PROTEST

Final Report

Project Results and Recommendations for Standardisation

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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 main results of the PROTEST project are summarised in this report. The PROTEST project was a pre-normative project focussing on improvement of the design and development procedure for the mechanical components in a wind turbine: the drive train, the pitch system and the yaw system.
The current state-of-the-art is discussed and the short comings are pointed out. It is clear that the level of analysis and the quality of modelling in the aeroelastic analysis of blades and tower is currently at a much higher level than the integrated analysis of the mechanical components.
When trying to set guidelines for the set-up of a measurement campaign on a wind turbine prototype to be used to validate or tune a model of one of the mechanical components, it became apparent that it would not be possible to describe a fixed method for this. A flexible six step approach is therefore suggested, that takes into account the limitations and uncertainties of the model that has been used.
After a general description of this approach, the results for each component are discussed, starting with the results for the drive train, followed by the pitch system and finally the yaw system. The design loads that are relevant for each component are summarised and additional load cases have been defined for the drive train. The loads at the interconnection points are described and the measurements definitions are given. Then the results of using the six step approach for are summarised.
Finally the recommendations for the standardisation are given.

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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 wind specific 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 was in fact a pre-normative project with the aim to 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].

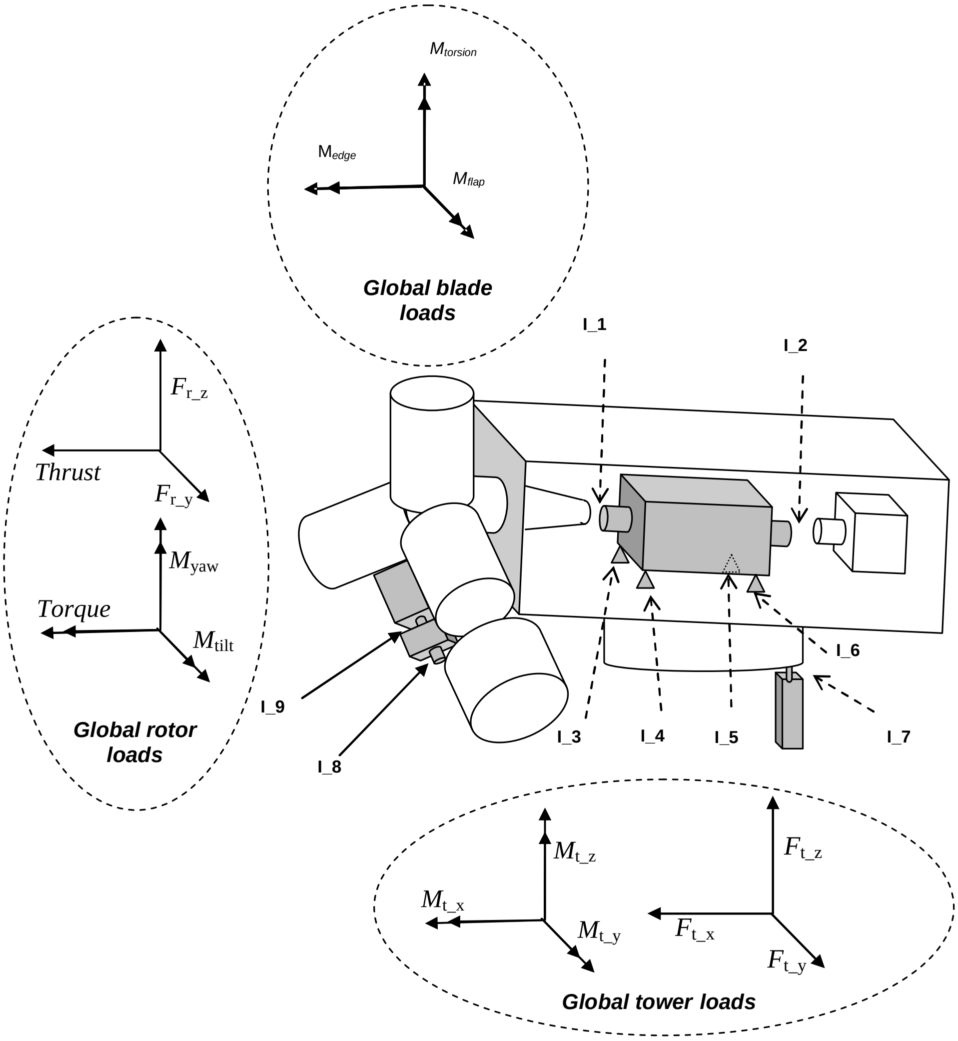


Figure 1-1: *Schematic presentation of transforming "global turbine loads" to "local components loads" at nine interfaces, (gearbox, pitch system and yaw system)*

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 the components drive train, pitch system and yaw system were 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 are discussed. First the results that are of importance for each of the three selected components (drive train, pitch system and yaw system) are reviewed. After these, results that are specific for each of these components are dealt with, enabling readers that have special interest for one component to skip the other parts.

Using this subdivision, the results for WP1 –state of the art – are first discussed, followed by an explanation of a six step approach that is proposed to be used as a guideline instead of trying to set fixed standards for each component, as for each components several different variations exist and at the same time the measurement campaign and its use is strongly dependent on the model used in the analysis.

After the description of the six step approach the main results are given first for the drive train followed by those concerning the pitch system and then the yaw system. Finally the recommendations for the standards for pitch and yaw systems are summarised.

2. Main results WP 1: State of the art

In this chapter the main results from Work Package 1 of the PROTEST project will be shortly described. For more details the interested reader is referred to the public deliverable of this work package [8].
First the state of the art of component design for each of the components examined within PROTEST will be discussed, followed by a description of measurement pitfalls. The comparison of simulations to measurements will be shortly presented and the main conclusions and outlook of this work package will be summarised.

2.1 State of the art of component design

2.1.1 Drive train

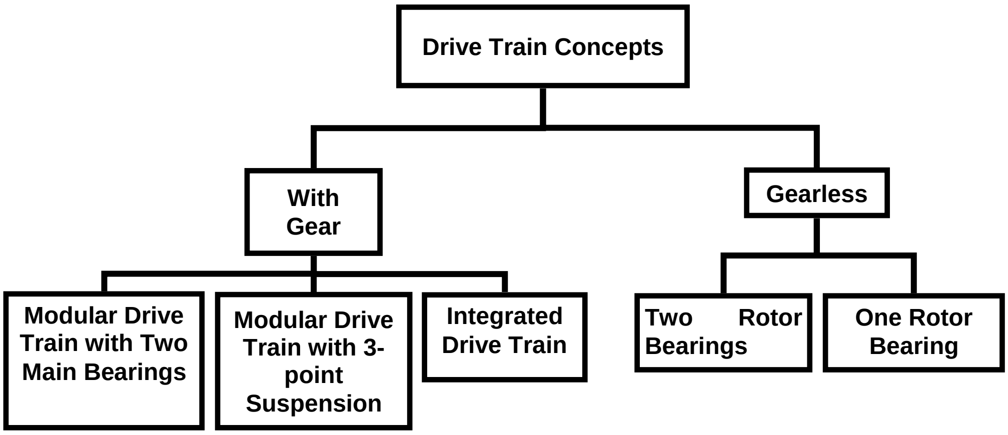


Figure 2-1: Overview of common drive train concepts

Currently there are roughly five different drive train concepts to be distinguished, as shown in Figure 2-1.
The turbine concepts that include gearboxes are of special importance within the PROTEST project. From various studies [1],[9],[10] it is concluded that the electrical sub assemblies in general have a higher failure frequency than the mechanical components. However, because the downtime and repair costs are typically higher for mechanical components than for electrical components, the perception of failure rates for gearboxes may differ from the actual values. As a consequence, this has led to many research activities to improve the reliability of gearboxes.
Many attempts have been taken by the industry and R&D institutes to improve the reliability of gearboxes. The most relevant ones are [8]:

1. Condition monitoring
   Condition monitoring systems can mitigate the consequences of damage, but not reduce the number of failures [4].
2. Failure analyses and gearbox measurements
   In the US NREL has set up the Gearbox Reliability Collaborative (GRC) to address the reliability of gearboxes [11].
3. Improved design software
   The current practice where a drive train is modelled as a single torsional spring with a fixed gear ratio, a mass moment of inertia, a spring constant and an estimated damping coefficient is recognised as insufficient. Therefore there are many activities towards more detailed modelling using multi body simulations [12][13][14][15].
4. International working groups and standardisation committees.
   A more complete inventory of the relevant load cases and design specifications has been created under international cooperation and new IEC standards have been created for the gearbox [6].

To determine the design loads and design stresses many different possible fault- and wind conditions are run in simulations and the maximum values for the loads are determined. In most aeroelastic codes the input for the drive train is very limited, e.g. in Flex5 only six input parameters are used. The limited number of input parameters also results in limited output parameters that can be used to design the drive train. It is possible to overcome this limitation by coupling more detailed drive train models (e.g. multi body) as a dynamic linked library (DLL) to the aeroelastic codes. This enables the reactions of the gearbox to be fed back into the aeroelastic code, resulting in a more accurate simulation. If a component design change modifies the reactions of the turbine, the new component design information needs to be included in further simulation runs to verify the design.
By simulating the turbine under the expected environmental conditions, a component fatigue analysis can be undertaken. Again, changes in the turbine design for not meeting the fatigue life criteria will necessitate further simulation runs to verify the new design.

2.1.2 Pitch system

Presently, most turbines in the multi-megawatt class are equipped with variable speed and pitch control. Pitch control can either be done by means of hydraulic pitch cylinders or by electrical motors. More and more, the option of individual pitch control is being investigated by the leading manufacturers to optimise power output and to reduce the mechanical fatigue loads. However, to reduce the mechanical loads by means of pitch control, it is necessary to obtain feedback from the loads in the blades. At present, the measurement techniques are not robust enough to be incorporated into the control loop reliably for a long period of time.

From publicly available data on failures and maintenance of wind turbines (e.g. [1][10][16]) it can be concluded that the failure behaviour of pitch systems (number of failures and their resulting downtime and repair costs) is not the real cost driver for maintenance and repair. However, there is a strong need to improve knowledge of loads and dynamic behaviour of the pitch system for new and large turbine designs. For example, larger wind turbines require larger pitch bearings which are relatively less rigid and thus more sensitive to deformations. To determine the lifetime of pitch systems it is necessary to understand the loads and deformations and their influence on friction and wear of the pitch system. If, in the future, load measurements are going to be incorporated in the pitch control loop, the number of pitch actions per blade and the pitch speed will probably increase. Under these circumstances the need to understand the loads on and wear of pitch systems is even higher. A first attempt to better understand the load pattern in the different components of a pitch system is presented in [17].

