Setting up a prototype measurement campaign for mechanical components

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Abstract

The reliability of mechanical components of wind turbines needs to be improved. In the PROTEST project, a pre-normative project, 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 has resulted in a suggestion for three new DLC's that are mainly due to the limitations of the current day tools. Next to this, a new approach of how to set up and use the prototype measurement campaign has been created. This approach consists of six steps that need to be taken in order to validate or improve the models used for the design of the mechanical components. This new approach is illustrated using the pitch system as an example. In this example the focus is on the friction in the pitch bearing, which is calculated and compared to measurements.

Keywords: Mechanical components, Measurement campaign, Reliability, Yaw system, Pitch system, Drive train

Definitions:

DLC

Design Load Case; the combination of operational modes or other design situations, such as specific assembly, erection or maintenance conditions, with the external conditions [6].

Design load

The design load is defined as the load for which the strength of any component has to be documented. It generally consists of the so-called characteristic load multiplied with the appropriate partial safety factors for loads and consequence of failure, see also IEC 61400-1

Limit state

State of a structure and the loads acting upon it, beyond which the structure no longer satisfies the design requirement [ISO 2394, modified] [Ref.6].

CDV

Critical Design Variable; A design variable that from experience is expected to strongly affect the design.

Failure mode

The mode of failure. Passing over a specific limit state described by a single equation could lead to different failure modes depending on the vector followed when passing from the safe state to the failure state.

1. Introduction

High reliability of wind turbines and their components is one of the pre-requisites for an economically viable exploitation of wind farms, especially for offshore. Therefore the reliability needs to be increased. 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 PROTEST project. Finally the outline of this paper is discussed.

1.1 Reliability of Mechanical Components

The reliability of wind turbines and their components is at present not yet at an acceptable level, wind turbines still show failure rates between 2 to 5 failures per year that need visits from technicians (derived from i.e. [1,2,3]). Although electrical components and control systems fail more often, the costs related to repair of failed mechanical systems (drive train, pitch and yaw systems and bearings) are dominating the O&M costs and downtime.

In-depth studies, e.g. [4] and discussions with turbine manufacturers, component suppliers, and certification bodies [5] revealed that one of the major causes of failures of mechanical systems is insufficient knowledge of the loads acting on these components. This lack is a result of the shortcomings in load simulation models and in

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load measurement procedures on the level of these components.

It was also concluded from a.o. [4] and expert discussions [5] that, at present, the procedures and standards for designing rotor blades and towers of wind turbines are much more specific than the procedures for designing other mechanical components such as drive trains, pitch and yaw systems, or main bearings. Lacking of clear procedures for designing mechanical components and specifying the loads on these components should no longer be the reason for early failures.

1.2 The PROTEST Project

PROTEST, the acronym of the full title "*PROcedures for TESTing and measuring wind energy systems"* is a pre-normative project that should result in complementary procedures to better specify and verify the local component loads acting on mechanical systems in wind turbines. PROTEST started in 2007 within FP7.

Within this project the following questions need to be answered [9]:

- How should the loads at the interfaces be derived from the global turbine loads?
- Which design load cases should be considered and measured and are relevant for the different components?
- Which signals should be measured during prototype testing (including sample frequency, accuracy, duration)?
- How should the loads at the interfaces be reported and communicated between turbine manufacturer and component supplier?
- How can design loads be compared with measured loads?
- Are the current practices of evaluating the experimental data in relation to their use for model tuning accurate?
- Do the assumptions in the model input yield to uncertainties which are higher than the ones achieved during the load measurements?
- What are the criteria to assess whether the measured loads are more benign than the calculated loads?
- Are the current practices of assessing the measured loads and the data post processing results adequate?

In this paper only the following three questions will be dealt with:

- Which design load cases should be considered and measured and are relevant for the pitch system?
- Which signals should be measured during prototype testing (including sample frequency, accuracy, duration)?
- How can design loads be compared with measured loads?

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, it can be used to tune and validate the models that have been used for the design of the mechanical systems.

1.3 Outline

First the question of possible additional design load cases that are needed, are discussed in section 2. Then the six step approach is described in section 3 followed by an example of this approach for the pitch system. In this example the method is used to analyse the friction in the pitch bearing. Finally the conclusions are given.

2. Additional Design Load Cases for Drive Train, Yaw and Pitch System

For designing wind turbines use is made of wind energy standards and/or guidelines, like the IEC-61400-1 [6] and the GL Guidelines for the Certification of Wind Turbines [7]. In these standards a number of Design Load Cases (DLC's) are specified which have to be analysed. DLC's for wind turbines are the combination of the design situations of a wind turbine (both operational modes and transient modes) with wind conditions (gusts) and other external conditions (e.g. grid failures and lightning). In these standards the procedures for designing wind turbine rotor blades and towers are much more specific and well documented compared to the design procedures of other mechanical systems, such as the drive train, pitch system, and yaw system.

