PROTEST

Recommended Practices for Measuring in Situ the 'Loads' on Drive Train, Pitch System and Yaw System.

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Acknowledgement/Preface

This report has been written as part of the project “PROTEST”, which in fact is a pre-normative project that should result in uniform procedures to better specify and verify the local component loads acting on mechanical systems in wind turbines.

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Abstract

The current standards and guidelines concerning prototype measurement campaigns are discussed. The experience in and requirements of load measurements is given concerning the three mechanical components: drive train, pitch system and yaw system. Based on this, a new method is proposed for setting up a prototype measurement campaign with the aim to validate and/or improve the models of the mechanical components that have been used during the design.

The method is flexible and consists of six steps that have to be taken. The flexibility is necessary as there are many different versions of each of these mechanical components and the models that are used also show large differences. Therefore the measurement campaign has to be specific, depending on the version of the component itself as well as the type of model used for the calculations. For example if high frequencies are not present in the model, it has no use to measure them, keeping in mind that the goal of the proposed measurement campaign set up is to validate the model. The six steps approach is illustrated by application of the method to these three components; drive train, pitch system and yaw system. The drive train has been modelled using the standard modelling in most aeroelastic tools as well as a more detailed multi body model. For the pitch system the friction is analysed as well as the ovalisation. As the analysis for the yaw system is very much similar to the pitch system, only the last steps (Step 4 to 6) are dealt with, concentrating on measurement aspects regarding the yaw system.

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1. Introduction

1.1 PROTEST project

High reliability of wind turbines and their components is one of the pre-requisites for an economic exploitation of wind farms. For offshore wind farms under harsh conditions, the demand for reliable turbines is even more relevant since the costs for repair and replacement are very high. Unfortunately, present day 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 load measurement procedures on the level of the components. Due to the rapid increase of wind turbines in size and power as a response to the market demands, suppliers of components are forced to (1) come up with new designs very often and (2) produce them in large numbers immediately. The time needed to check whether the components are not loaded beyond the load limits used in the design and to improve the design procedures is often not available or transparent to the component supplier. This leads to the unwanted situation that a large number of new turbines are equipped with components that have not really exceeded the prototype phase.

It was also concluded from a.o. [4] and expert discussions [5] that at present, the procedures 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. The design procedures for blades and towers are clearly documented in various standards and technical specifications. The reason for having extensive design standards for blades and towers is that these components are critical for safety: failures may lead to unsafe situations and designing safe turbines did have (and should have) the highest priority in the early days of wind energy. Parallel to the development of design standards, the wind energy community has developed advanced design tools and measurement procedures to determine the global turbine loads acting on the rotor and the tower. At present however, it is no longer acceptable to focus on safety only and neglect the economic losses. Lacking of clear procedures for designing mechanical components and specifying the loads on these components should no longer be the reason for early failures.

In 2007, ECN (NL) together with Suzlon Energy GmbH (DE), DEWI (DE), Germanischer Lloyd (DE), Hansen Transmissions International (BE), University of Stuttgart (USTUT; DE), and CRES (GR) decided to define the **PROTEST** project (**PRO**cedures for **TEST**ing and measuring wind energy systems) within the FP7 framework of the EU. The PROTEST project in fact is a pre-normative project that should result in uniform procedures to better specify and verify the local component loads acting on mechanical systems in wind turbines. The local component loads should be specified at the interfaces of the components. The relationship between global turbine loads acting on the rotor and tower and local component loads action on the interface of components is visualised in Figure 1-1. For gearboxes in common wind turbine architectures the special interfaces and load specification are explained in [7].

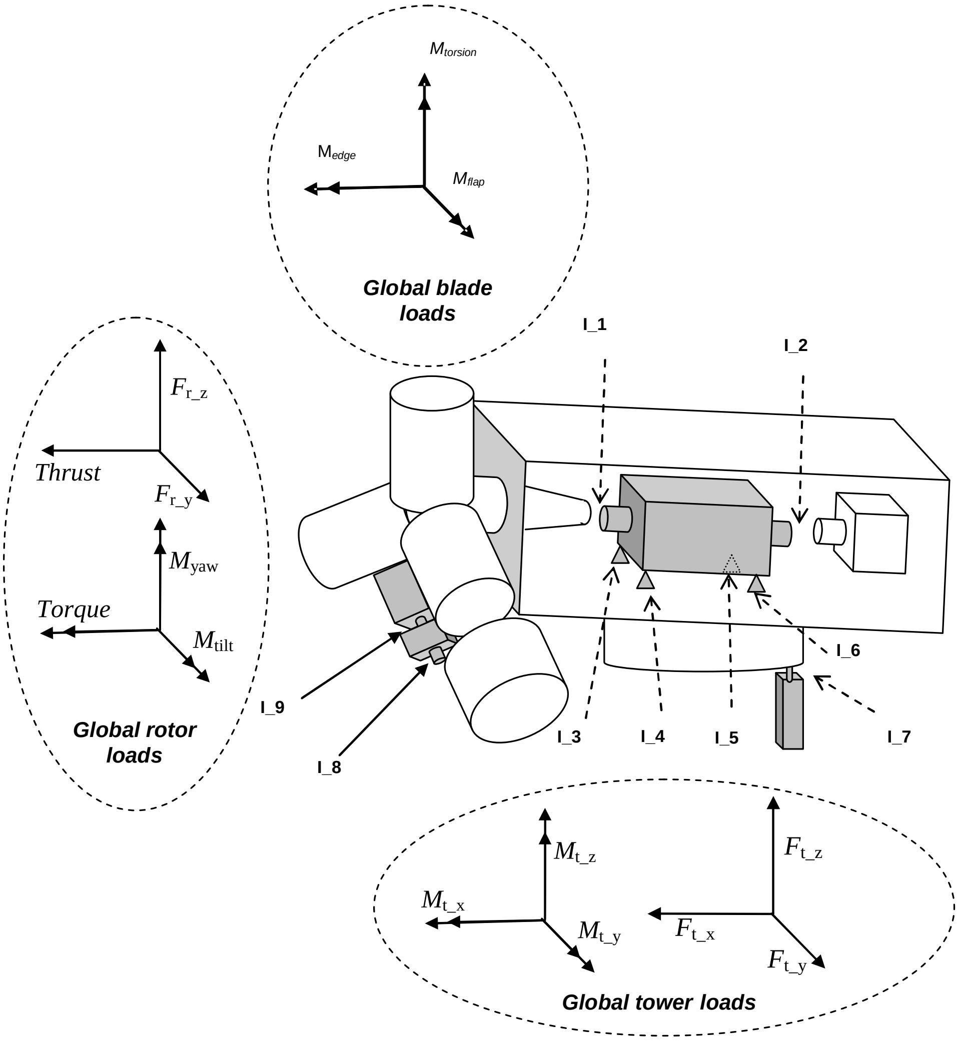


Figure 1-1: *Schematic presentation of transforming "global turbine loads" to "local components loads" at nine interfaces, (gearbox, pitch system and yaw system)*

The term "loads" should be considered broadly in this respect. It comprises not only forces and moments, but also all other phenomena that may lead to degradation of the components such as accelerations, displacements, frequency of occurrence, time at level, or temperatures. Within the PROTEST project initially the drive train, pitch system and yaw system have been selected for detailed investigation.

The uniform procedures to better specify and verify the local component loads should include:

(1) A method to unambiguously specify the interfaces and the loads at the interfaces where the component can be "isolated" from the entire wind turbine structure, and
(2) A recommended practice to assess the actual occurring loads by means of prototype measurements.

The following questions will be answered:

- 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?

To develop the procedures and to carry out the work within the PROTEST project, both analytical work and experimental work are foreseen. The analytical work is needed to determine the relevant load cases and to develop procedures to derive local component loads from global turbine loads during the design. The experimental work is needed to develop and verify new procedures for prototype measurements. In total nine work packages are foreseen.

1. State of the art report: An inventory will be made of the present day practice on turbine and component design and testing, including ongoing standardisation work and identification of areas for improvement.

2. Load cases and design drivers: For the selected components, it will be determined which load cases and design driving factors (external, operational or design inherent) should be considered

3. Loads at interfaces: For the selected components, it will be specified how the loads at the design points should be documented with the aim of being a meaningful improvement over the current state-of-the-art (reporting format, time series incl. synchronisation and minimum frequencies, statistics, spectra, time-at-level, etc.)

4. Prototype measurements definition: For each component, a recommended measurement campaign will be defined taking into account the following aspects: load cases, signals (torques, bending moments, forces, motions, accelerations, and decelerations), sensors, measurement frequencies, processing, uncertainties and inherent scatter, reporting.

Experimental verification is planned for the three components involved in the project. This work is defined in the Work Packages 5, 6, and 7.

5. Drive train: Suzlon S82 turbine in India with gearbox of Hansen Transmissions.

6. Pitch system: Nordex N80 turbine owned and operated by ECN at flat terrain.

7. Yaw system and complex terrain effects: NM 750 turbine in Greece in complex terrain.

In these three case studies, the initial procedures developed in task 1 through 4 will be applied. The initial design loads at the interfaces will be determined with state-of-the-art design methods and the measurement campaign will be executed to verify these design loads.

8. Evaluation and reporting: Based on the results of the design study and the measurement results, the procedures of task 2, 3, and 4 will be evaluated and if necessary improved.

9. Management, Dissemination and Exploitation

As mentioned previously, The PROTEST project in fact is a pre-normative project that should result in uniform procedures to better specify and verify the local component loads acting on mechanical systems in wind turbines. Ultimately, the procedures generated in this project should be brought at the same level as the state-of-the-art procedures for designing rotor blades and towers. If appropriate, the results of this project will be submitted to the (international) standardisation committees.

The project runs from March 2008 until mid 2010.

1.2 Work package 4: Definition of Prototype Measurement Campaign

1.2.1 Objective and background

The main objective of WP 4 is to set up a procedure (complementary to the procedures specified in IEC 61400-13 and chapter 8 of IEC 61400-4) for carrying out prototype measurements on the mechanical systems in such a way that characteristic design loadings (i.e. torques, bending moments, forces, accelerations, motions, rpm's, electrical power, etc.) at the interfaces can be derived from these measurements so that the results can be used (1) to verify the initial design loads, and (2) to tune and to validate simulation models used for the design of the mechanical systems. Especially the 2nd point was considered as very important by the project consortium at the kick off meeting of the project, and hence special attention should be given to measurements aimed at:

- the determination of specific turbine quantities, such as eigen frequencies to tune the models;
- the comparison of measured quantities with calculated quantities for validation purposes;
- to take into account guidelines on fault ride through (FRT) measurements for the measurement of torque at the high speed shaft.

Based on the results of WP 2 the following systems will be considered:

- Gearbox;
- Drive train apart from gearbox;
- Pitch system;
- Yaw system.

To set up a procedure complementary to IEC 61400-13 at least the following aspects should be considered:

- type of signal and corresponding sensor (i.e. torques, bending moments, forces, accelerations, motions, rpm's, electrical power, shaft misalignment, etc.);
- measurement frequency;
- load cases to be measured, including frequency and duration (and thus minimum duration of measurement campaign);
- data acquisition and processing, including generation of "calculated" or "pseudo" signals;
- dynamic analyses;
- uncertainty analyses;
- reporting format (digital time series, tables, plots, statistics, rain flow count with load spectra, time-at-level, etc.) and interpretation of results;
- assessment of measured loads in relation to the design loads.

To provide some background information a very brief outline of IEC 61400-13 and IEC 61400-4 is given in Chapter 2.

Although a large number of aspects have to be considered for the development of a procedure for prototype measurements, this activity is initiated by compiling an overview of a limited number aspects, viz.:

- the operational experience available with measurements on drive train, pitch system and yaw system;
- view on additional measurements that are required (both, *must-have* and *nice-to-have*).

For this purpose a questionnaire has been set. The questionnaire send out to the partners is part of Appendix A. The information provided by the partners is considered in Chapter 2. This overview will serve as a starting point for further specification by addressing all the other aspects mentioned above where also external parties (wind turbine manufacturers, suppliers of components, etc.) will be approached to provide further information. Furthermore the operational experiences obtained during the case studies (WP5, WP6 and WP7) will be incorporated.

1.2.2 Approach

To set up a procedure for carrying out prototype measurements on the mechanical systems (drive train, pitch system and yaw system) the following steps are carried out:

1. Initially the knowledge and experience within the PROTEST project team is collected. For this purpose a questionnaire has been set up, with the objective to make an overview of (1) the operational experience available with measurements on drive train, pitch system and yaw system (2) view on measurements that are required (both, *must-have* and *nice-to-have*)
2. The results of the questionnaire were processed and structured. Based on the results of the questionnaire a new approach has been developed to set up a prototype measurement campaign. This approach is first tried out for each of the mechanical systems treated in this project to investigate the effectiveness and possible shortcomings of this new approach. This proposal is presented to several specialists (wind turbine manufacturers, suppliers of components, etc.) not involved in the PROTEST project, and these specialists were asked to review this proposal.
3. With the feedback from the specialist the final proposal is drawn up.

1.3 Scope of the report

In this report a draft proposal is given for the set up of a prototype measuring campaign that can be used to validate the models that have been used for the mechanical systems. This draft proposal is based mainly on the knowledge and experience within the PROTEST project team, which has been collected on the one hand by means of a questionnaire set up for this work package and on the other hand by trying out the new method that was derived from this on the three systems.

In the current report the current standards and guidelines concerning prototype measurement campaigns are discussed in chapter 2. The questionnaire as submitted to the partners is part of Appendix A. The information provided by means of these questionnaires has been joined together in chapter 3. The new approach is described in chapter 4 and used for the analysis of a model of the drive train in chapter 5, for the pitch system in chapter 6 and for the yaw system in chapter 7. Finally the main conclusions are given in chapter 8.

2. Background information on standards and guidelines

In IEC 61400-13 and IEC 61400-4 standards for measurements are given. A short description of these two documents will be given in the next two sections.

2.1 IEC 61400-13

The following document has been considered:

IEC TS 61400-13, first edition 2001-06; Wind turbine generator systems –
part 13: measurements of mechanical loads.

The object of this technical specification is to describe the methodology and corresponding techniques for the experimental determination of the mechanical loading of key structural components in wind turbines. This technical specification is intended to act as a guide for carrying out measurements used for verification of codes and/or for direct determination of structural loading. So the goal of WP4 is in line with this objective therefore IEC 61400-13 seems suitable to serve as a guideline for the development of a procedure for carrying out prototype measurements for mechanical systems. W.r.t. the measurements itself 4 main topics are distinguished in IEC 61400-13, viz.:

- load measurement programmes
- measurement techniques
- processing of measured data
- reporting

These topics will be briefly outlined below, with emphasis on the application within the PROTEST project.

2.1.1 Load measuring programmes

The measurement program is meant to collect both a comprehensive statistical database and a set of time series, which represent sufficiently the behaviour of the turbine in certain specific situations. For this purpose (1) the quantities to be measured are specified and (2) a set of measurement load cases (MLC) is defined. These MLCs are defined such that they correspond with a selection of DLCs of IEC 61400-1 and from these MLCs the measurement campaign should be build up.

2.1.1.1 Quantities to be measured

In IEC 61400-13 the relevant physical quantities to be measured in order to characterise the loading of wind turbines are classified into: load quantities, meteorological parameters and operational parameters. To summarise the quantities to be measured in each of these classes table 8, 9 and 10 presented in IEC 61400-13 are given below.

Table 8 – Wind turbine fundamental load quantities

Load quantities	Specification	Comments
Blade root loads	Flap bending Lead-lag bending	Blade 1: mandatory Other blades: recommended
Rotor loads	Tilt moment Yaw moment Rotor torque	The tilt and yaw moment can be measured in the rotating frame of reference or on the fixed system (for example, on the tower)
Tower loads	Bottom bending in two directions	

Table 9 – Meteorological quantities

Quantity	Importance level	Comments
Wind speed	Mandatory	At hub height
Wind shear	Recommended	
Wind direction	Mandatory	At hub height
Air temperature	Mandatory	Influences material properties
Temperature gradient	Recommended	
Air density	Mandatory	Derived from air temperature and air pressure (which may be derived from the altitude taking into account ISO atmosphere)

Table 10 – Wind turbine operation quantities

Quantity	Importance level	Comments
Electrical power	Mandatory	
Rotor speed	Mandatory	
Pitch angle	Mandatory	Only for variable pitch turbines
Yaw position	Mandatory	
Rotor azimuth	Mandatory	If yaw and tilt moment are measured on the rotor shaft
Grid connection	Recommended	
Brake status	Recommended	
Wind turbine status	Useful	Relevant parameters may be derived from control panel of wind turbine

As WP4 of the PROTEST project is aimed at the definition of measurements (1) to validate simulation tools and (2) to verify design loads of the drive train, pitch system and yaw system it has to be determined which type of measurements should be carried out for this purpose in addition to the measurements summarized in table 8 of IEC 61400-13. Although table 9 and table 10 of IEC 61400-13 should be reviewed also, the emphasis will be on the specification of the load quantities, where loads should not be limited to forces and moments, but also comprehend quantities such as displacement, acceleration etc. Special attention should be given to the measurement of turbine specific quantities such as the natural frequencies which may be required to tune the simulation models.

2.1.1.2 MLCs

A standard load measurement campaign according IEC 61400-13 is built up from a number of prescribed measurement load cases. The MLCs prescribed in IEC 61400-13 define the main external conditions and the operational conditions of the turbine during which measurements should be made. The external conditions cover wind speed and turbulence intensity. The operational conditions are split up in steady state conditions and transient events.

The MLCs defined in IEC 61400-13 are given in table 1 and table 2 of this IEC specification, and are depicted below.

Table 1 – MLCs during steady-state operation related to the DLCs defined in IEC 61400-1

MLC number	Measurement load case MLC	DLC number (IEC 61400-1)	Wind condition at DLC	Remarks
1.1	Power production	1.2	$v_{in} < v_{hub} < v_{out}$*	In this mode of operation, the wind turbine is running and connected to the grid
1.2	Power production plus occurrence of fault	2.3	$v_{in} < v_{hub} < v_{out}$*	Any fault in the control or protection system, which does not cause an immediate shut-down of the turbine
1.3	Parked, idling	6.2	$v_{in} < v_{hub} < 0{,}75\ v_{e1}$*	When the wind turbine is parked, the rotor may either be stopped or idling
* Has to be divided further into wind speed bins and turbulence bins.				

Table 2 – Measurement of transient load cases related to the DLCs defined in IEC 61400-1

MLC	Measurement load case MLC	DLC	Target wind speed
2.1	Start-up	3.1	v_{in} and > v_r + 2 m/s
2.2.	Normal shut-down	4.1	v_{in}, v_r and > v_r + 2 m/s
2.3	Emergency shut-down	5.1	v_{in} and > v_r + 2 m/s
2.4	Grid failure	1.5	v_r and > v_r + 2 m/s
2.5	Overspeed activation of the protection system	5.1	> v_r + 2 m/s
Ideally the measurements should be taken at v_{out}. As this is impractical, the measurements are taken at wind speeds higher than v_r + 2 m/s.			

Measurements have to be made for several different wind speed bins and turbulence intensity bins. Due to the stochastic nature of the external conditions several measurements are required for a certain bin. To organise these measurements so called capture matrices are common practice. For each MLC the minimum number of measurements per bin and the bin sizes are prescribed by specification of the corresponding capture matrices.

For the PROTEST project it has to be determined whether these MLCs (depicted above) do cover the external and/or operational loads required for the drive train, pitch system and the yaw system and if needed additional MLCs should be defined. Herewith it should be kept in mind that with prototype measurements it is not possible to carry out all kind of measurements due to practical limitations (f.i. extreme wind speeds) or due to safety reasons (f.i. simulations of faulted situations).

At this stage it is sufficient to focus on the definition of the type of loads and the quantities to be measured. Based on the results it has to be discussed within the project team to what extend capture matrices especially for the drive train, pitch system and yaw system have to be prescribed.

2.1.2 Measurement techniques

In this clause of the IEC specification, the measurement techniques for the various types of quantities in load measurement programmes are described. These techniques include: instrumentation, calibration, and where relevant signal conditioning.

Furthermore, this clause gives recommendations with respect to the data-acquisition methods in load measurement programs.

At this stage of the PROTEST project it is sufficient to specify the instrumentation to be used to measure the loads on the drive train, pitch system or yaw system. In the stage described in chapter 5 of this report and further and especially in the reports concerning the PROTEST workpackages WP5, WP6 and WP7 details such as instrumentation, calibration and data acquisition will be worked out for the load measurements at the drive train, pitch system and yaw system.

2.1.3 Processing of measured data

For the measurements on the blades, rotor and tower it is outlined how the data should be validated and which type of analyses should be performed, f.i. determination of load spectra and equivalent loads.

Once it has been determined which loads should be measured for the drive train, pitch system and yaw system, similar specifications can be made for these measurements.

2.1.4 Reporting

A reporting format is given in IEC 61400-13. Once it has be determined which loads should be measured for the drive train, pitch system and yaw system, a reporting format can be compiled for these measurements also.

2.2 IEC 61400-4

In chapter 8 of IEC 61400-4 WD 3 [7] the minimum requirements for testing of new gearbox designs are defined. The following types of tests are considered:

- Workshop prototype testing (section 8.2);
- Field Test (section 8.3);
- Serial production testing (section 8.4);
- Robustness Test (section 8.5);
- Fleet Lubricant Temperature and Cleanliness (8.6).

For the PROTEST project the Field Test is of interest and especially the field tests aiming at the validation of the loads (section 8.3.1). The validation of loads is split up into:

- Validation of gearbox design loads
- Validating Wind Turbine Design Models
- Gearbox specific Field test requirements

Validation of gearbox design loads
In this section it is stated that as part of the certification process load measurements have to be carried out according to IEC WT01, Annex C. No further specifications of special measurements for the gearbox are given. IEC WT01 also refers to IEC/TS 61400-13 for the load measurement.

Validating Wind Turbine Design Models
Models to simulate the wind turbine response to prescribed design load cases exhibit uncertainty due to the fact that these models cannot normally be validated for all situations with field tests. In this clause of IEC 61400-4 some guidelines are given to reduce these uncertainties, viz.:

- In the WTG simulation codes, adjust turbine characteristics in order to accurately reproduce as –measured response using data from field tests
- Reproduce the simulations for design load cases not experienced in the field tests (as performed in original load determinations)
- Verify that loads used in the design are sufficiently conservative.

No further guidelines are given on the type of measurements that should be carried out.

Gearbox specific Field test requirements
Some design assumptions may have to be evaluated with specific testing, this could include torsional vibrations, combined structural response and reaction at the gearbox supports and

interfaces. These specific measurements shall be agreed upon between the gearbox manufacturer and the wind turbine manufacturer, but as a minimum it shall include:

- Time series during selected events
    - Run-up through all operating speed ranges
    - Cut-in at transition winds and high winds
    - Shutdown at low and high winds
    - Brake application
    - Emergency stops at high winds
    - Idling and backwind idling
- Measured Campbell diagram through the complete operating speed range to evaluate resonance risk

According to the document, the following signals should be measured at a sampling rate high enough to catch the mechanical vibrations with all relevant frequencies:

- High speed shaft torque
- Low speed shaft torque, if applicable
- Shaft speed

The sampling rate shall be selected in cooperation with the gearbox manufacturer. Typical sample rates will be in the range of 3 to 5 times the relevant vibration frequency.