For the determination of the design loads relevant for the pitch system, the rotor blade root pitch moment and the blade root bending moments are obtained from the aeroelastic simulations. To consider the loads which occur during pitch manoeuvres, the pitch actuator is modelled in those codes. The pitch actuator model is dependent on the wind turbine controller and its dynamics influence the blade root loads. The controller can either define the pitch angle or the pitch rate which is then modified by a transfer function in the pitch system model. However, the drive train dynamics of the pitch system are not modelled in detail.
For the determination of the pitch system design loads, combined analytical and empirical methods are used to transfer the blade root loads into the loads relevant for the design of the pitch system. This step is needed because the moments and forces acting on the toothing of the pitch system is highly dependent on the blade bearing friction and bearing deformations like ovalisation, which is again mainly caused by the blade root bending moments in edgewise and flapwise direction. These analytical models are a weak point in the current design process of the pitch system.

2.1.3 Yaw system

Similar to the pitch system, the yaw system is not a cost driver for the maintenance costs of wind turbines. However, a good understanding of the mechanical loads acting on the yaw system is needed to optimise the design of new turbines.
The concept of passive yawing has been an option for smaller wind turbines. The disadvantage of this concept is that one cannot control the nacelle during off-design conditions and that in case of extreme changes in wind direction, the turbines may continue running downwind instead of upwind (or vice versa). For larger turbines, the use of the active yawing concept is common practice. Several

(typically 4 to 8) yaw drives keep the rotor perpendicular to the oncoming wind. Some concepts allow the nacelle to yaw passively in case of small changes in wind directions. Only in case of extreme wind direction changes, in cases of low wind speeds, or during stand still, the yaw system is activated. A disadvantage of this concept is that passive yawing may lead to unwanted loading of the yaw drive and damage to the gearbox. Therefore, more and more turbines are equipped with yaw brakes to avoid passive yawing. This protects the yaw drives from high cycle fatigue due to turbulence induced yaw moments. The yaw brakes can be released during the yaw action itself. The most common concept is that the installed capacity of the yaw drives is sufficient to keep the yaw brakes closed during yaw manoeuvres.

In aeroelastic design codes the yaw moments are calculated for the relevant load cases. Therefore an active yaw system can be taken into account, if applicable. For wind turbines that are equipped with active yaw systems, a yaw manoeuvre can either be predefined during aeroelastic calculations or the control system can be used to specify the yaw angle or the yaw rate. The yaw can be defined as a rigid system that follows the predefined or controller demanded angles or rates. Most yaw systems are equipped with brakes to unload the yaw system when the system is not active. To exclude cyclic loads on the toothing, it is common practise to have a constantly acting residual braking moment which leads to a higher static loading during yawing. To take these loads into account the yaw system can also be modelled flexible to consider the yaw system dynamics. The yaw system drive train is usually not considered in this type of simulation.
For the fatigue analysis of the yaw system the load duration distribution (LDD) of the yaw moments derived from the aeroelastic simulations should be considered.
However there are numbers of issues still open for the determination of the loads acting on the yaw system, since the load transfer from the rotor to the tower top (through the yaw system), passes through the drive train and the nacelle, which, as already stated, are only very roughly modelled in the aeroelastic simulation tools.

2.1.4 Limitations

A root cause analysis for failures in gearboxes and bearings does typically not lead to one single reason and, thus, the solution should also be found in a combination of various root causes. Several possible root causes, seen from a designer’s point of view, are outlined below.

Given the open nature of the IEC design load cases, it is likely that the scenario that causes gearbox damage is included in the standard design load cases. However, as noted earlier, it is necessary for the designer to determine which load case is the critical one. This involves correctly modelling of the turbine and its components in one or more simulation tools, and correctly estimating the stress response due to the multi-axial loading. Any combination of loadings on a component could cause the maximum stress response, so all load case scenarios need to be simulated. The results then need to be analysed to find the maximum stress response.

If one does not know the exact design of the component, then only the loads are available, not the stress response. The loads from preliminary simulations are shared between the turbine designer and the sub-component suppliers. Ideally, the entire final design of the turbine would be available for simulations before the manufacturing process starts. This is often not possible or realistic.
Changes to the component design during the manufacturing process should be included in the simulation iterations. This is problematic when components are supplied by outside partners and manufacturers. Gearboxes and bearings are often manufactured by external suppliers. The simulation and dimensioning of gearboxes and bearings is dependent on the manufacturers. Their in-house proprietary simulation tools are assumed to be used in the design and selection of components. There is very little information coming back from the component supplier that can be used to re-simulate and fine tune the simulation model. The problem can also be in the opposite direction; if not enough loading information is given to the component engineering team.
For components that are designed in-house, the know-how to determine the critical load cases is also in-house. When a design needs modification due to a potential load case scenario, the design of the component can be modified and the turbine system re-simulated. Especially, in the field of rotor blades progress has been made due to the following facts:

- The blade response can be simulated rather accurately using beam models for an aero-elastic analysis to capture overall wind turbine performance; although for larger blades research is still ongoing.
- Important blade properties for the wind turbine overall behaviour have been identified in previous research. Therefore, if one of these properties is changed then the process of load calculations is repeated.
- The blades undergo a full-scale verification test, including identifying of properties affecting wind turbine behaviour.

For rotor blades there are still failure issues to be solved, but especially the so-called mechanical wind turbine components did not yet undergo any of the steps that resulted in the progress obtained in the reliability of the rotor blades.

It is also possible that the load cases that are relevant for the drive train, pitch system and/or yaw system may not be taken into account. This could be because the design load case combination (wind condition and fault) has not been assessed as relevant by the person simulating loads of the wind turbine. More information into the failure modes of failed turbines could be used in a 'reverse engineering' exercise to determine the possible fault condition that caused the failure. An example can be events in the generator (loss of electrical grid, short circuit) where a condition may cause a load going back through the drive train. This type of condition is not well understood or modelled. New operational requirements, such as low voltage ride through (LVRT) may also cause unforeseen loading and are not easy to model. This is also discussed in section 4.1.

The complexity of wind turbines does not allow easy dynamic simulations. The reactions of gears, bearings, etc. are complex in nature and cannot be modelled with 100% accuracy. Unexpected reactions of bearings might be beyond the ability of simulation codes to model in a realistic time window [18]. The specific failure modes of individual sub components (i.e. individual ball bearings) have effects throughout the entire system. The IEC design load cases make assumptions based on the wind and control actions or faults of the turbine, but individual component failures are often beyond the scope of current simulations.

2.2 Measurement pitfalls

The current state-of-the-art in measurement procedures is described in the WP1 report, chapter 3 [8]. Next to the description, some pitfalls in these procedures are identified in this WP1 report. To sum them up shortly:

- Uncertainty of the load measurements, the methods currently used lead to an uncertainty in the loads is in the order of 5% [19].
- Limited amount of experimental data due to cost considerations, limited time, wind regime etc. If the data base is larger, the design validation can be performed better.
- Limited description of the wind inflow.

For more details on the current state-of-the-art measurement approaches, the interested reader is referred to the WP1 report, chapter 3.

2.3 Comparison of simulations to measurements

Results of prototype measurement campaigns are being used by R&D departments among others to verify the design approach. In addition to that, measurement results are also being used by certification bodies to verify the design loads.
In the EU project VEWTDC (Verification of European Wind Turbine Design Codes) [20], various design codes have been compared with measurement results. During the verification process the following sources of discrepancies between measurements and calculations were observed:

1. Discrepancies due to errors in post-processing and coordinate systems
2. Uncertainties in machine description
3. Uncertainties in the prescribed external conditions
4. Uncertainties in the load measurements
5. Uncertainties due to different implementation and interpretation of the input description
6. Differences caused by fundamental model effects

These discrepancies need to be considered when verifying the design approach using prototype measurements.

The comparison between measured quantities and design values can be done in various ways, and at various levels. For instance, designers can compare the 20 years load spectra, or they can analyse time series. From the interviews and the authors' own experiences a kind of common approach could be determined. In general it can be said that first of all the statistical values (azimuthally binned, scatter plots, etc) and natural frequencies are being compared. If differences are observed, more detailed investigations are being done on the level of time series to explain the causes of the differences [8].

The following issues should be noted:

- The IEC-61400-13 standard [21] clearly describes how mechanical load measurement campaigns should be carried out. However, this standard does not include a procedure how to compare the measured data with the design data.
- From the VEWTDC project and from interviews with designers it is concluded that the measured data are being used for checking the correctness of design models, for quantifying input parameters of design models, for estimating uncertainties in the design models, and for completing the set of design loads in case the models are not suitable. A clear procedure for doing this has not been found.
- The information found about comparing design and measured data is limited to the global turbine loads. The authors have not found any publication or report on comparing measured data of mechanical components with component design data.

2.4 Standards and Certification Procedures

The certification and the design of turbines are based on several guidelines and standards specifically developed for the wind turbine industry [22],[23],[24]. Regarding the overall turbine safety the most frequently used are GL's wind guideline [22] and the IEC standard [7],[25]. In these standards the principal requirements for the analysis of the wind turbine regarding safety are given, mainly as requirements for the safety system, environmental conditions, load case definitions and safety factors. From these the loads and safety requirements for the different components under investigation are extracted.

2.5 Conclusions and outlook

The design process of wind turbine components is based on aeroelastic calculations of various DLCs that are described in standards and guidelines like the IEC 61400-1 or the GL guidelines for the Certification of wind turbines. These wind turbine standards have been developed to ensure the engineering integrity of wind turbines. Recently it became obvious that these DLCs are not sufficient for the design of machinery components, especially the drive train, but also for the pitch and yaw systems. For the drive train an additional standard (IEC 61400-4) is currently under development. IEC 61400-4 is a good starting point to define specific DLCs, but no detailed list of DLCs is provided. At present it cannot be concluded whether the DLCs specified in the wind turbine standards are specific enough for the drive train components. It can also not be evaluated whether the DLCs defined with the assistance of IEC 61400-4 are relevant and/or sufficient for calculating the design loads of drive train components.

Therefore, the PROTEST project has evaluated whether the currently considered load cases are sufficient for the drive train, pitch and yaw system, or which additional DLCs have to be considered. The results of the evaluation can be found in section 5.1.

It is also concluded that the state-of-the-art aeroelastic simulation codes use a simplified representation of the machinery components, esp. the drive train, that result in neglecting the interactions of the components. Instead, at present only the GL guideline requires consideration of the internal component dynamics by drive train resonance analyses to identify possible resonances. But the results are not linked to wind turbine loading. To overcome the shortcomings of this simulation approach, advanced wind turbine simulation codes could consider more detailed models of the drive train components also for aeroelastic simulations in the time domain.