Using the definition given above, possible Critical Design Variables (CDV) for the three selected components (yaw system, pitch system and drive train) have been determined by the project team, based on experiences of design and occurred failures. The CDV's need to be determined in order to be able to define the critical design load cases that need to be analysed in detail for each component and to set minimum demands on the models used in the simulations. If the CDV's are not determined first, it is difficult to judge the suitability of the model that is used.

Three short comings in the current procedure to validate the design of wind turbine drive trains have been identified. These three short comings can result in significant differences between the results of the analysis of the wind turbine model and the real turbine.

- Misalignment

Misalignment of the drive train may cause constraining forces in the gearbox. To analyse these constraining forces it should be prescribed to what extend misalignment in the drive train should be considered. A complicating factor to analyse drive train misalignment is that in the traditional wind turbine simulation tools in general a simplified model is used for the drive train with only a limited number of degrees of freedom, so that misalignment cannot be analysed with these models.

- Resonance

The drive train consists of a number of dynamic structures, which may show interference. This interference may lead to unexpected dynamic behaviour and therefore a resonance analysis has to be performed with models and tools that can be used within different frequency ranges e.g. [0 – 5 Hz] [5 – 50 Hz], [50 – 200 Hz], [200-500 Hz], [500-2000Hz]. The current tools are not yet accurate enough to enable these analyses.

- LVRT

Fault or loss of the electrical network connection is included in DLC's 2.3 and 2.4, however in practise the tools are not yet good enough to completely analyse these DLC's

Strictly speaking the first two discussed here are not new DLC's; a maximum misalignment should be taken into account in the analysis and the real misalignment should not exceed the assumed maximum, it must be assumed that a turbine is constructed according to the prescribed accuracy. Resonance should automatically show up during the analysis. However, many of the currently used general wind turbine simulation tools do not take the misalignment into account and the resonance could occur for frequencies that are much higher than can be analysed in current tools or not show up due to the limitations of the models used.
Loss or faults of the electrical network are already described in the DLC's, however this process is at this moment too complex for the state of the art tools to enable detailed enough analysis. Also a lot of different possible cases could be defined for LVRT and the most critical cases are not easily determined, they can even depend on the country due to the different grid codes in place.

A general conclusion with respect to the characteristic design loads is that for almost all design loads that are expected to be of importance for the three discussed components (drive train, pitch system, and yaw system), the external conditions are covered by the DLC's already specified in IEC-61400-1 and the GL guidelines. This means that in principle all loads acting on the structural components can be determined. However, in practice it is more complicated due the fact that the traditional wind turbine simulation tools are not able to model the structural components in sufficient detail.

