oscillating behaviour is essential for the pitch (bearing) design and life time.

The pitch system transfers axial and shear forces, bending moments and torsion from the blade to the hub. Bending moments are measured during conventional load measurement campaigns (as specified in IEC 61400-13[9]). The force measurements, however, are not required and, therefore, usually not performed. The performance of accurate torsion measurements is still very difficult or maybe even impossible.[22] It is possible to estimate the forces and torsion by using aeroelastic simulations, but it is difficult to simulate similar conditions because of wind speed variations in space and time.

In[22], it was concluded that the bearing friction and bearing deformation are responsible for the critical failure modes. The friction moment depends on the geometry of the pitch bearing (diameter), the friction coefficient of the pitch bearing and the bending moments, axial and shear forces transferred from the blade (root) through the pitch bearing.

Another issue of special importance to the pitch system is the effect of the elastic deformation (ovalisation and twisting) of the pitch bearing rings on the loading of the pitch components, as a result of the deformation of the blade root, the hub and the bearing rings themselves because of the acting loads on the blade. Table II presents a summary of the recommended quantities to be measured during an experimental campaign focusing on the pitch system components.

Additional measurements and analyses that are recommended in order to obtain more knowledge of the system and validate models and design calculations (e.g. fatigue life):

- For the pitch bearing: blade torsion and blade root axial and shear forces, estimated through simulations
- Estimation of the frictional torque
- For the wind turbine behaviour in relation to the pitch system, the time delay from the pitch control set-point to blade pitch angle/speed
- Pitch bearing deformation measurements (on bearing rings and/or blade flange and hub mounting): these can be used to investigate the influence of the stiffness of the mounting flanges and of the support structures (blade and hub). Also, the effect on bearing friction (and, thus, wear and pitch driver load) should be addressed.
- Lubrication contamination: lubrication (grease) of the pitch bearing is essential for the fatigue life, especially when the bearing is in oscillating motion. Also, the lubrication (oil) of the pitch transmission can be monitored to investigate the wear in the pitch drive train.
- Electrical load between bearing rings due to high voltage lightning strikes: lighting strikes (count) on the blades can cause bearing raceway degradation if no proper provision is available for routing the charge.

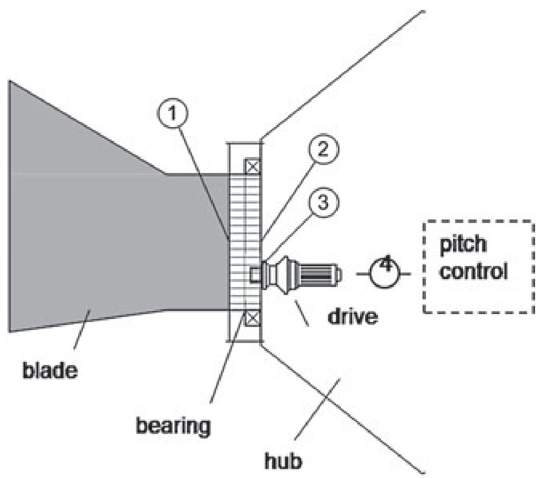

Figure 4. A simplified sketch of the pitch system showing its main components and the four interfaces.

Table II. Definition of the loads at the interfaces of the pitch system (internal and external).