For load measurements, performed to support the design process, and for certification various MLCs are defined in guidelines and standards, esp. IEC/TS 61400-13. At present measurement campaigns are used to validate the global design loads with measurement loads. Similar to the DLCs, it is not yet clear whether the provided MLCs are sufficient to validate the design loads of all the wind turbine components. Moreover, the guidelines and standards define the measurement campaign in detail, but no procedure is given on how to validate the global design loads with the measured loads. Furthermore, at present there are no guidelines or standards to define a measurement campaign for wind turbine components. Since no procedure is given for the validation of the global loads, this lack of information is even more relevant for the validation of component loads.
To compensate that, the PROTEST project has developed procedures for performing such a measurement campaign and for validating the loads, used for the component design, with the component measurement data described in chapter 3.

When looking at the future prospects for the design and development procedures for mechanical components, it is the expectation of the authors that it will follow a similar route as the design and development of rotor blades over the last 20 years. The design process for rotor blades (and also for the tower) is critical for safety: failures will lead to unsafe situations. Therefore in the past, safety standards have been developed for wind turbines [25] together with technical specifications on how to carry out full scale blade testing [26] and prototype measurements [21] in order to prevent critical failures. Failures of other mechanical systems however are mainly critical for reliability: failures will lead to standstill and economic losses only.
In short improvements to the design and development procedures for mechanical components in the following areas are expected [8]:

- Design approach.

  The design approach for blades and towers is much more extensive than the approach used for mechanical components. It should be assessed if the current DLCs cover everything for these components, where not only fatigue and ultimate strength should be considered. Also the dynamic properties should be modelled and tested in more detail, as is currently the case for blades, but not for the other components.
- Measurement procedures

  Similar to the standards for blade measurements, guidelines should be set up for analysing and reporting the measurement data for the components, but with more emphasis on using measured data in the design process.
- Standards and certification

  New standards and guidelines will differentiate based on size of the wind turbine and on onshore or offshore turbines. Redundancy or maintainability will become important. The certification process will focus more on supervising the quality of the manufacturing process.
- Data exchange

  After validation of the prototype turbine it is necessary that a certain minimum of data flows from wind farm owner to wind turbine manufacturer to component manufacturer.

3. The six step approach (WP4)

The objective of Work Package 4 was to define the prototype measurement campaigns for each component. Using the combined knowledge of the consortium members it soon became apparent that it would not be feasible to define a strict campaign for each component, but that a more flexible approach is needed, one that takes into account the limitations of the model that has been used as well as the differences between various concepts for each component.

There can be two different objectives for the measurement campaign. In the PROTEST 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 diverse 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: 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). The main results of these analyses are described in sections 4.6, 5.4 and 6.4 respectively.

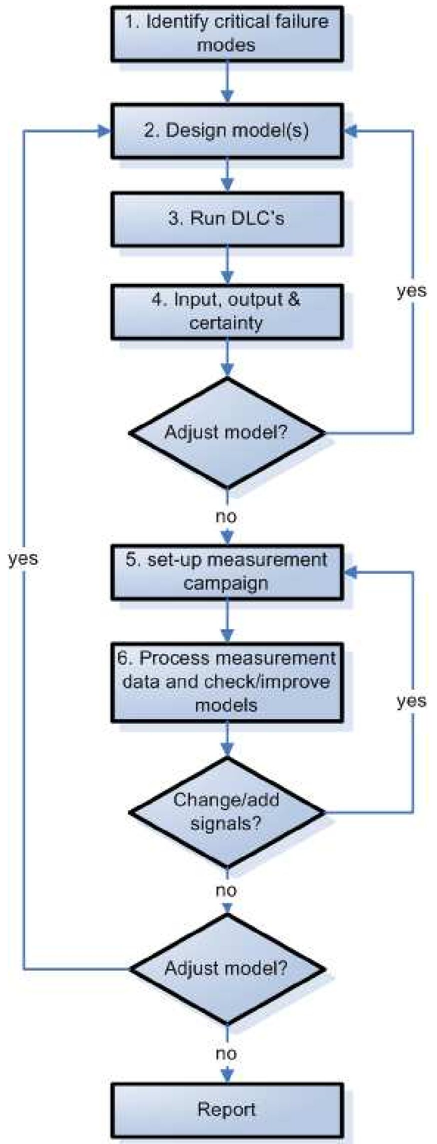


Figure 3-1: *Illustration of the six step approach.*

The remainder of this report will discuss the relevant results for WP 2, WP 3, WP 4, WP 5, WP 6 and WP 7 collected for each component, so not organised per Work Package. The drive train will be discussed first, followed by the pitch system and finally the yaw system results will be shown.

4. Main results for drive train measurements and analysis

The main results from the different Work Packages that concern the drive train are described in this chapter. First the Critical Design Variables and load cases are discussed in section 4.1. In the next section three new DLCs are suggested. In section 4.3 the sensitivity analysis is discussed. This is followed by a description of the loads at the interconnection points in section 4.4. The measurement definitions are given in section 4.5. Finally the results of the measurements and the analysis of the drive train are described in section 4.6.

4.1 Load cases and Critical Design Variables (CDVs) (WP2)

Though blades and tower are sufficiently covered in the standards, the mechanical components are less well presented. Because of the specific problems with gearboxes nowadays it is obvious that the gearbox should be treated separately from the other drive train systems. To determine the “loads” on the whole drive train properly it should be emphasized that the dynamics that may be introduced by the machine frame and by the generator support structure should be taken into account in the simulation models as well.

To enable the specification of the Critical Design Variables for the drive train use is made of the breakdowns given in Table 4.1 - Table 4.2. It should be noted that these breakdowns are limited to those components or subsystems which are relevant for the structural integrity and therefore should be aimed at when specifying CDVs and design loads.

Although not a structural component and not directly linked with “loads”, there is a strong argument to include lubrication in the breakdown of the gearbox. Lubrication is of great importance for the structural reliability of the gearbox, while the efficiency of the lubrication system may be strongly dependent on the external conditions, e.g. also during a cold start up the lubrication system should work properly, which may depend upon specific control strategies. Hence for the designer of a gearbox it is of importance that the minimum set of DLCs provided by the wind turbine manufacturer does cover the reliable working of the lubrication system also.

In general the components mentioned in Table 4.2 are present in a drive train. However, the exact drive train architecture should be considered to assess whether this list is still covering the actual design.

Table 4.1: *Breakdown of gearbox*

- Gears
- Bearings
- Shafts and shaft-hub connections
- Structural elements
  - Torque arm
  - Planet carrier
  - Any other structural components transferring major loads
- Lubrication

Table 4.2: *Breakdown of drive train apart from gearbox*

- Main bearing
- Main shaft
- Main shaft – gearbox connection
- High speed shaft
- High speed shaft – mechanical break coupling
- High speed shaft – generator coupling
- Generator

For the relevant Critical Design Variables (CDVs) for the drive train, the corresponding design loads are given below in Table 4.3 - Table 4.4. Beside the specification of the design loads, it is indicated whether the external condition introducing these design loads are covered by an existing design load case (DLC) in IEC-61400-1.

Table 4.3: *Design loads for gearbox*

General		
Name	**Type of load**	**Covered by DLC**
Rotor torque (Note 1)	Ultimate strength and fatigue	Yes: Note 2
Torque reversals	Ultimate strength, fatigue, and Hertzian stresses	Yes: Note 2
Relative displacements (axial, radial and angular) of: • main shaft – main bearing; • HSS - gearbox flange	Ultimate strength and fatigue	Yes: Note 2
Misalignment in drive train		No
External or internal electrical fault	Ultimate strength and fatigue	Yes: DLC 2.1, 2.2, 2.3, 2.4, 6.
LVRT (Low Voltage Ride Through)	Ultimate strength and fatigue	Yes: DLC 2.3, 2.4: Note 3

Gears		
Name	**Type of load**	**Covered by DLC**
Gear Rating • Pitting and bending stress • Scuffing • Micro pitting • Static strength	IEC-61400-4 WD3 – section 7.2.2	Yes: Note 2
Operation at overload	Ultimate strength and fatigue	Yes: Note 2

Bearing		
Name	**Type of load**	**Covered by DLC**
General design consideration • Subsurface initiated fatigue • Surface initiated fatigue • Adhesive wear • Frictional corrosion • Overload	IEC-61400-4 WD3 – section 7.3	Yes: Note 2
Relative displacement shafts/bearings	Ultimate strength and fatigue	Yes: Note 2
Temperature differences in the bearing	Ultimate strength and fatigue	Yes: Note 2

Note 1: Special attention should be given to the efficiency of the lubrication system

Note 2: All typical wind turbine operation modes are covered in the guidelines, so the external condition(s) introducing this design load is captured.

Note 3: LVRT is a complex situation which can contain many different DLC’s, the current representation of this in the guidelines is too limited. The current tools are also not able to perform the necessary detailed analysis.

It is currently assessed that, when the above mentioned DLCs will be considered for gears and bearings, they will also include the relevant CDVs with sufficient detail for the shafts, shaft-hub connections and structural element (and possibly the lubrication system).

Table 4.4: *Design loads for drive train apart from gearbox*

Name	**Type of load**	**Covered by DLC**
Torque and bending moments acting on main shaft, high speed shaft, couplings, bearings, etc.	Ultimate strength and fatigue	Yes, Note 1
Resonance	Different components interference	No
Torque oscillations with load reversals of high speed shaft	Fatigue	Yes, Note 1
Relative displacements (axial, radial and angular) of: • main shaft – main bearing; • HSS – generator flange	Ultimate strength and fatigue	Yes, Note 1

Note 1: All typical wind turbine operation modes are covered in the guidelines, so the external condition(s) introducing this design load is captured.

It appears that for almost all design loads the external conditions introducing these design loads are covered by existing DLCs. The following two design loads are not covered: (1) loads in gearbox due to misalignment in the drive train, and (2) loads due to resonance in the drive train. The fact that the external conditions are covered by existing DLCs does not mean that that these DLCs can be applied straightforward in the design process of the structural systems, mainly due to the fact that the traditional wind turbine simulation tools are limited in modelling the structural systems with sufficient detail. Application of other type of simulation tools (like multi body simulation) may provide the possibility to solve the problem of the lack of detail, however at the cost of increased computation time, so then the problem arises that not all required DLCs can be analysed in a reasonable time period anymore.

4.2 Proposal New Design Load Cases (WP2)

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.

4.3 Sensitivity analysis (WP2)

As far as the level of details of the model is concerned, a general rule to follow by the modeller is expressed by J. Peeters in his PhD thesis [44]:
An optimal target in the search for more advanced calculation methods is a combination of, on the one hand, accuracy and, on the other hand, workability, such as regarding time-efficiency and user-friendliness. This target is often aimed at in various applications and is well summarised by Einstein's quote: "Make everything as simple as possible, but not simpler".