3. Six Steps Approach

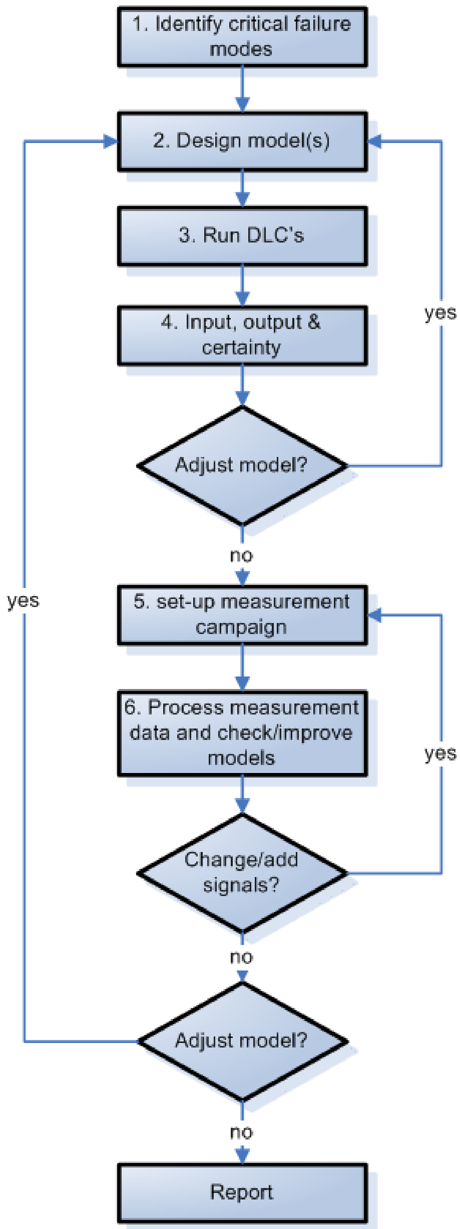

Figure 1: The six steps approach

The measurement campaign of the prototype has to be set-up such that the simulations can be verified. When focussing on the three discussed components, it is important that the loads on these components are validated. However, due to the large differences in these components between different wind turbine concepts as well as the differences in the corresponding models that need to be used, it appears to be impossible to set strict standards. For example it has no use to include measurements of variables that are not included in the model or do not exist in the chosen concept or to measure at frequencies that are much higher than those that would show up in the simulations. The model that is used determines the measurements that are 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 to set up a measurement campaign for a component are (see also Figure 1):

Step 1: Identify critical failure modes or phenomena for component

Step 2: Design the model (simple analytical to e.g. multi body)

Step 3: Run model for various DLC's (critical DLC's can be different for the different phenomena!)

Step 4: Determine input and output parameters of the model that has been used, determine how "certain" they are, and if they need to be verified/measured

Step 5: Design measurement campaign to verify models and quantify parameters

Step 6: Process measurement data and check or improve the models or the model parameters.

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.

4. Example of six step approach for pitch system

As an example, the pitch system of an N80 turbine has been analysed and measured using the above described approach.

4.1 Identify critical failure modes or phenomena for component

The pitch system can be subdivided into several subcomponents. A sketch of the pitch system in the N80 is given in Figure 2. For every subcomponent, an analysis can be made determining the failure mechanisms, failure modes or phenomena. The different subcomponents to be considered are:

1. pitch bearing
2. pitch gear
3. pitch gearbox
4. pitch brake
5. pitch motor
6. pitch controller / pitch electrons
7. pitch encoder

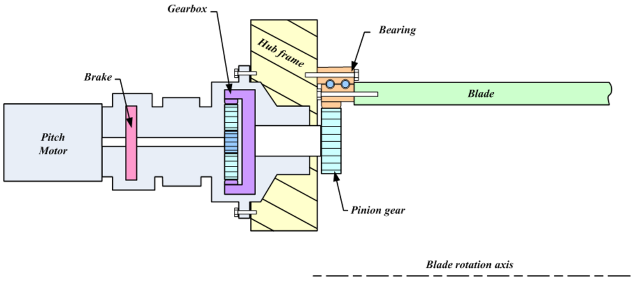

Figure 2: Sketch of the pitch system components

Considering the costs due to failure, the most relevant of the possible failure modes are those that are related to the bearing. These are determined to be friction and ovalisation. In this example of the six step approach only the friction will be analysed. It is apparent that other failure modes for other components of the pitch system should also be analysed when a prototype measurement campaign is being set-up, for example overheating of the pitch motor. Also the other mechanical components (yaw system and drive train) must be analysed. In this paper it is intended to only give an example of one possible failure mode as an illustration of the proposed six step approach.

4.2 Design the model

The primary function of the pitch system is to pitch the rotor blade. To rotate the blade, the pitch drive has to overcome the friction moment of the blade bearing, the resulting moment due to the aerodynamic forces and moments on the rotor blade and the inertia of the bearing and rotor blade. A blade bearing is designed based on the loads at the interfaces between the blade and the bearing and the interfaces between the bearing and the hub, as well as the required life time of the bearing. The life time is dependent on the friction of the bearing, which in itself is dependent on the loads at the interfaces. If the actual friction moment during operation does not correspond to the calculated friction moment, this may have consequences for the blade bearing life time and for the pitch drive (e.g. overload or fatigue).

As a starting point, a simple model will be used to determine the friction moment of the pitch slewing bearing. This model is taken from [10] and allows one to calculate the starting torque Mr of ball bearing slewing rings. The starting torque model is based on theoretical and empirical knowledge according to [10]:

$$M_r = \frac{\mu}{2}\left(4.4 \cdot M_k + F_a \cdot D_L + 2.2 \cdot F_r \cdot D_L \cdot 1.73\right) \quad (1)$$

with F_a the axial load on the bearing in *kN*, F_r the radial load in *kN*, M_k the resulting tilting moment in *kNm*, D_L the bearing race diameter in *m* and μ the friction coefficient. For a double-row ball bearing slewing ring [10] gives a friction coefficient of 0.004.

The friction model in equation 1 is valid for the starting torque, which means that once the bearing is rotating (e.g. pitching), the equation may not be valid. For those cases a function describing the friction in dynamic situations has to be used.