Interface	Loading	Synchronicity	Analysis
1. Blade and pitch system (bearing) - external	Loads: blade root forces (axial, radial, shear) and moments (bending, torsion) Kinematics: measured at 2 Dynamics: measured at 2	WTOD blade pitch angle and pitch speed in 2	extreme loads mean loads fatigue loads (LDD)
2. Hub and pitch system (bearing) - external	Loads: measured at 1 Kinematics: pitch angle and pitch speed Dynamics: acceleration on the blade in 2 perpendicular directions	WTOD with loads in 1	time at level of pitch angle (LDD) oscillation of the pitch angle (rain-flow)
3. Hub and pitch system (transmission and driver) - external	Loads (driver): reaction torque/force of pitch driver on hub Loads (transmission): reaction torque (or force at torque arms) on hub	WTOD	
4. Controller and pitch system (driver) - external	Loads: driver voltage and current / pressure and flow Kinematics: control set point (pitch angle/speed)	WTOD with loads in 1	thermal load (LDD of RMS value)
Bearing outer ring and bearing inner ring - internal	Kinematics: clearance (at the four quarters on the bearing)	blade pitch angle and blade root forces and moments	
	In case of an electric pitch actuator:		
Driver pinion and ring gear - internal	Kinematics: relative angle of rotation	blade pitch angle	
Gearbox and driver pinion - internal	Loads: driving torque		
Motor and transmission - internal	Loads: driving torque Kinematics: rotational speed	blade pitch angle	peak load
	In case of a hydraulic pitch actuator:		
Motor and transmission - internal	Loads: force in driving rod Kinematics: speed and position (nonlinear transmission)	blade pitch angle	

- Temperature on frictional parts: friction in the pitch drive train and the pitch bearing causes extra load on the pitch drive, which could lead to increase in temperature.

Regarding the presentation of the measurements, specifically for pitch components, the following presentations of measurements are recommended to be included in the test report for such a campaign:

- For the pitch actuator: torque of the pitch motor, time series and root mean square (RMS) per wind condition (wind speed and turbulence)
- For the pitch transmission system: meshing torque of the pitch bearing, time series and rain-flow-counting matrix (RFC) per wind condition
- For the pitch bearing:
  - Loads (forces and moments) time series and RFC per wind condition
  - Kinematics (mean/amplitude/speed) per wind condition
  - Temperature (if available) in relation to other measurements
  - Acceleration PSD per wind condition

Additional information in statistical terms per wind condition (wind speed, turbulence) and wind turbine condition (normal operation or standstill) should be provided regarding pitch operation. These, for example, can be:

- Start the pitch system within a 10 min captured file, time of operation, time duration up to next start
- Average angle of rotation for each single operation (i.e. average change in pitch angle for each pitch operation) and average speed of rotation per wind condition

It should be noted that the definitions and procedures of IEC/TS 61400-13 should be followed as close as possible for all measurements conducted and the presentation of output.

5.2. Results

The six-step approach, as explained in Section 3, has also been applied to the pitch system. As an illustration, some results will be shown here that concern the friction torque of the pitch bearing.[15]

For the pitch system, the two most relevant failure modes are a reduced performance of the electric motor, whereby less power is provided, and a reduced performance of the slewing bearing, increasing the power needed for pitch angle changes. The bearing deformation and the friction were found to be important causes for both failure modes, and the analyses was therefore limited to these two causes. However, other possible causes were also identified for the critical failure modes and should not be neglected.[23]

For the friction, a model has been designed which uses the outcome of PHATAS (ECN, Petten, the Netherlands) simulations[24] as the input in calculating the friction and the pitch motor torque, as illustrated in Figure 5.

The friction torque M_r refers to the starting torque and can be found in[25] to be

$$M_r = \frac{\mu}{2}(4.4M_k + F_a D_L + 2.2F_r + D_L \cdot 1.73) \tag{2}$$

with

M_r	starting torque	$[Nm]$
F_a	axial load	$[N]$
F_r	radial load	$[N]$
M_k	resulting bending moment	$[Nm]$
D_L	bearing race diameter	$[m]$
μ	friction coefficient	$[-]$

Reference[25] gives the friction coefficient μ for a double-row ball bearing slewing ring of 0.004.

The pitch motor torque T_m can be compared to the torque because of acceleration of the blade plus blade torsion and the friction torque

$$T_m i_{gbx} i_{gear} = T_b + \frac{\dot{\theta}}{|\dot{\theta}|} M_r + \ddot{\theta}\left\{i_{gbx}^2 i_{gear}^2 I_{gbx} + I_b\right\} \tag{3}$$

The measured pitch motor torque has to be multiplied by the gearbox and pinion gear ratios i_{gbx} and i_{gear}, respectively. T_b is the calculated torque acting on the blade; M_r is the friction torque (its sign is dependent on the direction of rotation), and $\ddot{\theta}$ is the acceleration of the rotor blade about the pitch axis. I_{gbx} and I_b are the inertia of the gearbox of the blade from root to calculation location.

