PROTEST

Suggestions for Improved Standards

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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 pre-normative PROTEST project looked at possibilities to improve the current standards by using wind turbine prototype measurements to validate the drive train, pith system and yaw system models. The main conclusions from this project that could be valuable for the standards committees are summarised in this report. These recommendations include new Design Load Cases, measurement procedures for model validation and load measurements at specific interconnection points.

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Terms and Definitions

DLC	Design Load Case; the combination of operational modes or other design situations, such as specific assembly, erection or maintenance conditions, with the external conditions [7].
Design Load	The load for which the strength of any component has to be documented. It generally consists of the so-called characteristic load multiplied with the appropriate partial safety factors for loads and consequence of failure, see also IEC 61400-1 and clause 6 [6].
Limit State	The state of a structure and the loads acting upon it, beyond which the structure no longer satisfies the design requirement [ISO 2394, modified] (*NOTE The purpose of design calculations (i.e. the design requirement for the limit state) is to keep the probability of a limit state being reached below a certain value prescribed for the type of structure in question (see ISO 2394).)* [7].
CDV	Critical Design Variable; a design variable that from experience is expected to strongly affect the design.
Failure Mode	The mode of failure. Passing over a specific limit state described by a single equation could lead to different failure modes depending on the vector followed when passing from the safe state to the failure state.

1. Introduction

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 (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 Annex B of [6].

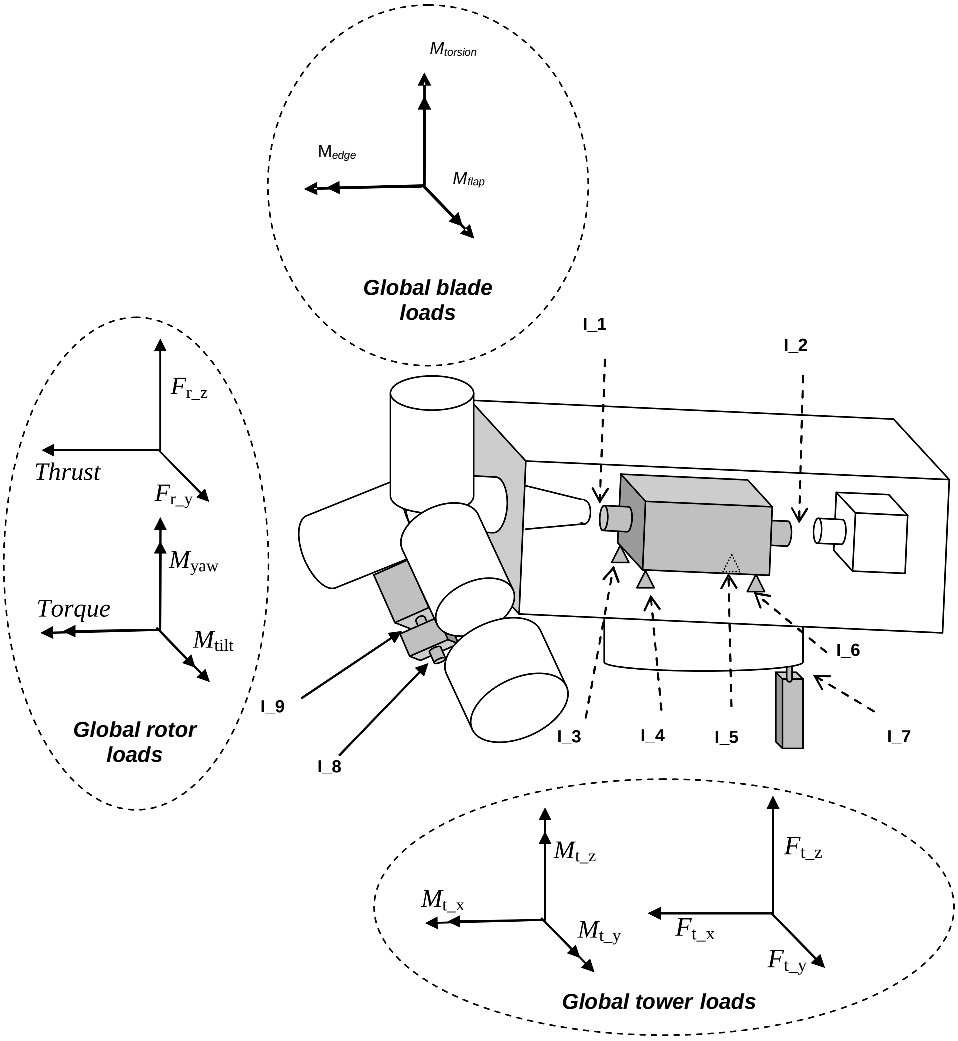

Figure 1-1: *Schematic presentation of transforming "global turbine loads" to "local components loads" at nine interfaces, (gearbox, pitch system and yaw system)*

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 components 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.

Answers to the following questions were sought:
- 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 was foreseen. The analytical work was 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 was needed to develop and verify new procedures for prototype measurements. The overall work was split in total in nine work packages.

1. State of the art report: An inventory has been taken 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: Including the determination of load cases and design driving factors (external, operational or design inherent) that should be considered for the selected components.

3. Loads at interfaces: Comprising the specification of 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.) for the selected components.

4. Prototype measurements definition: For each component, a recommended measurement campaign was 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 was planned for the three components involved in the project. This work was 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 were applied. The initial design loads at the interfaces were determined with state-of-the-art design methods and the measurement campaign was 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 ran from March 2008 until August 2010.

1.2 Scope of the report

In this report the main results of the PROTEST project that are relevant as recommendations for the standards are discussed. First suggestions that concern design recommendations are given, specifically for IEC 61400-1 first, followed by those for IEC 61400-4 and those concerning pitch system and yaw system respectively. Finally suggestions concerning the measurements, IEC 61400-13, are given.

2. Design recommendations

In this section, the suggestions that concern IEC 61400-1 are give first, followed by suggestions for IEC 61400-4. After this the implementation of the project results concerning pitch system and yaw system into practical certification procedures will be discussed for the single components of these systems.

2.1 Suggestions concerning IEC 61400-1

2.1.1 Proposal for new DLCs

For the design of the gearbox and the drive train the following three causes for higher loads should possibly be taken into account in the guidelines prescribed by IEC-61400-1 or the GL guidelines.

Three short comings in the current procedure to validate the design of wind turbine drive trains have been identified. These three short comings can result in significant differences between the results of the analysis of the wind turbine model and the real turbine, therefore for the design of the gearbox and the drive train the following three causes for higher loads should possibly be taken into account in the guidelines prescribed by IEC-61400-1 or the GL guidelines.

- DLC – Misalignment

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

  However, no clear information is available about the magnitude of drive train misalignment and deflection of supporting structures in practical situations. Therefore it is advised that the aspect of drive train misalignment and reactive forces should be discussed between the wind turbine designers, gearbox suppliers, coupling suppliers and bearing suppliers, with the aim to specify a target value for drive train misalignment that should be used in the design for the gearbox and other drive train components.

  A complicating factor to analyse the effect of drive train misalignment is that in the traditional wind turbine simulation tools in general a simplified model is used for the drive train with only a limited number of degrees of freedom, so that misalignment cannot be analysed with these models. This implies that for the design of the gearbox the constraining forces have to be specified on the interfaces. The consequences of specifying loads at the interfaces are discussed in the next section.

- DLC – Resonance

  The drive train consists of a number of subassemblies which together form a dynamic system. The intersection of the systems natural frequencies and excitation frequencies may lead to load increasing resonances that will affect the main drive train components. In order to identify and investigate resonances a resonance analysis has to be performed. Depending on the phenomena to be analysed and the frequency ranges, different models and tools with varying levels of complexity can be used. Depending on the excitation mechanisms different frequencies ranges needs to be analysed, e.g. [0 – 5 Hz] [5 – 50 Hz], [50 – 200 Hz], [200-500 Hz], [500-2000Hz].