4.3 Run model for various DLC's

The pitch system is used either to actively pitch the blade (during power production or stops, both emergency as going to idling) or keep the blade at a constant angle (power production, idling). Each of these actions could result in critical cases, therefore all these types of cases should be included in the DLC's that are run for the above described model. To run these DLC's a simulation tool such as PHATAS [8] can be used. This will return the forces and moments that are needed as input for equation (1). As this paper is only intended to give an illustration of the proposed new six step approach only one DLC will be used.

A model of the turbine, including the pitch controller, is created for PHATAS and the loads for DLC 1.2 (*power production, normal turbulence model, wind speed between cut-in and cut-out, fatigue analysis*) are determined for a time interval of 10 minutes. The results for 50 seconds of this simulation are shown in Figure 3. The forces and moments are determined at the root of the blade so that they correspond to the loads at the blade-bearing interfaces. The simulation results show that the pitch controller was not active for this specific wind speed range, therefore the pitch angle was kept constant.

4.4 Determine I/O parameters of the model, determine how "certain" they are, and if they need to be verified/measured

The input parameters for the friction model are the modelling parameters in equation (1). The blade loads and moments are time series output parameters from the PHATAS post-processor from which the resulting radial load (F_r) and tilting moment (M_k) on the blade can be determined. The bearing race diameter (D_L) is known and the friction coefficient is supplied in [10]. Apart from the input parameters prescribed by the friction model, specific information about the wind turbine operational condition is important for a comparison to measurement data in Step 6, namely wind speed, yaw angle, azimuth angle and pitch angle.

The output of the friction model is the friction starting torque (M_r) corresponding to the time series input. A summary of measurements and uncertainties for the parameters discussed above

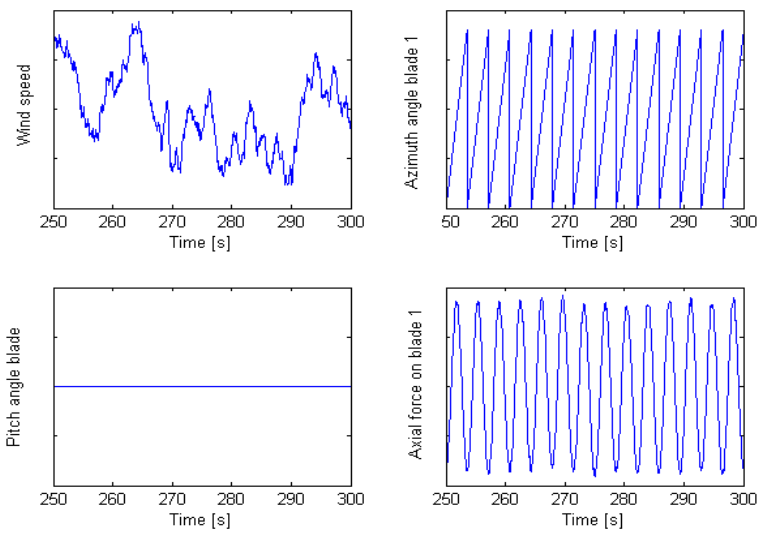

Figure 3: PHATAS output (wind speed, Azimuth angle, pitch angle and the axial force on the bearing) for DLC 1.2, for a 50 second interval of the simulated 10 minute time series.

Parameter	I/O	Measurement required	Uncertainty (%)
Axial load F_a	input	Yes, verify load	Unknown
Radial load F_r	input	Yes, verify load	Unknown
Resulting tilting moment M_k	input	Yes, verify load	Unknown
Bearing race diameter D_L	input	No	Est. < ±0.1 %
Friction coefficient μ	input	If possible	Unknown
Wind speed	input	Yes	Unknown
Yaw angle	Input	Yes	Unknown
Azimuth angle	input	Yes	Unknown
Pitch angle	input	Yes	Unknown
Friction torque, M_r	output	Yes, verify load output	± 25 % ([10])

Table 1: Input and output parameters or the friction model

is shown in Table 1.
The friction starting torque is calculated using equation (1) with the PHATAS output as its input. The starting torque corresponding to the PHATAS output for DLC 1.2 (Figure 3) is shown in Figure 4. It also shows the contributions of the tilting moment, axial force and radial force to the calculated starting torque. This illustrates that the tilting moment has by far the highest influence on the starting torque.