The analysis of the time series of the measured pitch motor torque data revealed that the maximum pitch motor torque always occurs at the start of a normal shutdown. This corresponds to the initial design consideration, where this starting torque was considered as a critical design parameter.

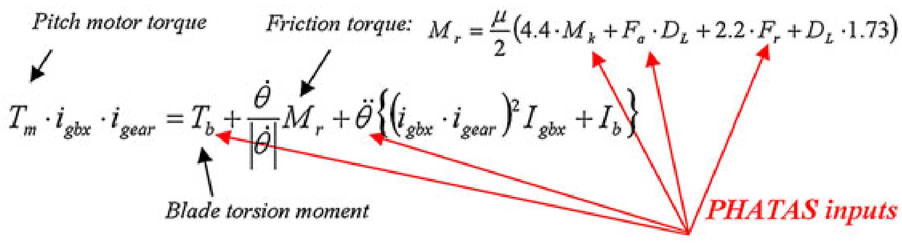

Figure 5. Model for the pitch torque calculation with μ as the friction coefficient, M_k the resulting bending moment (Kipp moment), F_a the axial force on the bearing, F_r the radial force on the bearing, D_L the bearing race diameter, i_{gbx} the gearbox ratio, and i_{gear} the pinion gear ratio, θ the pitch angle, I_{gbx} the inertia of the pitch gearbox, and I_b the blade inertia from the blade root up to the torsion calculation location.

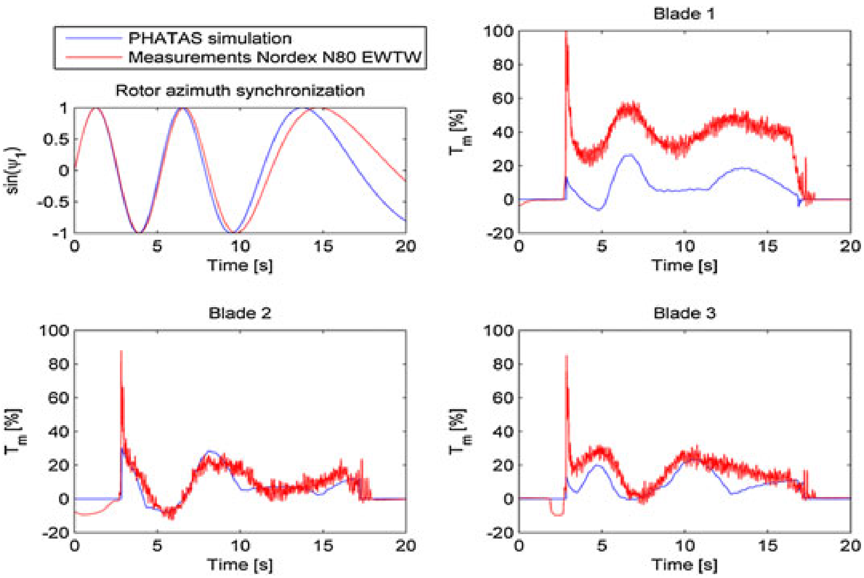

Figure 6. Synchronized pitch motor torque comparison, where the red lines indicate the measurements and the blue lines indicate the modelling results.

Figure 6 shows the results from the measurements and the model concerning a case of a normal shutdown operation, which is in fact the transition from running to idling at low wind speed. It illustrates that the pitch motor torque model has quite good qualitative similarities for all three rotor blades. The trend in the pitch motor torque during blade rotation follows a similar path. However, quantitatively, the model has fewer agreements with the measurements. Specifically, the starting torque peak seems to be much lower in the model compared with the measurements.