The dynamic behaviour of the drive train depends mainly on the mass, inertia and stiffness properties of the components in the drive train. Varying drive train configurations might cause variations of these properties. Hence, a new analysis of the drive train dynamics is necessary if different types of the following components are installed in the same type of wind turbine:

- rotor blades
- main shaft
- gearbox
- elastic gearbox and generator supports
- generator coupling
- generator
- type of main or gearbox bearings

A sensitivity analysis can be carried out in order to identify the contribution of individual components to the overall dynamic behaviour of the drive train. As a result, it might be possible to reduce the number of combinations to be investigated by separate resonance analyses of the drive train.

Results of the analysis are Campbell diagrams showing natural frequencies related to excitations. The investigation of the natural frequencies shall include an analysis of the energy distribution for each mode shape. In the case that the evaluation of these results shows potential resonances, more detailed investigations need to be carried out by the simulation of a e.g. a rpm-sweep that covers the operating speed range of the wind turbine. The results are to be evaluated with regard to the increase of local component loads and the load-carrying capacity of the components.

The analysis requires a linearised model for determining natural frequencies and mode shapes. In the case of non-linear simulation models, an adequate number of linearization states shall be considered.

More on the implementation of the resonance analysis can be found in Appendix A of this report.

- DLC –LVRT

  Fault or loss of the electrical network connection is included in DLCs 2.3 and 2.4, however in practise the tools are not yet good enough to completely analyse these DLCs. The LVRT should be described in more detail, many different shapes of the low voltage can be specified and have different effects on the turbine. The different grid codes that exist in different countries further complicate this DLC. This combination deems it impossible to prescribe the LVRT DLCs in detail. It is clear that it will also be very hard to find the most critical cases for a specific turbine. Combined with wind speed, a detailed approach of LVRT can result in a large number of DLCs to be analysed. The details of this process can therefore not be specified during this project. The LVRT DLCs are however of significant importance for both fatigue and ultimate strength.

Strictly speaking especially the first two cases are **not** new DLCs; a maximum misalignment should be taken into account in the analysis and the real misalignment shall not exceed the assumed maximum. A violation of the tolerance criterion cannot be accounted for in the DLC's, it must be assumed that a turbine is constructed according to the requirement specification. Resonance should show up during the analysis.

Since the currently used tools are not capable of taking into account the aspects of drive train misalignment, reactive forces and resonance, other means of analyses need to be implemented. In a first step it can be assumed that the additional loads originating from misalignment of the drive train and resonances in the drive train will only marginally affect the overall system response of the wind turbine. This means that the global loads at the current interconnection points will not shift significantly whereas the local component loads will be notably affected. From this assumption it appears acceptable to investigate the a.m. load cases by analysing the drive train and the supporting structure separated from the remaining wind turbine.

For this purpose a detailed calculation model of the drive train and adjacent components needs to be implemented and analysed. For the investigation of resonance phenomena the natural frequencies and excitation frequencies of the system needs to be evaluated. For the load case “misalignment” calculations in the time domain are necessary. Here, design loads obtained from the currently used simulation software will be applied to the detailed calculation model and the influence of misalignment and reactive forces can be investigated in order to obtain realistic component loads.

Certain transient events as well as DLCs of normal operating condition should be analysed by this means.

In a second step, once more insight and experience with complex models is gained, the entire wind turbine may be analysed with such models.

However, many of the currently used tools do not take the misalignment into account and the resonance could occur for frequencies that are much higher than can be analysed or are practically feasible in current tools. They could also not show up due to the limitations of the models used. Therefore it seems appropriate to specify new DLCs and/or introduce new analysis procedures as mentioned above where these aspects are taken into account, but not put the same demands on all simulations of existing DLCs. Loss or faults of the electrical network are already described in the DLCs, however this process is at this moment too complex for the state of the art tools to enable detailed enough analysis. Also a lot of different possible cases could be defined for LVRT and the most critical cases are not easily determined, they can even depend on the country due to the different grid codes in place.

2.2 Treatment of existing DLCs

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

- insufficient modelling of the drive train deformation and of the deflections of gearbox interior components making some underlying assumptions for the calculations invalid (load transmission lines in the gears may change to load transmission points under angular displacement of gears);
- insufficient modelling of the systems by omitting flexibilities and deformations in the wind turbine main frame that lead to constraining forces and moments at the system joints to the adjacent structural system/component.

Therefore the suppliers generally use special purpose tools for the design of their components and for this purpose the relevant loads acting at the interfaces have to be made available by the wind turbine manufacturer. For this reason a set of relevant loads has to be specified at the interfaces, which can be used for the design by the component supplier. For the calculation of this load set traditional wind turbine simulation tools are used.

For the specification of a set of relevant loads acting at the interfaces the following aspects have to be addressed.

- Definition of load set at interface

  The DLCs relevant for the design of the structural component have to be selected. One of the objectives of the PROTEST project is to provide guidelines for this selection, where the input from wind turbine designers and the designer of the structural components should be leading. As the case studies defined in the work packages 5 – 7 comprised measurements and analyses on existing wind turbines, the feedback from these case studies has been used to set up a proposal for such a guideline.

- Modelling of structural component in simulation tool

  To determine the relevant loads at the interfaces, all relevant DLCs should be simulated with the traditional wind simulation tools, where the modelling of the structural components is done with sufficient detail. As a part of the case study on the Suzlon S82 wind turbine, analysis has shown that the consideration of additional torsional degrees of freedom did not have any significant influence on the main shaft and generator torques during normal production load cases. Load cases experiencing higher transients (such as emergency stop or LVRT) should require a more detailed model. It is however not straightforward to draw conclusions for turbines of different classes or with others kinds of drive train concepts. Future standards should define what level of details need to be implemented for the modelling the drive train, depending on the load case and possibly drive train concept. Procedures for determining judicious degrees of freedom are defined in PROTEST's final report D19 and the final report of WP 5, always starting by the most complex model possible and suppressing the insignificant degrees of freedom.

- Determination of loads at interfaces

  Once the relevant DLCs have been analysed in sufficient detail with a traditional wind turbine simulation tool the loads acting at the interfaces have to be determined. A procedure has been developed in work package 3 of the PROTEST project, see section 3.2.

2.3 Suggestions concerning IEC 61400-4

Currently there are no DLCs in 61400-4, therefore a suggestion for new DLCs, even though they do concern the drive train, have been suggested in section 2.1.1. The implementation of the resonance analysis can be found in Appendix A of this report.

2.4 Pitch System (general)

Since the validity of all subsequent listed calculations depends on sufficient lubrication of the components, an automatic lubrication system shall be mandatory for the blade bearings and blade bearing gears of wind turbines.

2.4.1 Blade Bearing

2.4.1.1 Calculation of Friction and Bearing Deformation:

Due to the various different bearing, hub and blade designs in industry, which are strongly interfering with the friction and response of the blade bearings, it is not recommended to use a generalized model to calculate these effects. More appropriate is the use of a simplified FEM model (hub, bearing, blade root) to calculate the overall load distribution on the bearings contact elements and to derive by these results and subsequent well-known analytical methods a load dependent (axial and radial forces, bending torque at blade root) bearing friction. The common commercial load calculation software tools are already capable to implement load dependent friction torques for the blade bearings.