4.5 Design measurement campaign to verify models and quantify parameters

Within the PROTEST project, the main focus of the measurement campaign that has to be designed is to verify and improve models and parameters that have been used in the design process. In this example the friction model input and output parameters have to be verified using a measurement campaign. The measurements on the pitch system are conducted on a Nordex N80 wind turbine. The parameters to be measured for the friction model have been determined and were given in Table 1.
The friction coefficient cannot be measured directly. A value has been assumed and one of the goals in this approach is to verify this value. Direct measurement of the friction torque within the bearing itself is also not possible. However, the friction torque can be indirectly measured from the difference between the output torque of the pitch motor and the blade torsion moment. This means that measurements are required at the pitch drive and the blade root.
Due to practical limitations the blade strain measurements cannot be taken directly at the blade root, but only at some distance from the blade root. This means that a correction is needed using the mass moment of inertia of that part of the blade between strain gauges and

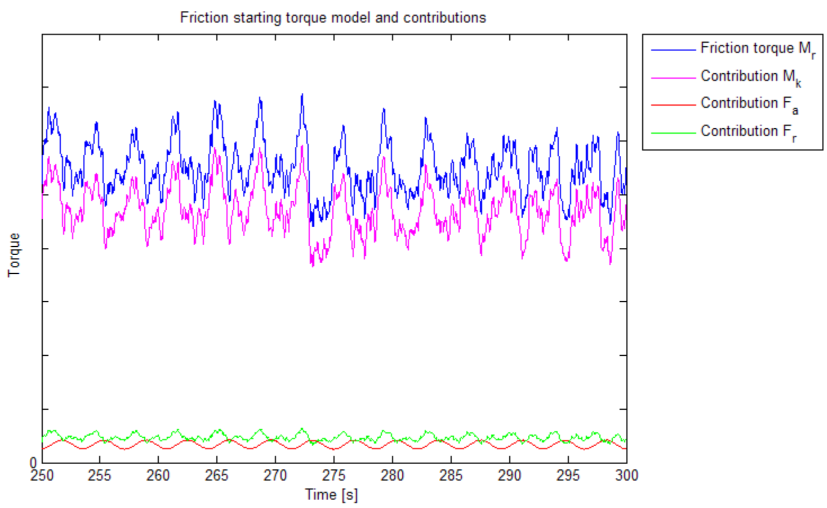

Figure 4: Friction starting torque and individual contributions of tilting moment, axial, and radial forces for 50 second interval

bearing, combined with the inertia of the rotating piece of the bearing. Furthermore, the pitch motor torque is measured before the pitch gearbox. This means that the resulting friction torque includes friction losses in the pitch gearbox and pinion gear.

A *measurement model* has been developed for a backward calculation of the friction torque from the blade torsion and pitch motor torque measurements. The following equilibrium equation for the pitch motor torque with respect to rotation of the blade is derived:

$$T_m \cdot i_{gbx} \cdot i_{gear} = T_b \pm T_f + \ddot{\theta}\left\{\left(i_{gbx} \cdot i_{gear}\right)^2 I_{gbx} + I_b + I_{br}\right\} \quad (2)$$

Where:

- T_m is the measured pitch motor torque. Notice that the pitch motor torque is increased by the gearbox and pinion gear ratios (i_{gbx} and i_{gear}).
- T_b is the measured torque acting on the blade
- T_f is the friction torque (its sign is dependent on direction of rotation)
- $\ddot{\theta}$ is the acceleration of the rotor blade about the blade axis.
- I_{gbx} is the inertia of the pitch gearbox
- I_b is the blade inertia from blade root up to the measurement location
- I_{br} is the pitch bearing inertia

Equation (2) now yields for the friction torque:

$$T_f = \pm\left(T_m i_{gbx} i_{gear} - T_b - \ddot{\theta}\left\{\left(i_{gbx} i_{gear}\right)^2 I_{gbx} + I_b + I_{br}\right\}\right) \quad (3)$$

In equation (3), the pitch motor torque and the blade torque shall be measured. The acceleration of the rotor blade can be calculated from the measured pitch angle.

The uncertainty in the inertia terms, gearbox and pinion gear ratios is assumed very low and thus no measurements will be performed for verification.

For the friction model, measurement data has to be provided as time series for different Measurement Load Cases (MLCs), e.g. start-up, emergency stop, idling, running to idling, running with pitching, and running without pitching. A complete overview of the necessary MLCs is given in Table 3. As described in this table, some of these cases need to be provided for a selection of wind speed bins.

Each time series should include a minimum of three full rotations while running (in the order of 10 or 20 seconds for the N80) and at least two full rotations while idling. For cases where special events occur (running to idling, start-up etc.) the time series should run at least two rotations while running (before or after the event) and the complete event. When time series are provided which suffice according to these requirements, enough information will be present to validate, verify or improve the model parameters.