The measurement campaign also revealed a large scatter in the maximum pitch torque measurements for the normal shutdown. An effort was made to understand the cause of this scatter by studying the relation between the pitch motor torque and other measured signals, but no clear correlation was found, other than the lagging of the pitch rotation with respect to the pitch setpoint. The peak torque includes high frequency dynamics and a relatively short time span (see Figure 6). As after a certain time, the pitch torque seems to be limited. This leads to the conclusion that the pitch system electronics and motor controller most likely play an important role here. It is therefore recommended to add these to the model for pitch drive train analysis if this effect is to be captured.

During the blade rotation, illustrated in Figure 6, the model fits relatively well to Blades 2 and 3, whereas Blade 1 seems to have a clear offset with respect to the model. Other time series showed similar offsets for the measured pitch motor torque of Blade 1. Moreover, the offsets of the pitch torque for the three blades changed significantly after maintenance was performed on the test wind turbine near the end of the measurement campaign. These observations illustrate the necessity of adding a load-independent term to the friction model. Also, a higher pitch speed has an influence on the pitch torque offset, which can be included by adding a speed dependent term to the friction model.

6. YAW SYSTEM

The final system that has been analysed is the yaw system. For the yaw system, no new DLCs were identified; the critical load cases are already included in the current sets of DLCs.[20] In Section 6.1, the interfaces and the loads transferred across these interfaces are discussed. An illustration of the results using the six-step approach is discussed in Section 6.2.

6.1. Yaw system interfaces and definition of transferred loads

Similar to the pitch system, when determining the procedure used to describe the method to define loads at the interconnection points of the yaw system, each system or subcomponent of the yaw system was isolated from the overall wind turbine structure, and an adequate description of the sectional loads at the interconnection points (interfaces) was created.

Again, for the yaw system, the working draft IEC 61400-4[6] was used as a starting point to determine the interfaces and the kind of information necessary at these interfaces.

Table III. Definition of the loads at the interfaces of the yaw system.

Interface	Loading	Synchronicity	Analysis
Tower top and yaw system	Loads: tower top axial and shear forces, bending moments and torsion Kinematics, dynamics: measured at the nacelle	WTOD	Mean loads fatigue loads (LDD)
Nacelle and yaw system	Loads: measured at the tower top Kinematics: nacelle yaw position and speed Dynamics: acceleration on the nacelle bearing in two perpendicular directions	WTOD	
Yaw transmission system (gear) and yaw system	Loads: torque (pressure) Kinematics, dynamics: measured at the nacelle	WTOD	Uneven torque distribution
WT controller and yaw system	Yaw system power consumption command	WTOD	
Yaw bearing and yaw system (internal system measurements)	Additional measurements: temperature at the yaw base and frictional parts		

WT, wind turbine.

Regarding the loads that are transferred through the interfaces of the yaw system, there are two distinct cases: (i) the loads transferred while the yaw system is used to keep the nacelle position at a pre-defined position, i.e. non-yawing (as defined by the controller); and (ii) the loads transferred while the yaw system is active and used to bring the nacelle into the required position, i.e. yawing. These two cases should be clearly distinguished and connected with conditions of the wind flow and the wind turbine.

For the loads to be transferred while the yaw system is used to keep the nacelle at a defined orientation angle, i.e. non-yawing, all loads acting on the nacelle end should be transferred to the tower. That is, the yaw system should transfer axial and shear forces, bending moment and torsion from the nacelle to the tower. These are already estimated through aeroelastic simulations. Tower top bending moments and torsion can be measured during conventional load measurement campaigns.

Torsional motion of the yaw system (while the system is maintaining nacelle position), i.e. non-yawing, could be measured on an operating wind turbine with vibration sensors positioned at the nacelle part of the yaw system, measuring possible small torsional vibrations (motion and acceleration).