2.4.1.2 Proof of static and fatigue strength of the blade bearings:

The most common standard to calculate the fatigue strength of roller or ball bearings is given by the ISO 281 standard. As shown by the measurement results generated within this project, the blade bearings are predominantly loaded during standstill and not during rotation, which is not in the scope of ISO 281. Applying ISO 281 for the fatigue calculations may therefore show misleading, non-conservative results regarding the lifetime of the bearings. In addition, blade bearings are not trough hardened as anticipated in ISO 281 but only surface hardened and have very big ball sizes compared to common rotating bearings.
The typical damage for these kinds of surface hardened, mostly non rotating bearings is known as "core-crushing", a crack initiation below the raceway tracks at the transition between hardened surface and tempered core, consequentially the analysis of this damage mechanism is recommended to calculate the fatigue strength of the blade bearings. For this kind of fatigue calculation the same simplified FEM-model as described above can be used to determine the contact inside the bearing with the highest loading. The contact forces generated by the production load cases (DLC 1.0 to

DLC 1.13) and derived for the highest loaded contact are used subsequently to calculate the fatigue strength depending on specified hardening depth/gradient by common analytical methods.
The standard ISO 76 is only appropriate to calculate the static strength of through-hardened bearings by calculation of the Hertzian contact pressure. This calculation may not be sufficient for the surface hardened blade bearing raceways, so a similar approach as described above for the fatigue calculation is recommended using the ultimate loads on the bearing as input data and the surface hardness and subsurface hardness distribution/hardening depth of the bearings as limiting material properties.

2.4.2 Blade Pitch Drive

2.4.2.1 Blade Gear

The blade gear is not equally loaded on all teeth of the gear, actually only a quarter (~90°) of the bearing's teeth comes in contact with the pinion teeth of the pitch drive during turbine operation. As shown by the measurements, the load on the teeth is not uniformly distributed between 0° and 90° blade position. In fact the teeth between 0° and ~30° (depending on turbine controller settings) have to bear higher fatigue loads than the rest of the teeth. Most relevant load cases for the fatigue calculations of the operating blade gear are the production load cases with wind speeds higher than rated wind speed of the turbine and lower than cut-out wind speed.
Therefore a valid fatigue calculation according to ISO 6336 for the blade gear has to implement cycle numbers that are calculated on a teeth-by-teeth basis. The easiest approach to achieve these results is to summarize the load cycle numbers for discrete pitch angles and the associated teeth meshes.
For the calculation of the blade gear the load dependent bearing friction (see 2.4.1.1), blade inertia and pitch drive inertia have to be included in the calculations. The common commercial load calculation software tools for wind turbines are already capable to implement these parameters in the calculations.
For the static gear calculation (ultimate loading) see 2.4.2.2

2.4.2.2 Pitch Gearbox

For the pinion and the output stage of the pitch gearbox it has to be verified whether a uniform load cycle distribution is achieved or not on all teeth of the pinion and the output planetary stage of the gearbox. If a non-uniform load distribution is to be expected, this has to be taken into account for the fatigue calculations of these gear meshes. All friction losses inside the pitch gearbox as well as all inertia and friction loads as mentioned in A2.1 must of course also be taken into account for the strength calculation of the pitch gear box meshes.
The static strength calculation for the pitch gearbox as well as for the blade gear have to be performed with the dynamic sliding torque of the pitch motor brake, as this is in common designs the highest possible torque to be applied. Nevertheless it has to be checked that this torque is not outpaced by the maximum pitch motor torque or the maximum toque resulting from the aerodynamic forces on the blade (ultimate load cases).

2.4.2.3 Pitch Motor

The basis for the thermal rating of the pitch motor shall be derived from the torque-time simulation run with the highest torque rms value. The averaging time shall be 600 seconds, or the overall time of the simulation run to be used.

2.5 Yaw System (General)

As well as in the Pitch System, the validity of all subsequent listed calculations depends on sufficient lubrication of the components. Therefore an automatic lubrication system shall be mandatory for the yaw bearing and yaw bearing gears of wind turbines.

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2.6.1 Yaw Bearing

2.6.1.1 Calculation of Friction and Bearing Deformation:

Similar to the blade bearings the use of a generalized model to calculate the effects of deformation and friction in yaw bearings is complicated by the various different bearing-, mainframe- and tower designs used in the industry, which are strongly interfering with the friction and deformation of the yaw bearings. Since the connecting structures on the yaw bearings have considerably higher stiffnesses than the connecting structures in blade bearings, the effect of deformation on the loading of the bearing is lower than in the blade bearings. Nevertheless, to derive suitable parameters for the load dependent (axial and radial forces, resulting bending moment at tower top) friction of the yaw bearing it is recommended to use a simplified FEM model (mainframe, bearing, tower top) to calculate the overall load distribution on the bearings contact elements. The common commercial load calculation software tools are already capable to implement these parameters for load dependent yaw bearing friction. A simplified FEM model as described above is anyway necessary to generate the required input data for the static and fatigue proof of strength for the bearing raceways (see section 2.6.1.2).

2.6.1.2 Proof of static and fatigue strength of the yaw bearing:

As already described in section 2.4.1.2 for the blade bearing, the yaw bearing is also predominantly loaded during standstill and not during rotation. This is not in the scope of ISO 281, the most common standard to calculate the fatigue strength of roller or ball bearings. Applying ISO 281 for the fatigue calculations of yaw bearing may therefore show misleading, non-conservative results. In addition, yaw bearings are not through hardened as anticipated in ISO 281 but are only surface hardened and have very big ball sizes compared to common rotating bearings.
The typical damage for these kinds of surface hardened, mostly non rotating bearings is known as “core-crushing”, a crack initiation below the raceway tracks at the transition between hardened surface and tempered core. Consequentially the analysis of this damage mechanism is recommended to calculate the fatigue strength of the yaw bearings. For this purpose the same simplified FEM-model as described in Section B1.1 can be used to determine the contact inside the bearing with the highest loading. The contact forces generated by the production load cases (DLC 1.0 to DLC 1.13) and derived for the highest loaded contact are used subsequently to calculate the fatigue strength depending on specified hardening depth/gradient by common analytical methods.
This calculation method is also sufficient for the calculation of yaw bearings on wind turbine locations with a distinctive mean wind direction, since the complete set of accumulated load cycles for the production load cases of full wind turbine lifetime will be applied to the bearing contact with the highest loading.

As already stated in Section 2.4.1.2 the standard ISO 76 is only appropriate to calculate the static strength of through-hardened bearings, therefore application of ISO 76 on the surface hardened yaw bearing raceways may not be sufficient. A similar approach as described above for the fatigue calculation is recommended also for the static proof of strength, using the ultimate loads on the bearing as input data and the surface hardness and subsurface hardness distribution/hardening depth of the bearings as limiting material properties.

2.6.2 Yaw Drive and Yaw Gear

2.6.2.1 Yaw Gear

For the fatigue proof of strength of the yaw gear the load dependent bearing friction torque (see 2.4.1.1), nacelle inertia, yaw drive inertia and the gyroscopic torque around tower axis induced by the wind turbine rotor have to be included in the calculations.
The common commercial load calculation software tools for wind turbines are already capable to implement these parameters in the calculations, except for the gyroscopic torque of the rotor. This torque may be implemented as a constant value assuming a constant rpm value for the rotor during all yawing operations. To achieve sound results for the calculation of the operational time of the yaw drives during wind turbine lifetime yaw controller settings like sampling time, minimum yaw misalignment for yaw start-up and yawing speed have to taken into account.
To address possible transient loads from switching events in fatigue calculation of the yaw gears, a detailed analysis of the electrical setup of the yaw system may be necessary.
This is the case if the induction motors of the yaw drives are switched directly to the grid during start-up, causing typically transient torques of about 3 times the motor mean torque for duration of about 30 ms during each start-up of the yaw system. These transient torques have to be included in the LDD´s for the calculation of the yaw gear. Another occurrence of transient loads in the yaw gear/yaw drives may result from the stop procedure, if the brakes at the end of the yaw motors are engaged in the same moment when the motor itself is switched off. These transient loads have also to be implemented in the LDD´s for yaw gear calculation.
In case of yaw motors driven by a frequency inverter and application of a delay time between switching off the yaw motors and brake engagement, transient loads from switching events are negligible.