3. Results questionnaires partners

In this questionnaire information is provided concerning:

- the operational experience available with measurements on drive train, pitch system and yaw system, section 3.1
- view on measurements that are required (both need to have and nice to have), section 3.2

3.1 Operational experience

In the tables below an overview is given of the experience available within the project consortium with load measurements on drive train, pitch system or yaw system in excess of the mandatory measurements according to IEC/TS 61400-13 at the time the questionnaire was filled in.
In these tables the following information is provided:

- Quantity: short description of type of measurement
- Specification: if possible, indication how measurement should be done, what kind of sensors should be used and at which locations sensors should be installed
- Objective: description of objective of measurement, e.g. frequency measurement to tune model or measurement for validation
- Comments: further information, e.g. whether measurement is needed or nice to have

Table 3-1: *Existing load measurements gearbox*

GEARBOX			
Quantity	**Specification**	**Objective**	**Comments**
Torque arm displacement	Optical or LIPS distance sensor to measure the displacement of the gearbox torque arm in axial and radial (horizontal; and vertical) direction	Research to learn about constraining load Improve modelling Certification	
Accelerations of housing	Acceleration sensor attached at different location of the housing of the gearbox to determine gearbox kinematic frequencies	Comparison to book values	
Gearbox dynamics	High frequency accelerometers to measure vibrations at gearbox bearing castings	Research	
	Accelerometers at mounting points to identify eigenfrequencies and modes	Validation of calculation tools	
Rpm at input, output and internal stages	Rpm sensor (incremental encoder or similar) to determine: (1) rpm characteristics vs power, and (2) angular displacement of the output shaft vs input shaft	Improve modelling Validate plant control	

Shaft loading	Strain gauges on pinion shaft to determine torque	Research	
Temperatures bearing components	Temp. sensor at various bearing components to determine/monitor: (1) temperatures inside the bearings (2) differential temperatures	Validate operating limits	
Oil temperatures	Temp. sensor to determine/monitor temperatures of oil in different locations of the lubrication system	Validate operating limits Dimensioning of cooling system	
Oil pollution / particles	Particle sensor to determine purity of oil	Validate operating limits	
Tooth base strain in specific gear wheels	Strain gages to determine loads inside the gearbox at specific locations	Validation of calculation results	

Table 3-2: *Existing load measurements drive train apart from gearbox*

DRIVE TRAIN APART FROM GEARBOX			
Quantity	**Specification**	**Objective**	**Comments**
LSS loading	Strain gauges to determine torque and bending loads	Validations of loads Tuning of models	Load duration distribution
LSS displacement	Displacement sensor to measure axial and radial displacement of LSS w.r.t. main bearing, 2nd bearing and gearbox. For radial displacement sensors at 0deg and 90deg.	Research to learn about constraining loads	
HSS loading	Strain gauges to determine torque	Validation of loads System development FRT tests [1])	Load duration distribution
HSS displacement	Displacement sensor to measure axial and radial displacement of HSS w.r.t gearbox/coupling. For radial displacement sensors at 0deg and 90deg.	Research to learn about constraining loads	
Rpm HSS & angular position	Optic sensor	System development FRT tests	
Generator dynamics	Accelerometers to identify eigenfrequencies and modes. Location of sensors must be suitable, to capture major modes identified in theoretical analyses.	Validation of calculation tools	

[1]) At CRES, the telemetry allows a 2kHz bandwidth for monitoring fast transient phenomena

Table 3-3: *Existing load measurements pitch system*

PITCH STYSTEM			
Quantity	**Specification**	**Objective**	**Comments**
Pitch drive train loading	Strain gage at shaft to determine torque		
Pitch motor loading	Current transformer and power transducer measure current and voltage to determine power consumption		
Rpm of pitch drive motors			Nice to have, difficult to get

Table 3-4: *Existing load measurements yaw system*

YAW SYSTEM			
Quantity	**Specification**	**Objective**	**Comments**
Yaw drive train loading	Strain gage at shaft to determine actual torque		
Yaw motor loading	Current transformer and power transducer to measure current and voltage to determine power consumption		
Rpm of yaw drive motors			Nice to have, difficult to get
Yaw activity		Load validation	
Yaw loads	Tower torsion measurement; see tower (top) loads	Load validation	
Tower (top) loads	Strain gauges in tower (top and/or bottom) to determine tower torsion and tower bending loads.	Load validation	
Nacelle bed dynamics	Low frequency accelerometers	Load validation	

3.2 Required measurements

According to the project partners the load measurements summarized in the tables below should be carried out for drive train, pitch system or yaw system. .
In these tables the following information is provided:

- Quantity: short description of type of measurement
- Specification: if possible, indication how measurement should be done, what kind of sensors should be used and at which locations sensors should be installed
- Objective: description of objective of measurement, f.i. frequency measurement to tune model or measurement for validation
- Comments: further information, f.i. whether measurement is needed or nice to have

In addition to the type of measurements, the quality of the measurements should be addressed also. Particular focus should be on time resolution of speed signals and on calibration stability of strain measurements. Data acquisition has to be capable of capturing high-frequency signals with high accuracy for low-noise signals.

Table 3-5: *Required load measurements for gearbox*

GEARBOX			
Quantity	**Specification**	**Objective**	**Comments**
Dynamics (vibrations/acceleration)	Accelerometers on gearbox housing	Evaluation of meshing frequencies	needed
	Accelerometers on support arms	In conjunction with dynamics of bed plate: (1) Validation of bed plate stiffness (2) Research to determine influence of elastomer	nice to have
		Model validation and tuning	
Kinematics (displacements)	Rotational speed intermediate shafts		nice to have
		Rotational speed measurement can be used to determine rotational accelerations	
	Axial displacement intermediate shafts	Validation	optional acc. to IEC 61400-4
	Displacement w.r.t nacelle (measured at torque arms)	Validation	needed
	Relative deflection of planet carrier side walls		
Shaft loads	Torque of intermediate shafts and sun shaft	Validation	
Temperatures			temperatures play an important role in the setting (play / pre-load) of bearings and can therefore be “needed” to validate loads and/or load simulation models
Grid frequency			nice to have

Table 3-6: *Required load measurements for drive train apart from gearbox*

DRIVE TRAIN APART FROM GEARBOX			
Quantity	**Specification**	**Objective**	**Comments**
Main bearing ovalisation	strain gauges to measure tangential strain at three annual positions on bushing: 0deg, 45deg, 90deg	identification of the loads, that are induced by main bearing	nice to have
Bearing loads	Radial and axial forces		
LSS loads	Strain gauges to determine torsion and bending moments of LSS	Load validation	According to IEC 61400-13
LSS kinematics	Decoder to measure azimuth position and	Azimuth angle is needed for transformation of load	Needed

(displacements)	rotational speed of LSS	vector	
		Rotational speed is used in conjunction with rotational speed of HSS for overall validation of gearbox model	Needed
		Rotational speed measurement can be used to determine rotational accelerations	
	Axial displacement w.r.t. gearbox	Validation of simulation model	
	Strain gauge to measure axial elongation	Validation of simulation model	
HSS loads	Strain gauges to determine torsion and bending moments of HSS	Validation and tuning of simulation model	nice to have
HSS kinematics (displacements)	Decoder to measure rotational speed of HSS	Rotational speed is used in conjunction with rotational speed of LSS for overall validation of gearbox model	
		Rotational speed measurement can be used to determine rotational accelerations	
	Decoder to measure azimuth position speed of HSS	In conjunction with azimuth position of LSS to determine stiffness properties	nice to have
	Radial displacement	Validation	
Loading of generator coupling	Radial and axial force acting on generator coupling		
Deflection of generator coupling	Sensor to measure deflection of generator coupling	Validation	
Dynamics of bed plate (vibrations/acceleration)	Accelerometers at several positions: (1) near support arms gearbox (2) main bearing (3) generator platform	In conjunction with dynamics of bed plate: (1) Validation of bed plate stiffness (2) Research to determine influence of elastomer (position (1))	nice to have
		Model tuning and validation	
Temperatures			Locations to be specified

Table 3-7: *Required load measurements for pitch system*

PITCH SYSTEM			
Quantity	**Specification**	**Objective**	**Comments**
Pinion rod loading			Depending on the pitch system, electric or hydraulic different measurements are

			needed on the pinion or rod.
Blade loads	Strain gauges at blade root to measure: (1) blade bending moments (2) blade torsion moment	. (1) load validation (2) research w.r.t. model development and validation	. (1) IEC 61400-13 (2) needed
Pitch angle	Pitch encoder at pinion gear to measure number of revolutions encoder disc Velometer on each blade piston	Transformation of load vectors	Needed
Pitch drive load	Strain gauges to measure torque	Load validation Development and validation of model for pitch system	
Pitch drive current	Current transformers	electrical power consumption: needed for load validation	needed
Pitch piston shaft loads	Axial force and bending moments on piston shaft	Validation	
pitch bearing ovalisation	optic sensors strain gauges to measure tangential strain at three annual positions on bushing: 0deg, 45deg, 90deg Strain gauged studs or bolts	validation of pitch bearing model Detect bending and tension	Needed
Pitch speed	Encoder	Validation	
Pitch acceleration	Encoder or differentiating the pitch speed	Validation	Needed for the model
Desired pitch angle	PLC	Validation of pitch response	
Pitch drive voltage	Voltage transformers		
Pitch drive power	Power transducer		

Table 3-8: *Required load measurements for yaw system*

YAW SYSTEM			
Quantity	**Specification**	**Objective**	**Comments**
Tower loads	Strain gauges near tower top to measure axial and shear (radial) loads, turning and bending moments	Load validation	
Yaw motor gear loads	Torque on driving gear shaft	Validation	
Vibration	Vibration on slewing ring	Validation	Design parameter
Shear	Cup anemometer in meteo mast at different heights Additional wind vanes	Validation	A higher shear might lead to a higher tilting moment of the rotor and a higher friction For complex terrain, identify directional shear
Yaw brake pressure	Pressure sensor of PLC	Validation	The yaw brake pressure is applied to keep the nacelle in position. During yawing, the brake pressure is released. Is the brake pressure sufficient to keep the nacelle in place? It should be noted that measuring the pressure may not answer this question, and measurement of vibration on slewing ring in combination with accurate measurement of the yaw position may be required.
Yaw misalignment	Spinner anemometer or a special probe in front of the hub to measure the wind speed / wind direction / wind components in front of the hub	Research	How accurate is the wind turbine aligned with the wind, how does the wind fluctuate in front of the turbine, and how does the turbine react on these fluctuations.

3.3 Specifications of MLCs

A straightforward approach to include the above mentioned required load measurements, may be that these measurements are carried out for the same internal and operational loadings (MLCs) as specified in IEC 61400-13. However it may occur that the loadings considered by the MLCs in IEC 61400-13 are not sufficient and additional internal or operational loadings should be considered for the drive train, pitch system or yaw system. F.i. for validation purposes of simulation models applied to analyse specific conditions.

Below the results provided by the project partners are summarised. First, it is relevant to distinguish between two different possible goals for a measurement campaign:

1. MLCs to verify the initial design loads (and complete the design load set where necessary)
2. MLCs to tune and validate the simulation models used for the design of the mechanical systems

For the PROTEST project, the second objective is the objective that the focus will be on. The first objective is typically performed and described in the traditional “load validation” approach such as clause 8.3.1 in IEC-61400-4 (p.76-78).

3.3.1 Summary

Table 3-9: *required internal and external loadings*

Name of (sub)system or component drive train			
Operational mode of turbine	**Additional requirements w.r.t. internal loading**	**Wind conditions**	**Remarks**
power production		v_in < v < v_out	
power production	emergency stop	tbd	
power production	yawed inflow	v_in < v < v_rated	
Idling	Sudden application and release of brake to create torque pulse excitation		
Low voltage ride through (LVRT)			Loading conditions at coupling / high speed shaft
			MLCs recommended in IEC 61400-13 are otherwise sufficient. The problem is the “recommended” which usually leads to a very shortened list of measured MLCs.
Power production	Faults according to grid requirements	Acc. IEC 61400-13 (faults)	These may be performed during certification tests
			Existing MLCs sufficient, turbine specific operational modes are normally tested regardless. Such measurement is normally requested by the certifying body. In particular transient operation is used for validating model assumptions (i.e. first drive train frequency).

Name of (sub)system or component yaw/pitch system			
Operational mode of turbine	**Additional requirements w.r.t. internal loading**	**Wind conditions**	**Remarks**
power production		v_in < v < v_rated	
power production		v_rated < v < v_out	
power production	emergency stop	tbd	
Power production	System engagement with reduced capacity	Acc. IEC 61400-13 (faults)	

Except from the description of internal loading cases (faults, operation cases where components undergo adverse loading), there is the need for more detailed description of the external loading, and this may be investigated. For instance:

- description of turbulence intensity, taking out the low frequency content
- procedure for identification of gusts/shear/direction changes and provision of specific experimental time series for model validation

This will enhance the exploitation of the experimental campaigns and the reliability of the validation procedure.

MLCs specified in IEC 61400-13 are considered sufficient for validating load assumptions of theoretical tools.

4. A Six Steps Approach

As discussed in the previous chapter, there can be two different objectives for the measurement campaign. In this project the focus is on a measurement campaign of the prototype that can be used to verify the model assumptions that have been used in the simulations of the different components. The measurement campaign therefore has to be set-up such that these 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 becomes 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. A procedure similar to IEC61400-13 would prescribe exactly the number of measurements, frequencies, etc. which may lead to an unnecessary amount of measurements without validation possibilities for the models used. It is not the intention of these new guidelines to replace the existing IEC61400-13; the new ones should be considered complementary.

To solve the problem of the model determining the measurements that are needed, a completely new and more flexible approach is suggested, a six steps 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:

Step 1: Identify critical failure modes or phenomena for component
Step 2: Design the model (simple analytical, multi body, FEA)
Step 3: Run model for various DLCs (critical DLCs can be different for the different phenomena!)
Step 4: Determine input and output parameters of model, determine how “certain” they are, and if they need to be verified/measured (spring constant, damping, axial motions, nat. frequencies, etc.)
Step 5: Design measurement campaign to verify models and quantify parameters (parameter, sensor, frequency, duration, processing, etc.)
Step 6: Process measurement data and check/improve models/ model parameters.

These 6 steps will not always be performed sequentially, as illustrated in Figure 4-1, it is possible to have one or more loops in the process. As illustrated in this figure, once the model is designed, the DLCs are run and the (un)certainty of different parameters has been investigated, it is possible that the model proves to be inadequate and needs to be altered, for example when it is realised that it will not be possible to determine enough parameters in the measurements or if it becomes clear that the uncertainty of certain input parameters is too large. It is also possible that, after measuring and processing the data, the measured signals appear to be incorrect or that more signals are needed, which results in the loop illustrated, going back to step 5, ‘setting up the measurement campaign’. Another possible outcome after the final step is the need to return to the design of the model, if the approach that has been followed turns out to be unsuccessful or if some parameters need to be improved, which calls for a small change in the model and rerunning the critical DLCs again. However, in that case it should be possible to skip repeating step 5. These are a few of the possibilities of going through the six steps approach and they illustrate that the order is not always sequential and as long as all steps are performed at least once, differences with the illustrated order are realistic possibilities.

To investigate this suggested six steps approach, it will be applied to the three different components in WP 5, 6 and 7. In this report the approach is described for each of the different components for a single case, purely as an illustration of the method. The complete results will be discussed in the before mentioned work packages.

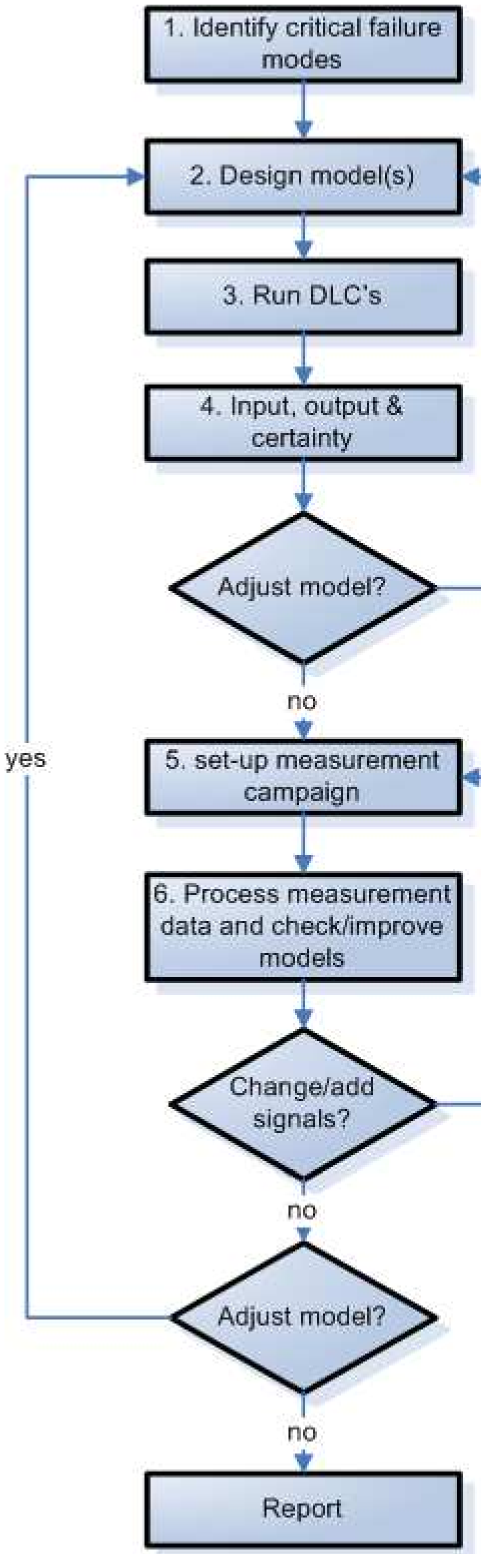


Figure 4-1: *Illustration of the six steps approach.*

5. Drive Train

In this chapter the six-step approach outlined in chapter 4 is applied to the drive train.

<u>Introduction</u>

Figure 4-1 shows the six-step approach as it is currently presented in WP4.

The application of this six-step approach for the drive train is carried out as follows:

- *<u>step 1:</u>*To identify relevant failure modes only the power transmission function of the drive train is considered which encompasses increasing the slow rotational speed of the rotor driven by the wind to the fast rotational speed required by the generator connected to the grid. Doing so, the high torque available at the rotor is reduced to the lower torque required at the generator to deliver the demanded grid load.
- *<u>step 2:</u>* As design models a state-of-the-art Flex5 model and a torsion SimPack model are considered. For the Flex5 model the drive train is simulated as an equivalent stiffness, damping and inertia to represent the mechanical connection from wind turbine rotor to generator rotor. In the Simpack model the individual components in the drive train are modeled in more detail using 1DOF rotational bodies.
- *<u>step 3:</u>* To run DLC's a selection is made according to the considered failure mode and occurrence of the DLC in the field.
- *<u>step 4:</u>* A sensitivity analysis can be applied to assess the impact of individual model parameter uncertainty on the eigenfrequencies, eigenmodes or outputs of the time simulations with respect to the considered failure mode calculation.
  Note that following this formalism for the drive train, there is no justification for adjusting the model at this stage as indicated in the 6-step approach of figure 4.1. The selection of the proper model is based on the considered failure mode in step 1.
- *<u>step 5:</u>* Assessment of the 2 model outputs with respect to the considered failure mode should indicate which interconnection loads need a dedicated experimental **load validation** campaign. The sensitivity analysis of the model parameters should indicate which parameters are to be experimentally identified following a dedicated experimental **model validation** campaign.
- *<u>step 6:</u>* The measurements should be processed to provide an answer to the following question:
  - Load validation: is the simulated load used for reliability calculation of the considered failure mode accurate enough to guarantee the reliability?
  - Model validation: is the accuracy of the model parameter high enough to guarantee that the sensitivity impact on the simulated loads for the considered failure mode is acceptable?

Note that the question to change or add signals after step 6 is justified if the questions above cannot be answered properly at this stage. A model parameter update is included in step 6. Further need to adjust the model after step 6 implies that the wrong model is chosen for the considered failure mode. In that case the process loops back to step 1 where calculations with the candidate models according to the considered failure mode reveal what model is required for this failure mode.

The six steps are discussed in more detail in the following sections.

5.1 **Step 1:** Identify critical failure modes within the drive train

The drive train is the assembly of components of the wind turbine that transforms the mechanical energy from the rotor into electrical energy. This definition includes the wind turbine's rotor, gearbox and generator, all interconnecting shafts and couplings as well as all bearings and supports. There are many possible configurations for the drive train depending on the manufacturer's general concepts and to a certain extend on the size of the turbine (for more details about the different drive train concepts and examples, see the state of the art report [19]). Among the four main types of concepts (modular drive, integrated drive, partially integrated drive train and direct drive train), the Triple-Point Suspension drive train (type modular) is the most common one. The drive train implemented into the Suzlon S82 wind turbine is a typical and hence relevant example for this class of drive trains.

As far as the technical realisation of the concepts is concerned, it only differs in the number, size and configuration of the components. To transmit power from the slow rotating turbine rotor to the fast rotating generator rotor a gearbox typically uses gears, splines, shafts, keys and key ways, bearings and structural components. Beside this, there are other components and subsystems which are relevant for the proper functioning of a gearbox but are considered out of the scope for the PROTEST project.