For the loads to be transferred while the yaw system is operating (driving the nacelle to the requested nacelle position), i.e. yawing, the loads to be transferred through the yaw system are again all axial and shear forces, as well as bending moments acting on the tower top, whereas torsion will be transferred to the tower distorted through the action of the yaw actuator (driver).

This last load component (torsion) should form the load at the relevant interfaces for the yaw system, which affects the loading on the gear of the yaw system (the transmission subsystem) as meshing torque and the torque on the driver of the yaw system.

For the load-dependent frictional moment, several practical estimates are available,as discussed in the previous section.

On the basis of the results for the loads transferred across the interfaces of the yaw system, Table III presents a summary of the recommended quantities to be measured during an experimental campaign focusing on the yaw system components.

Regarding the presentation of the measurements, specifically for yaw components, the recommendation is identical to the recommendation given for the pitch system in Section 5.1, except for the yaw components and yaw angles.

6.2. Results

On the basis of the determined loads to be transferred across the interfaces of the yaw system, as described previously, a measurement campaign was designed and carried out within the PROTEST project. The campaign was carried out on the NM44/750 wind turbine installed at the Centre for Renewable Energy Sources and Saving wind farm in Lavrion, Greece. To the authors' knowledge, it is the first time that shear and axial forces were directly measured on a wind turbine tower. The specific measurements were performed near the tower top. These are to be used for the direct estimation of the yaw bearing loads, as well as incorporated within the estimation of the frictional moment to determine the actual loads on the yaw motor.

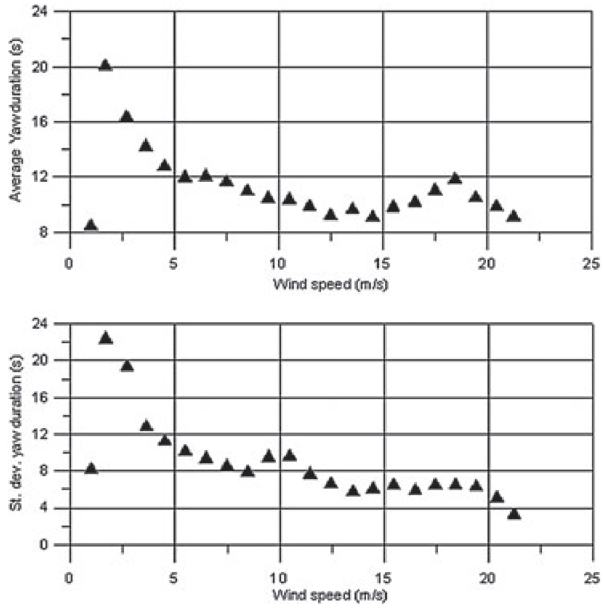

Figure 7. Yaw duration statistics binned with respect to wind speed (complete data base)

Regarding the kinematic statistics necessary in designing the yaw system actuator, analysis has been conducted to determine the duration of yaw, time of start between yaw motions, etc. An indicative plot is shown in Figure 7, where the mean of the yaw duration and the respective standard deviation is presented binned, with respect to the 10 min mean wind speed. Although the results are site-specific, as well as dependent on the controller settings of the wind turbine, these can be used in assessing the design load cases taken into account for the yaw system.[15]

7. CONCLUSIONS

The relevance of improved reliability of wind turbines from an economic point of view, the significant costs involved in operation and maintenance and downtime when mechanical components need to be replaced initiated the PROTEST project. This pre-normative project, aimed at extending the procedures concerning the design of the wind turbine, as well as prototype testing with regard to the mechanical systems, has resulted in the suggestion of three new DLCs. These three DLCs are relevant for the drive train. Including these new DLCs in the analysis will result in a higher accuracy of the analysis of the loads on the drive train components. First, misalignment may cause among other constraining forces on the gearbox, reducing the expected life time compared to the analysis that does not take misalignment into account. Misalignment is not only due to incorrect assembly, it can also be caused by the flexibility of the structure. Second, resonance of the drive train should be investigated in more detail. The frequency ranges that are relevant in the drive train components are very wide, and relevant ranges are well outside the scope of traditional wind turbine tools. By extending the analysis to different frequency ranges, the risk of failure due to resonance can be reduced. Third, the low voltage ride through should be analysed in more detail as it is known to result in significant loads on the drive train. Applying these three extensions to the current set of DLCs is expected to improve the reliability of the drive train. During the project, they have been included in the latest GL guidelines.[16]