In comparison to the blade bearing the yaw bearing gear has a more uniform load cycle distribution on all teeth of the circumference, even at locations with a distinctive mean wind direction, since there are three or more yaw drives nearly symmetrically engaged around the circumference of the yaw bearing gear. A simple sensitivity check for critical wind turbine locations with distinctive mean wind directions may be performed by multiplying the load cycle numbers for the yaw gear teeth with the factor 2.

To address possible loadings on the yaw gear teeth during standstill of the yaw drives, an analysis on aerodynamically induced vibrations of the nacelle around tower axis has to be performed. These vibration torques have to be include in the LDD´s for the yaw gear calculation if they exceed the friction torque of the yaw bearing. To avoid fretting at the gear flanks under these conditions operation of the yaw system for lubrication purposes may be enforced by the yaw controller in adequate intervals.

If a separate yaw brake system is installed which operates independent from the yaw gear, any influence of vibration torques on the yaw gear is assumed as negligible.

For the static gear calculation (ultimate loading) see 2.6.2.2

2.6.2.2 Yaw Gearbox

All friction losses inside the yaw gearbox as well as all inertia and friction loads as mentioned in B2.1 must of course also be taken into account for the strength calculation of the yaw gear box meshes.
The static strength calculation for the yaw gearbox as well as for the yaw gear have to be performed with the maximum torque of the yaw motor brake, as this is in common designs the highest possible torque to be applied. Nevertheless it has to be checked that this torque is not outpaced by the maximum yaw motor torque, the maximum torque resulting from the aerodynamic forces or by transient torques occurring during yaw system operation (See 2.4.2.1).

2.6.2.3 Yaw Motor

The basis for the thermal rating of the yaw motor shall be derived from the torque-time simulation run with the highest torque rms value. The averaging time shall be 600 seconds, or the overall time of the simulation run to be used.

3. Test and Measurement Recommendations (IEC 61400-13)

3.1 Measurement procedures for model verification

There can be two different objectives for the measurement campaign. In the PROTEST project, and therefore also in the method that will be described here, 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 diverse components. The measurement campaign therefore has to be set-up such that these simulations can be verified.

When focussing on the three components; drive train, pitch system and yaw system, 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 was concluded that it becomes impossible to set strict standards. 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; these 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 [16]. 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: Set up the calculation model (simple analytical to e.g. multi body)
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, natural 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 3-1, it is possible to have one or more loops in the process. As illustrated in this figure, once the model is set up, 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 specific input parameters is too large. It is also possible that, after measuring and processing the data, the 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 redoing step 5. These are a few of the possibilities of going through the six step 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 step approach, it has been applied to the three different components in WP 5 (drive train), WP 6 (pitch system) and WP 7 (yaw system), see [15].

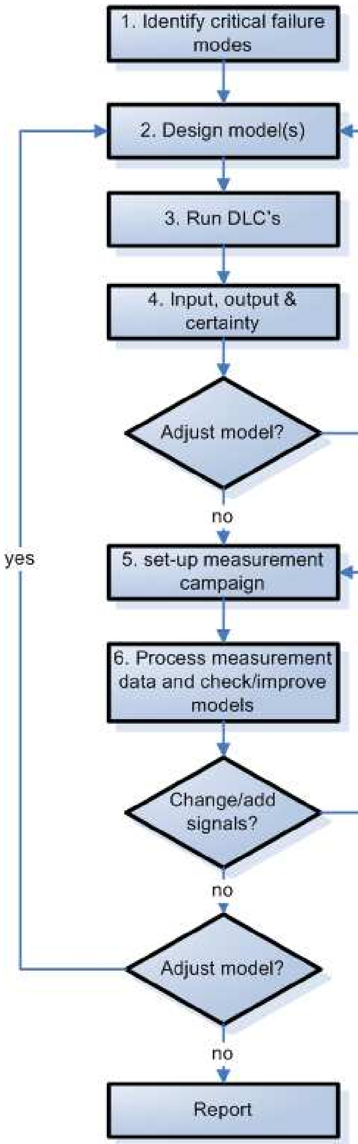

Figure 3-1: *Illustration of the six step approach.*

3.2 Load measurement at specific interconnection points

3.2.1 Drive train

3.2.1.1 Loads at interconnection points

To determine the procedure to describe how the loads at the interconnection points should be defined, the specification of the interfaces of the gearbox and the drive-train and its sub-components if necessary, is required. That includes isolation of each system or sub-component from the overall wind turbine structure and further building on the adequate description of the sectional loads at the interconnection points (interfaces) the overall wind turbine loads need to be transferred to design parameters. Within WP3 an assessment followed regarding which knowledge of loading (i.e. torques, bending moments, accelerations, motions, deformations etc.) is considered as a valuable improvement over the current state-of-the-art.

The results presented in [8] as well as the findings of work package 2 of the PROTEST project [14] regarding the design load cases for the gearbox and the drive train that should be considered, discussed in 2.1.1 of the present report, were further developed to define the procedure for determining the loads at the interfaces of the considered components. For the gearbox and the drive

train the determination of the necessary information at the interfaces for designing the mechanical components was based on IEC 61400-4 [6].
The details of the findings were reported within [11]. In here only a summary of the findings will be presented, regarding the loads at the interconnection points of the gearbox, the drive-train and specific components of the drive-train, such as the main shaft.
For the gearbox, IEC 61400-4 [6] identifies the interconnection points (interfaces), commonly applied in modern wind turbine designs. Depending on the arrangement of the wind turbine the following sketch shows the relevant interconnection points (interfaces) relevant to the gearbox only. In the configuration shown in Figure 3-2 it is supposed that the gearbox does not support other systems (i.e. that no additional systems are directly mounted on the gearbox). Accordingly, following interfaces can be identified for this configuration:

1. The low speed shaft to the gearbox (specifically the gearbox entrance stage)
2. The high speed shaft to the gearbox (specifically the gearbox output stage)
3. The nacelle main frame through the supporting positions of the gearbox to the gearbox (specifically the gearbox housing)
4. The mounting positions of the gearbox on the nacelle main frame via torque arms to the gearbox (specifically the gearbox housing)

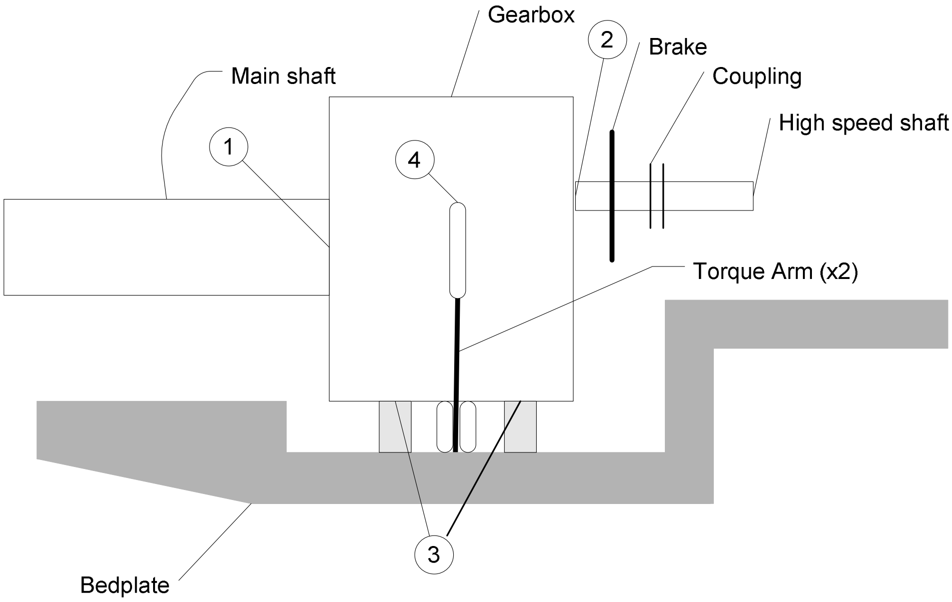

Figure 3-2: *Simplified sketch of gearbox layout without supported systems showing interfaces.*