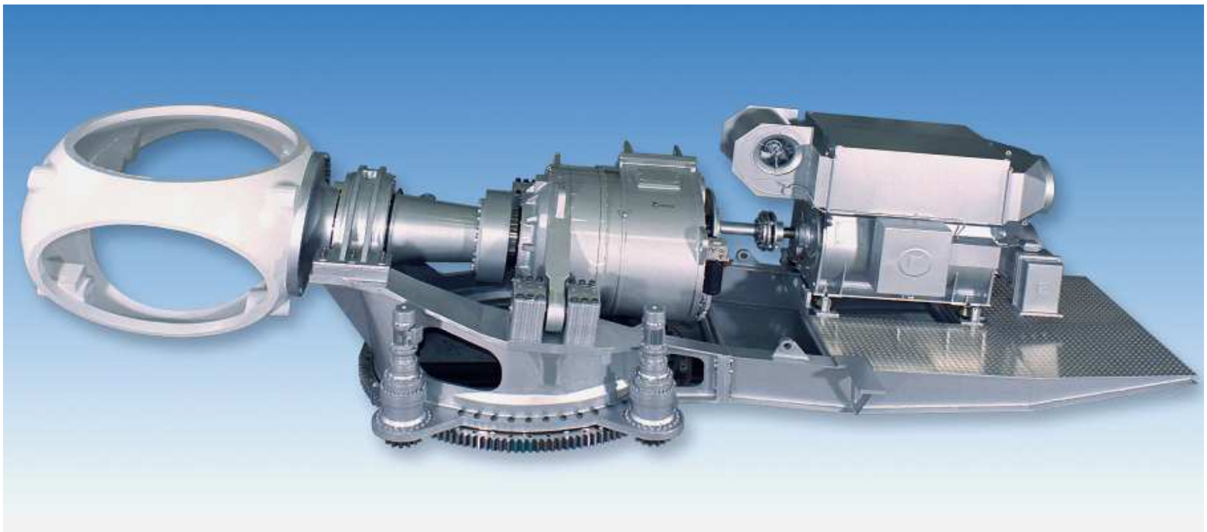


Figure 5-1: *Photograph of a partly integrated drive train concept from DeWind (2002), similar to the Suzlon S82: 3 point suspension*

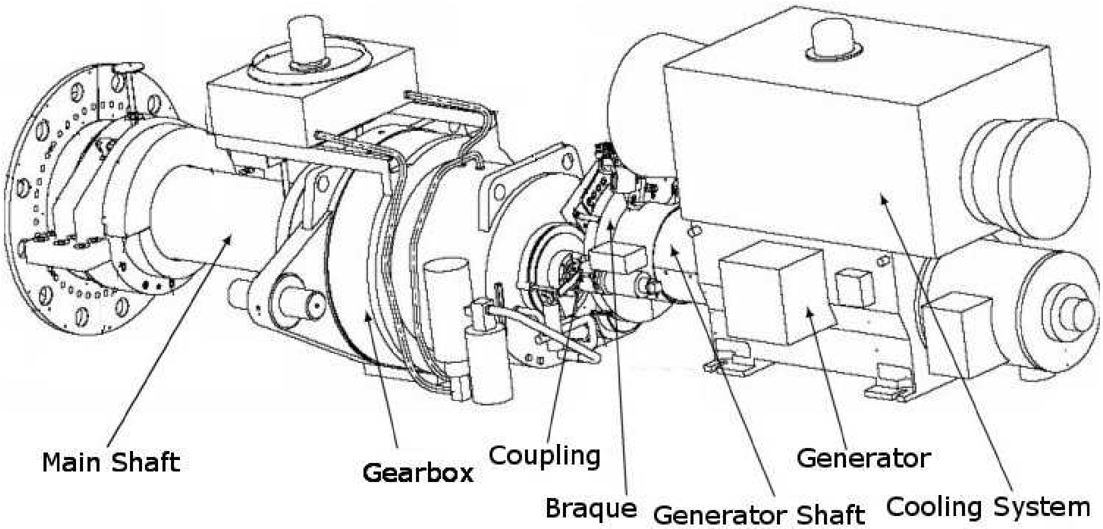


Figure 5-2: *Sketch of a typical modular drive train (triple point suspension) similar to the SUZLON S82 configuration [source: www.wind-energie.de]*

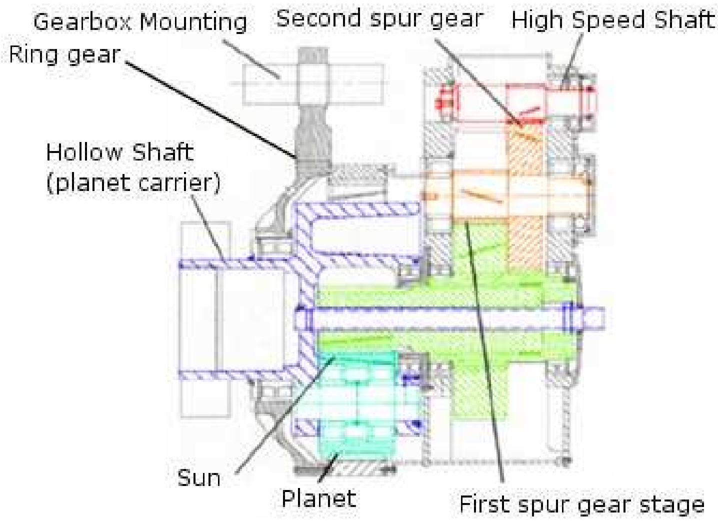


Figure 5-3:: *Cross section of a 3 stage gearbox (one planetary stage and 2 helical gear stages) for wind turbine [source: www.wind-energie.de]*

5.1.1 The gearbox (A)

For each of these components many non-uniformly standardized lists of failure modes exist in literature. Most of these failure modes can typically be assigned to one of the next categories according to their load related root cause:

- gears and splines:
  - pitting fatigue
  - bending fatigue
  - surface distress
  - overload fracture

- shafts:
  - bending fatigue

    - overload fracture

- keys and key ways
    - fatigue fracture
    - overload fracture

- bearings:
    - subsurface fatigue
    - surface distress
    - overload fracture

- structural components:
    - fatigue fracture
    - overload fracture

5.1.2 The drive train without the gearbox (B)

The drive train without the gearbox is broken down into its components and interconnection points. The following list specifies the relevant failure modes for the PROTEST project for each of its components. Furthermore examples for DLCs (defined by the GL guideline [19]) which can be used to check the component load assumptions are listed.

- connection to main shaft
    - bolt failure (fracture)

      load cases: extreme (for example GL-DLC1.5, DLC 1.6, DLC 9.1)
- low speed main shaft
    - fatigue fracture
    - damage due to extreme loads

      load cases: extreme (for example GL-DLC 1.5, 1.6, 9.1), fatigue
- main bearing(s)
    - bending fatigue main bearing housing
    - pitting fatigue

      load cases: fatigue
- connection to gearbox low speed shaft (shrink disc)
    - bending fatigue
    - slippage due to excessive torque

      load cases: extreme (for example GL-DLC 1.5, DLC-1.6, DLC-9.1), fatigue
- connection to gearbox high speed shaft
    - bending fatigue
    - slippage on shrink disc

      load cases: extreme (for example GL-DLC-1.5, DLC-1.6, DLC-9.1), fatigue
- mechanical brake
    - overload (thermal)

      load cases: extreme (for example DLC-5.1)
- high speed coupling
    - fatigue fracture
    - slippage on all frictional connections

      load cases: extreme (for example GL- DLC-1.5, DLC-1.6, DLC-9.1), fatigue
- generator
    - bending fatigue

      load cases: fatigue

For the gearbox (A.) and the drive train (B.) fatigue driven failure modes are supposedly avoided by checking the ratings using safe estimates of a full life time equivalent component

load assumption. Overload driven failure modes are considered by checking the extreme component load assumptions. Surface distress driven failure modes are taken care of by making sure that the proper lubrication and lubrication film thickness is present to avoid metal to metal contact and provide the required heat removal.
Based on these failure modes and assuming a design life of 20 years it can be concluded that the model should be such that it is able to

- estimate a 20 year equivalent load up to component level.
- estimate the extreme load values encountered within a 20 year life span up to component level.

For fatigue and overload driven failure modes, the required component load is mainly determined by rotation speed and torque, while for surface distress driven failure modes also lubrication flow rate, lubrication film thickness, temperature and displacements are to be considered.

The subsequent steps of the six steps approach are further demonstrated for the drive train based on an example: "Calculation of structural components". Taking into account only the power transmission function of the drive train (i.e. neglecting other loads than torque), the required model output load quantities are:

- o torque variations as a function of time and rotor position
- o extreme torque values

To go exemplary through the 6 steps approach, the failure of the high speed shaft coupling is chosen.

5.2 **Step 2**: Design models

To investigate properly the chosen failure mode, it seems to be appropriate to concentrate on the loads suspected to be responsible of the fracture: the torsional loads. As a consequence, we are focusing in improving the level of details of the drive train torsional modes.

The so-far commonly used approach for load-simulations of wind turbines design and the selection of components has been to calculate the loads by globally modeling the turbine with aeroelastic codes (e.g. Flex 5, GH-Bladed, see state of the art report, [19]). It has been extremely popular, due to its ability of capturing the interactions between inertial, elastic and aerodynamic forces and with relatively short computational times and high reliability. The typical model topology is presented in Figure 5-4 and Figure 5-5.

On the other hand, drive train component manufacturers sought to integrate several numerical models which capture the dynamic nature of the drive train.

However, due to the reciprocal characteristics of the dynamical interactions of the components and the external loads (e.g. wind, generator air gap moment, wave's loads or earthquakes), a fully-coupled model, which integrates all of the components with more details, is valuable to *complement* the existing methods with analysis, so that any significant interactions can be identified. It also avoids possible pitfalls in uniting uncoupled models.

This approach has been adopted in the case-study of PROTEST through integrating a detailed model of the drive train featuring one rotational DOF per component. The used tool is SIMPACK, a commercial software implementing the Multi Body Systems method (MBS) to model a mechanical system and determine the motions of the bodies in a 3D-space as well as the forces acting on them. The approach represents each component of the system as a rigid body and defines the interaction between them. An extended application of the method enables

the modeling of some components as flexible bodies and is called *Flexible* Multi Body System (FEMBS) simulation, which is also possible with SIMPACK.
The load flow in the wind turbine is schematically represented in Figure 5-5.
Two different modeling stages are being implemented under SIMPACK, see sections 5.2.1 and 5.2.2.

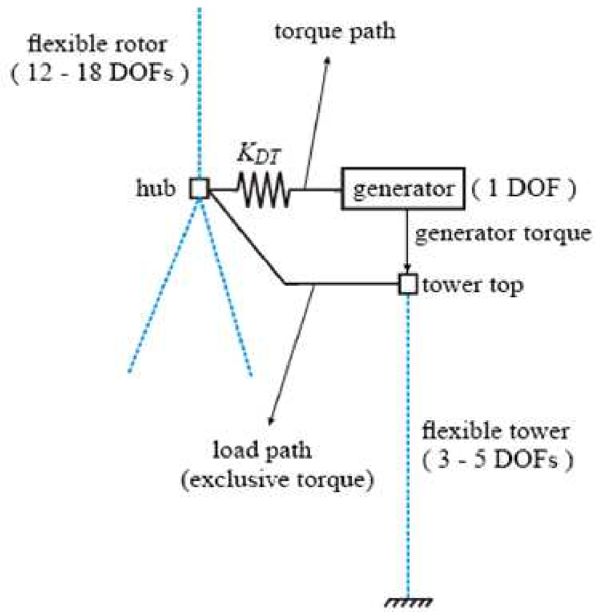


Figure 5-4: *Schematic representation of the DOFs in a structural model of a three-bladed wind turbine in a traditional design code (from [20])*

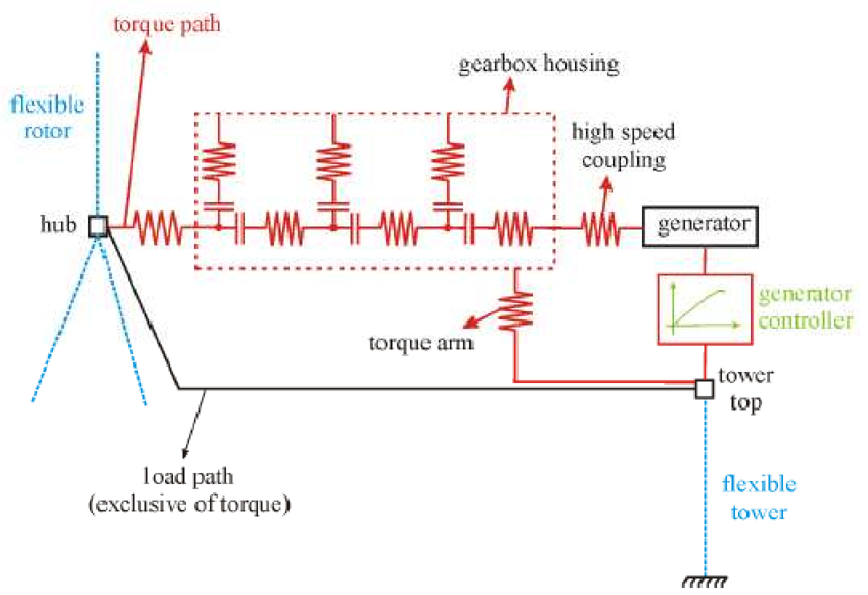


Figure 5-5: *Schematic overview of the load flow in the wind turbine (from [20])*

5.2.1 Modeling stage 1: based on standard aero-elastic tools

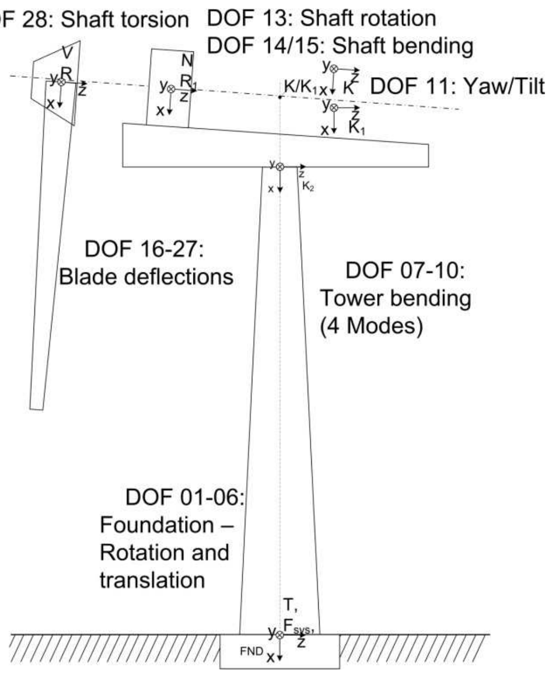


Figure 5-6: *Degrees of freedom of a Flex 5 model, with coordinate systems*

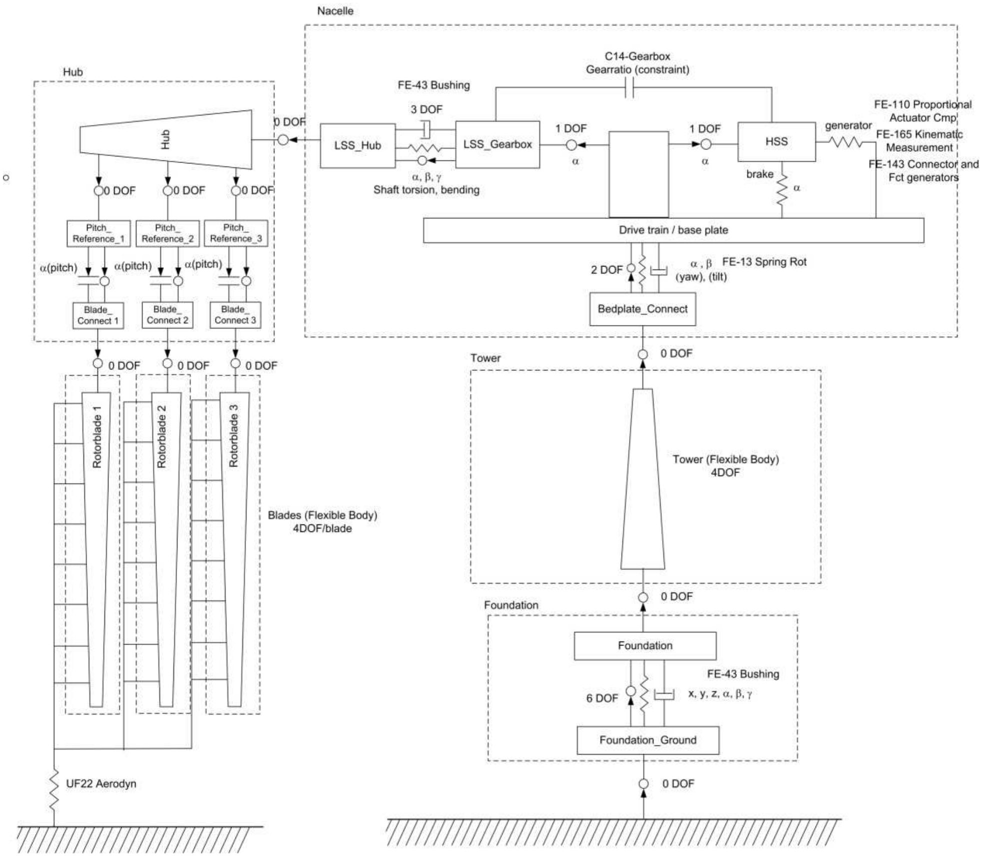


Figure 5-7: *Topology of the model similar to Flex-5's under SIMPACK*

The drive train of the first modelling stage (based on Flex 5, see Figure 5-6) contains 4 degrees of freedom: the rotation of the low speed shaft (“LSS”), two bending degrees of freedom of the supporting parts of the hub (bodies “Hub” and “LSS Hub”) relative to the tower top and the torsion of the connection between the hub and the generator rotor. The rotation of the high speed shaft is defined through a constraint to the low speed shaft (stiff connection with the gear ratio as transmission ratio). The input of the overall torsional stiffness and damping for the drive train and its transmission ratio are the only required parameters for the drive train. The inertias of all the rotating parts but the rotor blades and the hub are modelled in the “HSS” Body. The influence of the individual rotating bodies, which transmit the torque from the rotor to the generator, is thus not considered. To what extend these elements are influencing the loading of the drive train itself and all other turbine components can only be investigated when the detailed gearbox is integrated in the model of the whole turbine. This is implemented in the next modeling stage, the stage 2.

5.2.2 Modeling stage 2: sophisticated drive train model

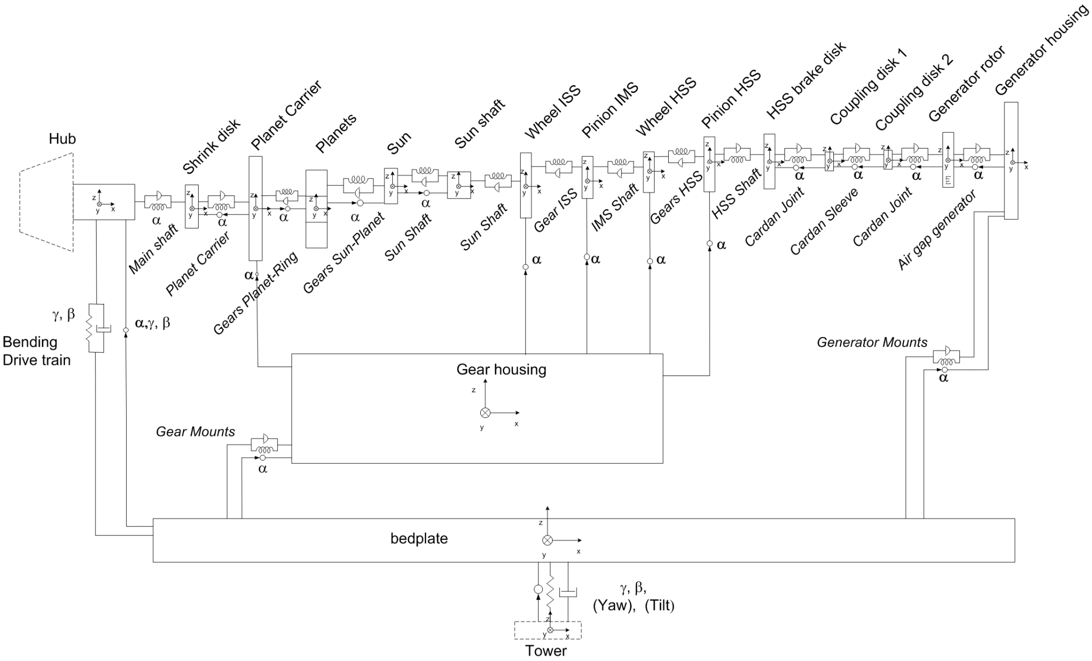


Figure 5-8: *Topology of the advanced torsional model of the drive train, modelled under SIMPACK*

It is possible to reach a very detailed modelling using multi body simulation software, representing for example the tooth contacts, bearing stiffnesses or even implementing flexible bodies. In practice it is, however, on one hand difficult for turbine manufacturers to have access to the needed data of the gearbox or of other components. Also the validation of the model parameters may be linked with a lot of work and effort. On the other hand the drive train manufacturer does not have access to the model data of the whole wind turbine, such as rotor blade data, controller model, tower data, etc. The possibility to extend the model for simulation of the load cases with the corresponding degrees of freedom is hence typically not considered. However, the data of the rotating bodies of the drive train are available.

Along with the masses and the inertias in particular the following information is of concern:

- Torsional stiffness of the gear box mounts.
- Torsional stiffness of the generator mounts.
- Torsional stiffness of the shafts and gear teeth
- Stiffness of the different gear stages
- Transmission ratios at the different gear stages.

By using these data it is possible to model the drive train in a more detailed manner than in the first stage. It thus leads to more precise simulation results for the whole system. In the drive train 14 supplementary degrees of freedom have been considered. The gear box housing and the generator housing are both connected with one rotational degree of freedom to the base plate. The whole drive train is being subdivided into 13 bodies between the rotor hub and the generator rotor. The torsional stiffnesses are being modelled using force elements at the interfaces between the elements and the rotational stiffnesses between the interfaces are being allocated to the corresponding bodies.

5.3 **Step 3**: Run model for various DLCs

Even though more investigations will be made in work package 5 (case study, drive train) where in particular more load-cases are simulated and results are extensively presented, it is explained how this step must be executed.

After completion of step 2, where the model has been designed, the first simulations (DLC's) must be run. It consists in calculating the relevant loads which are suspected to be at the origin of the investigated failure mode.

The corresponding loads are computed using the model under diverse operating conditions (design load cases), which have been determined in step 1 (Identify critical failure modes within the drive train).

The aim is to have enough data, to be able to check the validity of the model. In our case it consists in comparing the results of the simulation between Flex5 and Simpack. Flex 5 being a tool widely validated.

The Flex 5 outputs will be compared with the Simpack stage 1 outputs on the one hand, the Simpack stage 1 model and the Simpack stage 2 model on the other hand.
It is the way we did it in our case, however, the goal of the step is first and foremost to check the plausibility of the new model for the relevant load cases, so that it also could consist in comparing a baseline SIMPACK model, with an extended one.

Since the turbulent wind fields generated for the Flex 5 model are not compatible with the Simpack Aerodynamic module, the aerodynamic loading of the wind turbine models from a turbulent wind field cannot be identical for both models and time signals cannot, as a matter of fact, be compared directly.

However, a direct comparison between the time signals can be done at load cases using deterministic wind, (e.g. GL-DLC1.0, normal power production, Normal Wind Profile) since the corresponding wind fields are defined identically for both aerodynamic modules.

As a reminder, the selection of the DLC's to be run has to be made according to the investigated failure mode and the occurrence of the DLC in the field (see step 1 "identify critical failure modes"). That is why we should in our case run the following DLCs – see also table 5.1 for load case definition:

GL-DLC 1.0 for validation of the new Simpack Stage 1 model, using Flex 5 as a reference.
GL-DLC 1.5, since this load case is critical for the investigated failure mode

Design situation	DLC	Wind conditions[1]	Other conditions	Type of analysis	Partial safety factors
1. Power production	1.0	NWP $V_{in} \le V_{hub} \le V_{out}$		U	N
	1.1	NTM $V_{in} \le V_{hub} \le V_{out}$		U	N
	1.2	NTM $V_{in} < V_{hub} < V_{out}$		F	*
	1.3	ECD $V_{in} \le V_{hub} \le V_r$		U	N
	1.4	NWP $V_{in} \le V_{hub} \le V_{out}$	External electrical fault	U	N
	1.5	EOG_1 $V_{in} \le V_{hub} \le V_{out}$	Grid loss	U	N
	1.6	EOG_{50} $V_{in} \le V_{hub} \le V_{out}$		U	N
	1.7	EWS $V_{in} \le V_{hub} \le V_{out}$		U	N
	1.8	EDC_{50} $V_{in} \le V_{hub} \le V_{out}$		U	N
	1.9	ECG $V_{in} \le V_{hub} \le V_r$		U	N
	1.10	NWP $V_{in} \le V_{hub} \le V_{out}$	Ice formation	F / U	* / N
	1.11	NWP $V_{hub} = V_r$ or V_{out}	Temperature effects	U	N
	1.12	NWP $V_{hub} = V_r$ or V_{out}	Earthquakes	U	**
	1.13	NWP $V_{hub} = V_r$ or V_{out}	Grid loss	F	*
2. Power production plus occurrence of fault	2.1	NWP $V_{in} \le V_{hub} \le V_{out}$	Fault in the control system	U	N
	2.2	NWP $V_{in} \le V_{hub} \le V_{out}$	Fault in the safety system or preceding internal electrical fault	U	A
	2.3	NTM $V_{in} < V_{hub} < V_{out}$	Fault in the control system or safety system	F	*
3. Start-up	3.1	NWP $V_{in} < V_{hub} < V_{out}$		F	*
	3.2	EOG_1 $V_{in} \le V_{hub} \le V_{out}$		U	N
	3.3	EDC_1 $V_{in} \le V_{hub} \le V_{out}$		U	N
4. Normal shut-down	4.1	NWP $V_{in} < V_{hub} < V_{out}$		F	*
	4.2	EOG_1 $V_{in} \le V_{hub} \le V_{out}$		U	N
5. Emergency shut-down	5.1	NWP $V_{in} \le V_{hub} \le V_{out}$		U	N
6. Parked (standstill or idling)	6.0	NWP $V_{hub} < 0.8\ V_{ref}$	Possibly earthquake; see Section 4.4.3.3	U	N / **
	6.1	EWM Recurrence period 50 years		U	N
	6.2	EWM Recurrence period 50 years	Grid loss	U	A
	6.3	EWM Recurrence period 1 year	Extreme oblique inflow	U	N
6. Parked (standstill or idling)	6.4	NTM $V_{hub} < 0.7\ V_{ref}$		F	*
	6.5	EDC_{50} $V_{hub} = V_{ref}$	Ice formation	U	N
	6.6	NWP $V_{hub} = 0.8\ V_{ref}$	Temperature effects	U	N
7. Parked plus fault conditions	7.1	EWM Recurrence period 1 year		U	A
8. Transport, erection, maintenance and repair	8.1	EOG_1 $V_{hub} = V_T$	To be specified by the manufacturer	U	T
	8.2	EWM Recurrence period 1 year	Locked state	U	A
	8.3		Vortex-induced transverse vibrations	F	*

* Partial safety factor for fatigue strength (see Section 4.3.5.3)
** Partial safety factor for earthquakes (see Section 4.4.3.3)
1 If no cut-out wind speed V_{out} is defined, V_{ref} shall be used.