The first guideline, to assess how advanced should the model be, is to start with the most advanced model possible, using all the model data which is available, in our case represented by the so-called "stage 2", and to reduce it until any significant change can be noticed.
The second guideline is to use a strict criterion to assess the results, such as:

1) a resonance analysis up to a certain frequency
Possible techniques for doing the assessment are modal analyses of the models or resonance analysis based on FRF's (Frequency Response Functions), explained in J. Peeters' work [44].

2) take a relevant Design Load Case, usually a -highly - dynamic transient event (e.g. a low voltage ride through) and compare absolute load levels on certain component for various modeling detail;

A sensitivity analysis has been carried out as a part of the modeling works of PROTEST to estimate what was is the influence of the degree of detailing of the model topology on the simulation results. Two different stages have been modeled. The first stage is a reproduction of the Flex5 like model and the second one proposes an extension of this model, exclusively over its drive train, based on the data available from a Dresp like Model (multi-torsional model of the drive Train), with an additional 14 Degrees of Freedom. The topologies of these models have been explained in details in work package 5.
Investigations have been carried out, both in the frequency and the time domain, using respectively modal analyses and time simulations. They aimed at observing the influence of the consideration of additional torsional degrees of freedom (e.g. shaft stiffness and gear stiffness) within the model. The results of these investigations cannot be extrapolated or generalized to models of any other wind turbine, however some methods for running a sensitivity analysis on the level of details of the model are proposed below and they have been partly exemplary tested on the drive train model integrated in the model of the Suzlon S82 Wind Turbine. The comparison of the outputs of the modal analyses of stages 1 and 2 showed a slight difference on the second and third eigenfrequencies where the drive train torsion is significant (deviation from 1,5% to 3%), and as a matter of fact the stage 2 model contains the additional eigenfrequencies coming from the gear box (housing mounts, shafts, gears) and the further components of the high speed shaft (cardan sleeves, brake disk etc. …) which are well above 10Hz. Based on these informations, time simulations of production load cases under deterministic wind (Normal Wind Profile for GL-DLC 1.0) and turbulent wind (GL-DLC 1.2) were run but they didn't not show any significant differences as far as torques on the main shaft and high speed shaft are concerned. As a consequence, it can be considered that in the case of the Suzlon S82, the most basic model (Stage 1) can be used to simulate torques on both sides of the gear box during the production load cases with a sufficient confidence. As far as transients (e.g. emergency stop, Low Voltage Ride Through) are concerned , the designer has to observe them case by case, however more advanced torsional models are expected to bring significant improvements, since they can reproduce higher modes which tend to be excited under these extreme conditions.
During model reductions, among others, the following rules can be followed:
When rotational springs (with stiffness K_1 and K_2) are connected in series and their dynamic influence on the system can be neglected for a particular load case, they can be merged to a single spring with an equivalent stiffness of $(K_1.K_2)/(K_1+K_2)$, the inertia of the intermediary element can be integrated to the border bodies.
A more straightforward way of reducing the number of DOFs is to fully neglect its corresponding stiffness, by e.g. changing the torsional DOFs to a zero DOF joints or by adding a constraint to torsion without calculating and inserting an equivalent stiffness, in that case the equivalent stiffness becomes the stiffness of the element which is left (in general the most flexible one).
The flexibility of the gear teeth can be also neglected by replacing the force element reproducing it by a constraint with a fixed gear ratio (stiff link).

As far as the question of how to define which DOFs can be suppressed is concerned, the following procedures are proposed, based on the guidelines previously described.

As explained above, starting from the most advanced model (which could be the stage 2 with 14 additional DOFs), the following propositions deal with

<u>1st proposition</u>

- E.g. run a **modal analysis or use Frequency Response Functions** and identify the DOFs responsible for the higher modes
- **Set up a threshold** frequency above which harmonics are considered as uninteresting. If certain eigenfrequencies can be excited, the frequency range that is subject to investigation should be chosen wide enough that the highest relevant excitation frequency to be covered.
- **Suppress the DOFs** which can be identified as responsible for the harmonics above this threshold
- **Run DLC's** (depending on the investigated critical design load cases)

- Define if the **differences between the different models are significant** in comparison to the additional computational time

2nd proposition:

- **Run the most advanced model** at the different **relevant critical load cases**
- **Run a FFT of the relevant signals**
- **Check in the frequency spectra** above what frequency the components can be neglected in the analysis
- Suppress the DOFs responsible for the eigenfrequencies above that threshold

3rd proposition

- **Suppress some DOFs systematically** (for example shafts' torsional DOF, then gear teeth flexibility)
- **Run the relevant DLCs** for the Critical Failure Mode
- Notify and stop reducing the DOFs as soon as this reduction has a noticeable influence on the results of the time signals

4.4 Loads at interconnection points (WP3)

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 regarding the design load cases and design drivers for the gearbox and the drive train that should be considered, discussed in 4.1 and 4.2 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 [27]. 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 4-1 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)

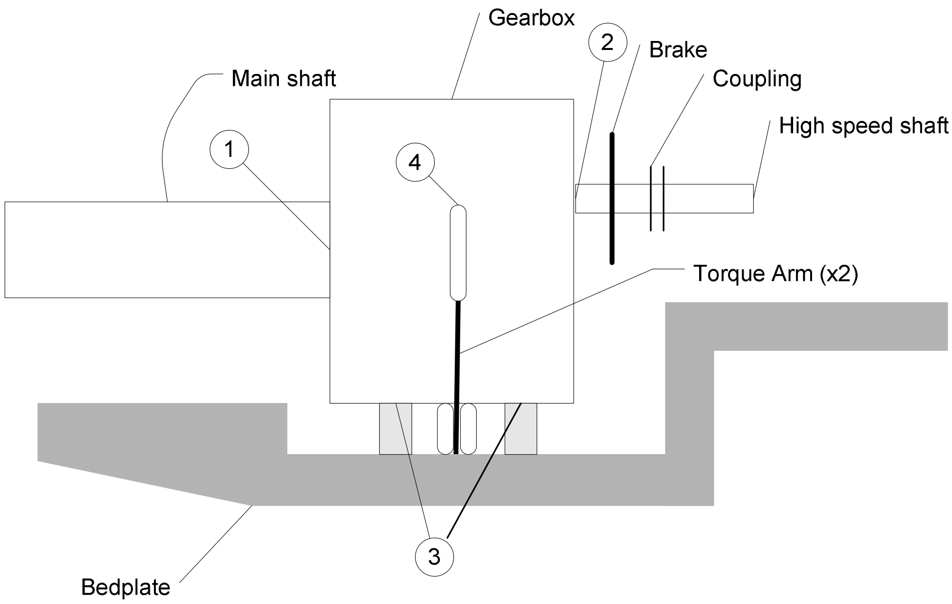


Figure 4-1: *Simplified sketch of gearbox layout without supported systems showing interfaces.*

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 4-2, 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 4-2

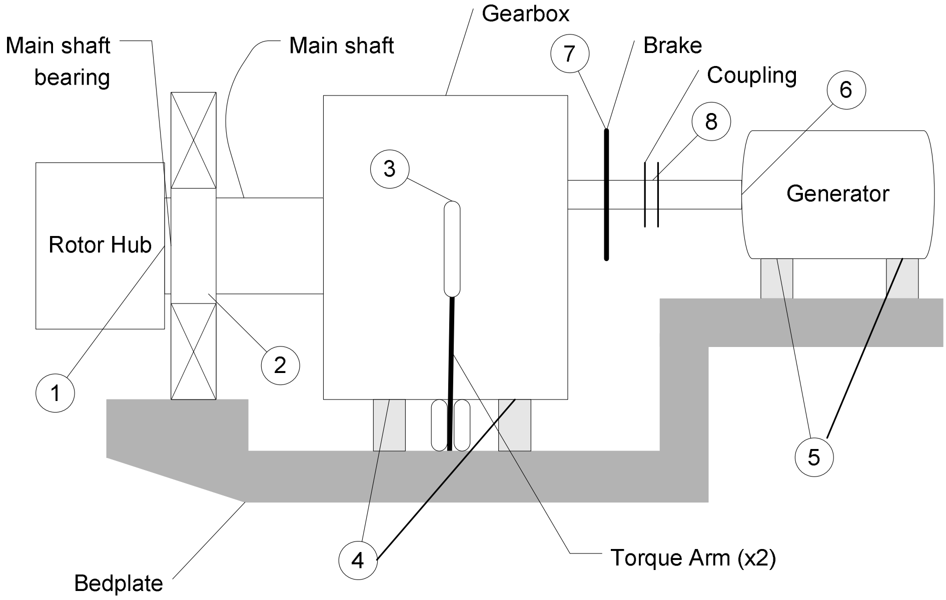


Figure 4-2: *Schematic of nacelle layout with Triple-point suspension showing interfaces.*

Other drive train configurations are discussed in [27].

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.

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 [27], 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.

4.5 Measurement definitions (WP3)

Based on the results for the loads transferred across the interfaces of the gearbox, Table 4.5 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 4.5: *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 [21] 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.

[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

- 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 4.5. 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 4.5, 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.

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.

4.6 Measurement and analysis results (WP4, WP5)

4.6.1 Introduction

In the PROTEST case study on drive train testing the focus is set on a measurement campaign that shall be used to verify the assumptions that have been used in the simulations for inertia, damping and stiffness. The measurement campaign therefore has to be set-up such that these simulations can be verified. As the employed models determine the measurement requirements, a completely new and more flexible approach is suggested in WP4 letting go of the current, less flexible, approach in the guidelines and standards but allowing to adapt measurements *to the model's needs.*

The idea of the six–step-approach as suggested in WP4 is to give guidance in testing the wind turbine component in question – the drive train in this case. Although component validation comprises regularly two parts, i.e. model and load validation, the main focus in this case study is placed on model validation. Hence, measurement requirements and evaluation procedures are sought that promise the highest efficiency in terms of delivering the crucial parameters needed for model validation.

It is desired that the simulation models describe the behavior of the turbine as precisely as possible. To achieve this, the simulation models need the feedback of the measurements to be tuned.

Any model can be as complex as the designer decides it to be, nevertheless as much as the number of degrees of freedom and parameters in them increases the more complex it will be to adjust the model to the real world (measurements).

There are several parameters widely used in all the commercial simulation tools such as FLEX5, SIMPACK, DRESP or Bladed that describe basically the dynamics of the drive train. Those parameters can be considered as the global descriptors of the drive train. Once obtained from the measurements, they can be used to tune the simulation tools.