Loads transferred across the gearbox system, depend on the configuration of the wind turbine. Therefore, detailed analysis would have to be based on detailed configurations. In an ideal situation the purpose of the gearbox would be to transmit the torque and the rotation (revolutions) of the rotor to the generator through the high speed shaft, counter-acting all other loads arriving at the gearbox from the rotor through the low speed part of the drive train. To this end, the torque arms of the gearbox are used to counter-act the torque reaction of the gearbox from the rotor. The forces and bending moments are either counter-acted through the main bearing(s) of the main shaft or (depending on the configuration) through bearings of the gearbox. Bending moments and torsion (torque) are usually measured on the main shaft during conventional load measurement campaigns (as specified in IEC/TS 61400-13). The force measurements, however, are not required and usually these measurements are not performed. The forces (and moments) on the main shaft can be estimated through aero-elastic simulations. But to obtain the forces and moments on the high speed shaft or the forces on the torque arms through aero-elastic simulation detailed information on the gearbox and the drive train is necessary.
Classifying the general loads transferred across the interfaces of the gearbox as loads, kinematics and dynamics, the following parameters should be defined.
Loads:

- Axial and shear loads, bending moments and torsion of the low speed shaft (at the gearbox interface).
- Axial and shear loads, bending moments and torsion of the high speed shaft (at the gearbox interface).
- Forces at the torque arms.

Kinematics:

- Position (including angle, rotational speed and axial displacement) of the low speed shaft (at the gearbox interface)
- Position (including angle, rotational speed and axial displacement) of the high speed shaft
- Displacement of the torque arms.

Dynamics:

- Accelerations

Furthermore, synchronization of the general loading conditions is required with the wind turbine operational parameters, such as status, rotor revolution speed, power production, azimuth position.
For the drive train, IEC 61400-4 [6] identifies the interconnection points (interfaces), depending on the wind turbine configuration, similar to the gearbox. As an example for the configuration using a modular drive train with a 3-point suspension, as shown in Figure 3-3, the following interfaces (interconnection points) can be identified:

1. The rotor hub to the drive train (on the low speed – main shaft)
2. The Main bearing of the drive train (on the low speed shaft) to the nacelle main frame
3. The torque arm on the gearbox to the nacelle main frame
4. The nacelle main frame to the support points of the gearbox of the drive train
5. The nacelle main frame to the support points of the generator of the drive train
6. The generator (on the high speed shaft) to the drive train – internal interface of the drive train
7. The mechanical brake to the drive train (on the high speed shaft) – internal interface of the drive train
8. The coupling on the high speed shaft of the drive train - internal interface of the drive train
9. Other (e.g. interfaces for lubrication systems, sensors) – Not shown in Figure 3-3

Other drive train configurations are discussed in [11].

Loads transferred across the drive train on specific interface points, depend on the configuration of the wind turbine. Similar to the case for the gearbox, detailed analysis of the loads transferred through each component of the drive train would have to be based on the specific configuration of the wind turbine. In an ideal situation the purpose of the drive train would be to transmit the torque and the rotation (revolutions) of the rotor to the generator, counteracting all other loads of the rotor through the interfaces with the nacelle bed. Therefore, all axial and shear forces and the bending moments of the rotor will have to be transferred to the nacelle bed (and from there to the tower top), while the rotor torque should pass through the drive train to the generator, leaving the torque reactions of the gearbox on the nacelle bed.
Classifying the general loads transferred across the interfaces of the drive train as loads, kinematics and dynamics, the following parameters should be defined for the drive train.

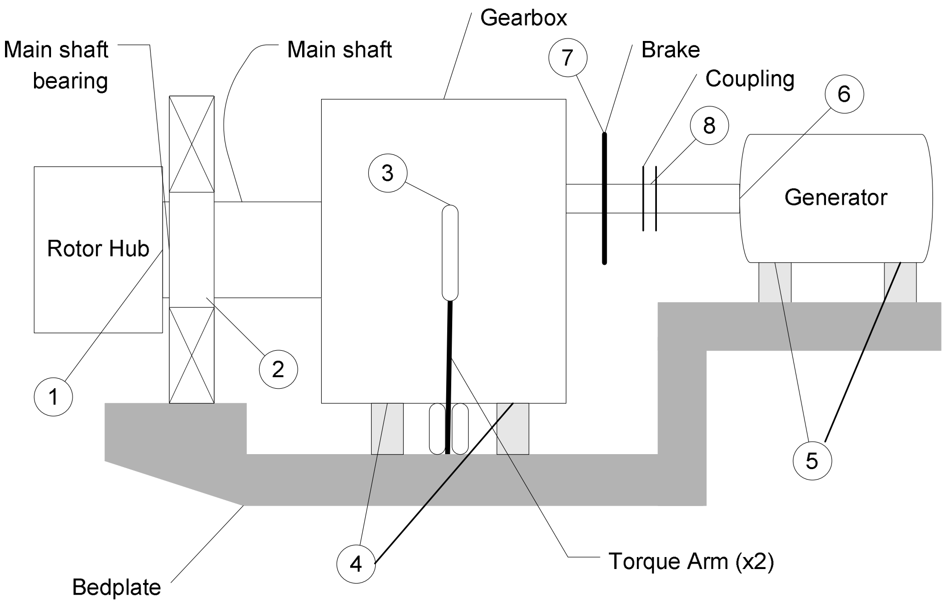

Figure 3-3: *Schematic of nacelle layout with Triple-point suspension showing interfaces.*

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

Kinematics:
- Displacements at the supports
- Positions (angle, speed of rotation and axial displacement) of moving (rotating) elements (e.g. shafts)

Dynamics:
- Accelerations

Furthermore, synchronization of the general loading conditions is required with the wind turbine operational parameters, such as status, rotor revolution speed, power production, azimuth position.
Since the drive train comprises components that are treated as separately as stand alone systems within drive train simulations, such as the main shaft, in a comprehensive analysis the interfaces of such systems should be also identified. The case of the main shaft and the main bearing was investigated within [11], with analysis examples for specific configurations. Yet, since the PROTEST project these components are treated as components of the drive train the analysis was kept to a minimum.

3.2.1.2 Measurement definitions

Based on the results for the loads transferred across the interfaces of the gearbox, Table 3.1 presents a summary of the recommended quantities to be measured during an experimental campaign focusing on the gearbox. The same table can be used as a starting point for the definition of loads transferred across the interfaces of the drive train, properly adjusted for the specific wind turbine configuration.

Table 3.1: *Definition of loads at interfaces of the gearbox*

Interconnection point	Loads	Synchronicity	Analysis
Main shaft & gearbox	Loads: [Main shaft Axial and Shear forces][1], Bending moments and Torsion (Torque) Kinematics: main shaft angle & speed, axial displacement Dynamics:	WT status WT operational magnitudes (Power, RPM) Azimuth position Wind inflow (Wind speed & Wind direction)	Mean loads Fatigue loads (RFC, LDDs)
High speed shaft & gearbox	Loads: [Axial & Shear forces, Bending Moments][2] and torsion (torque) Kinematics: High speed shaft angle & speed, axial displacement Dynamics:		Mean loads Fatigue loads (RFC, LDDs)
Torque arms & gearbox	Loads: Kinematics: Axial, Vertical & tangential Displacement Dynamics:		
Gearbox housing	Accelerations on bearings		
Additional measurements (internal to the gearbox system)	Lubrication temperature on Gearbox bearings, gear meshes or overall volume temperature		

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

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

According to IEC 61400-4 in addition to load measurements prescribed in the IEC/TS 61400-13 the torque on the low and the high speed shaft should be measured in experimental campaigns requiring the verification of the gearbox and the drive train. Additionally, the shaft speed should be also measured. Both measurements are foreseen in Table 3.1. According to IEC 61400-4 additional load measurements for forces and bending moments may be required for the evaluation of the gearbox interface loads and design assumptions. These however, are also foreseen in Table 3.1, such as the bending moments and forces on the two shafts (main shaft and high speed shaft).
Following IEC 61400-4 sampling rate should be adequately selected (in cooperation with the gearbox manufacturer) for each application, higher than 3 to 5 times the relevant vibration frequency.