Table 5.1 Design load cases, defined by GL-Guideline (see [21])

5.4 **Step 4**: Assess results, determine input and output parameters, determine how “certain” they are, and if they need to be verified/measured

The input parameters for the drive train model are the model parameters of the Multi body System, as seen in step 2.
As far as the structural part is concerned, it represents the mass and inertias of the different bodies, the stiffness and damping values of the force elements connecting them.

	Component	**Uncertainty**
Mass	Blades	Max +-3% deviation
	Components Drive Train	Max +-3% deviation
	Tower	Max +-3% deviation
Inertias	Components Drive Train	Max +-3% deviation
Stiffnesses	Blades	?
	Tower	?
	Drive Train shafts	Max +-10% deviation
	Gear mesh Stiffnesses	Max +-20% deviation
	Gear Box mounting	Max +-20% deviation, particularly the non-linearity should be considered with care
Damping values	Blades	?
	Tower	?
	Drive Train shafts	Structural damping can be taken from literature, however, the consideration of friction losses as damping in all contacts is not straightforward, i.e. uncertainty of -+100%
	Gear mesh Stiffnesses	

Table 5.2 Uncertainty of the model input parameters

The table 5.2 shows that some structural characteristics of the dynamical model can be determined more precisely than others: e.g. the drive train components masses and inertias can be derived very precisely, since the geometry and the material densities are well known. It becomes more difficult when it concerns e.g. the tower or in particular the blades, which have relatively high fabrication tolerances, due to labour intense manufacturing.

Note that not absolute uncertainty but manufacturing deviations leading to differences e.g. in the blade masses are already taken into consideration in the simulation, by for example adding unbalanced in the blades (one blade with +3% mass and one with -3%).

Concerning the stiffness values, the situation is similar. For example, the gear mesh stiffness can be theoretically known exactly, based on the component geometry and the material properties but due to the involute tooth profiles and cyclic multi-tooth contacts, it witnesses further non-linearities (which are not taken into consideration in our model, all stiffnesses in the model being assumed to stay constant over displacements, velocities or accelerations).

The most difficult parameters to determine precisely are the damping values. Approximated values that have been determined empirically depend on the material properties (material damping) but also greatly on the component geometry (structural damping) or on the medium in which the mechanical parts are moving (viscous damping).

In other words, some structural data are difficult to be determined precisely. Moreover the different structural data act quantitatively different on the overall behaviour of the system. Note for example that in the approximation of overall stiffness of a drive train with one stage at ratio *n*, the stiffness of the high speed shaft has to be considered with a factor *n^2*, showing that identical uncertainty for different parts can have very different effects in the dynamics, only due to the kinematics.

In addition, the complexity of the equations of motion behind the multi body system makes it impossible to derive the uncertainty of the output result analytically from the uncertainties of the different inputs. That is why it is not practical to compute the uncertainty of the simulation results.

An alternative is to carry out a sensitivity analysis on the different input parameters.
It gives a rough approximation on the influence on input uncertainties on the simulation results. This can be realized with different approaches:

<u>1st approach:</u>

1. Vary a given parameter which ought to influence the relevant load (e.g. high speed shaft stiffness)
2. Run a modal analysis
3. Observe the change of the resulting eigenfrequencies and eigenmodes
4. Judge what uncertainty is acceptable

<u>2nd approach</u>:

1. Vary a given parameter which ought to influence the relevant load (e.g. high speed shaft stiffness)
2. Run load simulation for relevant DLC’s (determined in the previous step)
3. Analyse the results (e.g. capture matrix, or the outputs of the Rainflow Count)
4. Judge what uncertainty is acceptable.

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

The definition of the measurement campaign is the logical next step in our approach. It aims at verifying the model: its topology or input parameters.

IEC/TS 61400-13 [13] (Wind Turbine Generator Systems – Part 13: Measurement of mechanical loads) is commonly used for field testing of Wind Turbines, it defines what quantities should be measured, how to do it properly and how to analyse the data. This procedure, as well as the IEC61400-4 WD3 [7], are formally further to be followed. However, for our example, and the validation of our sophisticated drive train model, further signals are required.

At this stage of the 6-step approach, a good communication is necessary between the model experts and the person conducting field testing. Together, they must define the best combination between what signals are necessary, “nice to have” and what is technically feasible. During this process every party must be aware of the importance of each signal, to pay particular attention e.g. to their robustness, sampling rate, eventual data-processing (such as filtering), location of the sensor or to eventually drop some signals which are no must. Additional signals can have expensive consequences in terms of sensors, acquisition system, or data processing, that is why, they must be chosen with parsimony, defined after the early design phase (steps 1 to 4), so that no data which are useless for model validation or other design-relevant issues are produced.

In that sense, the prototype measurement campaign has been designed, as a part of PROTEST, to try to verify the sophisticated drive train model. The measurements have been realized on a SUZLON S82 1500kW wind turbine situated nearby Sankeneri, Tamil Nadu in India. The assessment and documentation of the load measurements have been performed according to the IEC 61400-13 (1. ed. 2001). Transients and normal power productions operation modes have been recorded. The relevant quantity are described in the table 5.3

At the beginning of the field testing, all load quantities are calibrated either employing analytic/nominal calibration or external load calibration. In most cases where geometry of the measured component at the measurement cross section is well known along with the material properties an analytic calibration scheme is applied for the strain gage sensors.
Specific load quantities have been calibrated by applying external loads on the complete measurement chain as it is common practice to use the blades dead weight for calibrations. In this scheme a minimum of two reference loads are applied on the corresponding component or in case of rotor blade calibration the rotor is idled at low wind speed and low rotational speeds several times to load the blade root strain gage sensors with dead weight load moment of the corresponding blade. Assuming a linear relationship between external loads and measured values, sensitivity and offset of the measured quantity can be evaluated. This scheme has been applied to calibrate the rotor blade root bending moments. Details on calibration are to be seen in the appendix A.2 of the instrumentation report, a deliverable of WP5.

The error estimation executed on the blade root moments and main shaft torque signals are also derived in part 6 of the S82 instrumentation report: "*To limit the effort involved in full scale load measurement campaigns to a practical degree, calibration checks cannot be carried out in a frequency that allows for statistical analysis of the calibrated sensitivities and offsets. Applying the provisions of the technical specification IEC TS 61400 - 13 (Section B.1.1 and B.2.2) the relevant data of the devices used in the measurement chain, estimates of uncertainties for material constants and estimates of uncertainties for component geometry and masses have been used to estimate the uncertainties for the measurement quantities. Standard uncertainties of type B ISO1993 for applied calibration values Ucal (i.e. combined uncertainty due to material constants, component geometry and masses, bridge scheme, used strain gages and completion resistors) as well as for the measurement output value Uov (i.e. combined uncertainty due to uncertainties of signal sensors, transducers and transmission lines in the measurement chain) have been derived*."

Unfortunately calibration and error estimate are in general a challenging processes. Strain gauge measurements are sometimes difficult to calibrate and it is indeed often impossible to load the gauged component properly in the field to calibrate the complete measurement chain. However, analytical calibration combined with a
shunt calibration check, (which verifies the functionality of the measurement Wheatstone bridge and the connected wiring by switching a shunt resistance in parallel to one of the strain gages in the Wheatstone bridge) is always practicable and even a prerequisite to every strain gauge test.

Most of the other sensors, however, (s.a. vibration sensors, torque measurement shaft, rotation velocities sensors, temperature sensors) will come with calibration documentation given by the sensor manufacturer or by a calibration institute. In some cases the sensor can be calibrated in the lab, before the field test.

Load quantities				
Quantity	**Name**	**Sensor**	**Sample Rate**	**Uncertainty**
Operational quantities				
Main Shaft Rotor Speed	**N_rot**	Inductive pick up	50hz	<2%
Generator Speed	**n_LSHSS**	Inductive pick up	200hz	<1%
Nacelle Wind Speed	**V_nac**	Controller Turbine	50Hz	1%
Electrical Power Output	**F_grid_r**	Active Power transducer	50Hz	0.5%
Pitch Angle of Blade 1	**Alpha_p**		50Hz	<1%
HSS torque (calculated)	**--**	Calculated	200Hz	<2%
Torque HSS Coupling (KTR)	**T gen**	DATAFLEX 140	200Hz	<5%
Torque HSS axis	**Taxis**	Manner system (strain into Volts)	200Hz	<5%
Power delivered (controller)	**P_WT_SCS**		50Hz	<1%
RPM (controller)	**n_rot**		50Hz	<1%
RPM HSS (KTR)	**N_Gen_p/D_ n_Gen**	Keyence laser	50 Hz	<1%
RPM HSS – pulses as a digital channel	**D_n_HSLP**	Encoder	50 Hz	n.a.
RPM ISS – pulses as a digital channel	**D_n_LSLP/D_n_IM**	Keyence laser sensor	50Hz	n.a.
RPM LSS – pulses as a digital channel	**D_n_LSLP**	Keyence laser sensor	50Hz	n.a.
RPM IMS	**n_LSHSS**		50Hz	<1%
Torque LSS	**Trot**	Sensor	50Hz	<5%
Displacement gearbox in supports vertical (left)	**dp_g_v_l**	LVDT inductive sensor	200Hz	<1%
Displacement gearbox in supports vertical (right)	**dp_g_v_r**	LVDT inductive sensor	200Hz	<1%
Bending Y LSS	**Mhy**	Sensor	200Hz	<5%
Rotor Azimuth	**Ar_rot**	Sensor (proximity switch 1ppr)	200Hz	<1%
Axial displacement HSS	**dp_HS**	Eddy current sensor	200Hz	<1%
Axial displacement ISS	**dp_IS**	Eddy current sensor	200Hz	<1%
Axial displacement LSS	**dp_LS**	Eddy current sensor	200Hz	<1%
Displacement gearbox in support axial (left)	**dp_g_a_l**	LVDT inductive sensor	200Hz	<1%
Displacement gearbox in support axial (right)	**dp_g_a_r**	LVDT inductive sensor	200Hz	<1%
Displacement gearbox in support tangential	**dp_g_t_r**	LVDT inductive sensor	200Hz	<1%
Axial play of planet carrier bearings	**Play_bea**	Eddy current sensor	200Hz	<1%

Table 5.3 Used measurement signals on the S82 for verification of the sophisticated model – campaign 12 2008

In addition, further signals, giving important information on the external conditions must as well be recorded, such as the yaw angle, generator speed, electrical power output, blade pitch angle, wind speed at nacelle, diverse status signals (such as “ Grid connection”, “Availability”, “Emergency stop”, “Manual Stop”, “Manual Stop”, “Fault”, “Yawing Error”). The resulting information is important to ensure that the loading environment in which the dynamic mechanical system is operated is similar for the model and the measurement (consistency of DLC –Design Load Case- and MLC –Measurement Load Case-).

5.6 **Step 6**: Process measurement results

The analysis and documentation of the measured signals should be executed following the IEC/TS 61400-13 standard. This includes 10-minute-statistics of the external condition descriptors (operational quantities and meteorology) and load quantities in terms of time series of 10-min-averages, scatter plots and time at level histograms. The way of representing measurement data is of course very much depending on the actual type of data and the type of analysis applied to come up with comprehensive documentation.

On the level of statistics and scatter plots a standard IEC load report format is recommended to describe the overall behavior of the tested wind turbine throughout the measurement campaign.

Since the loading of the wind turbine cannot be exactly defined in the model as it is on site (turbulent wind field), it is not possible to compare high resolution time signals with each other between the measurements and the model itself. As a consequence, the concept of the capture matrix (as given by IEC/TS 61400-13) should be used to categorize measured and simulated data w.r.t. external conditions of wind speed and turbulence. By using the corresponding data from the measured and simulated data base the consistency between the measurements (MLC) and the model results themselves (DLC) can be established and maintained throughout the various data post processing schemes.

Such post processing evaluations like *fatigue analysis* based on Rainflow counted load cycles or *time at level analysis* , however, do not give any precise hint about how to redesign the model, but only that it should be redesigned in case of important deviations between simulations and results.

Coming back to the validation targets of the prototype/component testing the aim is to employ the measurements in *model validation* and in *load validation*. Depending on the target of the validation approach specific evaluation techniques are applied.

In this demonstration the focus is placed on model validation and only the power transmission function of the drive train (i.e. neglecting other loads than torque) shall be considered. Hence, the relevant model output load quantities and measurement load quantities are:

- torque variations as a function of time and rotor position

The time series data obtained during manual testing and during an automatic monitoring campaign can be evaluated for

- extreme torque values (load validation)
- natural frequencies
- model parameters stiffness, damping and inertia

5.6.1 Extreme torque analysis

For extreme value analysis it is proposed to evaluate a min-max-matrix that relates the maximum and minimum loads observed for the individual load quantities to the simultaneous loads at all other load quantities. In the validation step the tables are set up for measured and simulated loads and compared.

MAX	$F_{x,LSS}$	…	$P_{m,LSS}$	$F_{x,HSS}$	…	$d_{z,TAR}$
$F_{x,LSS}$	$F_{x,LSS,max}$	…	$F_{x,LSS}$ @ $P_{m,LSS,max}$	$F_{x,LSS}$ @ $F_{x,HSS,max}$	…	$F_{x,LSS}$ @ $d_{z,TAR,max}$
…	…	…	…	…	…	…
$P_{m,LSS}$	$P_{m,LSS}$ @ $F_{x,LSS,max}$	…	$P_{m,LSS,max}$	$P_{m,LSS}$ @ $F_{x,HSS,max}$	…	$P_{m,LSS}$ @ $d_{z,TAR,max}$
$F_{x,HSS}$	$F_{x,HSS}$ @ $F_{x,LSS,max}$	…	$F_{x,HSS}$ @ $P_{m,LSS,max}$	$F_{x,HSS,max}$	…	$F_{x,HSS}$ @ $d_{z,TAR,max}$
…	…	…	…	…	…	…
$P_{m,HSS}$	$P_{m,HSS}$ @ $F_{x,LSS,max}$	…	$P_{m,HSS}$ @ $P_{m,LSS,max}$	$P_{m,HSS}$ @ $F_{x,HSS,max}$	…	$P_{m,HSS}$ @ $d_{z,TAR,max}$
$F_{x,TAL}$	$F_{x,TAL}$ @ $F_{x,LSS,max}$	…	$F_{x,TAL}$ @ $P_{m,LSS,max}$	$F_{x,TAL}$ @ $F_{x,HSS,max}$	…	$F_{x,TAL}$ @ $d_{z,TAR,max}$
…	…	…	…	…	…	…
$d_{z,TAL}$	$d_{z,TAL}$ @ $F_{x,LSS,max}$	…	$d_{z,TAL}$ @ $P_{m,LSS,max}$	$d_{z,TAL}$ @ $F_{x,HSS,max}$	…	$d_{z,TAL}$ @ $d_{z,TAR,max}$
$F_{x,TAR}$	$F_{x,TAR}$ @ $F_{x,LSS,max}$	…	$F_{x,TAR}$ @ $P_{m,LSS,max}$	$F_{x,TAR}$ @ $F_{x,HSS,max}$	…	$F_{x,TAR}$ @ $d_{z,TAR,max}$
…	…	…	…	…	…	…
$d_{z,TAR}$	$d_{z,TAR}$ @ $F_{x,LSS,max}$	…	$d_{z,TAR}$ @ $P_{m,LSS,max}$	$d_{z,TAR}$ @ $F_{x,HSS,max}$	…	$d_{z,TAR,max}$
Event	event @ $F_{x,LSS,max}$	…	event @ $P_{m,LSS,max}$	event @ $F_{x,HSS,max}$	…	event @ $d_{z,TAR,max}$

MIN	$F_{x,LSS}$	…	$P_{m,LSS}$	$F_{x,HSS}$	…	$d_{z,TAR}$
$F_{x,LSS}$	$F_{x,LSS,MIN}$	…	$F_{x,LSS}$ @ $P_{m,LSS,MIN}$	$F_{x,LSS}$ @ $F_{x,HSS,MIN}$	…	$F_{x,LSS}$ @ $d_{z,TAR,MIN}$
…	…	…	…	…	…	…
$P_{m,LSS}$	$P_{m,LSS}$ @ $F_{x,LSS,MIN}$	…	$P_{m,LSS,MIN}$	$P_{m,LSS}$ @ $F_{x,HSS,MIN}$	…	$P_{m,LSS}$ @ $d_{z,TAR,MIN}$
$F_{x,HSS}$	$F_{x,HSS}$ @ $F_{x,LSS,MIN}$	…	$F_{x,HSS}$ @ $P_{m,LSS,MIN}$	$F_{x,HSS,MIN}$	…	$F_{x,HSS}$ @ $d_{z,TAR,MIN}$
…	…	…	…	…	…	…
$P_{m,HSS}$	$P_{m,HSS}$ @ $F_{x,LSS,MIN}$	…	$P_{m,HSS}$ @ $P_{m,LSS,MIN}$	$P_{m,HSS}$ @ $F_{x,HSS,MIN}$	…	$P_{m,HSS}$ @ $d_{z,TAR,MIN}$
$F_{x,TAL}$	$F_{x,TAL}$ @ $F_{x,LSS,MIN}$	…	$F_{x,TAL}$ @ $P_{m,LSS,MIN}$	$F_{x,TAL}$ @ $F_{x,HSS,MIN}$	…	$F_{x,TAL}$ @ $d_{z,TAR,MIN}$
…	…	…	…	…	…	…
$d_{z,TAL}$	$d_{z,TAL}$ @ $F_{x,LSS,MIN}$	…	$d_{z,TAL}$ @ $P_{m,LSS,MIN}$	$d_{z,TAL}$ @ $F_{x,HSS,MIN}$	…	$d_{z,TAL}$ @ $d_{z,TAR,MIN}$
$F_{x,TAR}$	$F_{x,TAR}$ @ $F_{x,LSS,MIN}$	…	$F_{x,TAR}$ @ $P_{m,LSS,MIN}$	$F_{x,TAR}$ @ $F_{x,HSS,MIN}$	…	$F_{x,TAR}$ @ $d_{z,TAR,MIN}$
…	…	…	…	…	…	…
$d_{z,TAR}$	$d_{z,TAR}$ @ $F_{x,LSS,MIN}$	…	$d_{z,TAR}$ @ $P_{m,LSS,MIN}$	$d_{z,TAR}$ @ $F_{x,HSS,MIN}$	…	$d_{z,TAR,MIN}$
Event	event @ $F_{x,LSS,MIN}$	…	event @ $P_{m,LSS,MIN}$	event @ $F_{x,HSS,MIN}$	…	event @ $d_{z,TAR,MIN}$

Table 5.4: Example of Min-Max-Matrix for load validation

5.6.2 Natural frequency analysis

The MLC’s shall be investigated in the frequency domain (frequency spectra of the relevant MLC’s) to establish the relevant natural frequencies of the drive train system. This typically requires measurements of the relevant load and operational quantities at sufficiently high sampling rates. For suitable time series the amplitude spectra (FFT-analysis) and Campbell plots are evaluated for compliance with the model parameters and for analysis of resonance risks.

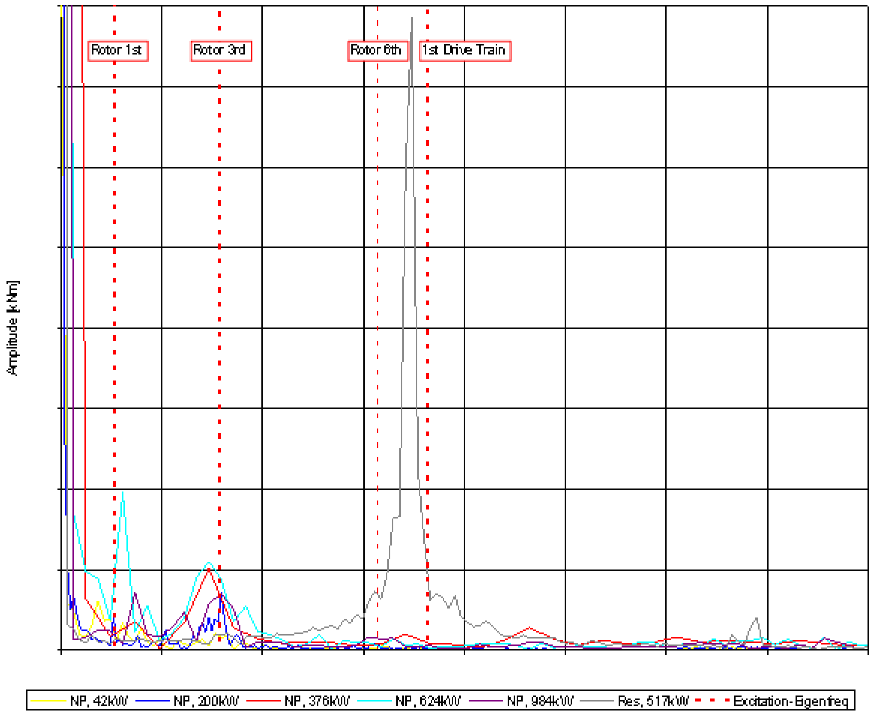


Figure 5-9: FFT analysis results for low frequencies for the normal production and resonance cases and the excitation eigenfrequencies.

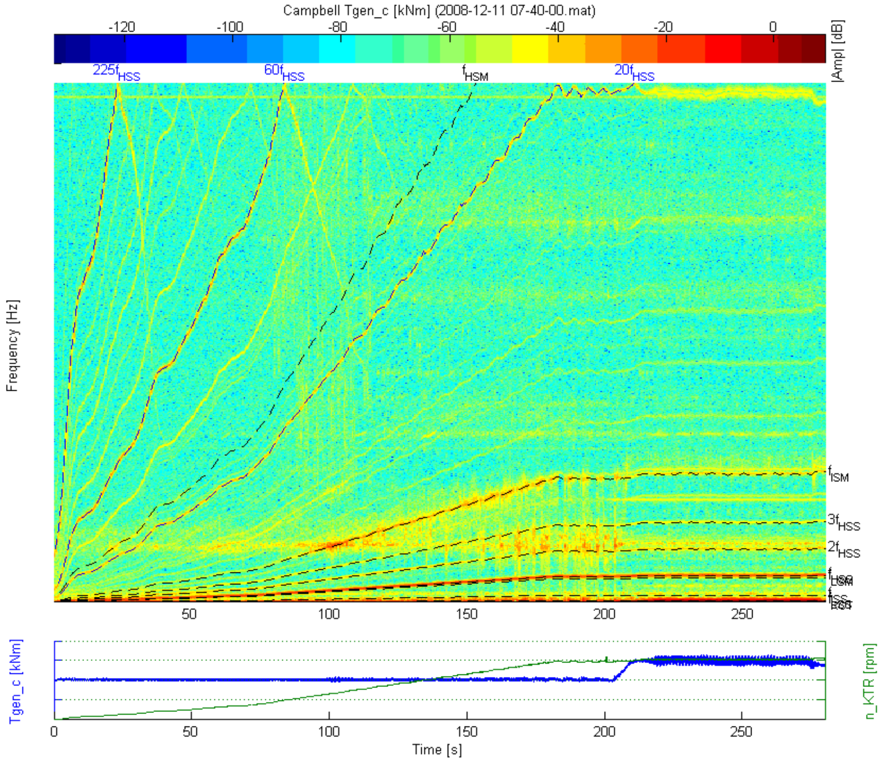


Figure 5-10: Example for Campbell diagram for the calibrated generator torque signal (Tgen_c)

If harmonics, that are suspected to have a high impact on the fatigue characteristics of the drive train, are present in the measurement and not in the model, it should be attempted to redefine the original model to try to reproduce them. e.g. by adding degrees of freedom, tuning parameters (stiffness or eventually damping values), or adding excitations. At this stage, it should be checked , if such harmonic content is only present during one particular Measurement Load Case or if its consideration is realistic for all the relevant DLCs.