Parameter in model	Measured signal types	Frequency	Method	Input for pseudo-signal(s)
1. Axial load Fa 2. Radial load Fr 3. Resulting tilting moment Mk	Blade 1, Root, flap moment	128 Hz	T-shape strain gauges	Leadwise & flapwise blade bending, In plane & out of plane blade bending, Rotor Thrust
	Blade 1, Root, edge moment	128 Hz	T-shape strain gauges	Leadwise & flapwise blade bending, In plane & out of plane blade bending, Rotor Thrust
	Blade 2, Root, flap moment	128 Hz	T-shape strain gauges	Leadwise & flapwise blade bending, In plane & out of plane blade bending, Rotor Thrust
	Blade 2, Root, edge moment	128 Hz	T-shape strain gauges	Leadwise & flapwise blade bending, In plane & out of plane blade bending, Rotor Thrust
	Blade 3, Root, flap moment	128 Hz	T-shape strain gauges	Leadwise & flapwise blade bending, In plane & out of plane blade bending, Rotor Thrust
	Blade 3, Root, edge moment	128 Hz	T-shape strain gauges	Leadwise & flapwise blade bending, In plane & out of plane blade bending, Rotor Thrust
	Pitch angle blade 1	32 Hz	PLC	In plane & out of plane blade bending
	Pitch angle blade 2	32 Hz	PLC	In plane & out of plane blade bending
	Pitch angle blade 3	32 Hz	PLC	In plane & out of plane blade bending
	Tower bottom bending N-S	128 Hz	T-shape strain gauges	For-aft tower bending
	Tower bottom bending E-W	128 Hz	T-shape strain gauges	For-aft tower bending
	Main shaft torsion	128 Hz	+45° – 45° strain gauges	Main shaft torque
4. Bearing race diameter DL	None			
5. Friction coefficient μ	None			
6. Wind speed	Wind speed	32 Hz	PLC	
7. Yaw angle nacelle	Yaw angle	32 Hz	PLC	For-aft tower bending
8. Azimuth angle rotor	Azimuth angle	128 Hz	Incremental encoder	Rotor thrust
9. Pitch angle blade	Pitch angle blade 1	128 Hz	Absolute encoder	
10. Friction torque, Mr	Blade torsion	128 Hz	+45° – 45° strain gauges	Blade torque
	Pitch motor torsion	128 Hz	Based on strain measurement	Pitch motor torque

Table 2: Required measurement signals for verification of the friction model

Description	Comments
Not pitching, $V_{hub} < V_{rated}$	Pitch system has to keep constant pitch (MLC 1.1)
Pitching, $V_{hub} > V_{rated}$	Pitch system has to adjust pitch angle (MLC 1.1)
Emergency stop	Large forces go through the bearing and pitch system has to pitch towards vane quickly, large dependency on controller. For different wind speeds. (MLC 2.3)
Start-up	Pitch from idling to small pitch angle, large dependency on controller. For different wind speeds. (MLC 2.1)
Running to idling	Pitch system will have to pitch to vane, large dependency on controller
Power production + fault	Any fault in the control or protection system which does not cause immediate shut down (MLC 1.2)
Stand still, blade vertically downwards, pitching	Measurement for friction model
Stand still, blade horizontally, pitching	Measurement for friction model

Table 3: Measurement load cases required for validating and tuning the friction model

To organise these measurements for different wind speed and turbulence intensity bins so called capture matrices are common practice. For each MLC the minimum number of measurements per bin and the bin sizes should be prescribed by specification of the corresponding capture matrices.

4.6 Process measurement data and check/improve models/ model parameters

A measurement campaign on the Nordex N80 turbine is initiated and ongoing. The theoretical analysis of the starting friction torque was made for DLC 1.2 (normal power production). MLC 1.1 corresponds to this DLC, see also Table 3. For illustration purposes, a *single* measurement for a matching wind speed bin (12 m/s) is selected. Some of the measured pseudo-signals are plotted in Figure 5. The wind speed is measured on the nacelle of the wind turbine, and is thus somewhat distorted by the wind turbine itself. Due to the wind resource stochastic nature, the PHATAS simulation will never result in an exact match to the measured wind speed. For a more exact wind speed bin and turbulence intensity determination, information from the on-site meteo mast could be used.