[1] These loads are not usually measured but are estimated during aeroelastic simulations
[2] These loads are not usually measured and are estimated during aeroelastic simulations only when adequate data are provided for the gearbox

Additionally, following IEC 61400-4, a Campbell diagram (plot of system forcing and response frequencies) should be provided through the complete operating speed range to evaluate resonance risk.
Finally, measured temperatures at specified locations on the gearbox and lubrication system should be reported with emphasis on maximum temperatures and maximum temperature durations.
If applicable, during the measurement campaign lubricant analysis shall also be performed and reported.

3.2.2 Pitch system

3.2.2.1 Loads at interconnection points

To determine the procedure to describe how the loads at the interconnection points should be defined, the specification of the interfaces of the pitch system, is required. That includes isolation of the pitch system from the overall wind turbine structure and further building on the adequate description of the sectional loads at the interconnection points (interfaces) the overall wind turbine loads need to be transferred to design parameters. An assessment followed regarding which knowledge of loading (i.e. torques, bending moments, accelerations, motions, deformations etc.) is considered as a valuable improvement over the current state-of-the-art.
Within WP3 the results presented in [8] as well as the findings of work package 2 of the PROTEST [14] project regarding the design load cases and design drivers for the pitch system that should be considered, discussed in 5.1 of the present report, were further developed to define the procedure for determining the loads at the interfaces of the considered components. On the topic of the pitch system the working draft IEC 61400-4 [6] where the relevant issues of the wind turbine gearbox are discussed, was used as a starting point to determine what kind of information are necessary at the interfaces for designing the mechanical components of the pitch system.
The details of the findings were reported within [12]. In here only a summary of the finding will be presented, regarding the loads at the interconnection points of the pitch system.
As an outcome of the work the interconnection points (interfaces) for the pitch system are:
The pitch system specific interconnection points (interfaces) are:

1) The interface between the blade & the pitch system (bearing)
2) The interface between the hub & the pitch system (bearing)
3) The interface between the hub & the pitch system (transmission & drive)
4) The interface between the controller & the pitch system (drive)

A simplified sketch of an example (electric) pitch system is shown in Figure 3-4. The schematic diagram of the pitch system, its components/subsystems and the relevant interfaces are shown in Figure 3-5 (internal interconnection points are indicated as i).
Specifically for the pitch system there are two distinct cases: The pitch system is used to keep the blade at a predefined position (as defined by the controller) or the pitch system is used to bring the blade into the required position (pitching). The two modes should be clearly discerned and connected with wind flow conditions and operating states of the wind turbine, as the intermittent/oscillating behaviour is essential for pitch (bearing) design and life time.
The pitch system transfers axial and shear forces, bending moments and torsion from the blade to the hub. Bending moments are measured during conventional load measurement campaigns (as specified in IEC 61400-13 [9]). The force measurements however, are not required therefore and usually not performed. The loading can be estimated with aeroelastic simulations, but it is difficult to simulate the exact same conditions.
Torsion should form the load at the relevant interfaces for the pitch system, which affects the loading on the gear of the pitch system (the transmission sub-system) as meshing torque and the torque on the driver of the pitch system. The equation providing equilibrium for the pitch system, involving the meshing torque of the pitch bearing, M_{Mp}, the torsion of the blade, M_{Bx}, the friction M_{fric} and the torque of the pitch motor, M_{Ap}, through the pitch angle of the blade (speed and acceleration), θ_B, the blade and pitch system inertia, J_B and J_{Ap}, respectively, and the gear ratio of the pitch system, i_p is following [12]:

$$M_{Mp} = M_{ZB} - \mathrm{sign}(\dot{\theta}_B) M_{fric} + J_B \cdot \ddot{\theta}_B = i_p \cdot M_{Ap} + i_p^2 \cdot J_{Ap} \cdot \ddot{\theta}_B \qquad (5.1)$$

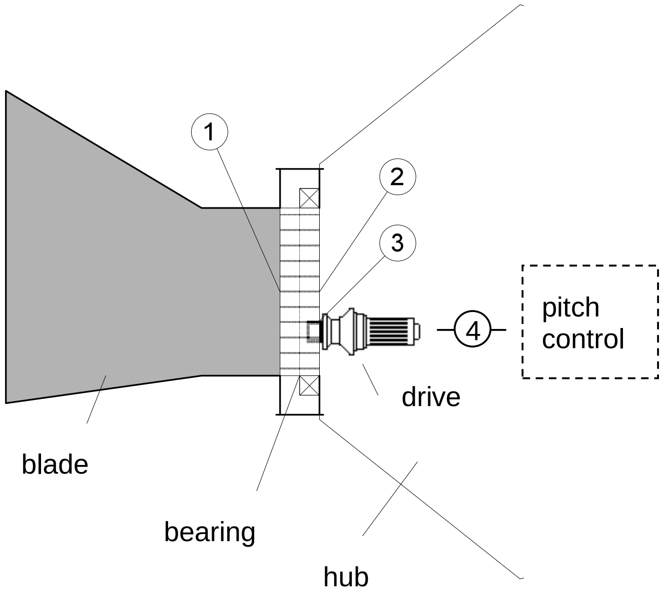

Figure 3-4: *Simplified sketch of pitch system showing interfaces.*

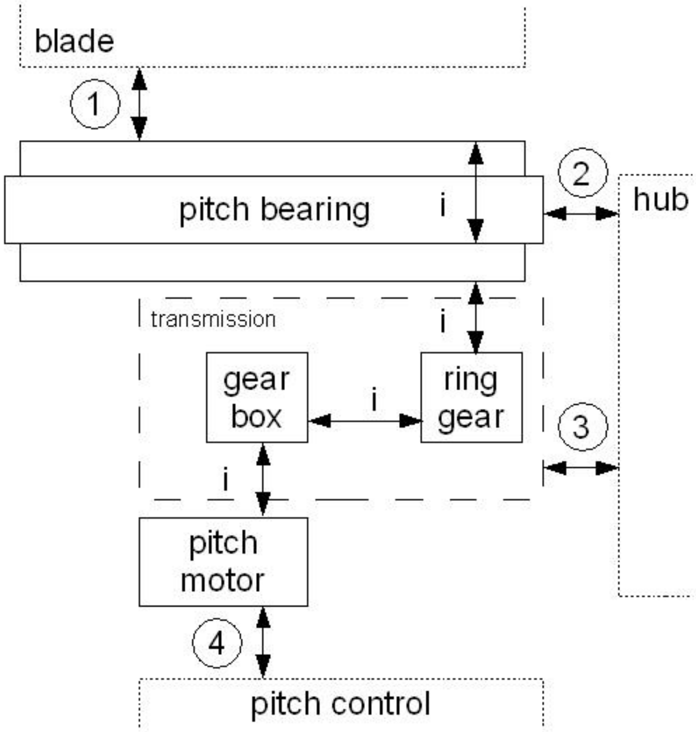

Figure 3-5: *Schematic diagram of the pitch system and its interfaces.*

The friction moment, M_{fric}, depends on the geometry of the pitch bearing (diameter), friction coefficient and the bending moments, axial and shear forces transferred from the blade (root) through the pitch bearing as described in detail in [12]. Although this equation neglects elasticity of the pitch system drive train and the gear mesh free play it is considered as a good approximation of the loads that are transferred through the pitch system components.
Another issue of special importance to the pitch system is the effect of the elastic deformation (ovalisation) of the pitch bearing on the loading of the pitch components, as a result of the deformation of the blade root and the hub due to the acting loads on the blade.