If harmonics can be seen in the model but not in the measurement –or suspected to have a negligible contribution in the investigated failure mode, it can be assumed that the model is not realistic e.g. by underestimation of the damping in the model. In that case, the model can be simplified by reducing the degrees of freedom responsible for the corresponding frequencies or eventually damping values increased.

5.6.3 Model parameter analysis

Based on the specific measurement data the overall model parameter stiffness, damping and inertia are derived. In the project team two different approaches have been employed. Details on these approaches and their results can be taken from the report of WP5 Drive Train Case Study.
In the following only the essentials are outlined:

5.6.3.1 Deterministic Approach

Stiffness:

The stiffness of the drive train can be estimated from measurement data considering assumptions in drive train dynamics. The inertia of the rotor is high compared to the drive train and the stiffness of the main shaft is known to be high.

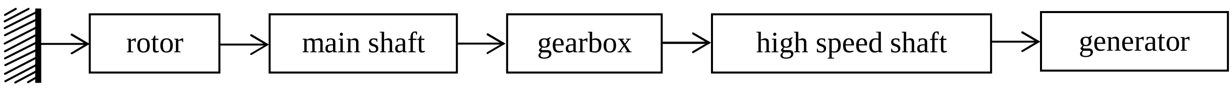


Figure 5-11: *Assumption of drive train model.*

Essential measurement results for a stiffness analysis are the rotary speeds and the angles of the low speed and the high speed shafts, respectively, and the torque of the low speed or high speed shaft, which are assumed to be completely transmitted by the gear box due to the gear box ratio.
Determination of stiffness involves relation of the drive train torque variation (averaged or steady state) to a correspond variation of the angular difference of the rotation angle of the high speed shaft and the low speed shaft.

$$stiffness = \frac{torque}{angle_{rel}} \quad [1]$$

Damping:

Two different methods are applied in this analysis in the project team:
To estimate damping values (logarithm decrement) measurement data is investigated for events where the drive train is excited to oscillate with its eigenfrequency. A damped oscillation is observed. These events are for example produced during emergency stops. The logarithm decrement is estimated by the natural logarithm of the quotient of two adjacent amplitudes. (eq. 2).

$$\Lambda = \ln\left(\frac{u(t_i)}{u(t_{i+1})}\right) \quad [2]$$

5.6.3.2 Statistic Approach

A time domain approach is used in the attempt to determine stiffness, damping and inertia. This method can be used depending on the signal availability, sampling rate, accuracy and reliability of the measured quantities.

T_{rot} φ_{rot} w_{rot}	T_{gen} φ_{gen} w_{gen}	φ_{gen} w_{gen}

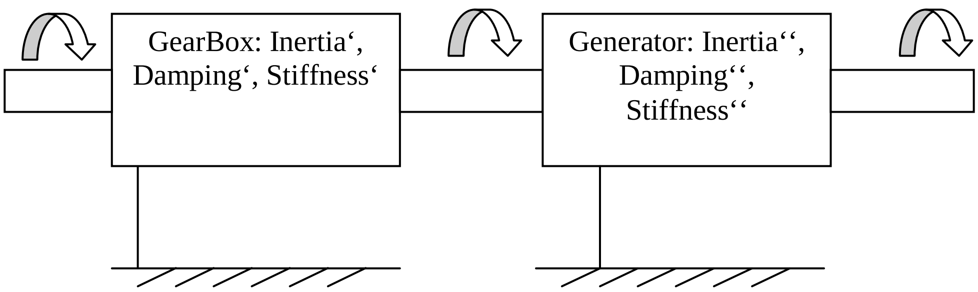


Figure 5-12: *Schematic conception of the drive train*

First of all the main equation of motion of the drive train is presented:

$$T_{rot}\big|_{high} = \left[\frac{\partial w_{rotor}\big|_{high}}{\partial t} - \frac{\partial w_{high}}{\partial t}\right] \cdot Inertia + \left[w_{rotor}\big|_{high} - w_{high}\right] \cdot Damping + \left[\varphi_{rotor}\big|_{high} - \varphi_{high}\right] \cdot Stiffness \quad [3]$$

Where:

$T_{rot}\big\|_{high}$	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
$\frac{\partial w_{rotor}\big\|_{high}}{\partial t}$	angular acceleration of the low speed shaft converted to the high speed side using the gear ratio
$\frac{\partial w_{high}}{\partial t}$	angular acceleration of the high speed shaft
$Inertia$	overall inertia of the drive train
$w_{rotor}\big\|_{high}$	speed of the low speed shaft converted to the high speed side using the gear ratio

w_{high}	speed of the high speed shaft
$Damping$	damping of the drive train
$\varphi_{rotor}\big\|_{high}$	angle of the low speed shaft converted the high speed side using the gear ratio
φ_{high}	angle of the high speed shaft
$Stiffness$	overall stiffness of the system

The approach assumes that any measured data will contain the information about the inertia, damping and stiffness. Depending on the individual method (there are three) a multitude of solutions will be derived for each (ten-minute-) time history that is processed. Hence, the centered value of the frequency distribution of all solutions or the median value of all solutions derived from one 10-min-data set will be considered as the most probable solution.

Stiffness:

In stationary operation that also excites the 1st drive train resonance frequency it is assumed that the effects of the inertia and damping are small and can be neglected. This means that the quasi-steady state variation of the torque will be just driven by the stiffness and the differential angle of both shaft ends. This simplifies the equation to this minimal expression:

$$T_{rotor}\big|_{range} = \left[\varphi_{rotor}\big|_{high} - \varphi_{high}\right]_{range} \cdot Stiffness \qquad [4]$$

Stable solutions for stiffness can best be found given that the first drive train natural frequency is excited. For this reason the “resonance” data have been used as obtained in a dedicated trial carried out in the test campaign in December 2008. Generally, the described methods can be used on any data set that contains drive train oscillations anticipating that the first natural frequency has a dominant contribution.

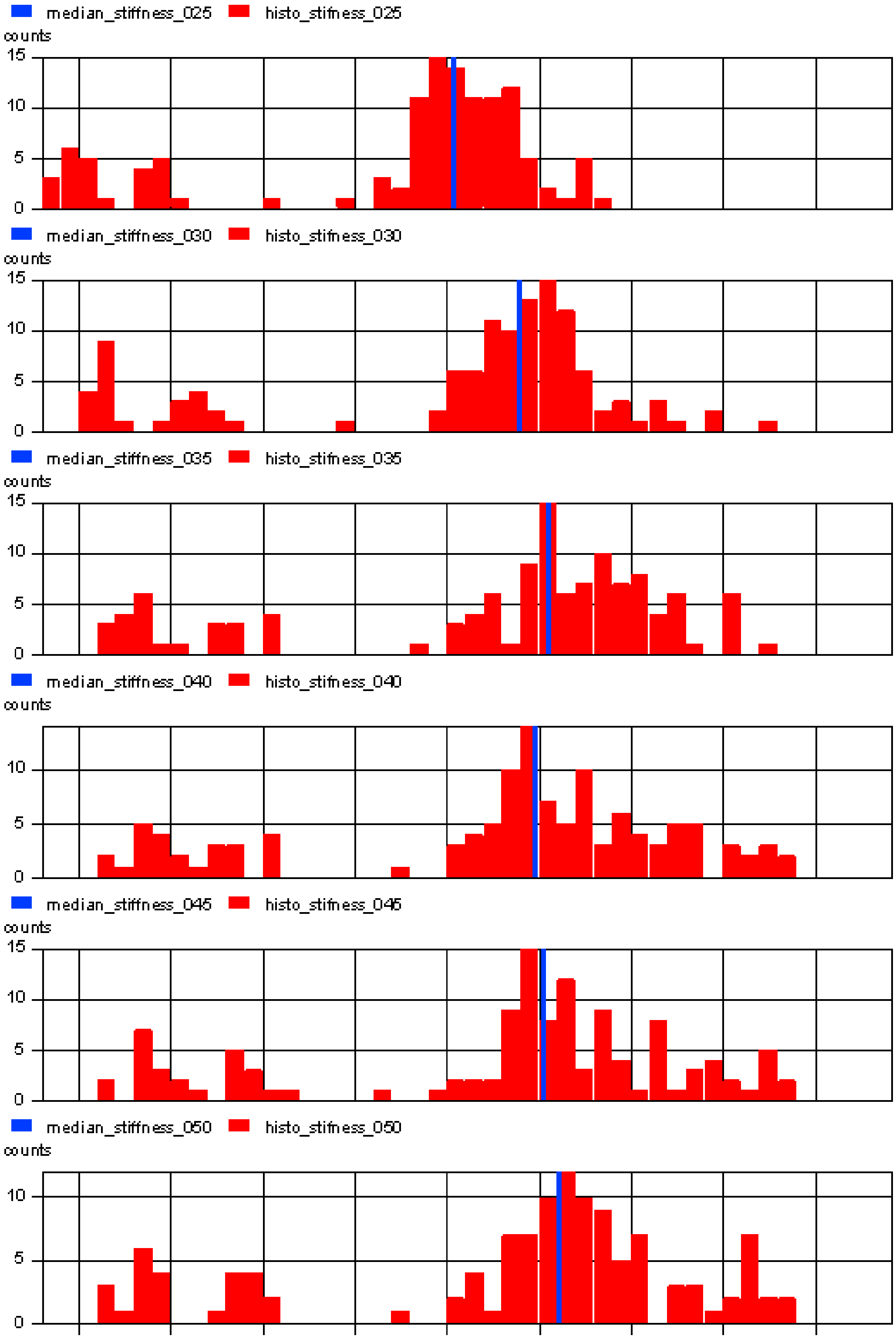


Figure 5-13: *stiffness solution median value and frequency distributions when modifying the cut-off frequency of the high-pass filter between 0.25 and 0.50 Hz*

Damping:

Once having overall drive train stiffness determined, the ratio between the damping and inertia can be obtained. Keeping in mind the findings of the analysis for stiffness the method is applied

- to “resonance” data - this means that the same time series used for analyzing the stiffness must be used,
- applying the same signal treatment (high-pass and low-pass filters) .

Processing the same file containing the drive train resonance, the statistical distribution of the ratio returns a damping ratio of 0.018544 as shown in the following figure:

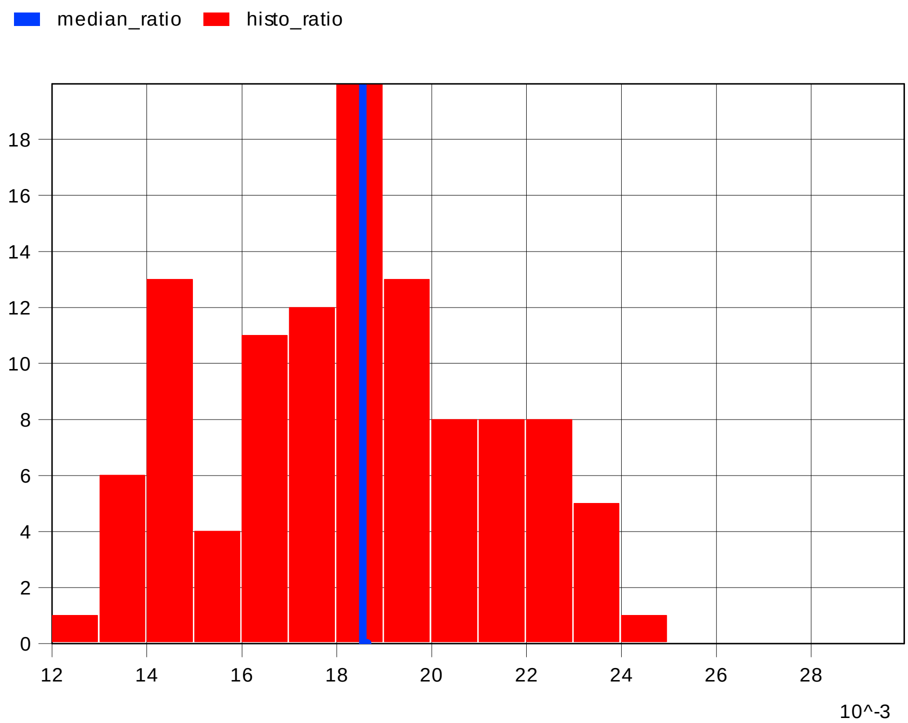


Figure 5-14: *frequency distribution of the solutions for the damping ratio, median: 0.018544*

Inertia:

There have been three different methods developed to determine damping and inertia. Depending on the availability, reliability of the signals a suitable method can be chosen. The assessment of these methods is still under progress. For more details the reader is referred to WP05 report.

5.6.3.3 Results

Assessment of the results that can be obtained by the “deterministic and statistical” approach is still under progress. For more details the reader is referred to WP05 report.

6. Pitch system

In this chapter an example of the application of the six steps approach is discussed for a pitch system. There is a number of different pitch systems used in today's wind turbines. The pitch system discussed below is the pitch system of the Nordex N80, it is *an electric driven pitch system*, individual pitching, no cyclic pitching and pitch to vane.

6.1 **Step 1:** Identify critical failure modes or phenomena for component

The pitch system can be subdivided into several subcomponents. A sketch of the pitch system is depicted in Figure 6-1. For every subcomponent, an analysis can be made to determine the failure mechanisms, failure modes or phenomena. The different subcomponents considered are:

1. pitch bearing
2. pitch gear
3. pitch gearbox
4. pitch brake
5. pitch motor
6. pitch controller / pitch electronics (not depicted in Figure 6-1)
7. pitch encoder (not depicted in Figure 6-1)

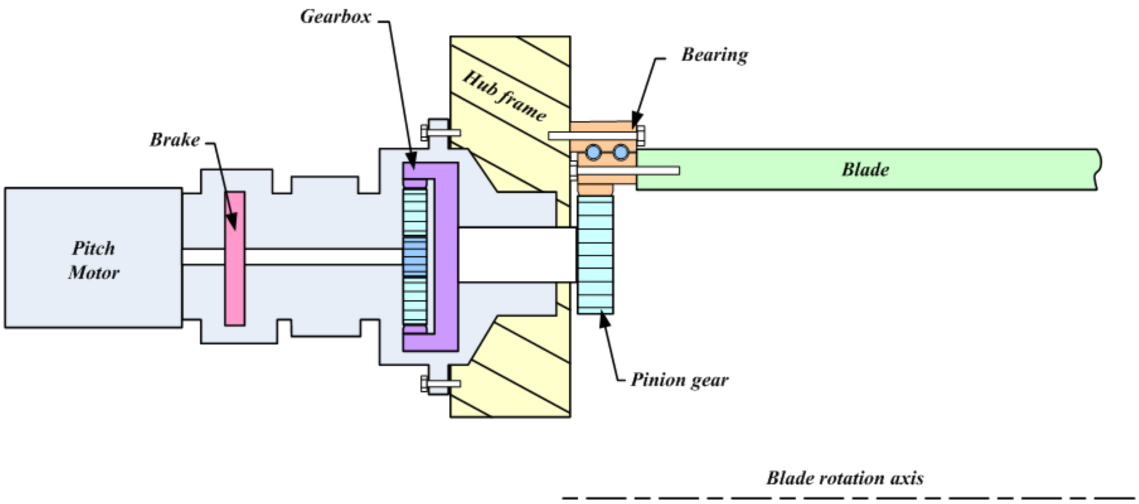


Figure 6-1: *Sketch of pitch drive and connection to blade via slewing bearing (cross-sectional)*

For each of these subcomponents, critical failure modes or phenomena can be determined. To identify these critical failure modes, it is common in industry to perform a *Failure Mode and Effect Analysis (FMEA)*. An FMEA is used to collect and identify possible failures of system components and the impact on the functioning of the entire system. It is common to assign weight factors of each possible failure by estimating the severity of impact on the system level. Below a summary of failure modes and their effects on the pitch system is given. A more extensive FMEA description will be part of WP 6 of the PROTEST project.

Overall pitch system:
If due to wear, ovalisation or friction the difference between the pitch set-point and the actual pitch angle becomes too large, this will lead to a (emergency) shutdown of the turbine.

Pitch bearing:
A typical bearing used to connect a blade to the hub and pitch drive is a so-called slewing bearing. The sketch in Figure 6-1 contains a cross-sectional view of this bearing. The bearing analysed here is a double-row ball bearing slewing ring. Critical failure modes for this bearing are:

- Elastic deformation or ovalisation due to loading (getting stuck, high friction). Ovalisation may also lead to a deformation of the pitch shaft or damage of the pitch bearing, either the slewing ring or the bearing of the pitch shaft.
- Friction (excessive wear, getting stuck) causing fatigue of the bearing.

Both the pitch gear and the pitch bearing are actively lubricated.

Pitch gear:

- Play due to excessive wear results in inaccurate pitch angles and high torque transients in the pitch gearbox and the pitch motor shaft. This play occurs at the gears which are loaded with the blade in the zero position (working point below v-rated). The play also results in high forces on the teeth.
- Not enough play will lead to high friction between the pitch pinion and the pitch gear, which leads to higher loads on the pitch gearbox and pitch motor.

Pitch gearbox:

- Maximum rpm due to internal lubrication and lubricant film
- Maximum torque, if exceeded this will result in damage.
- Oil leakage

Pitch brake:

- Failure of electric brake system, causing the brake to apply (i.e. by wire breakage or excessive vibrations).

Pitch motor.

- The pitch motor is allowed to run at a higher capacity than rated for short times.
- The pitch motor may have a rotational speed which is faster than rated.
- The motor may overheat.

Pitch controller / pitch Electronics

- If one or more blades are no longer controlled, an emergency stop will occur.
- Pitch batteries for the emergency stop system are sensitive for maintenance, low temperatures and trickle charges.

Pitch encoders:

- Calibration of the encoders has to be performed regularly, if the blade is not correctly calibrated, the power curve will not be achieved.

The identified failure mechanisms are summarized in Table 6-1.

Table 6-1: *Summary of failure mechanisms in pitch drive components*

Component	**Failure mechanism**
Pitch bearing	Too much ovalisation
	Too much friction
Pitch gear	Too much play
	Too little play
Pitch gearbox	Too high rotational speed
	Too high torque applied
Pitch brake	Electric system malfunction
Pitch motor	Overloading
	Too high rotational speed
	Overheating
Pitch controller	Control loss for one blade
	Control loss for two blades
Pitch encoders	Calibration offset

The mechanisms which will have the largest impact on the structure and are in need of further analysis are:

1. friction (excessive wear, getting stuck)
2. ovalisation (getting stuck, high friction)

These are the mechanisms that need to be analysed further. If a bearing fails, the replacement costs will be significant; therefore the life of the bearing has to be at least equal to that of the complete turbine. The other failure mechanisms should not be neglected, but in current practice, possible problems will show up during the standard prototype test. Therefore within the PROTEST project, the ovalisation and the friction of the pitch bearing are the mechanisms that the focus is on in setting up the measurement campaign.

Ovalisation results in a deformation of the bearing race way, which can lead to higher friction, the bearings being unable to run at all, higher wear as well as earlier than expected bearing failure. Friction in the bearing itself is also dependent on the loads on the bearing, regardless of whether any ovalisation occurs (though ovalisation is likely to contribute to friction). Generally, higher friction results in higher wear. Obviously, it is likely that there is a cross-correlation between these two, as the ovalisation will probably cause extra friction. To know which is which the models need to be studied first to see if load cases or measurement data can be selected to differentiate between the two.

Steps 2 to 6 of the Six Steps Approach will be performed first for the friction and second for the ovalisation. Therefore, the discussion of the friction models starts in the following section and the discussion of the ovalisation will start in section 6.7.

6.2 **Step 2:** Design the model, Friction

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 torque (moment) of the bearing, the aerodynamic forces (moment) on the rotor blade and the inertia (moment) of the bearing and rotor blade. A pitch bearing is designed based on the loads at the interface with the blade and the bearing, and 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 interface. If the actual friction torque during operation does not correspond to the calculated friction torque, this may have consequences for the pitch 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 torque of the pitch slewing bearing. This model is taken from [9] and allows one to calculate the *starting torque* M_r

of ball bearing slewing rings. The starting torque model is based on theoretical and empirical knowledge according to [9]:

$$M_r = \frac{\mu}{2}\left(4.4 \cdot M_k + F_a \cdot D_L + 2.2 \cdot F_r \cdot D_L \cdot 1.73\right) \tag{6-1}$$

Symbols used in this equation are:

F_a	is the axial load	[kN]
F_r	is the radial load	[kN]
M_k	is the resulting bending moment	[kNm]
D_L	is the bearing race diameter	[m]
μ	is the friction coefficient	[-]

The loads at the blade-bearing interface in this equation are depicted in Figure 6-2.

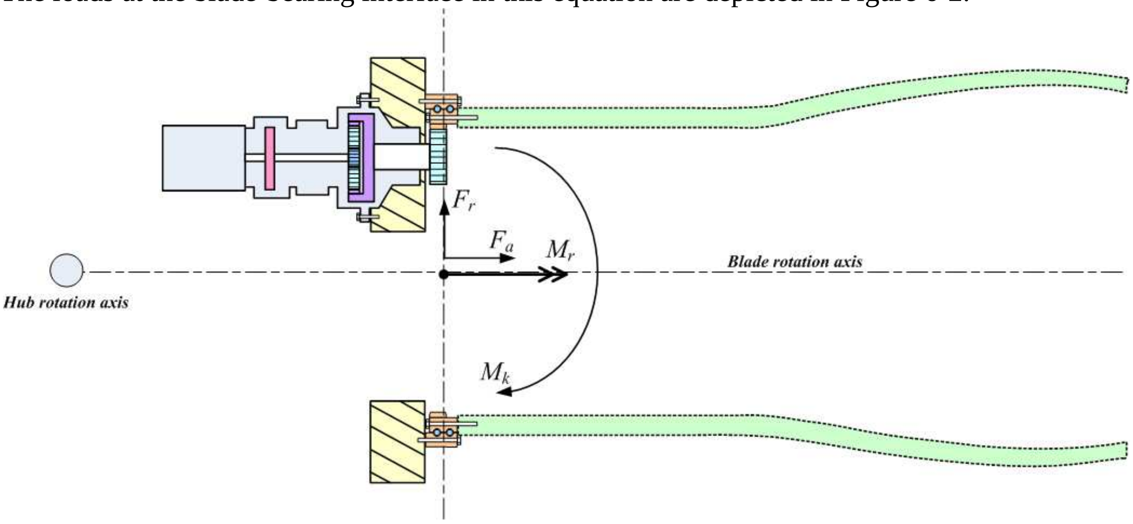


Figure 6-2: *Loads at the blade-bearing interface*

[9] gives the following friction coefficient for a double-row ball bearing slewing ring:

$$\mu = 0.004 \tag{6-2}$$

The friction model equation in (6-1**)** is *valid* for the starting torque, which means that once the bearing is rotating (e.g. pitching), the equation *may* not be valid. Possible improvements of the friction model will be part of WP 6 of the PROTEST project.

6.3 **Step 3:** Run model for various DLCs, Friction

The pitch system is used either to actively pitch the blade (during power production, start up and (emergency stop) or keep the blade at a constant angle (power production, idling). For the Nordex N80, active pitching starts at an average wind speed of around 14 m/s.