4.6.2 Applying the six-step-approach

4.6.2.1 STEP 1: Identify critical failure modes

Employing the suggested six step approach in the drive train case study of WP5 no special failure mode is chosen. As the main task is to test the process of model design, the measurement set up, the data analysis process and to validate the designed models the work group has come to the conclusion that not only one failure mode / design load case is sufficient to serve all these targets.

4.6.2.2 STEP 2: Design models

Within WP5 three different models are used:

- FLEX5 model (figure Figure 4-3)
- SIMPACK model stage 1: similar to FLEX5 (Figure 4-4)
- SIMPACK model stage 2: sophisticated drive train model (Figure 4-5)

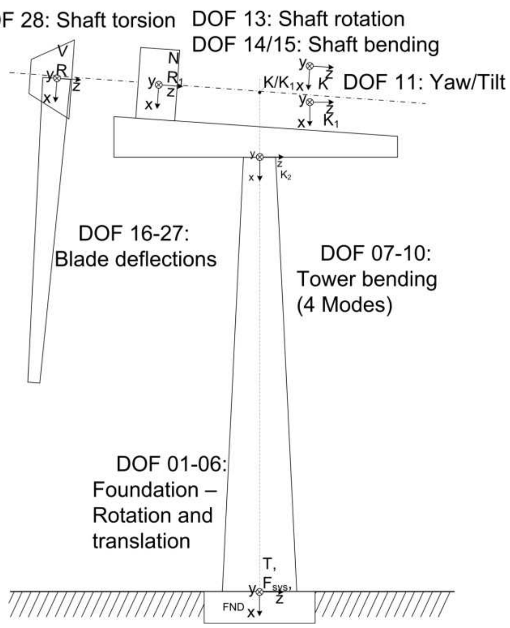


Figure 4-3: *Degrees of freedom of a FLEX5 model, with coordinate systems*

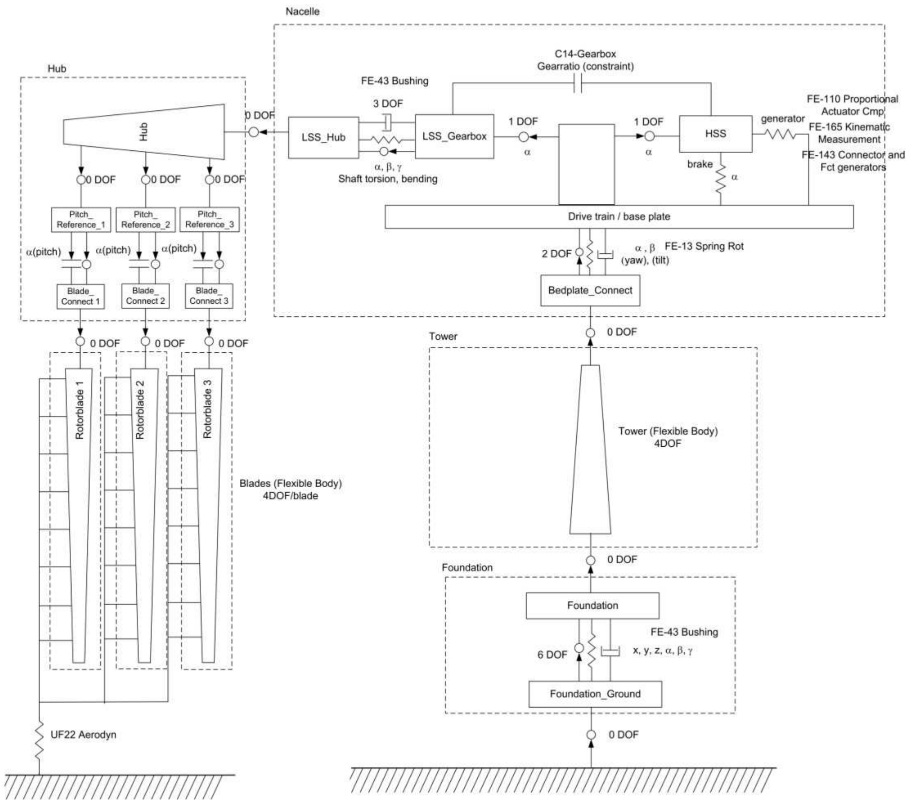


Figure 4-4: *Topology of the model similar to FLEX5's under SIMPACK*

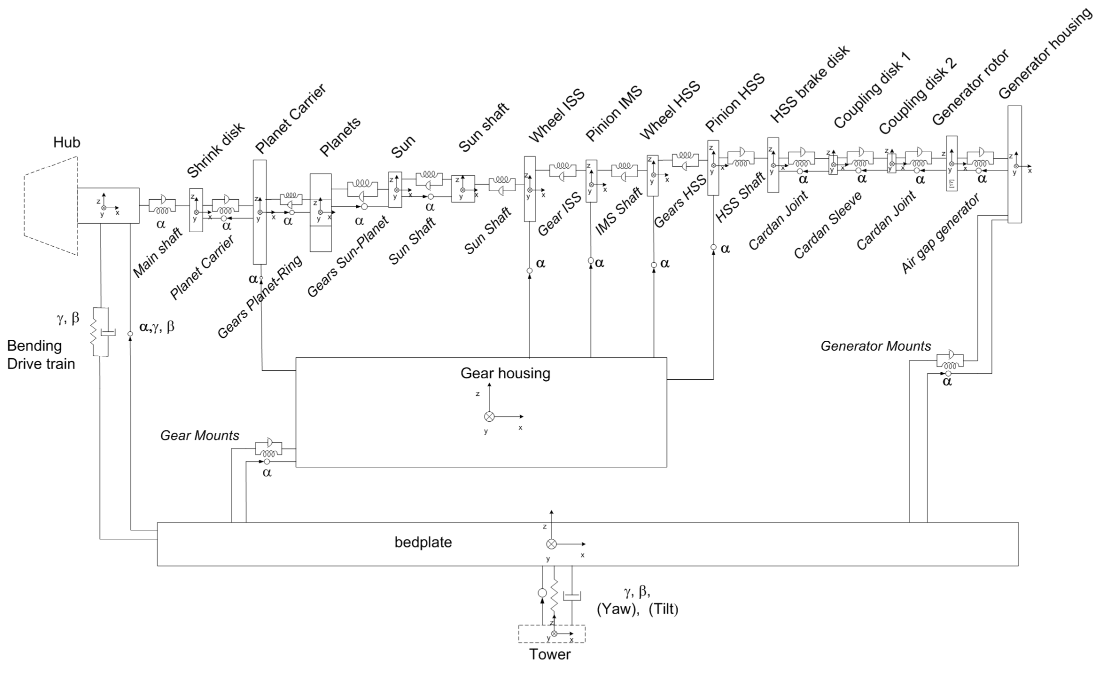


Figure 4-5: *Topology of the advanced torsional model of the drive train, modelled under SIMPACK*

The drive train of the first modelling stage (based on FLEX5 (Figure 4-3), see Figure 4-4) 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 modelling stage, the stage 2 (Figure 4-5).
It is possible to reach 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 mounting
- torsional stiffness of the generator mounting
- 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 bedplate. 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.

4.6.2.3 STEP 3: Run model for various DLCs

The validation of the model has been executed following two steps. First of all the drive train total stiffness has been compared between the models, and adjusted. Then a modal analysis has been made under SIMPACK, eigenfrequencies and eigenmodes have been compared with the FLEX5 results. In the next step simulations have been run with the FLEX5 model and the two SIMPACK models, with deterministic wind fields and time series of the three models have been compared. The same process has been made with turbulent wind fields. In that case, statistical values have been compared between the three types of model. Finally the extended measurements have been used to identify resonance frequencies and compute the corresponding stiffness to then adjust the model.

Comparison of the drive train stiffness (static)

The measurement of the total rotational stiffness can be realized under SIMPACK, by blocking the drive train on one side (for example the generator rotor) with a constraint and applying a given torque on the low speed shaft and measuring the resulting total torsional deformation angle.
The drive train total stiffness is calculated by:

$$stiffness = \frac{torque}{angle_{rel}}$$

with:

torque [kNm]: input torque on the low speed shaft (static loading)

$angle_{rel}$ [rad]: rotation angle of the low speed shaft, relatively to the high speed shaft.

If there is a noticeable deviation between the different models, the origin should be investigated and the models eventually adjusted.

Comparison in the frequency domain

Along with the investigations in the time domain the eigenfrequencies and the corresponding eigenshapes have also been analyzed. It is particularly relevant to prevent resonance during the design process, by e.g. selecting the appropriate components, changing the stiffness or inertias or avoid operating ranges where excitations are ought to meet an eigenfrequency.

It is also a quite straightforward method to validate or tune a model. In our case it constitutes the next step to validate our new model and also to define the so-called “target frequencies” for the measurements, which are the frequencies where more attention should be paid.
The first step in the validation of the new model is to compare the eigenfrequencies and eigenmodes of the new SIMPACK models and the FLEX5 results.

Comparison in the time domain

Deterministic wind field

Comparison of the time signals

The comparison of the output signals of the new model and the benchmark (FLEX5) model have the purpose to validate the whole aeroelastic model, with the consideration of the aerodynamic forces, the controlling (pitching), the generator and eventually the braking forces. The comparison of the results of the modal analyses indeed only enables to check the consistency of the structure.
Since the benchmark model (FLEX5) uses turbulent wind fields produced by a different turbulent wind field generator than SIMPACK, the resulting time signals from a simulation can only be realized using deterministic wind fields, e.g. for the load case DLC1.0 defined by the GL Guideline.

As an example GL-DLC 1.0 has been simulated at 5 different wind speeds (Table 4.6). It uses the BEM method with the correction of tip losses according to Prandtl, considers dynamic stall (Beddoes model) and tower shadow.
The rotor is imbalanced, by increasing the weight of blade 3 by 1%. Aerodynamic imbalance is also considered by reducing the pitch angle of blade 2 by 0.3° and by increasing the pitch of blade 3 by 0.3°.
In our case, the simulation times were shorter than the 600s foreseen in the GL-Guideline: 100s. It is indeed sufficient for validation.
The following wind speeds have been used for the comparison FLEX5 and the stage 01 model of SIMPACK.

Table 4.6 *Design load cases used for the comparison of FLEX5 and the stage 1 SIMPACK model.*

GL-DLC 1.0	NWP (Normal Wind Profile)	Average Wind Speed at hub height: vhub[m/s]
	T_DLC1P0_VN1	5
	T_DLC1P0_VN2	11
	T_DLC1P0_VN3	15
	T_DLC1P0_VN4	19
	T_DLC1P0_VN5	20

Turbulent wind field

The same analysis for the time signals of FLEX5 and SIMPACK were made for a turbulent wind field. The comparisons of FLEX5 and the stage 1 SIMPACK model are shown for DLC1P2_VN1 (7m/s) in Figure 4-6 for the rotor speed and the electrical power.