3.2.2.2 Measurement definitions

Based on the results for the loads transferred across the interfaces of the pitch system, Table 3.2 presents a summary of the recommended quantities to be measured during an experimental campaign focusing on the pitch system components.

Table 3.2: *Definition of loads at interfaces of the pitch system (external & internal)*

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

Specifically for the pitch system components, the following measurements are recommended:

- For the pitch actuator: P_{pA} (power consumption of the pitch driver) as measured in (4) of Table 3.2, M_{Ap} calculated from measurements using Eq.(5.1)
- For the pitch transmission system: M_{Mp} as calculated during pitch motion through Eq.(5.1)

[3] The Wind Turbine Operation Data (WTOD) consists of the status, hub wind speed and direction, rotor angular speed and azimuth angle, pitch angle, yaw angle and generator power.
[4] These loads are required to separate loading of the components and to determine elasticity, hysteresis (free play) and friction in the pitch system drive train.

- For the pitch bearing: Blade Bending moments as measured, Blade torsion and Blade root axial & shear forces, either directly measured or estimated through simulations
- Estimation of the frictional torque, M_{fric} through measurements of blade root and actuator moments and use of Eq.(5.1).
- Estimation of bearing friction coefficients through relation between pitch bearing loading and friction torque
- For the wind turbine behaviour in relation to the pitch system, the time delay from pitch control set-point to blade pitch angle/speed

Additional measurements/analyses that are recommended to obtain more knowledge of the system and validate models and design calculations (fatigue life for instance):

- **Pitch bearing deformation** measurements (on bearing rings and/or blade flange and hub mounting: These can be used to investigate the influence of the stiffness of the mounting flanges and of the support structures (blade and hub). Also the effect on bearing friction (and thus wear and pitch driver load) should be addressed.
- **Lubrication contamination**: Lubrication (grease) of the pitch bearing is essential for the fatigue life, especially when the bearing is in oscillating motion. Also the lubrication (oil) of the pitch transmission can be monitored to investigate the wear in the pitch drive train.
- **Electrical load** between bearing rings due to high voltage **lightning strikes**: Lighting strikes (count) on the blades can cause bearing raceway degradation if no proper provision is available for the routing the charge.
- **Temperature** on frictional parts: Friction in the pitch drive train and the pitch bearing causes extra load on the pitch drive, which could lead to increase in temperature.

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

- For the Pitch Actuator: M_{Ap} time series & Root mean square (RMS) per wind condition
- For the Pitch Transmission system: M_{Mp} time series & Rain-flow-counting matrix (RFC) per wind condition
- For the Pitch bearing:
    - Loads (Forces and moments) time series & RFC per wind condition
    - Kinematics (θ mean/amplitude/speed) per wind condition
    - Temperature (if available) in relation to other measurements
    - Acceleration PSD per wind condition

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

- Starts within 10-min captured file, time of operation, time duration up to next start
- Average angle of rotation for each single operation & speed

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

3.2.3 Yaw system

3.2.3.1 Loads at interconnection points

To determine the procedure to describe how the loads at the interconnection points should be defined, the specification of the interfaces of the yaw system, is required. That includes isolation of the yaw system from the overall wind turbine structure and further building on the adequate description of the sectional loads at the interconnection points (interfaces) the overall wind turbine

loads need to be transferred to design parameters. An assessment followed regarding which knowledge of loading (i.e. torques, bending moments, accelerations, motions, deformations etc.) is considered as a valuable improvement over the current state-of-the-art.
Within WP3 the results presented in [8] as well as the findings of work package 2 of the PROTEST [14] project regarding the design load cases and design drivers for the yaw system that should be considered, discussed in 6.1 of the present report, were further developed to define the procedure for determining the loads at the interfaces of the considered components. On the topic of the yaw system the working draft IEC 61400-4 [6] where the relevant issues of the wind turbine gearbox are discussed, was used as a starting point to determine what kind of information are necessary at the interfaces for designing the mechanical components of the yaw system.
The details of the findings were reported within [13]. In here only a summary of the finding will be presented, regarding the loads at the interconnection points of the yaw system.
As an outcome of the work the interconnection points (interfaces) for the yaw system are:

1. The interface between the Tower Top to the yaw system (specifically the yaw bearing)
2. The interface between the Nacelle to the yaw system (specifically the yaw bearing)
3. The interface of the yaw transmission system, i.e. the yaw gear(s), where a tangential force is introduced to the system to rotate the nacelle relative to the tower
4. The (electrical) interface of the wind turbine controller to the yaw system (specifically the yaw actuator/driver)

An oversimplified sketch of the yaw system and the relevant interfaces is shown in Figure 3-6.

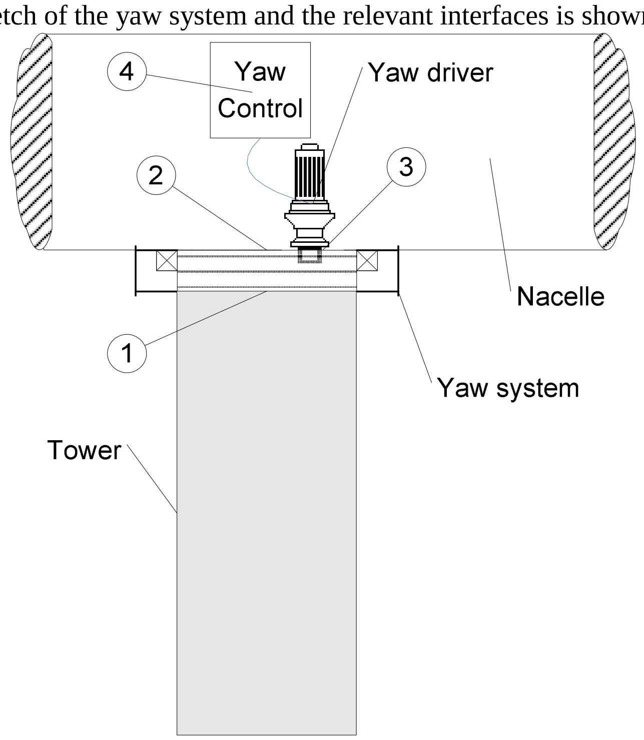

Figure 3-6: *Simplified sketch of yaw system showing interfaces.*

In addition to the interconnection points (interfaces) described above, internal interconnection points of a system can be introduced by dividing a system into its sub-systems. This approach leads to a set of systems on different detail levels which are connected by interconnection points. At each interface, the 6 fundamental loads can be defined. Additionally, characterising values for each load can be added (e.g. rotational speeds).
This leads to a matrix of interconnection loads. Each interconnection load has to be assessed with respect to the individual importance of the value towards the overall result as well as the effort required for the determination of the value.
Regarding the loads that are transferred through the interfaces of the yaw system, there are two distinct cases: 1) the loads transferred while the yaw system is used to keep the nacelle position at a pre-defined position, i.e. Non-yawing (as defined by the controller) and 2) the loads transferred while the yaw system is active and used to bring the nacelle into the required position, i.e. yawing.