Therefore, several DLCs should be run for the model as described in the previous section. A selection of DLCs for fatigue and ultimate loads is made; therefore DLCs 1.1, 1.2, 1.3, 1.4 and 1.5 are suggested to be used. Also the emergency shutdown (DLC 5.1) should be included. Due to the influence of the controls it can also be important to also simulate cases where the details of the controller are of importance e.g. running to idling (DLC 4.1) and start up (DLCs 3.1, 3.2 and 3.3). All load cases are prescribed in the IEC design requirements as specified by [6].

These DLCs can be used as input for ECN's aeroelastic simulation program PHATAS (Program for Horizontal Axis wind Turbine Analysis and Simulation). For this simulation program a model of the Nordex N80 turbine (including pitch controller) exists that has been validated at ECN [10]. PHATAS is able to simulate amongst others the loads acting on blade segments *in* *time* for the selected DLCs. The results are used as input for the friction torque calculation in Step 2, see equation (6-1**)**. The friction torque can then be calculated as a time series in Step 4 (see section 6.4).

The azimuth angle in PHATAS is defined clockwise with 0° for blade 1 in a vertical position as seen from the hub. A definition of directions of forces and moments acting on a blade segment, which are output from PHATAS, is given in Figure 6-3. The PHATAS output will be selected in such a way that *forces and moments correspond to the position at the blade root segment* in the model. This will ensure that the forces and moments output corresponds to the blade-bearing interface.

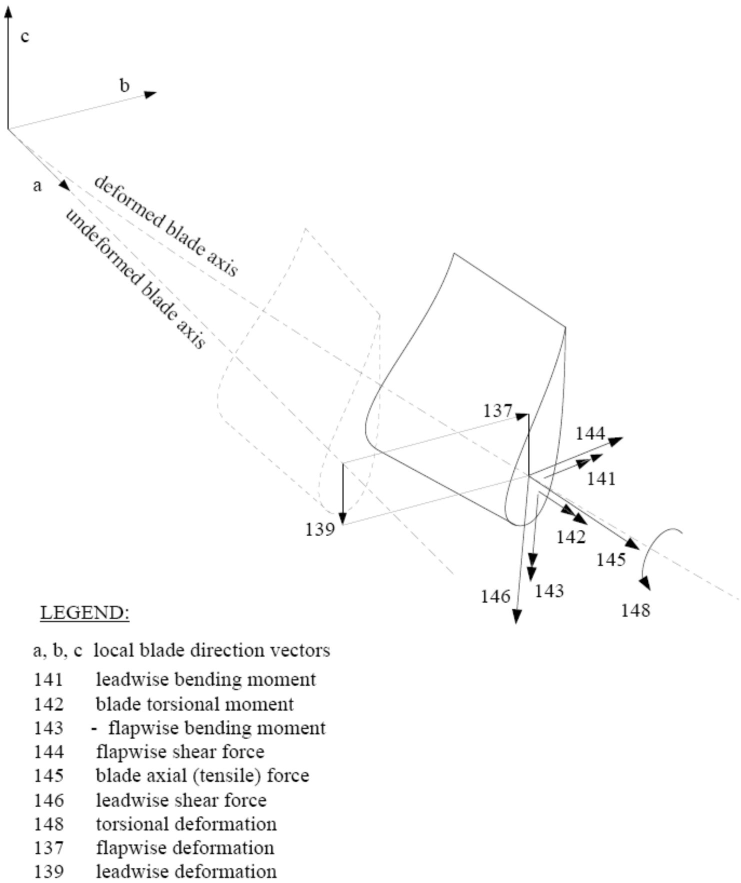


Figure 6-3: The *PHATAS definition of the blade loads output in the deformed rotor plane reference frame* [12]

Since fatigue of the bearing by friction will be investigated in the present example, it is chosen to start the analysis for DLC 1.2 as specified in [6]. This DLC is specifically meant for a fatigue analysis. The design situation is power production and the wind conditions are specified for a normal turbulence model as:

$$v_{in} < v_{hub} < v_{out} \qquad \textbf{(6-3)}$$

Equation (6-3**)** can be analyzed with PHATAS for a range of wind speed bins v_{hub} within this interval. Each PHATAS simulation per wind speed bin is run for a simulation length of 10 minutes. The PHATAS output can then be used to determine the friction model input and output parameters in Step 4.

The PHATAS simulations results for each DLC are stored in a post-processed comma-separated file. An example of some PHATAS output for the wind speed bin of 12 m/s, with a turbulence intensity of 10.1% is plotted in Figure 6-4.

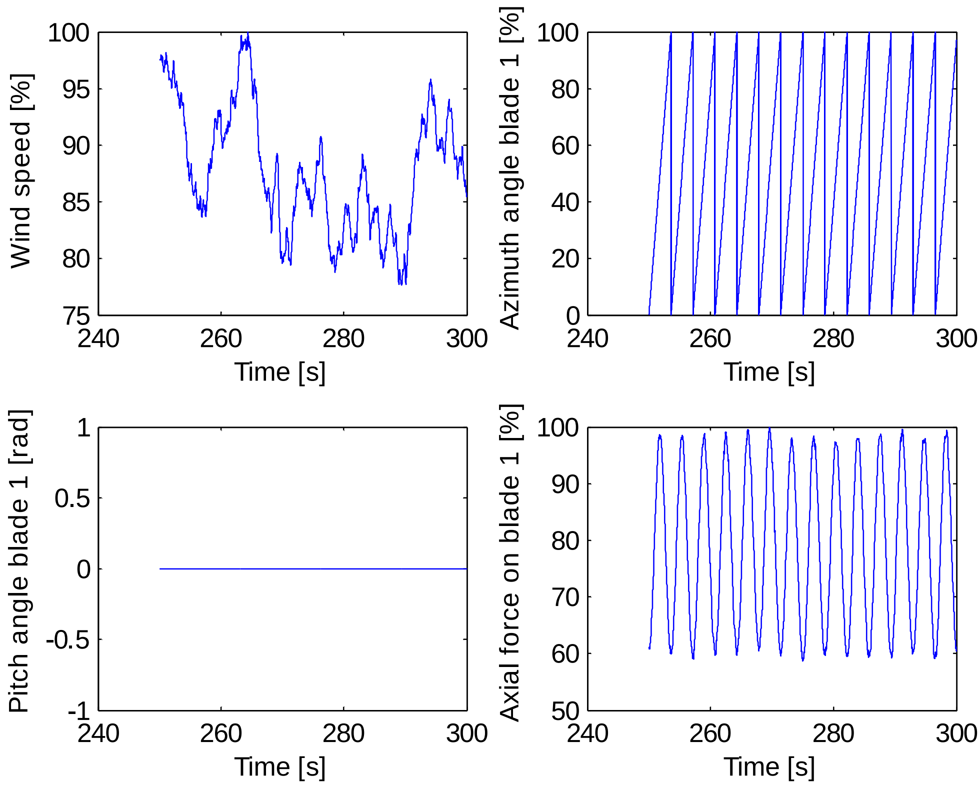


Figure 6-4: *PHATAS output DLC 1.2 for a 50 second interval of the simulated 10 minute time series: wind speed, azimuth angle, pitch angle, and axial force*

The PHATAS results show that for the simulated wind speed bin, the pitch controller was not active and the pitch angle remained constant. The axial force in blade 1 clearly varies with the rotor azimuth angle, caused by the reversal of gravitational forces due to the blade azimuth position. More in depth analysis and analysis of other DLCs and more wind speed bins will be part of WP 6 of the PROTEST project.

6.4 **Step 4:** Determine input and output parameters, determine how “certain” they are, and if they need to be verified/measured, Friction

The input parameters for the friction model are the modelling parameters in equation (6-1**)**:

- F_a is the axial load [kN]
- F_r is the radial load [kN]
- M_k is the resulting bending moment [kNm]
- D_L is the bearing race diameter [m]
- μ is the friction coefficient [-]

The blade loads and moments are time series output parameters from the PHATAS post-processor as shown in Figure 6-4. PHATAS has no resulting radial load (F_r) and bending moment (M_k) on the blade as output, however it does give the leadwise and flapwise forces and moments on the blade as a result. The resulting radial force at the blade-bearing interface can be calculated by adding the vectors of the leadwise and flapwise blade forces:

$$F_r = \sqrt{F_{lead}^2 + F_{flap}^2} \tag{6-4}$$

Similarly, the resulting bending moment can be calculated by adding those two moment vectors:

$$M_k = \sqrt{M_{lead}^2 + M_{flap}^2} \tag{6-5}$$

The bearing race diameter (D_L) is specified in a drawing supplied by Nordex [11]. The friction coefficient is supplied in equation (6-2**)** by [9]. 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:

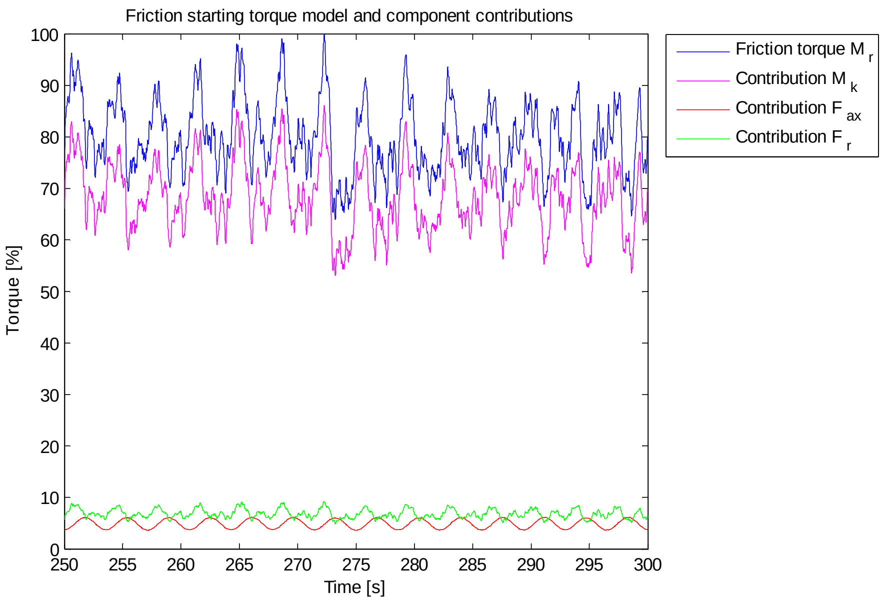
- Wind speed [m/s]
- Yaw angle [deg]
- Azimuth angle [deg]
- Pitch angle [deg]

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 is shown in Table 6-2.

Table 6-2: *Input and output parameters for the friction model*

Parameter	**Input/Output**	**Measurement required**	**Uncertainty (%)**
Axial load F_a	input	Yes, verify load	Unknown
Radial load F_r	input	Yes, verify load	Unknown
Resulting bending 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 %[1] [9]

The friction starting torque is calculated from (6-1**)** in MATLAB by reading the time-series from the PHATAS output and applying equations (6-2**)**, (6-4**)**, and (6-5**)**. The resulting starting torque corresponding to the PHATAS output for DLC 1.2 shown in Figure 6-4 is plotted in time and depicted in Figure 6-5.



[1] With respect to the model, not taking into account any uncertainty in the input of the model.

Figure 6-5: *Friction starting torque and individual contributions of tilting moment, axial, and radial forces for a 50 second interval*

Figure 6-5 also shows the contributions of the bending moment, axial force and radial force to the calculated starting torque. We see that the bending moment has by far the highest influence on the starting torque. Analysis of the friction torque model for other DLCs and more wind speed bins will be part of WP 6 of the PROTEST project.

6.5 **Step 5:** Design measurement campaign to verify models and quantify parameters, Friction

The prototype measurement campaign will be designed to verify the friction model input and output parameters (see also project objectives in 1.2.1). Measurements will be conducted on a Nordex N80 wind turbine at ECN's wind turbine test park Wieringermeer (EWTW). The parameters to be measured for the friction model have been determined and are given in Table 6-2.

The diameter of the raceway does not need to be measured, since it is given by the manufacturer on a component design drawing and a high certainty is assumed. The friction coefficient cannot be measured directly. A value has been assumed and it might be possible to try to indirectly verify this value by measuring the friction torque. The measured friction torque might then be used to 'tune' the friction model in Step 6.

Direct measurement of the friction torque within the bearing itself is not possible. However, the friction torque can be indirectly measured from the difference between the output torque of pitch motor and the blade torsion moment. This means that measurements are required at the pitch drive and the blade root. The suggested location of these measurements is sketched in Figure 6-6.

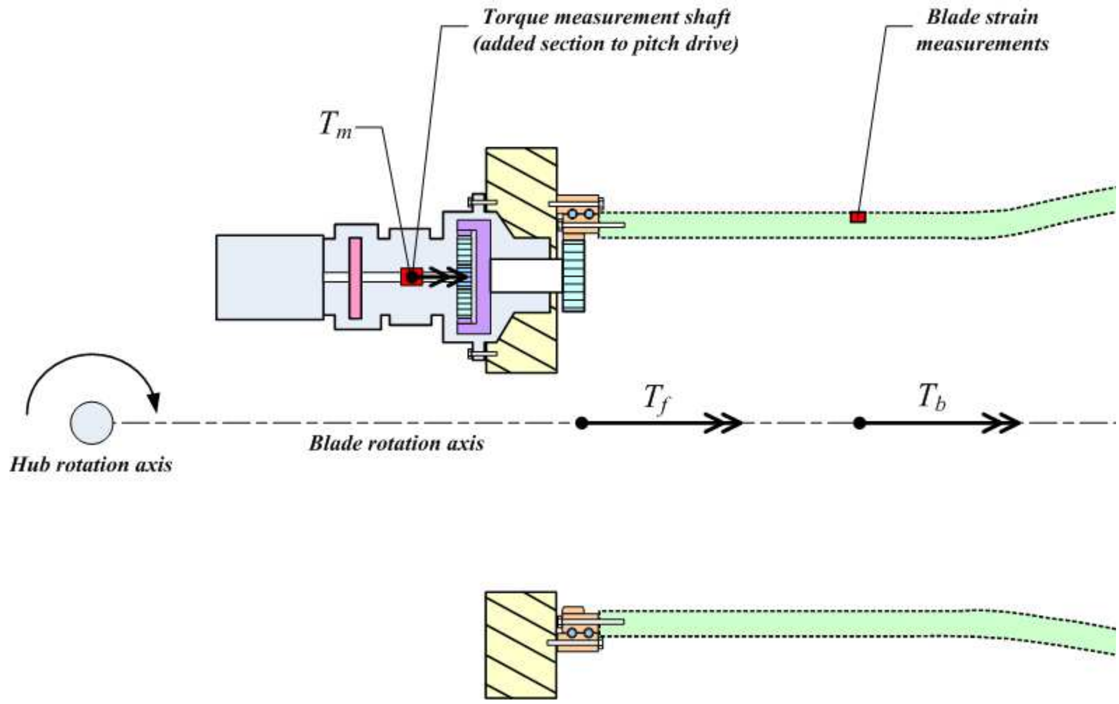


Figure 6-6: *Location of measurements to compute friction torque T_f*

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 some compensation has to be applied for the inertia of the mass of the blade part and rotating piece of the bearing not taken into account. Furthermore, the pitch motor torque is measured before the pitch gearbox. This means the resulting friction torque includes friction losses in the pitch gearbox and pinion gear.

Therefore, a *measurement model* has been developed for a backward calculation of the friction torque from the blade torque and pitch motor torque measurements. Considering positive measurement direction definitions in Appendix B, the following equilibrium equation for the pitch motor torque with respect to the rotation of the blade is derived:

$$T_m \cdot i_{gbx} \cdot i_{gear} = T_b \pm T_f + \ddot{\theta}\left\{\left(i_{gbx} \cdot i_{gear}\right)^2 I_{gbx} + I_b + I_{br}\right\} \tag{6-6}$$

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}). [Nm]
- T_b is the measured torque acting on the blade [Nm]
- T_f is the friction torque (its sign is dependent on direction of rotation) [Nm]
- $\ddot{\theta}$ is the acceleration of the rotor blade [rad/s^2]
- I_{gbx} is the inertia of the pitch gearbox [kgm^2]
- I_b is the blade inertia from blade root up to the measurement location [kgm^2]
- I_{br} is the pitch bearing inertia [kgm^2]

Equation (6-6**)** now yields for the friction torque:

$$\begin{aligned} & T_m \cdot i_{gbx} \cdot i_{gear} = T_b \pm T_f + \ddot{\theta}\left\{\left(i_{gbx} \cdot i_{gear}\right)^2 I_{gbx} + I_b + I_{br}\right\} \Leftrightarrow \\ & T_f = \pm\left(T_m \cdot i_{gbx} \cdot i_{gear} - T_b - \ddot{\theta}\left\{\left(i_{gbx} \cdot i_{gear}\right)^2 I_{gbx} + I_b + I_{br}\right\}\right) \end{aligned} \tag{6-7}$$

In equation (6-7**)**, the pitch motor torque T_m and the blade torque T_b shall be measured. The acceleration of the rotor blade can be calculated from the 2nd derivative of the pitch angle:

$$\ddot{\theta} = \frac{\partial^2 \theta}{\partial t^2} \tag{6-8}$$

The inertia terms in equation (6-7**)** can be calculated from the manufacturer's design drawings. The gearbox and pinion gear ratios are supplied by the manufacturer. The uncertainty in these terms is assumed very low and thus no measurements will be performed for verification. In contrast to the theoretical model, the measurement model should be valid for calculation of the friction torque during blade rotation as well. A more extensive assessment of the validity of this equilibrium equation and uncertainties is input for WP6 of the PROTEST project.

To verify the loads acting on the blade-bearing interface, the axial force in the blade, the radial forces (leadwise and flapwise, see equation (6-4**)**), and the blade bending moments (leadwise and edgewise, see equation (6-5**)**) *at the blade-bearing interface* are required. The blade bending moments are identified as required measurements in Table 3-7, section 3.2. The axial force in the blade may be derived from the centrifugal forces and gravity forces. The radial forces at the blade-bearing interface can be estimated from the measured bending moments.

An overview of which measurements are required for verification of the design loads and validation of the friction model is given in

Table 6-3. In this overview, measurement signal types defined in the 2nd column correspond to model parameters in the 1st column. Some of the model parameters have been grouped, since they share measurement results.

The raw measurements usually need to be corrected for offsets and gains. According to the definition in [8] *any calculation from a measured signal including calibration corrections for offsets and gains is a pseudo-signal*. Some pseudo-signals can be completely calculated from calibrated measurements (e.g. the leadwise and flapwise bending moments). The calculated pseudo-signals for which the measured signals are input are described in the 3rd column.

The signals and pseudo-signals are stored in ECN's database system "Informatix". The sensor points of installation, specifications, calibrations, and equations of the pseudo-signals are completely described in a dedicated measurement report in [8].

Table 6-3: *Required measurement signals on N80 for verification of friction model* [2] *[8]*

Parameter in model	Measured signal types	Frequency	Method	Input for pseudo-signal(s)
1. Axial load Fa 2. Radial load Fr 3. Resulting bending moment Mk	Blade 1, Root, flap moment	128 Hz	T-shape strain gauges	Leadwise & flapwise blade bending, In plane & out of plane blade bending
	Blade 1, Root, edge moment	128 Hz	T-shape strain gauges	Leadwise & flapwise blade bending, In plane & out of plane blade bending
	Blade 2, Root, flap moment	128 Hz	T-shape strain gauges	Leadwise & flapwise blade bending, In plane & out of plane blade bending
	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
	Blade 3, Root, edge moment	128 Hz	T-shape strain gauges	Leadwise & flapwise blade bending, In plane & out of plane blade bending
	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
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	Yaw angle
8. Azimuth angle rotor	Azimuth angle	128 Hz	Incremental encoder	Azimuth angle, Rotor thrust
9. Pitch angle blade	Pitch angle blade 1	128 Hz	Absolute encoder	Pitch angle blade 1
	Pitch angle blade 2	128 Hz	Absolute encoder	Pitch angle blade 2
	Pitch angle blade 3	128 Hz	Absolute encoder	Pitch angle blade 3
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
	Pitch motor current	32 Hz	PLC	
	Pitch motor voltage	32 Hz	PLC	

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 6-4. As described in this table, some of these cases need to be provided for a selection of wind speed bins. These different cases are required as it is likely that the model will be valid for one case, but invalid for another case, or the tuned parameters might be dependent on the specific MLC. For example, it is clear that there will be a difference between keeping the pitch constant and a dynamic case where the pitch is changing as there is a difference between the start-up friction and the dynamic friction.

Table 6-4: *Measurement load cases that are required for tuning and validating the friction model of the pitch system*

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)

[2] The measurement values from the wind turbine PLC are all calibrated externally and stored directly in the database

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 down, pitching	Measurement for friction model
Stand still, blade vertically up, pitching	Measurement for friction model
Stand still, blade horizontally, pitching	Measurement for friction model
Pitch tests, continuously pitching the blades of the turbine with triangular desired pitch angle	Measurement to calibrate torsion measurements
Removing the pitch pinion and operate the pitch motor	Measurement to determine the friction and inertia of the pitch gearbox

Each time series should include a minimum of three full rotations while running, therefore this will be in the order of 10 or 20 seconds for the N80, and at least two full rotations while idling, which results in much longer time series that are required, e.g. 200 seconds. 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.

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 are prescribed by specification of the corresponding capture matrices (see section 2.1.1.2). As an example the layout of such capture matrices is illustrated in Table 6-5.

Table 6-5: *Capture matrices*

Normal power production								
Time series length	10 minutes							
Wind [m/s] / I (%)	$v_{in} \rightarrow v_{in}+2$	$v_r-2 \rightarrow v_r$	$v_r \rightarrow v_r+2$	$> v_r+5$				
<8								
8-15								
>15								

Power production plus occurrence of fault			
Time series length	2 minutes	2 minutes	2 minutes
Wind [m/s] / fault condition[3]	$v_{in} \rightarrow v_r-2$	$v_r-2 \rightarrow v_r+2$	$>v_r+2$
fault 1			
fault 2			

Parked (stand still and/or idling)			
Time series length	20 minutes	2 minutes	2 minutes
Idling			

Normal start-up and shut-down events			
Event	$v_{in} \rightarrow v_r-2$	$v_r-2 \rightarrow v_r+2$	$> v_r+2$
Start-up			
Normal shut-down			

[3] Any fault not resulting in immediate shut down

Other transient events	
Event	
Emergency shut down	3 times

Specific measurement conditions	
Event	
Parked, blade pointing downwards, pitching, $v < v_{in} + 2$	3 times min-max-min pitch angle

Further investigation of required MLCs and further specification of the capture matrices will be part of WP 6 of the PROTEST project.

6.6 **Step 6**: Process measurement data and check/improve models/ model parameters, Friction.

A measurement campaign at the Nordex N80 turbine is initiated and ongoing including the signals specified in

Table 6-3. Once the measurements have been performed, these need to be analysed for (see section 1.2.1):

1. Verification of the model input parameters (validate simulation tool)
2. Analysis of the friction torque model output (verify friction torque as design load on the pitch bearing)

The 2nd objective is considered as very important and in the present example *the friction torque model output from equation* (6-1**)** *using the simulation model inputs* (in this case by PHATAS) *will be compared to the measurement model output for the friction torque from equation* (6-8**)**.

The theoretical analysis of the starting friction torque was made for DLC 1.2 (normal power production) in section 6.4. MLC 1.1 corresponds to this DLC, see also Table 6-4. For illustration purposes, a *single* measurement for a matching wind speed bin (12 m/s) is selected. The measurement pseudo-signals are read-in from a comma-separated file which was extracted from the measurement database. Some of the measured pseudo-signals are plotted in Figure 6-7.

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 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 will be required.

Compared to the PHATAS simulation result for DLC 1.2 in Figure 6-4, the number of rotor rotations during the 50 second interval is almost the same. The measurement shows that the pitch motor is delivering some torque continuously to maintain the 0° position of the rotor blade. In Figure 6-7, the motor delivers the maximum torque if the pitch angle is near -0.2%, and minimum torque if the pitch angle is near -0.17%. This situation simulates the blade just overcoming the starting friction torque. The friction torque is calculated in Matlab by substitution of the measurement signals in equation (6-7**)**. The resulting friction torque corresponding to MLC 1.1 in Figure 6-7 is plotted in time in Figure 6-8.