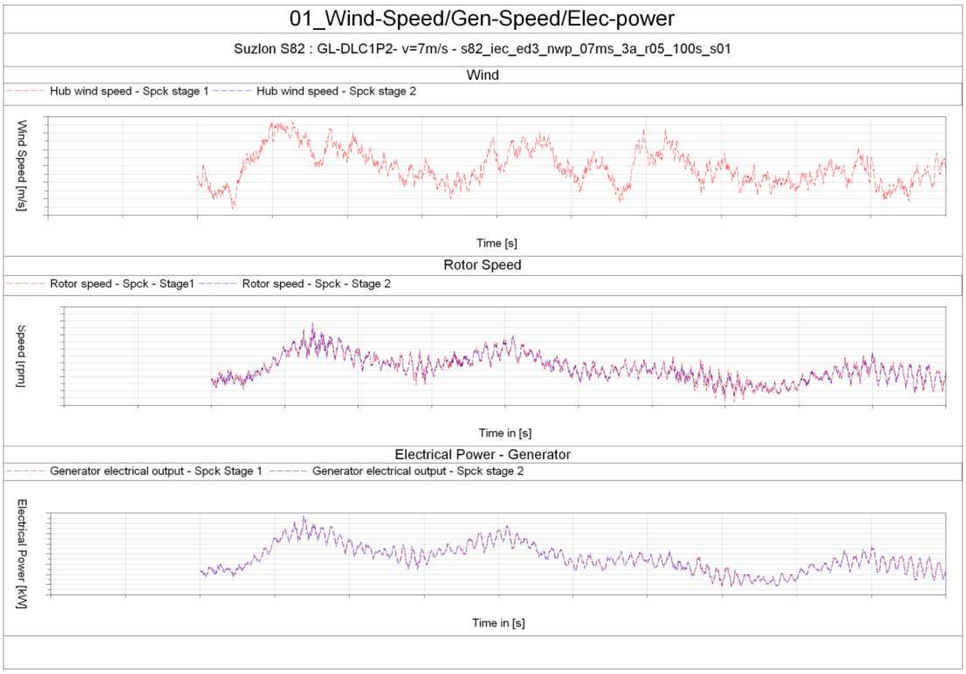


Figure 4-6: *Comparison of the FLEX5 and SIMPACK stage 1 models with design load case GL_DLC1P2_VN1 for wind speed, rotor speed and electrical power.*

4.6.2.4 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.

Table 4.7: *Uncertainty of the model input parameters*

	Component	**Uncertainty**
Mass	Blades	Max +-3% deviation
	Components Drive Train	Max +-3%
	Tower	Max +-3%
Inertias	Components Drive Train	?
Stiffnesses	Blades	?
	Tower	?
	Drive Train shafts	?
	Gear mesh Stiffnesses	?
	Gear Box mounting	?
Damping values	Blades	?
	Tower	?
	Drive Train shafts	?
	Gear mesh Stiffnesses	?

Table 4.7 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, 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 in e.g. the blade masses are already taken into consideration in the simulation. For example a blade mass unbalance has been added by assuming 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. Analyze the results (e.g. capture matrix, or the outputs of the Rainflow Count)
4. Judge what uncertainty is acceptable.

4.6.2.5 STEP 5: Design measurement campaign to verify models and quantify parameters

According to the experiences in load testing and evaluations in the context of turbine certification a measurement campaign has been planned to verify models and quantify parameters. The scheme [33] encompasses four steps as given in Table 4.8.

Table 4.8: *Planning of a measurement campaign*

step	Quantity to Check	Example for Methods	Objective of Validation Step
1	• Documentation • Selected Time Series	• Comparison of model data against weighing log • Spectral analysis of selected time series for various operational states (e.g. in partial and full load)	• Main structural properties like masses, stiffnesses, eigenfrequencies and coupled modes
2	• Characteristic Curves	• Visual comparison of curves of operational parameters (e.g. speed, power) and loading for several environmental conditions	• Validation of basic control characteristics and rotor aerodynamics as well as mechanical and electrical parameters (e.g. losses)
3	• Time Series of various operational states, like power production start stop emergency stop	• Visual comparison of data in time and frequency domain • Check of statistical properties of data • Analysis of decay rates of oscillations during stopping procedures	• Dynamic behaviour all important and assessable operational states with focus on aerodynamic mode, controller model and actuator models • Structural and aerodynamic damping
4	• Post-Processed Data	• Comparison of loading spectra like • rainflow distribution • load duration distributions • damage equivalent loads	• Final check of turbine behaviour and dynamic properties • Check of all previously performed validation steps

For the design of the measurement campaign and the subsequent data processing the simpler models FLEX 5 and SIMPACK stage 1 have been considered. As discussed before in these drive train models the input of the overall torsional stiffness and damping for the drive train and its transmission ratio are the required parameters. The inertias of all the rotating parts in the drive train except for the rotor blades and the hub are modelled in a single “HSS” body (see Figure 4-42). Hence, for measurement data evaluation the drive train is thought of as a single mass rotatory oscillating system according to the following Figure 4-7:

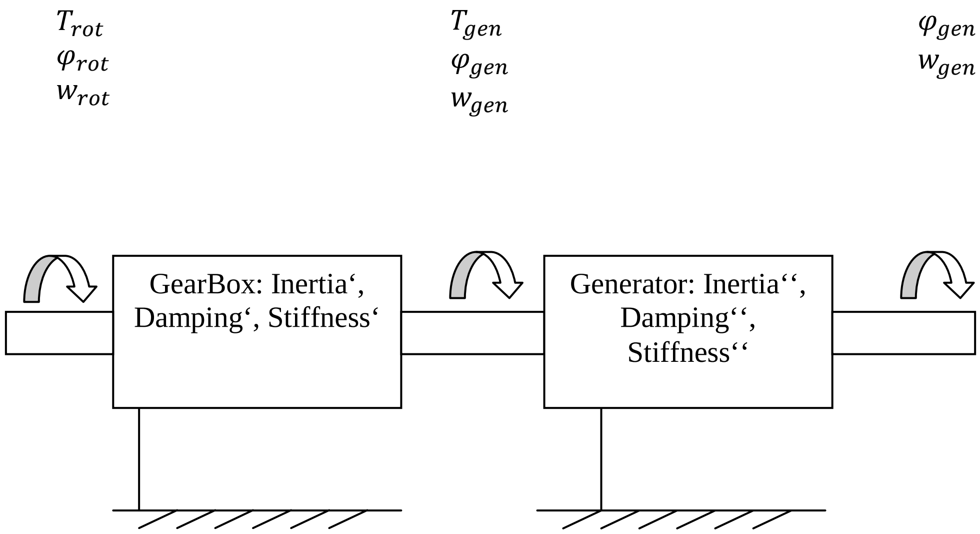


Figure 4-7: *Schematic concept of the drive train*

For this concept the equation of motion of the drive train is given as:

$$T_{rot}\big|_{high} = \left[\frac{\partial w_{rotor}\big|_{high}}{\partial t} - \frac{\partial w_{high}}{\partial t}\right] \cdot Inertia + \left[w_{rotor}\big|_{high} - w_{high}\right] \cdot Damping + \left[\varphi_{rotor}\big|_{high} - \varphi_{high}\right] \cdot Stiffness \quad (1)$$

Where:

$T_{rot}\big\|_{high}$	: is the mechanical torque in the rotor, at the low speed shaft converted to the high speed side using the gear ratio and by making assumptions for gearbox and generator losses
$\frac{\partial w_{rotor}\big\|_{high}}{\partial t}$	: 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

Measurement Setup

The measurement campaign setup to verify the FLEX5 and SIMPACK stage 1 models and quantify model parameters consisted of load measurements according to IEC 61400-13 (1. ed. 2001) [21] and additional measurements for the experimental validation of drive train overall stiffness, damping and inertia. To this end measurements of at least rotational speeds and torques at input and output shaft of the gearbox where considered necessary. More signals may become necessary when model complexity increases as discussed before.

A detailed description of the measurement setups for several campaigns that were carried out between May 2008 and June 2010 can be found in [34]-[37]. Beside the standard IEC 61400-13 meteorological, operational and mechanical load measurement signals (s.a. global blade, rotor and tower loads) the measurements include a number of signals with special relevance for drive train load assessment:

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

In the context of drive train model validation and in the attempt to quantify the relevant parameter stiffness, damping and inertia of the drive train the focus has been placed on the rotational speed, angle and torque measurements at the main shaft (= Low Speed Shaft) and the generator shaft (= High Speed Shaft).

Speed measurements

For measurements of the rotor shaft, intermediate shaft and generator shaft speeds incremental encoder signal conditioning modules have been used. Such modules are capable of high precision speed measurements using either event counting or pulse width techniques applied on a pulse train signal generated by a suitable sensor. The internal processor samples on the incremental encoder input signal at very high frequencies (e.g. 32 MHz for the used model INC4 of imc [38]). This high sampling allows detection of the upward-slope of a pulse train signal as it is typically produced from an angular encoder sensor or another pulse generating sensor. Depending on signal quality and settings of the comparator at the input stage of the INC4 the pulses are detected and counted in event counting mode or are evaluated for the time difference to the previous detected upward slope in the pulse width mode. In case of many and heavy disturbances false transitions may be counted as they trigger the counter/comparator leading to false speed signals in terms of spikes or even drifts in the recorded speeds. In fact quality and stability of the pulse train signal turned out to be the major problem in this application.

The pulse trains where created by

- high resolving optical laser pick-ups looking at bar codes applied to the shafts covering the full circumference at a given cross section
- proximity probes that deliver a voltage signal when metallic objects pass by the sensor at short distance of a few mm.
- industrial incremental encoder sensors

Angle increment measurements

To avoid error accumulation due to integration of troubled rotation speed signals it is recommended to also measure the rotation angle of the shafts directly at a sufficiently high resolution. Typically such incremental encoder sensors come with various numbers of increments per revolution giving a corresponding resolution of the rotation angle. In the case study incremental encoder featuring 60 increments on the high speed shaft (make: KTR[39]), 256 increments on the intermediate shaft (make: Baumer Thalheim [40]) and 4096 increments on the low speed shaft (make: TWK [41]) have been used with a resolution of less than 0.1 degrees at the low speed shaft.