The two load cases should be clearly distinguished and connected with wind flow conditions and conditions of the wind turbine.
For the loads to be transferred while the yaw system is used to keep the nacelle at a defined orientation angle, i.e. **non-yawing**, all loads acting on the nacelle end should be transferred to the tower. That is the yaw system should be used to have axial and shear forces, bending moment and torsion transferred from the nacelle to the tower. These are already estimated through aeroelastic simulations. Tower top bending moments and torsion can be measured during conventional load measurement campaigns.
Torsional motion of the yaw system (while the system is maintaining nacelle position), i.e. **non-yawing**, could be measured on an operating wind turbine with vibration sensors positioned at the nacelle part of the yaw system, measuring possible small torsional vibrations (motion and acceleration).
For the loads to be transferred while the yaw system is operating (driving the nacelle to the requested nacelle position), i.e. yawing, the loads to be transferred through the yaw system are again all axial and shear forces, as well as bending moments acting on the tower top, while torsion will be transferred to the tower distorted through the action of the yaw actuator (driver).
This last load component (torsion) should form the load at the relevant interfaces for the yaw system, which affects the loading on the gear of the yaw system (the transmission sub-system) as meshing torque and the torque on the driver of the yaw system.
For the meshing torque, M_M, acting on the yaw gear teeth, the following relationship can be applied:

$$M_M = M_{ZNT} + \mathrm{sign}(\dot{\alpha}_y) M_R + J_{NR}\ddot{\alpha}_y = i_y M_{yD} - i_y^2 J_{yD}\ddot{\alpha}_y \quad (6.1)$$

Where M_{ZNT} the yaw moment on the nacelle, α_y the yaw angle, J_{NR} the nacelle yaw inertia, J_{yD} the inertia of the yaw driver and the yaw transmission system (as one system), i_y 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 the drive pinion, M_{yD} the torque of the yaw driver and M_R is the frictional moment of the yaw bearing. For the load dependent frictional moment several practical estimates are available, all involving the bending moments, the radial and axial forces applied on the yaw bearing.
Detailed equations for the definition of the loads transferred across the various interfaces have been described within [13], where the interested reader is referred to.

3.2.3.2 Measurement definitions

Based on the results for the loads transferred across the interfaces of the yaw system, Table 3.3 presents a summary of the recommended quantities to be measured during an experimental campaign focusing on the yaw system components.
Specifically for the yaw system components, the following measurements are recommended:

- For the yaw actuator: P_{yD} (power consumption of the yaw driver) as measured, M_{yD} calculated from measurements using Eq.(6.1)
- For the yaw transmission system: M_M as measured during yaw motion
- For the yaw bearing: Tower top axial & shear forces, Bending moments and Torsion, as measured
- Estimation of the frictional torque during yaw motion.

Table 3.3: *Recommended measurements during an experimental campaign for the yaw system*

Interconnection point	Loads	Synchronicity	Analysis
Tower top & Yaw System	Loads: Tower top Axial & Shear forces, Bending moments and Torsion Kinematics: (measured at nacelle) Dynamics: (measured at nacelle)	WT status WT operational magnitudes (Power, RPM) Wind inflow (speed & direction)	Mean loads Fatigue loads (LDDs)
Nacelle & Yaw System	Loads: (measured at tower top) Kinematics: Nacelle yaw position & speed Dynamics: acceleration on nacelle bearing in two perpendicular directions		..
Yaw transmission system (gear) & Yaw System	Loads: Torque (Pressure) Kinematics: (measured at nacelle) Dynamics: (measured at nacelle)		Uneven torque distribution
WT controller & Yaw System	Yaw system power consumption Command		..
Yaw bearing & Yaw System (internal system measurements)	Additional measurements: Temperature at yaw base and frictional parts		..

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

- For the Yaw Actuator: M_{yD} time series & Root mean square (RMS) per wind condition
- For the Yaw Transmission system: M_M time series & Rain-flow-counting matrix (RFC) per wind condition
- For the Yaw bearing:
    - Loads (Forces and moments) time series & RFC per wind condition
    - Kinematics (α_y mean/amplitude/speed) per wind condition
    - Temperature (if available) in relation to other measurements
    - Acceleration PSD per wind condition

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

- Starts within 10-min captured file, time of operation, time duration up to next start
- Average angle of rotation for each single operation & speed

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

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Appendix A. Implementation of resonance analysis

A.1 General

Requirements and recommendations regarding the definition of the objective, type and scope required for the resonance analysis of the drive train as well as modelling aspects are given in this appendix. The necessary extent of analysis and modelling detail level depends on the particular design and can vary from case to case.

A.2 Scope

The following refers primarily to conventional drive train designs using a gearbox to increase the rotational rotor speed. For drive trains using a slow speed generator or other methods of power transmission, the statements shall apply with the necessary adaptations. In general, the analysis consists of the following steps:

- simplification of the complex drive train into an equivalent model
- determination of the required input for stiffness, mass, inertia and damping values
- set-up of the simulation model
- execution of the analysis
- verification of the model
- evaluation, assessment and documentation of the results

A.3 Modelling of the system

The technical data from the wind turbine manufacturer and component suppliers should be used to build the simulation model.

A.3.1 Discretization of the model

The simulation model should include all major drive train components. When using multi body systems for the analysis the individual component is subdivided into segments represented by rigid bodies. Gears and bearings can be modelled as single bodies, whereas for shafts and rotor blades finer discretizations are recommended. Interaction between the bodies is modelled by force elements (e.g. spring/damper elements). For the most flexible shafts and complex parts, the use of elastic bodies is recommended.

All relevant natural frequencies of the drive train need to be considered. Thus, all relevant mechanical properties (mass, inertia, stiffness) shall be included in the model.
The discretization of the major drive train components shall be attuned to the shape of the respective component. Moreover, it shall be selected in a way that allows identifying all natural frequencies of the component at or below the second harmonic of the highest excitation frequency.

Depending on the excitation mechanisms, the extent regarding the number of DOFs (degrees of freedom) of each individual component shall be chosen adequately. Torsional, axial and bending DOFs should be considered.

A.3.2 Model input parameters

The model input data consist of mass, inertia, stiffness and damping values of the components. The required input for masses and inertias can be derived from CAD data, by analytical calculation or by measurement. The elasticity of complex parts can be determined by finite element analysis, by measurement or, in cases of simple geometries, by analytical formulae.

For the gears, the meshing stiffness can be calculated on the basis of ISO 6336-1, Method B, or by measurement.

Stiffness properties of bearings are available from the bearing supplier.
Damping properties can be determined by measurements or, if applicable, data from the literature can be used. The final adjustment can be made by measurements on the actual drive train. Damping should only be applied to parts of the model where it will occur in wind turbines e.g. bearings, toothings.

If the analysis is carried out in the time domain, sources of excitation due to variations in the component stiffness and component inertia values need to be considered. These are at least:

- blade passing
- variations in tooth meshing stiffness
- imbalance of major drive train components (rotor, brake disc, coupling, and rotor of generator)
- communication frequencies of controllers (e.g. pitch and yaw controller)

A.3.3 Boundary conditions
The frequency range for analysis in the frequency domain should be chosen wide enough to cover the relevant excitation frequencies.

The analysis range for the time domain simulation shall be chosen in accordance with the operating range of the wind turbine.

In order to impose all operating conditions on the drive train, the simulation of a run-up by steadily increasing the rotational speed is an appropriate procedure. The run-up can be carried out in the speed- or torque-driven mode.

A.4 Calculation and evaluation of the results
For the time domain calculation, the time range and sampling rate should be chosen large enough so that, for each level of rotational speed, a steady state will be reached and reliable Fast Fourier Transformations (FFT's) with 2n supporting points can be performed. n shall be chosen in such a way that an appropriate resolution will be obtained and that the necessary frequency range can be analysed.

Calculated time series of e.g. rotor speed and torque and the load levels in all springs should be checked with respect to the correct reproduction of e.g. transmission ratio, rotational direction, angular displacement of shafts etc.

The results need to be checked for plausibility. This involves checking of natural frequencies and mode shapes to see whether their magnitude and shape, respectively, are credible in comparison to similar drive train layouts and to experience.

A.5 Extended evaluation
In the event that the analysis shows abnormalities in terms of e.g. resonances that occur in the operating range of the wind turbine, extended evaluations might become necessary. These can be performed by applying a even more detailed simulation model or by measurement on the actual drive train.

It is recommended that the simulation model of the drive train be used to analyse transient dynamic loading caused by extreme load cases (e.g. DLC 1.4, DLC 1.5, DLC 2.2, DLC 9.2; see [10]) that are relevant for the drive train.