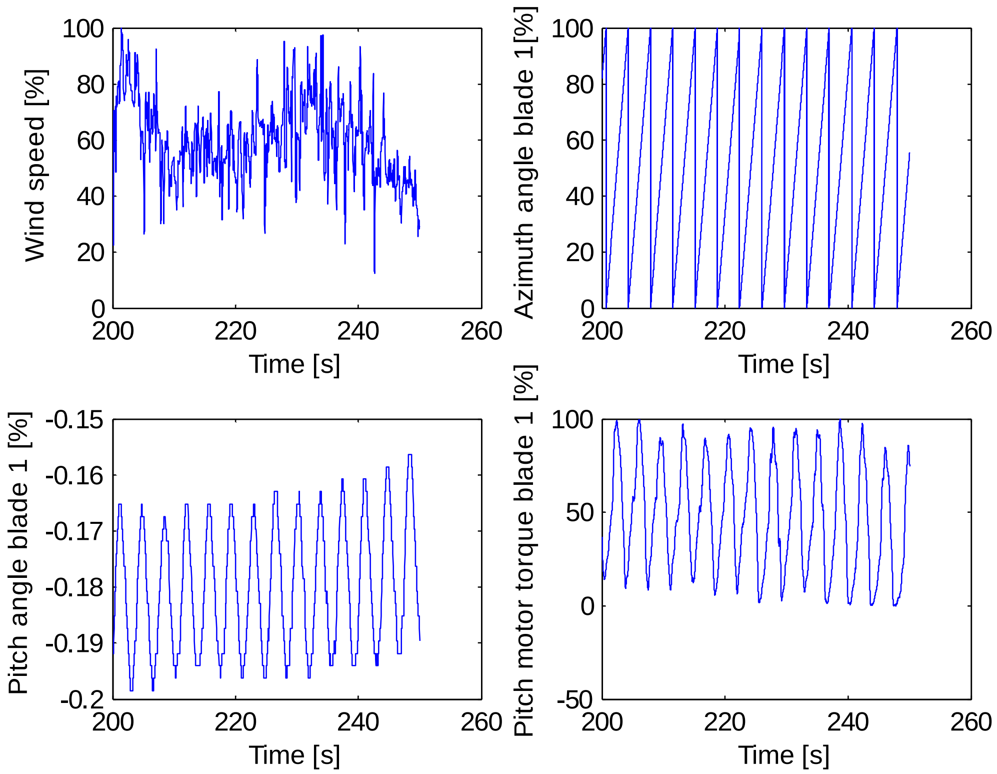


Figure 6-7: *Measured pseudo-signals MLC 1.1 for a 50 second interval of a recorded 10 minute time series: wind speed, azimuth angle, pitch angle, and pitch motor torque*

Apart from the friction torque, the pitch motor torque, the blade torque and the torque by inertia terms in (6-7**)** are also shown in Figure 6-8. Since the rotor blade has little rotation (difference between pitch angles is less than 0.04 degrees), the torque due to inertia is close to zero.
To compare the starting friction torque calculated with PHATAS input in Figure 6-5 and the friction torque calculated according to the measurement model in Figure 6-8 , the rotor azimuth angle shall be used as input to synchronize the time results. For both 50 second intervals plotted, the rotor azimuth angle is used to define each start of a new rotation (ψ_1=0°). The friction torque from the first full rotation within both intervals is compared side-by-side for a 10 second period in Figure 6-9. The selected time intervals are [253.58 : 263.58] for the starting friction torque model and [200.66 : 210.66] for the friction torque measurement model. As a reference, the measured pitch motor torque is plotted as well.

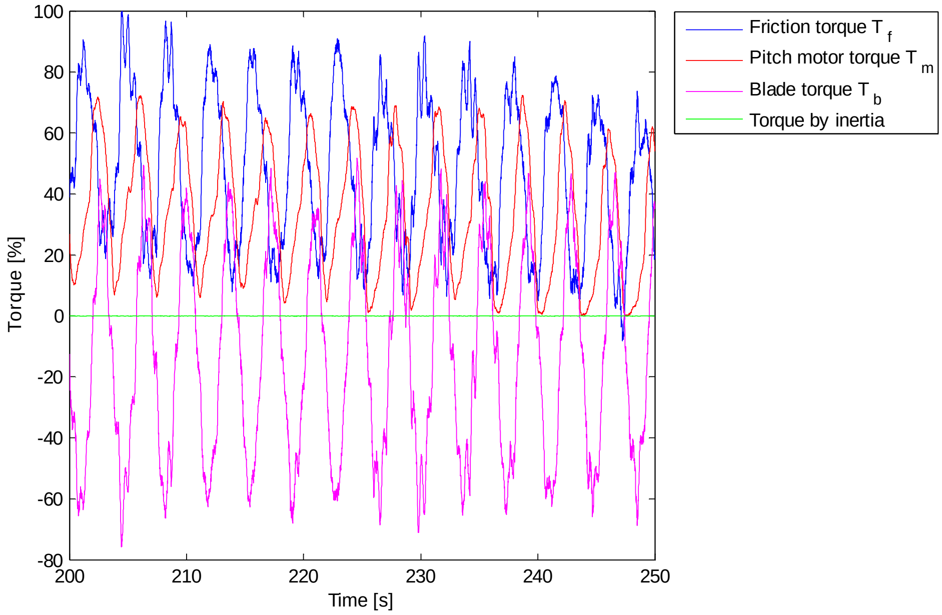


Figure 6-8: *Friction torque calculated with measurement model for 50 second interval*

The azimuth angle comparison in Figure 6-9, show 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 seems to correlate relatively well with the starting torque friction model, specifically for azimuth angles near 90°. This is illustrated by the three peaks of the friction torque that correspond to azimuth angles of 90° (blade horizontal and going down) in Figure 6-8. It may be also seen that the shape and peaks of the graph correlate well, albeit with some large differences in the friction torque values. The starting friction torque model seems to overestimate the measured friction torque for azimuth angles other than 90°. The maximum difference is seen at azimuth angles of 270°.

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 4-1, 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 WP 6 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. (**6-1**)) is not applicable straightforward anymore, and may need to be adjusted by considering a non constant friction coefficient.

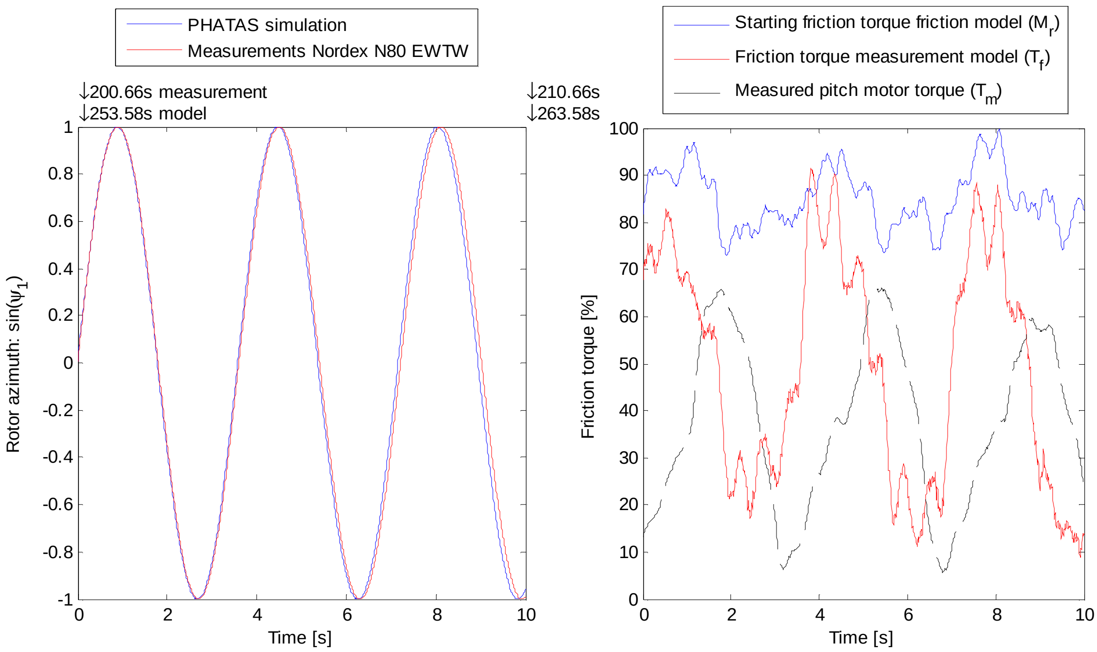


Figure 6-9: *Synchronized friction torque output comparison for 10 second interval (1) Starting friction torque model and (2) Friction torque measurement model*

More in depth analysis and comparisons between the starting friction torque model and the measurements will be part of WP 6 of the PROTEST project.

Based on experiences of the friction torque in the bearing during research, the following recommendations for further research within the scope of WP6 of the PROTEST project may be given:

- Analysis of the loads at the blade-bearing interface with PHATAS for the envisaged design load cases and wind speed bins.
- Analysis of starting friction torque model results corresponding to the PHATAS load simulations.
- Investigation of the validity of the measurement model equation for the friction torque
- Assessment of uncertainties of the measured (pseudo-)signals, with special attention to the blade torque calibration, determination of pitch acceleration and pitch motor torque calibration.
- Specification of the number of measurements in the capture matrices and collection of matching measurement data for envisaged measurement load cases.
- Investigate possible improvements to the friction model for calculation of the friction torque based on comparisons to measured torque (e.g. by tuning the model's friction coefficient, select and compare a different model).

6.7 **Step 2:** Design the model, Ovalisation

It should be noted that the analysis of the ovalisation at this stage is not complete yet. The work is being continued and the final results will be published in the reports of work packages 6 (confidential) and 8 (public).

Ovalisation is a deformation of the bearing that occurs under loading. It is difficult to estimate the deformation due to the given loads and vice versa, since a detailed model of the blade and the hub is necessary for this.

It is first assumed that the deformation of the bearing is mostly in the plane of the bearing and that, as a starting point, the bearing takes an oval shape under deformation. Once it became apparent that this assumption did not suffice, other possible theoretical shapes were also examined, including the shapes of a ring loaded with a point force, a ring loaded with a distributed force and an asymmetric shape. The shape and corresponding strain for the three different estimation models are shown in Figure 6-10. Combinations of different shapes were also examined, but fitting these to the data resulted in numerical difficulties.

As none of the above strategies created an accurate fit of the data, the model was changed to a finite element model of the structure. This is not convenient because of its complexity, but a simpler model based on the results from this detailed model may be created later on.

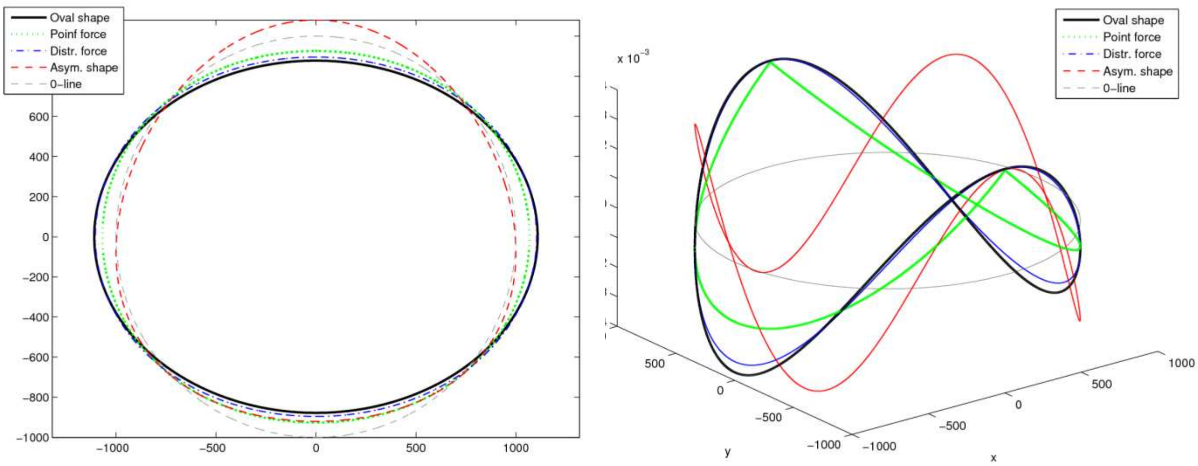


Figure 6-10: *Shape of the deformed ring for different models on the left and the corresponding strains on the right.*

6.8 **Step 3:** Run model for various DLCs, Ovalisation

For the ovalisation there are no upfront calculations to be performed to the model. However, the forces acting on the bearing should be determined using another model and tool, in this case PHATAS [12]. Running these DLCs in PHATAS results in the forces and moments that will act on the bearing in different situations, see also section 6.3.

Once a correlation is established between the deformation model and the applied loads, then the calculations can show which load cases are the most critical.

6.9 **Step 4:** Determine input and output parameters, determine how “certain” they are, and if they need to be verified/measured, Ovalisation

For the ovalisation, the most important assumption is the deformation of the shape. The shape needs to be verified. Apart from the shape, the magnitude of the deformation and the correlation to the loads should be established.

A complete overview of the discussed input and output parameters is given in Table 6-6

Table 6-6:*Input and output parameters for the ovalisation model*

Parameter	**Input/Output**	**Verification/measurement needed**
Deformation shape	Input	Verification needed
Correlation loads and deformation	Input	Verification needed
Strains in bearing	Output	Measurements needed
Ovalisation	Output	

6.10 **Step 5:** Design measurement campaign to verify models and quantify parameters, Ovalisation

The input parameters for the ovalisation model, as described in the previous step, are the deformation shape and the correlation between the loads and the deformation. These are yet unknown and therefore need to be determined using the measurements. This means that there is a strong preference to measure the deformation of the bearing at more positions than are used to fit the model. This allows one to assess the quality of the fit; the deformation according to the model fitted on other measurement points can be compared to the measured deformation in the point that the model has not been fitted to.

Measurements that should allow validation of the model are the strains on the bearing. The choice was made to measure the strains at 8 different locations on the bearing, as illustrated in Figure 6-11. The strains are measured using full bridge strain and only measure the in-plane strains, i.e. the strains tangential to the bearing. As a further verification, pitch bearing deflection measurements are used. The pitch bearing deflection is measured across the pitch bearing by means of LVDT displacement sensors. A steel wire is placed in the pitch root between the leading and trailing edge. This steel wire is pulled by means of two springs to avoid vibrations of the steel wire. The LVDT sensor is located at the end of this steel wire. The same is done perpendicular to this between the suction side and compression side of the pitch bearing. Unfortunately it was not practically possible to place more LVDT’s in the bearing.

Because the goal of the measurement campaign is to tune and validate the simulation models that have been used for the design of the mechanical components, the load cases that are of importance are those where pitching takes place, especially those above rated wind speed. However, to establish the correct model, measurements are first compared to the model for idling cases, first for the default pitch angle at idling. If these can be fitted, then different pitch angles and loads should be examined.

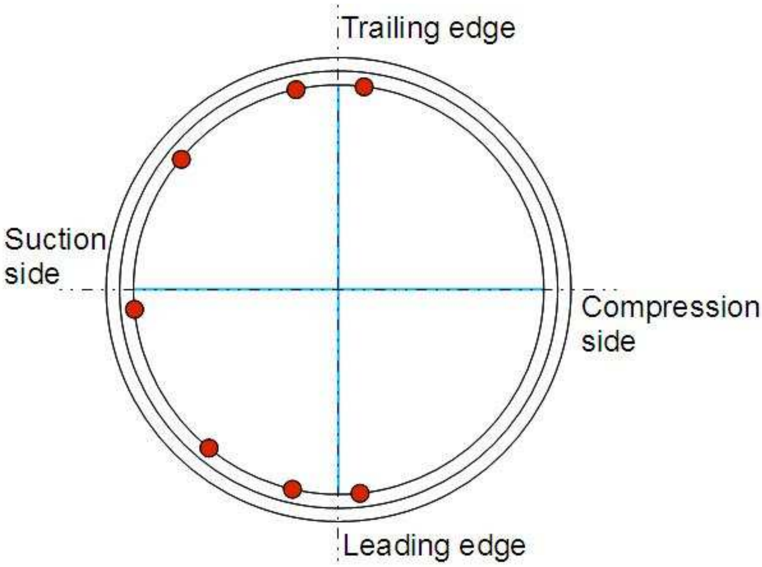


Figure 6-11 *Location of pitch strain gauges*

To give an overview of which measurements are needed for the ovalisation model Table 6-7 and Table 6-8 give the required measured and pseudo signals for the model.

Table 6-7: *Required measured signals for the ovalisation model*

Measured signal	**Frequency**	**Comments**
Strains on bearing	128 Hz	Full bridge strain sensors
Pitch bearing deflection	128 Hz	LVDT’s : LE-TE, SS-CS
Blade root bending flap	128 Hz	T-shape strain gauges
Blade root bending edge	128 Hz	T-shape strain gauges
Pitch bearing temperature	4 Hz	SS, CS, LE, TE : Pt 100
Pitch motor temperature	4 Hz	
Pitch gearbox temperature	4 Hz	
Wind speed	32 Hz	PLC
Wind direction	32 Hz	PLC
Electric active power PLC	32 Hz	PLC
Pitch angle PLC	32 Hz	PLC
Azimuth position of blade		

Table 6-8: *Required pseudo signals for the ovalisation model*

Pseudo signal	Comments
Blade root moments	These are needed to establish correlation between the deformation and the loads.

For the ovalisation, the analysis is purely fitting the deformation model to the measurements. These results can then be supplied to the bearing manufacturer to make sure that the bearing is behaving in accordance with expectations. The effect of the ovalisation may be noticeable in the friction of the bearing, ovalisation is likely to increase the friction. If this effect is noticeable, this could be included in the equation for the bearing friction, equation 6.1.

The measurement cases for this model are the same as those discussed for the friction model, due to the close relation between friction and ovalisation. Therefore these cases can be found in

Table 6-4 and Table 6-5. These different cases will be analysed in WP 6, where also the measurements are performed on the turbine.

6.11 **Step 6**: Process measurement data and check/improve models/ model parameters, Ovalisation.

The models will first be fitted for idling measurements. If these can be fitted, other load cases should be examined that are more critical for the pitch bearing design.

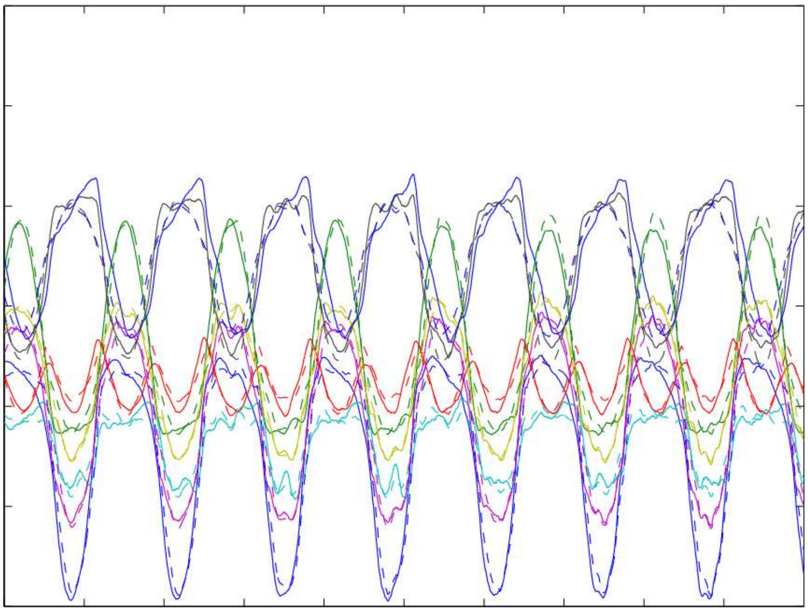


Figure 6-12: *Measured data (solid lines) and data reconstructed on the basis of correlation (dashed lines) match well if wind conditions do not change much*

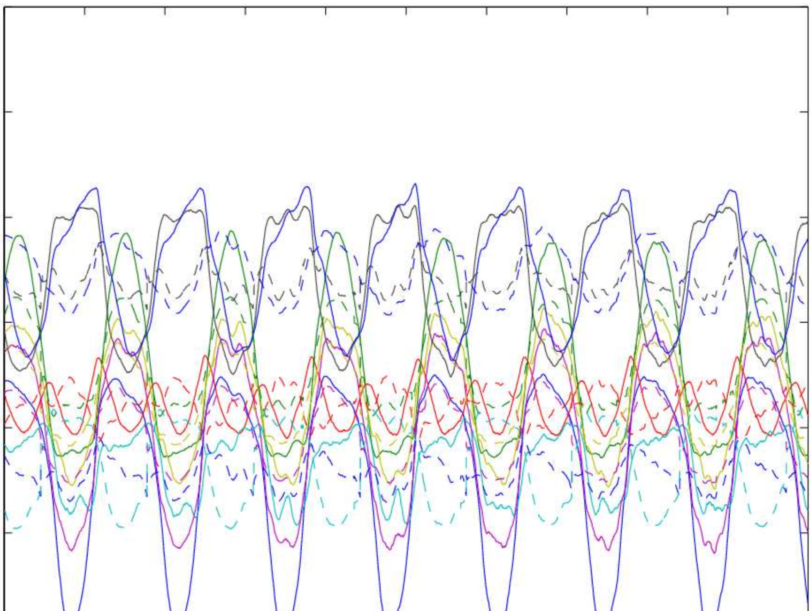


Figure 6-13: *Measured data (solid lines) and the fitted model based on the assumption of an oval shape (dashed lines) do not match well.*

Figure 6-12 shows the strain measurements for an idling case and a reconstruction based on a correlation with the blade loads. The strains and the loads correlate well. This suggests that it should be possible to predict the ovalisation on the basis of the loads if a model can be found that fits the data well. Figure 6-13 shows how the data that was reconstructed on the basis of a fitted ovalisation model. The fitting procedure allowed correction for constant offsets in the sensor signals. It was assumed that the bearing deforms as a perfect oval. The model clearly does not fit the data well.

Other models, based on assumptions of simple shapes, did give somewhat better fits, but still insufficient to be a good model of the deformations.

That none of the assumed shapes gives a good approximation of the data indicates that the basic assumptions are not valid. The basic assumptions are that the deformation only occurs in the plane of the bearing and that the measured strains are due to in-plane deformations. To establish whether this is correct, a finite element model will be created to examine the stress and strain distributions on the bearing. The results of the FEM analysis and the ovalisation model will be described in the final reports of WP 6 and WP8.

7. Yaw system

The description for the recommendations of the in-situ measurements of load for the wind turbine yaw system closely follows IEC/TS 61400-13[13], while the measured signals follow the recommendations of within the PROTEST project Deliverable [14]. In Figure 7-1 a sketch of the yaw system components of interest are shown. The main components considered are:

- Yaw bearing
- Yaw transmission system (including the pinion gear)
- Yaw motor
- Yaw brakes

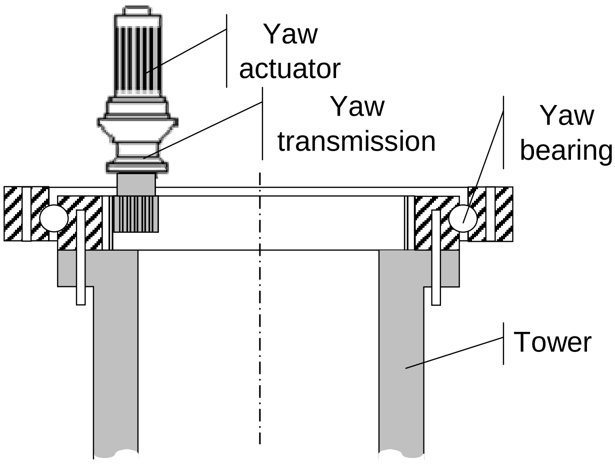


Figure 7-1 *Schematic lay-out of the yaw system*

Within the PROTEST project the interest lays mainly in the mechanical load carrying elements of each system, while the electrical and electronic subsystems are only limited treated.
In a six steps approach the issues that should be covered regarding Step 1 to 3 of the approach, closely resemble the issues covered for the electrical pitch system, as presented in the previous chapter, chapter 6 of the current document. Therefore, in the current section focus will be on the last Steps (Step 4 to 6), concentrating on measurement aspects regarding the yaw system.