Compared to the PHATAS simulation results for DLC 1.2 in Figure 3, the number of rotor rotations during the 50 second interval is almost the same. The measurement shows that for each full rotation of the blade, the pitch motor is just delivering some torque to maintain the 0° position. This situation simulates the blade

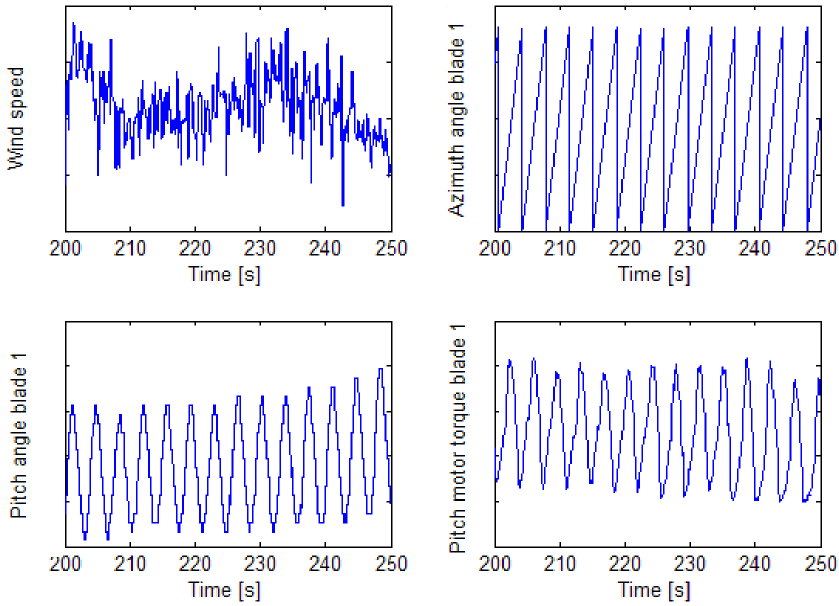

Figure 5: Measured signals MLC 1.1 for a 50 second interval of a recorded 10 minute time series: wind speed, azimuth angle, pitch angle, and pitch motor torque

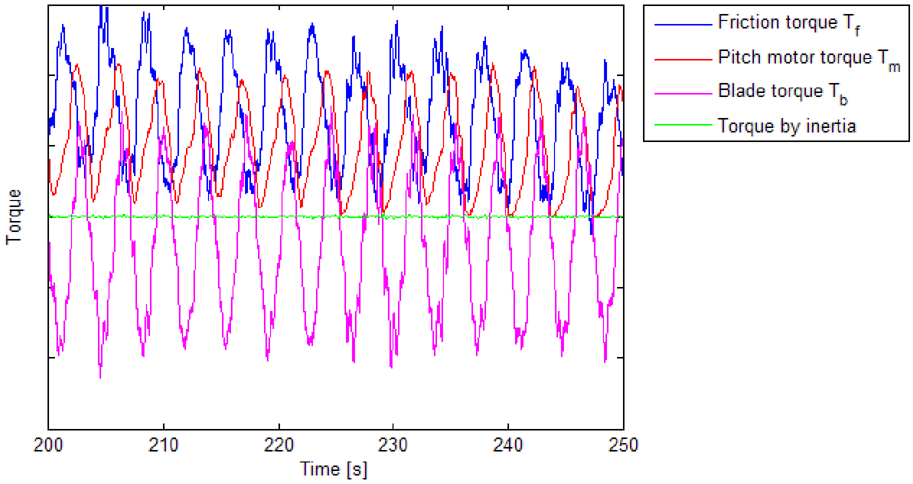

Figure 6: Friction torque calculated using measurements for a 50 second interval

overcoming the starting friction torque due to gravity (the blade has a slight pre-bend). It may be assumed that each time the pitch motor delivers some torque to maintain the 0° position of the blade, the friction torque should be at or near to the starting torque.
The friction torque is calculated by substitution of the measurement signals in equation (3). The resulting friction torque corresponding to MLC 1.1 in Figure 5 is plotted in time in Figure 6.
Next to the friction torque, the other terms in equation (3): the pitch motor torque, the blade torque and the torque by inertia are also shown in Figure 6. Since the rotor blade is not really pitching, the torque due to inertia is close to zero. The measured changes in the pitch angle shown in Figure 5 are in the order of 0.03 degrees.
To compare the starting friction torque calculated with PHATAS input in Figure 4 and the friction torque calculated according to the measurement model in Figure 6, the rotor azimuth angle shall be used as input to synchronise the time results.
The friction torques from the first full rotation within both intervals are compared side-by-side for a 10 second period in Figure 7.
The azimuth angle comparison in Figure 7 shows a small phase delay of the measured rotor azimuth angle compared to the PHATAS simulation after 10 seconds. The friction torque from the measurement model correlates quite well with the starting torque friction model, especially when the pitch motor is applying a little torque to maintain the blade position. This is illustrated by the three peaks of the pitch motor torque that correspond to azimuth angles of 90 degrees (blade horizontal and going down) in Figure 7. When the pitch motor is not supplying torque, the measured torque is lower than the starting friction torque which is logical (i.e. the blade does not require additional pitch torque to maintain its position due to the starting friction torque). This does lead to the preliminary conclusion that, in this case, the starting friction torque model overestimates the actual friction