Torque measurements

The torque on the main shaft (low speed shaft) and on the generator shaft (high speed shaft) have been measured using in situ bonded strain gage sensors on the LSS and on the HSS between gearbox and brake disk as well as a encased torque and speed measurement shaft.
Based on the long standing experience of DEWI the accuracy of the nominal calibration for rotor torque measurements by means of in situ bonded strain gage sensors is very much depending on the geometry of the measurement cross section and may be at times considered to be problematic.

In the case study the main shaft torque has been measured using a standard set-up of strain gages on a cross section directly in front of the adapter sleeve connecting the main shaft to the gearbox. This location normally is ideal for torque measurements as the shaft comes close to the assumption of a rotating beam of constant diameter as assumed in theory.
Nevertheless, the specified losses of the generator and the mechanical losses of the drive train have been used for converting the electrical output into mechanical power at the main shaft and at the high speed shaft. Using mean values of the electrical power output, the gearbox and generator efficiencies and mean values of the measured rotational speeds at the low and high speed shaft corresponding mean values of the LSS-torque and the HSS-torque have been derived in each 10 minutes file. These derived torque values have been applied for the purpose of scaling the values recorded from the strain gage sensors.

Campaigns

The measurements have been realized on a SUZLON S82 1500kW wind turbine situated nearby Sankaneri, Tamil Nadu in India (Figure 4-8).

Figure 4-8: *SUZLON S82 1500kW wind turbine situated nearby Sankaneri, Tamil Nadu in India*

The measurements have been taken in manual campaigns for recording time series of transient events and from steady state operation. See Table 4.9 for an exemplary list of measurement load cases (MLCs) as applied during manual testing. In monitoring campaigns during normal power production operation a population of data sets is recorded for normal operation – see capture matrix Table 4.10. While in manual testing the main focus was placed on time series recording and analysis the monitoring campaign focuses on assessment of long term statistical (processed) data s.a. scatter plots, frequency distributions, load duration distributions and load spectra. There have been several such campaigns between May 2008 and June 2010 [34]-[37].

A list of measurement load cases has been defined and carried out during manual testing at site:

Table 4.9: *Exemplary list of measurement load cases MLC*

NTA	Normal start-up (+ grid connection)
NTB	Normal stop (run to pause)
NTD	Pause (idling at low speed => 2 RPM)
NTE	Idling at high speed => 16 RPM ("waiting for wind without grid connection")
NTF	Stand still without rotor lock
NTH	Constant speed at X Geno-RPM - idling state (=> X is in the range [100 - 1750 RPM] / speed ramp up)
STB	E-stop without mechanical brake disk (= equivalent to grid loss)
STC	E-stop after activation of overspeed guard when generator is NOT connected (e.g. set blade pitch angle to 20 degrees and wait)
STD	E-stop after activation of overspeed guard during power production
STF	Constant speed at Y Geno-RPM - power production (=> Y is in the range [1500 - 1600 RPM])
STG	Slow reverse rotation

Table 4.10: *Capture matrix of recorded data sets in a monitoring campaign*

Capture matrix

Windturbine:	**SUZLON**	**Data:**	**10.12.2008 19:50-14.12.2008 07:10**
Wind speed bin size (x-axis):	**1 m/s**		
Turbulence bin size (y-axis) :	**2%**		

V(m/s)	0	1.5	2.5	3.5	4.5	5.5	6.5	7.5	8.5	9.5	10.5	11.5	12.5	13.5	14.5	15.5	16.5	17.5	18.5	19.5	20.5	21.5
I(%)	1.5	2.5	3.5	4.5	5.5	6.5	7.5	8.5	9.5	10.5	11.5	12.5	13.5	14.5	15.5	16.5	17.5	18.5	19.5	20.5	21.5	22.5
0- <3																						
3- 5						5	7	7	22	2												
5- 7					16	10	16	12	14	3												
7- 9					15	17	11	18	16	3												
9- 11				1	6	13	25	21	14													
11- 13				1	7	14	22	27	13	1												
13- 15				1	5	16	9	13	8													
15- 17					4	6	4	2	2													
17- 19					1	5		1														
19- 21						1	2															
21- 23						1																
23- 25																						
25- 27																						
27- 29																						
>29																						
	0	0	0	3	54	88	96	101	89	9	0	0	0	0	0	0	0	0	0	0	0	0

Datasets :	**440**
Mean Turbulence	**9.78**

4.6.2.6 STEP 6: Process measurement results

The processing of the measured data is accomplished in the frequency and in the time domain. Different approaches are used to estimate the model parameters from the measurements.

Natural frequency analysis

For a first observation a FFT analysis is done on time sequences of reasonable length to get an overview about the frequencies which are included in the measured signals. The analysis is applied to signals from different normal production modes producing a power of 42kW, 200kW, 376kW, 624kW and 984kW and for a resonance case producing 517kW for the sensor measuring the high speed shaft torque (Figure 4-9). Additionally the resulting frequencies from the FFT analysis are compared to the excitation frequencies of the generator and the rotor and to the drive train eigenfrequency derived from the modeled drive train. Analyzed frequencies which tend to be a multiple of the rotation speed should be identified as excitation frequencies while the ones which are approximately constant over the operation range or change with the rotational speed, but are not a multiple of the rotational speed (so not tending to zero for 0 rpm), should be suspected to be eigenfrequencies.

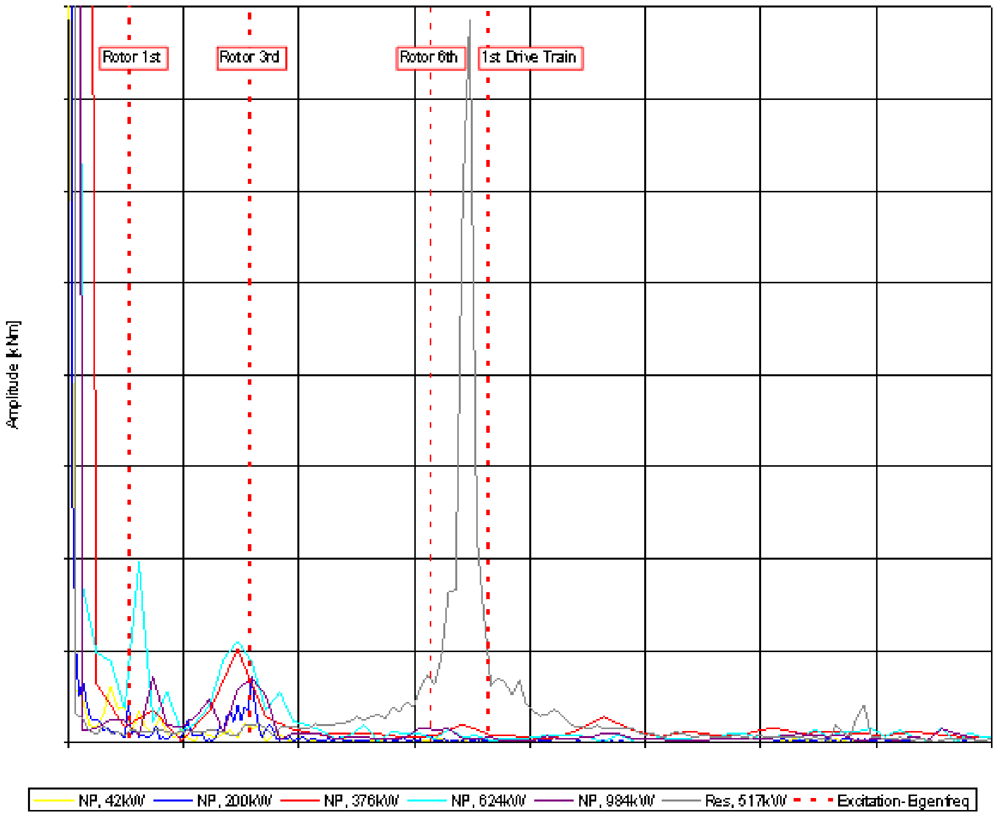


Figure 4-9: *FFT analysis results for low frequencies for the normal production and resonance cases and the excitation eigenfrequencies.*

Further to investigate the presence of torsion eigenfrequencies and estimate their corresponding frequency for the drive train Campbell representations of the frequency-time domain using a short FFT analysis of the torsion related measurement signals are used (Figure 4-10). Based on a visual inspection of the Campbell plots it can be decided around which frequency an eigenfrequency is suspected to be present in the drive train.

Campbell representations reveal increasing frequency responses during speed variations excited by, for example, gear meshing or shaft rotations. The amplitude of the frequency response increases if one of these excitation frequencies coincides with an eigenfrequency of the drive train.

Assuming the eigenfrequency is within the frequency measurement range of the sensor, whether or not this increase is captured by a sensor depends strongly on the sensor location with respect to the corresponding eigenmode. The amount of increase when exciting an eigenfrequency also depends on the amount of damping which can be actively controlled or inherently present, and on the load. For the exciting load it holds that high loads yield better excitations and thus better responses since the signal to noise ratio increases.

At dedicated test facilities a controlled speed ramp-up or ramp-down can be applied to the drive train while maintaining controlled and constant load conditions. During field tests the load and speed are affected by the wind conditions and are difficult to control accurately. Also, the S82 turbine is a constant speed turbine such that speed ramp-ups or ramp-downs are always during no load conditions. For these reasons the interpretation of Campbell representations of measured field data is even more subject to discussion and personal interpretation compared to Campbell representations of measurement data obtained under controlled test conditions.

Nevertheless, for the PROTEST project a selection of the available measurement data is made based on available speed variations in the data. For this selection Campbell representations of the torsion related measurement signals are made and possible torsion eigenfrequencies are indicated.