Complementary procedures are developed to improve the specification of the design loads at the interfaces, where the mechanical components (pitch and yaw system, as well as the drive train) are attached to the wind turbine. This is essential in enabling use of the results from current wind turbine simulation tools as input for more detailed mechanical system models, e.g. finite element method (FEM) models. Several improvements are suggested in order to enable the improved analysis of the drive train, pitch system and yaw system, using the prototype measurements. Next to this, a procedure on how the prototype test could be used to validate the models of these mechanical systems have been developed, the six-step approach; as at the moment, the prototype test seems to have a focus that limits itself to the blades and tower. As the reliability of the mechanical components needs to be improved, it is clear that there is a lot to be gained from improving the definition of the loads transferred across the interfaces and extending the prototype measurements to include the mechanical systems. By using the suggested improvements in analysis and prototype measurements, it becomes possible to increase the accuracy of the predicted loads on the components and, thereby, enhance the component designs, which should prevent the need of early replacements.

DOI: 10.1002/we

ACKNOWLEDGEMENTS

The PROTEST project was co-funded by the European Commission within the Seventh Framework Programme, under grant number 212825, for which the authors are very grateful. All partners of the PROTEST project are also acknowledged for their input during the project.

ABBREVIATIONS

WTOD: Wind Turbine Operation Data
LDD: Load duration distribution
RMS: Root mean square
RFC: Rain flow count
DLC: Design load case
LVRT: Low voltage ride through
PSD: Power spectral density

REFERENCES

1. BTM Consult. International wind energy development; World Market Update 2005. *Technical Report*, BTM Consult, Ringkøbing, Denmark March 2006.
2. ISET, Wind energy report Germany 2001. *Technical Report ISET*, Kassel, Germany 2002.
3. Energi- og miljødata (EMD). Database with energy production figures, incidents and accidents.
4. Wiggelinkhuizen E, Verbruggen T, Braam H, Rademakers L, Xiang J, Watson S, Giebel G, Norton E, Tipluica MC, Maclean A, Christensen AJ, Becker E, Scheffler D. Conmow: condition monitoring for offshore wind farms, *EWEC 2007 Proceedings*, Milan, Italy, 2007.
5. Link H, LaCava W, van Dam J, McNiff B, Sheng S, Wallen R, McDade M, Lambert S, Butterfield S, Oyague F. Gearbox reliability collaborative project report: findings from Phase 1 and Phase 2 testing. *Technical Report NREL/TP-5000-51885*, NREL, USA, June 2011. [Online]. Available: http://www.nrel.gov/wind/pdfs/51885.pdf.
6. IEC 61400-4 WD3 2008-06. *Design requirements for wind turbine gearboxes*. IEC: International Electrotechnical Commission: Geneva, 2008-06.
7. IEC-61400-1. *Wind Turbines—Part1: Design Requirements*, (third edn). IEC: International Electrotechnical Commission: Geneva, 2005-08.
8. Germanischer LL. *Guideline for the Certification of Wind Turbines*, (2003 edn). GL: Hamburg, Germany, 2003. With supplement 2004.
9. IEC-61400-13. *Wind Turbines Generator Systems—Part 13: Measurement of Mechanical Loads*, (first edn). IEC: International Electrotechnical Commission: Geneva, 2001-2006.
10. IEC-61400-1. *Wind Turbine Generator Systems—Part 1: Safety Requirements*, (second edn). IEC: International Electrotechnical Commission: Geneva, 1999-02.
11. IEC/TS 61400-23. *Wind Turbines Generator Systems—Part 23: Full-scale Structural Testing of Rotor Blades*, (first edn). IEC: International Electrotechnical Commission: Geneva, 2001-04.
12. Argyriadis K, Capellaro M, Hauptmann S, Kochmann M, Mouzakis F, Rademakers L, Ristow M. Deliverable D1: state-of-the-art-report. *Technical Report*, University of Stuttgart, Stuttgart, Germany, 2009. Available: http://www.protest-fp7.eu/publications/ (Accessed 18 May 2010).
13. *Standard DNV-DS-J102, Design and Manufacture of Wind Turbine blades, Offshore and Onshore Wind Turbines*. DNV: Høvik, Oslo, 2001-2004. http://www.dnv.com.
14. Tavner PJ, Xiang J, Spinato F. Reliability analysis for wind turbines. *Wind Energy* 2007; **10**: 1–18. DOI: 10.1002/we.204. http://dx.doi.org/10.1002/we.204.
15. Holierhoek JG (ed.). Protest, final report, project results and recommendations for standardisation. *Technical Report ECN-E–10-100*, ECN, Petten, the Netherlands, 2010. Available: http://www.protest-fp7.eu/publications/ (Accessed 18 November 2010).
16. Germanischer LL. *Guideline for the Certification of Wind Turbines*, (2010 edn). GL: Hamburg, Germany, 2010.
17. Hansen AD, Cutululis NA, Markou H, Sørensen PE. Impact of fault ride-through requirements on fixed-speed wind turbine structural loads. *Wind Energy* 2011; **14**(1): 1–11. DOI: 10.1002/we.398. http://dx.doi.org/10.1002/we.398.