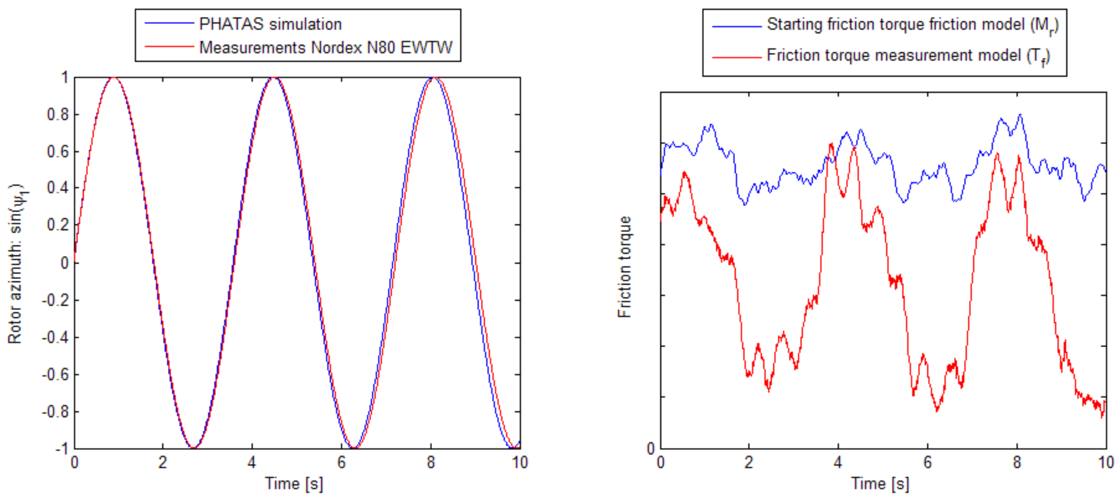

Figure 7 Synchronised friction torque output comparison for 10 second interval. Sine of azimuth angle (left) and starting friction torque model and friction torque measurement model (right).

torque. However, during the analysis it has become clear that unfortunately measuring the blade torsion is not straightforward. Due to the anisotropy of the blade as well as the large differences between the size of the deformation due to torsion compared to the much larger size of the deformation due to the bending moments, it is very difficult or impossible to calibrate these measurements, while it has a significant effect on the outcome of the model that has been used in this example. As shown in Figure 1, a possible outcome of step 6 is that it is necessary to go back to the step 2 and redesign the model. This action will be taken in the continuation of the PROTEST project, as the analysis of the friction coefficient following this model does not really allow for a quantitative judgement which inhibits the envisioned tuning of the input parameters. The improved model will compare the power from the pitch motor to a calculated power using the friction model and use this comparison to tune the model parameters. Next to this improvement, an improvement is also needed in the calculation of the friction moment from the starting friction to a dynamic function for this friction. In cases such as from running to idling, the dynamic effects cannot be neglected, therefore the equation for the starting torque (Eq. 1) is not applicable straightforward anymore, and may need to be adjusted by considering a non constant friction coefficient.

5. Conclusions

The PROTEST pre-normative project should result in complementary procedures to better specify and verify the local component loads acting on mechanical systems in wind turbines. This should enable improvements of the reliability of the mechanical components (pitch system, yaw system and drive train). First the possibility of necessary additional design load cases has been looked at. This has resulted in the suggestion of adding three new cases to the current standards: misalignment, resonance and Low Voltage Ride Through (LVRT). For all three cases the main problem is that the current state of the art codes do not enable the necessary detailed analysis, while it has become clear that these cases can result in relevant loads.

To enable an improvement in setting up the prototype measurement campaign, a new approach is suggested that contains six steps. The main focus of this approach is to enable validation and improvements of the model and its input parameters. As an illustration of this approach, these six steps are followed for the pitch system resulting in a quantitatively good comparison for the friction. However, due to the uncertainty in the blade torsion measurements, it was not possible to tune the parameters, which was one of the objectives. By improving the model further and therefore following one of the loops in the suggested approach, back to the second step, it is expected that it will become possible to do a quantitative comparison and tune the parameters of the friction model.

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