7.1 Load measurement programmes for the yaw system

The load measurement program follows IEC/TS 61400-13 [13], as also shown in Table **7-1**, yet a distinction is being made for cases where operation of the yaw system is captured and cases without, as proposed in [14]. Thus, for the concrete description of the measurement load cases (MLCs) the reader is referred to the relevant IEC technical specification IEC/TS 61400-13 [13].
Ideally the capture matrices including the minimum recommended number of time series as set in IEC/TS 61400-13[13] should be used for yawing and non-yawing conditions, that is for cases with no activity (operation) of the yaw system within the time series and with activity (operation) of the yaw system within the time series. For the yaw system empirical load determination or model validation both cases (i.e. yaw system operation and yaw system non-operation) are of interest, since some parts of the yaw system are affected only in cases of yaw system activity, while other parts are affected by both yaw system activity and non-activity. For example yaw system activity obviously affects the loading on the yaw actuator (driver), while both cases are important for the life estimation of the yaw bearing with a particular emphasis on the statistics of yaw operation and yaw standstill, since this affects the lubrication of the system.

Table 7-1: *MLCs targeted for the yaw system*

MLC number[4]	Yaw MLC	Short description	Target wind speed[5]	Notes
1.1	1.1.1	Power Production (yawing)	$v_{in} < v_{hub} < v_{out}$[6]	Referring to normal operation (if yaw system is active then the file is recorded under this classification)
	1.1.2	Power Production (non-yawing)	$v_{in} < v_{hub} < v_{out}$	Referring to normal operation (yaw system in-active)
1.2	1.2.1	Power Production plus occurrence of fault	$v_{in} < v_{hub} < v_{out}$	If necessary split to yawing and non-yawing condition
1.3	1.3.1	Parked, idling, non-yawing	$v_{in} < v_{hub} < 0.75v_{e1}$	(if possible include yaw misalignment)
	1.3.2	Parked, idling, yawing	$v_{in} < v_{hub} < 0.75v_{e1}$	
2.1	2.1.1	Start-up non-yawing	v_{in} and $>v_r$+2m/s	
	2.1.2	Start up yawing	v_{in} and $>v_r$+2m/s	
2.2	2.2.1	Normal shut-down non-yawing	v_{in}, v_r and $>v_r$+2m/s	
	2.2.2	Normal shut-down yawing	v_{in}, v_r and $>v_r$+2m/s	
2.3	2.3.1	Emergency shut-down	v_{in} and $>v_r$+2m/s	If necessary split to yawing and non-yawing condition
2.4	2.4.1	Grid failure	v_r and $>v_r$+2m/s	
2.5	2.5.1	Over-speed activation of the protection system	$>v_r$+2m/s	

It should be noted that for transient load cases 2.1-2.5 ideally the measurements should be taken at v_{out}, following the recommendations of IEC/TS 61400-13.

7.2 Quantities to be measured for the yaw system

Specifically for the yaw system and its components, **in addition to** the quantities specified within IEC/TS 61400-13 [13] as mandatory, the quantities classified into load quantities and operational parameters shown in following table should be considered.

[4] As per IEC/TS 61400-13
[5] Target wind speed as per IEC/TS 61400-13
[6] Has to be further divided into wind speed bins and turbulence bins

Table 7-2: *Quantities to be measured for the yaw system*

Quantity	Specification	Comments
Yaw system loads	Bearing Bending in two (perpendicular) directions	Mandatory; measured at the tower top
	Bearing Torsion	Mandatory; measured at the tower top
	Bearing Axial force	Mandatory; measured at the tower top
	Bearing Radial force in two (perpendicular) directions	Recommended; measured at the tower top
	Gear Torque	Recommended; measured at the pinion shaft or the input shaft of the yaw transmission system
Yaw actuator status	Power consumption of yaw actuator	Recommended
Local temperature	Temperature on yaw system bearings and frictional parts	Recommended

It is noted that the yaw position (related to the kinematics of the yaw system) as well as the wind inflow are considered as mandatory within the IEC/TS 61400-13[13].

7.3 Measurement techniques

Following the IEC/TS 61400-13 in this sub-section the measurement techniques for the various types of quantities in load measurement programmes regarding the yaw system are described in terms of instrumentation, calibration and signal conditioning. Furthermore, recommendations for data acquisition methods relevant to the load measurement programmes specifically intended for the yaw system will be provided.

7.3.1 Yaw bearing bending moments

The yaw bearing bending moments are measured at the tower top. Requirements for the types of sensors used, the selection of their location and their calibration are the same as for measuring bending moments on the tower top, described in IEC/TS 61400-13.

In summary, typically the measurement for the bending loads at the tower top performed in two perpendicular directions is performed using strain gauge bridges. Wire temperature effects and cross sensitivity should be avoided and proper temperature compensation should be ensured by selecting an appropriate full strain-gauge bridge design. A full T Type strain gauge bridge employing T strain gauge rosettes with two measuring grids perpendicular to each other at 0° and 90° would be therefore appropriate, following the recommendations of [15] and [16]. To the extend possible the strain gauge bridges should be applied at a location within a region of uniform stress, avoiding localized stress concentrations. The location selected on a material having uniform properties should be preferred.

For the calibration of the sensor (the full strain gauge bridge) in order to determine the sensor sensitivity it is preferred to apply quasi-static calibration loads on the wind turbine as described in IEC/TS 61400-13[13]. For the case of tower top bending moments the mass of the nacelle and the rotor combined with their centre of gravity, while yawing could be also used for the calibration, a procedure that would increase, nevertheless the load measurement uncertainty. Electrical (shunt) calibration could be also performed for the calibration as described within IEC/TS 61400-13. The load measurement uncertainty as a result of the various calibration methods will be discussed in a following section.

Calibration checks for the tower top bending moments can be performed by yawing the turbine through 360° as described within IEC/TS 61400-13.

For the yaw system bearing the signal that is of interest is actually the RMS signal of the two perpendicularly measured bending moments on the tower top.

7.3.2 Yaw bearing Torsion

The yaw bearing torsion is measured at the tower top. Requirements for the type of sensor, the selection of its location and its calibration are the same as for measuring the torsion on the tower top, described in IEC/TS 61400-13.

In summary, typically the measurement for the torsion at the tower top is performed using a strain gauge bridge. Wire temperature effects and cross sensitivity should be avoided and proper temperature compensation should be ensured by selecting an appropriate full strain-gauge bridge design. A full V Type strain gauge bridge employing shear/torsion strain gauge rosettes with two measuring grids perpendicular to each other, yet at ±45° with respect to the measuring (tower) axis would be therefore appropriate, following the recommendations of [15] and [16]. The strain gauge bridge should be applied at a location within a region of uniform stress, avoiding localized stress concentrations. The location selected on a material having uniform properties should be preferred.

For the calibration of the sensor (the full strain gauge bridge) in order to determine the sensor sensitivity it is preferred to apply quasi-static calibration loads on the wind turbine as described in IEC/TS 61400-13 [13]. Electrical (shunt) calibration could be also performed for the calibration as described within IEC/TS 61400-13. The load measurement uncertainty as a result of the two calibration methods will be discussed in the relevant section.

Calibration checks should be performed for identification of possible secondary effects (e.g. bending moment cross-talk).

Two cases should be distinguished when measuring the tower top torsion: 1) the case with the yaw system holding nacelle position constant (i.e. not yawing) and 2) the case with the yaw system in operation. The first case is referring to loads acting through the yaw bearing and the brakes of the yaw system. The latter refers to the yaw gear meshing torques. However, the treatment of these two cases depends on the configuration of the specific wind turbine (independent yaw brakes or yaw motor with internal brakes).

7.3.3 Yaw bearing axial force

The axial force acting on the bearing is measured on the tower top.

Typically the measurement for the axial force at the tower top can be performed using a strain gauge bridge. Wire temperature effects and cross sensitivity should be avoided and proper temperature compensation should be ensured by selecting an appropriate full strain-gauge bridge design. A full T Type strain gauge bridge employing strain gauge rosettes with two measuring grids perpendicular to each other at 0° and 90° with respect to the measuring (tower) axis would be appropriate, following the configuration recommended in [17]. The strain gauge bridge should be applied at a location within a region of uniform stress, avoiding localized stress concentrations. The location selected on a material having uniform properties should be preferred.

For the calibration of the sensor (the full strain gauge bridge) in order to determine the sensor sensitivity it is preferred to apply quasi-static calibration loads on the wind turbine as described in IEC/TS 61400-13 [13]. However, the applied loading for this method is expected to be of limited magnitude. Therefore, electrical (shunt) calibration should be performed for the calibration as described within IEC/TS 61400-13. The load measurement uncertainty as a result of the two calibration methods will be discussed in the relevant section.

7.3.4 Yaw bearing radial forces

The bearing radial force acting on the bearing is measured through measuring the shear forces on the tower top.

Typically the measurement for the shear loads near the tower top performed in two perpendicular directions is performed using strain gauge bridges. Wire temperature effects and cross sensitivity should be avoided and proper temperature compensation should be ensured by selecting an appropriate full strain-gauge bridge design. Two full V Type strain gauge bridges employing shear/torsion strain gauge rosettes with two measuring grids perpendicular to each

other, yet at ±45° with respect to the measuring (tower) axis would be therefore appropriate, following the configuration recommended in [17]. The strain gauge bridges should be applied at a location within a region of uniform stress, avoiding localized stress concentrations. The location selected on a material having uniform properties should be preferred.
For the calibration of the sensors (the full strain gauge bridges) in order to determine the sensor sensitivity it is preferred to apply quasi-static calibration loads on the wind turbine as described in IEC/TS 61400-13 [13]. However, the applied loading for this method is expected to be of limited magnitude. Therefore, electrical (shunt) calibration should be performed for the calibration as described within IEC/TS 61400-13. The load measurement uncertainty as a result of the two calibration methods will be discussed in the relevant section.
For the yaw system bearing the signal that is of interest is actually the RMS signal of the two perpendicularly measured shear forces on the tower top.

7.3.5 Yaw actuator loads

For the yaw actuator loads there are several options. These can be:

- calculated through the yaw gear loads (meshing torque)
- estimated through measurement of the electrical power consumption of the actuator
- measured through application of an appropriate torque measuring sensor in the yaw transmission system

In the first case measurements of the tower top torsion are required as described in Section 7.3.2. Then the yaw actuator loads can be estimated through:

$$M_{yD} = \frac{M_M}{i_y} + i_y J_{yD} \ddot{\alpha}_y$$

where M_M is the torsion moment measured at the tower top, α_y the yaw angle, i_y is the gear ratio of the entire yaw system (including the gear ratio of the yaw gearbox and the gear ratio of the yaw bearing and J_{yD} the inertia of the yaw driver and the yaw transmission system (as one system).
In case the electrical power consumption of the actuator is measured, then the yaw actuator loads can be estimated through use of the yaw system efficiency as:

$$M_{yD} = \frac{\eta P_{el} 60}{2\pi n_{rot}}$$

where η is the total efficiency of the electrical yaw motor system, P_{el} is the electrical power and n_{rot} is the revolution speed of electrical drive (in rpm).
The measurement of the torque directly requires an appropriate torque sensor. These could be either a strain gauge bridge measuring torque mounted on the shaft or a torque transducer directly mounted on the shaft. In the first case the available dimensions of the shaft, as well as the shaft diameter could limit the application. In the latter the direct mounting of the transducer on the drive train of the yaw system would require an intervention on the yaw system.

7.3.6 Yaw bearing local temperature

Since it is anticipated that temperature will have an effect on the kinematics and therefore also the loads of the yaw system during operation it is recommended to measure the temperature of yaw base and frictional parts. To this end, temperature sensors should be used and applied in relevant locations on the yaw system bearing.

7.3.7 Data acquisition

For the yaw system loads monitoring the same requirements as those set for the tower bending moments following IEC/TS 61400-13 should be applied.

7.3.8 Sensor uncertainty & resolution

Recommendations outlined in IEC/TS 61400-13 should be followed wherever possible to assure that the load measurement uncertainty is kept under the set target limit of 3%. The procedure, which should be followed for the evaluation of the uncertainty in load measurements, is described in IEC/TS 61400-13 [13]. The differences between electrical and mechanical calibration have been discussed in [18]. There the performance of mechanical calibrations is favoured against electrical calibrations, due to the fact that there is an increase in uncertainty when performing electrical calibrations.

Sensor accuracy specifically for the yaw system regarding nacelle position should be at least 3° (better than that recommended in IEC/TS 61400-13). This accuracy is governed by the equipment (compass) used to perform the signal calibration of the nacelle position. The resolution of the nacelle position measurement should be better than 1°. Typically used proximity sensors based equipment might not be adequate for measurements regarding the loads on the yaw system. A rotary encoder may be therefore preferred.

7.4 Processing of measured data

In general the recommendations described within IEC/TS 61400-13 should be followed. Specifically for the analysis aimed at the yaw system the measurements identified as affected due to obstacle shadowing should be marked adequately. Their statistical analysis however, could be used to have a more complete picture of the yaw system operation and loading.

7.4.1 Time series and load statistics

Plotting of measured and calculated load time series should be performed following IEC/TS 61400-13. The reporting should include the loads at the interfaces of the yaw system as described in the relevant sections:

- Yaw bearing bending moment
- Yaw bearing axial force
- Yaw bearing radial force
- Yaw torsion (distinguishing between yaw operation status)
- Yaw actuator loads

The statistical information of all measured and calculated loads should also follow directions described within IEC/TS 61400-13.

Regarding the meteorological quantities statistics, a more elaborate analysis is advised for the characterisation of the site with respect to the wind conditions, including not only a presentation of the mean wind direction with respect to the wind speed, but also a statistical analysis regarding the direction changes with respect to the mean wind direction and the wind speed. For this a 3-D plot could be used, with horizontal axis the mean wind speed and the wind direction and the vertical the wind direction change (average and if possible min-max).

Variation of measured loads versus nacelle position, including mean, standard deviation, maximum and minimum values could be shown. But it is realized that parts of the graph would be either noted as affected by obstacle shadowing, or would be not captured at all during the measurement campaign, depending on the site conditions with respect to wind inflow. Nevertheless for the loads relevant to the yaw system this could reveal trends that might be useful to identify specific asymmetries of the wind turbine.

Regarding the load measurements for the yaw system it is recommended to perform a statistical analysis with respect to the wind inflow conditions including the number of yaw actuator starts (within the 10-minute captured file), duration of operation and amplitude of yaw movement (average, minimum, maximum and standard deviation). If the yaw system is designed to operate under a constant speed, as for most of the wind turbines, then only one of the two parameters (duration of operation or amplitude of yaw movement) needs to be presented. It should be noted

that the number of starts should be normalized over the whole data set of captured wind condition.
As an example following table could be used:

Table 7-3 *Example of Yaw operation statistical analysis presentation*

Wind speed (m/s)	Starts	Duration of operation (s) or Yaw movement (°)			
		Average	St. Deviation	Maximum	Minimum
0-3.5	=Starts/(total 10-min files in bin)				
3.5-4.5					
…					

If a more elaborate analysis is required for the wind inflow conditions, then each wind speed bin could be further analysed in turbulence bins and the data can be classified according to the turbulence intensity.
For the frequency analysis presentation of results should follow the description of IEC/TS 61400-13. However, specifically for the parameters monitored for the yaw system the frequency analysis should be performed for cases involving yaw operation and cases not involving yaw operation at similar wind inflow conditions. This should be conducted for identifying possible changes in the frequency spectrum due to the operation of the yaw system.

7.4.2 Load spectra

For the estimation of load spectra the description of IEC/TS 61400-13 should be followed. Yet yaw specific loads should be presented together with the loads required through the IEC/TS 61400-13.

8. 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). 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 drive train, pitch system and yaw system

The six steps approach constitutes for the drive train a very practical process which can be integrated into the design process flow chart. It enables the designer to focus on the component, taking into account the relevant boundary conditions (identifying the failure cases and the critical design load cases). Investigations aiming at determining any required precision on the input parameters must be led as a part of a sensitivity analysis, unfortunately it is not possible to make any general statement from the current case study. The iteration loops considered in the approach between the design of the model and the field testing enables reaching an optimised combination between a detailed model and realistic measurement procedures. The tuning of the model parameters, using the measurements' results, will however be done in the Drive Train Case Study, in workpackage 5.

For the pitch system the six steps approach has resulted 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. This work will be done in workpackage 6 of the PROTEST project.

For the yaw system, the work within the PROTEST project resulted in suggesting improvements both for the "generalized" load components measurements and the analysis techniques, specifically targeted at the yaw system components. In particular, by introducing load measuring sensors in the yaw actuator system (e.g. actuator torque or electrical power) in combination with the usually measured tower load components (tower top bending and torsion), estimations for the loading of the various yaw system parts is possible, including friction effects. These measurements combined with proper analysis of the statistics for the operation of the yaw system (e.g. starts, stops and duration per wind speed) can be effectively used in the design of the yaw system components (bearing, actuator, etc.). Issues of accuracy and uncertainty will be further addressed and refined within WP7 of the PROTEST project.

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Appendix A Questionnaire

In this questionnaire you are asked to provide information (as far as possible) concerning:

- the operational experience available with measurements on drive train, pitch system and yaw system
- view on measurements that are required (both need to have and nice to have)

For this purpose a number of tables have to be filled out. The information required is explained in *italic text*, and sometimes an example is given just for illustration. The italic text inside the tables can be skipped or deleted when filling out these tables.

A.1 Operational experience

In case you have experience with load measurements on drive train, pitch system or yaw system please fill out the table below.

Table: existing load measurements

Name of (sub)system or component			
Quantity *(f.i. drive train loading or gearbox displacement)*	Specification *(please indicate what is measured by which sensor at what location)*	Objective *(Please describe the objective f the measurement, f.i frequency measurement to tune model or measurement for validation)*	Comments *(additional information, f.i. is possible how data is processed)*

Please copy above table to fill out information for another (sub)system or component.

A.2 Required measurements

Please indicate which kind of load measurements should be carried out to your opinion for drive train, pitch system or yaw system and for what reason. Please fill out the table below.

Table: required load measurements

Name of (sub)system or component			
Quantity *(f.i. drive train loading or gearbox displacement)*	Specification *(if possible please indicate how it should be measured, which sensor at what location, etc.)*	Objective *(Please describe the objective of the measurement, f.i frequency measurement to tune model or measurement for validation of specific simulation model)*	Comments *(additional information, f.i.whether the measurement is needed or nice to have)*

Please copy above table to fill out information for another (sub)system or component.

A straightforward approach may be that the above mentioned measurements are carried out for the same internal and operational loadings (MLCs) as specified in IEC 61400-13. However it may occur that the loadings considered by the MLCs in IEC 61400-13 are not sufficient and additional internal or operational loadings should be considered for the drive train, pitch system or yaw system. F.i. for validation purposes of simulation models applied to analyse specific conditions.

Please indicate in the table below which additional internal or operational loadings should be included in addition to the MLCs specified in IEC 61400-13 (see section 2..2).

Table: required internal and external loadings

Name of (sub)system or component			
Operational mode of turbine *(f.i. power production or idling)*	Additional requirements w.r.t. internal loading *(f.i. a specific faulted condition)*	Wind conditions *(indication of range of wind speeds for which measurements should be carried out)*	Remarks

Please copy above table to fill out information for another (sub)system or component.

Appendix B Definition of positive measurement directions

A definition of positive measurement directions is given in Figure B-1 below:

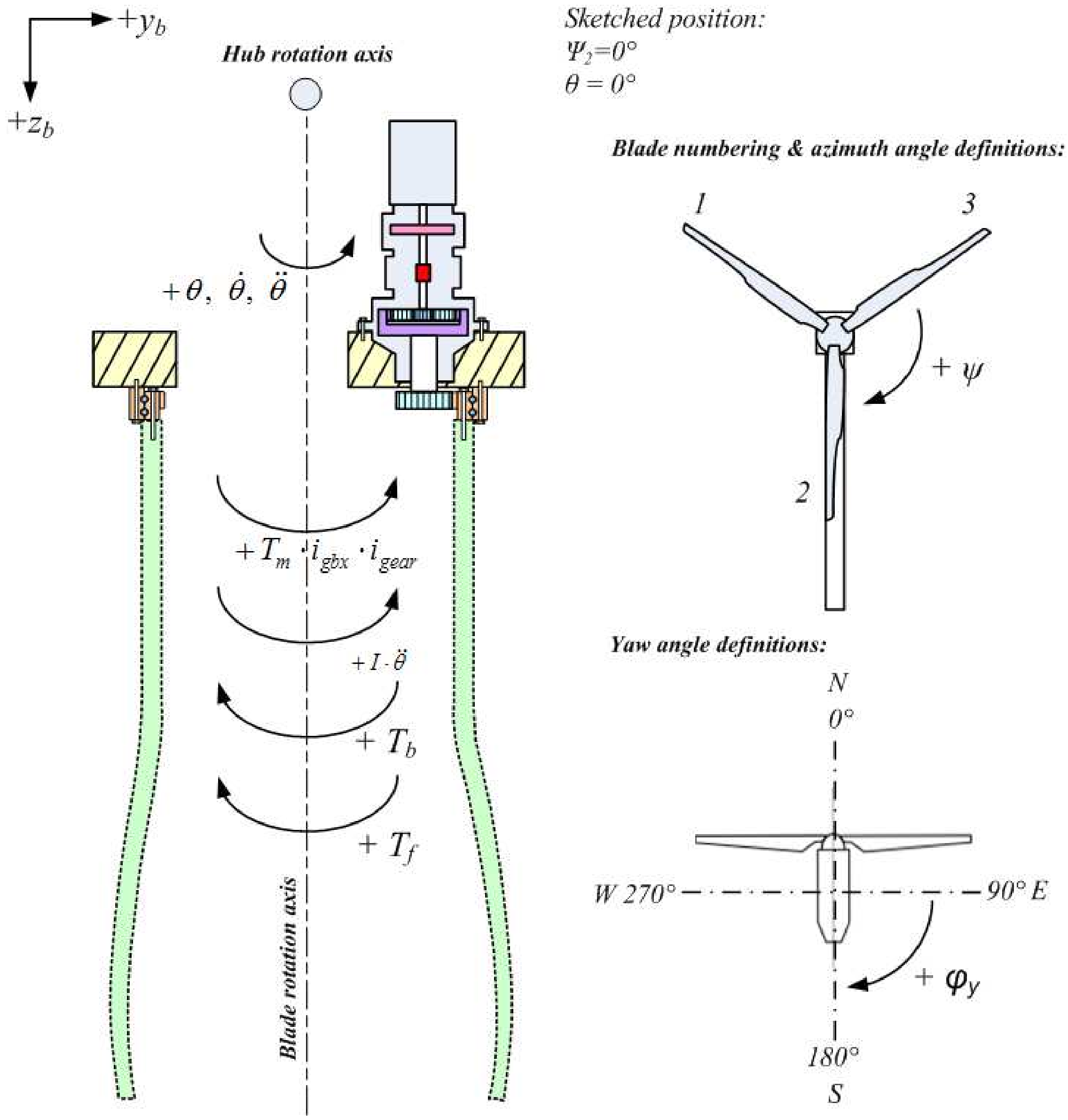

Figure B-1: *Definition of positive measurement directions for pitch system*