18. Lekou DJ, Mouzakis F. Template for the specification of loads necessary for designing drive train systems. *Technical Report Deliverable D3*, CRES, Pikermi Attiki, Greece, 2010. Available: http://www.protest-fp7.eu/publications/ (Accessed 11 November 2010).
19. Söker H, Monux O, Ehlers BM, Stache F, Smolders K, Peeters J, Hecquet T. PROTEST—Procedures for Testing and Measuring Wind Energy Systems Drive Train Case Study II. *DEWEK 2010, Bremen, Germany*, 2010. Available: http://www.protest-fp7.eu/publications/ (Accessed 11 February 2011).
20. Holierhoek JG, Braam H, Rademakers LWMM. Protest, determination of load cases and critical design variables. *Technical Report ECN-E–10-007*, ECN, Petten, the Netherlands, 2010. Available: http://www.protest-fp7.eu/publications/ (Accessed 18 May 2010).
21. Lekou DJ, Mouzakis F, Savenije FJ. Template for the specification of loads necessary for designing pitch systems. *Technical Report Deliverable D4*, CRES, Pikermi Attiki, Greece, 2010. Available: http://www.protest-fp7.eu/publications/ (Accessed 11 November 2010).
22. Holierhoek JG, Korterink H, van de Pieterman RP, Braam H, Rademakers LWMM, Lekou DJ. Protest, recommended practices for measuring *in situ* the 'loads' on drive train, pitch system and yaw system. *Technical Report ECN-E–10-083*, ECN, Petten, the Netherlands, 2010. Available: http://www.protest-fp7.eu/publications/ (Accessed 11 November 2010).
23. Holierhoek JG, van de Pieterman RP, Braam H, Savenije FJ, Korterink H. A six steps approach to set up a measurement campaign to validate or improve the wind turbine component model. *Wind Engineering* 2011; **35**(4): 381–396. DOI: 10.1260/0309-524X.35.4.381.
24. Lindenburg C, Surname2 FS. PHATAS release "NOV-2003" and "APR-2005" USER'S MANUAL. *Technical Report ECN-I-05-005*, ECN, Petten, the Netherland, 2005.
25. Rothe Erde slewing bearings, catalogue ThyssenKrupp, 2007. Available: www.rotheerde.com (Accessed 17 May 2010).