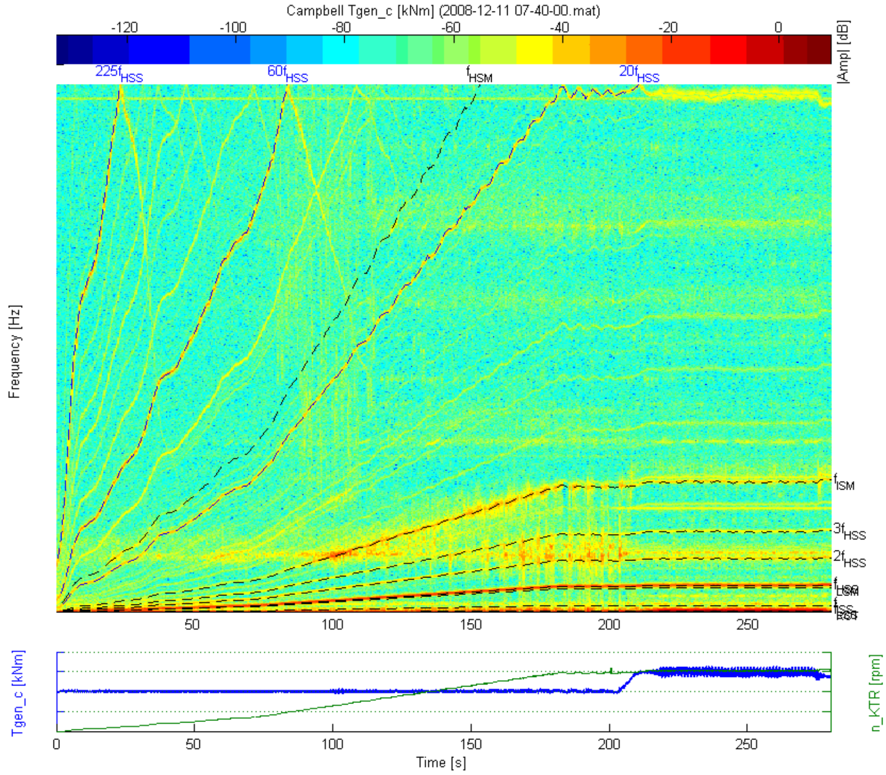


Figure 4-10: *Example for Campbell diagram for the calibrated generator torque signal (Tgen_c)*

Determination of overall drive train stiffness and ratio of inertia to damping

The approach for determination of drive train stiffness, damping and inertia assumes that in general any measured data will contain this information. To extract this information from the measured data a statistical or deterministic approach can be chosen to come up with estimates of inertia, damping and stiffness. In the deterministic approach a suitable set of equations is derived and solved once, using measurement data with suitable operating condition. In the statistic approach the intention is to draw advantage from solving the relevant equations numerous times for the measurement data of suitable operating conditions. This way sets of solutions will be derived for each (ten-minute-) time history that is processed. To find the most likely solution, the centered value of the frequency distribution of all solutions or the median value of all solutions of one or more 10-min-data sets will be determined. Different approaches are used to estimate the model parameters from the measurements.

Deterministic approach

Stiffness

The stiffness of the drive train can be estimated from measurement data considering assumptions in drive train dynamics (Figure 4-11). The inertia of the rotor is high compared to the drive train and the stiffness of the main shaft is known to be high.

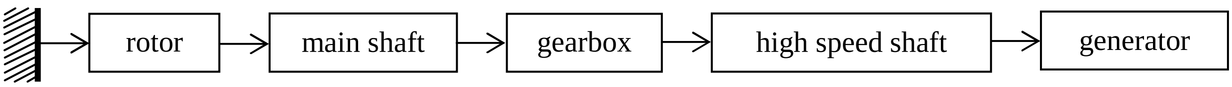


Figure 4-11: *Assumption of drive train model.*

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 (drive train twist angle).

$$stiffness = \frac{torque}{angle_{rel}} \quad (2)$$

Damping
To estimate damping values (logarithmic decrement) measurement data is investigated for events where the drive train is excited to oscillate at its eigenfrequency. A damped oscillation is observed. These events are for example produced during emergency stops. The logarithmic decrement is estimated by the natural logarithm of the quotient of two adjacent amplitudes:

$$\Lambda = \ln\left(\frac{u(t_i)}{u(t_{i+1})}\right) \quad (3)$$

Statistic approach

Stiffness

In stationary operation near the 1st drive train resonance frequency it is assumed that the effects of inertia and damping are small and can be neglected when looking at consecutive periods of steady state operation. This means that the quasi-steady state variation of the torque will be determined by drive train stiffness and the differential angle of both shaft ends. This simplifies the equation of motion (1) to the following minimal expression:

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

In the campaign of December 2008, more precisely the file 2008-12-10 13-42-53, the angles of the shafts have been derived based on the measured speeds. These data were used to obtain the following set of solutions.

The first thing to do is to obtain the torque variation range ($T_{rotor}|_{range}$) through a given period of time. This time must be long enough to allow the measured torque signal to include at least one full swing from the minimum to the maximum torque of that quasi-stationary oscillation. The time period has been chosen to 5 seconds which approximately is 1.3 times longer than the period of one low speed shaft revolution and is several times longer than the drive train natural frequency.

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

Solving equation (4) provides a solution for the overall stiffness for every 5 seconds or 120 solutions in for a ten minute time series in which the drive train natural frequency is excited according to the initial assumption for equation (4) to hold true. Figure 4-12 shows the median value and frequency distribution for stiffness solutions.

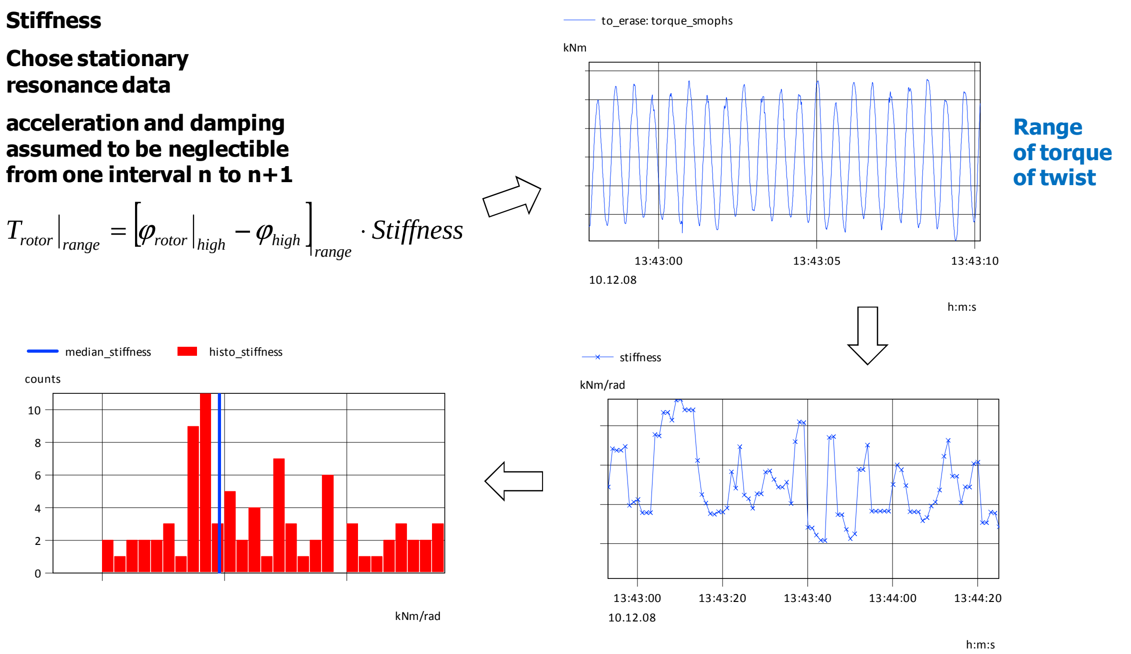


Figure 4-12: *Median value and frequency distributions for stiffness solutions*

Ratio of inertia to damping

Knowing the stiffness, the equation of motion (1) can be evaluated for the ratio of inertia to damping. Assuming the resonance quasi-steady state operation as before the left hand side of equation (1) becomes zero and equation (5) can be solved:

$$0 = \left(\left[\frac{\partial w_{rotor}\big|_{high}}{\partial t} - \frac{\partial w_{high}}{\partial t} \right] \right)\Bigg|_{range} \cdot Inertia + \left(\left[w_{rotor}\big|_{high} - w_{high} \right] \right)\Big|_{range} \cdot Damping \qquad (5)$$

Again a frequency distribution of the set of solutions can be obtained (Figure 4-13):

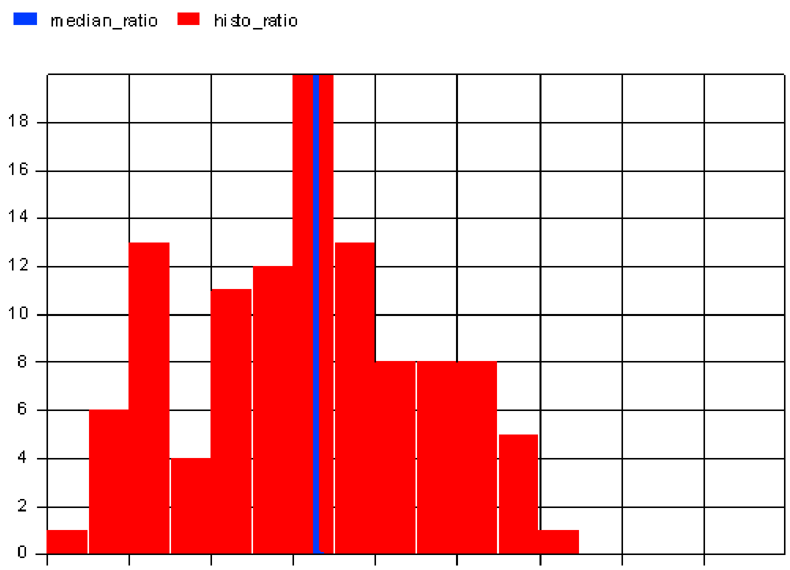


Figure 4-13: *Median value and frequency distributions for solutions of ratio of inertia to damping.*

Three Methods to determine inertia and damping

Three different methods to determine damping and inertia have been applied.

A) Time Method

This approach requires stiffness to be known and employs determination of the ratio of inertia to damping in a second step to solve the complete system. First stiffness and ratio of inertia to damping are determined from a special drive train resonance data file (2008-12-10 13-42-53.dat).

Applying the ratio of inertia to damping in equation (6) allows solving for one of the unknowns in time domain sample by sample. This process generates a solution for each time step / sampling step. For example in data sets with a sample rate of 200 Hz as used in the case study measurements some 120000 solutions are calculated. Using a statistical approach on all results will deliver a frequency distribution of all solutions in a data set and a median value. This method is denominated "time" for easy recognition.

$$\left(T_{rot}\right)\Big|_{real\_sample} = \left(\left[\frac{\partial w_{rotor}\big|_{high}}{\partial t} - \frac{\partial w_{high}}{\partial t}\right]\right)\Bigg|_{real\_sample} \cdot Inertia + \left(\left[w_{rotor}\big|_{high} - w_{high}\right]\right)\Big|_{real\_sample} \cdot Damping + \left(\left[\varphi_{rotor}\big|_{high} - \varphi_{high}\right]\right)\Big|_{real\_sample} \cdot Stiffness$$

(6)

B) Differential Method

This approach can be used when stiffness and the ratio of inertia to damping have not been obtained from a resonance test.

A time step δt used in this method is defined as the average time for the rotor to complete one rotation in a selected 10-min-data set. The twist of drive train is assumed to be constant throughout one time step δt. For each of the time steps δt the min-max-ranges of the changes (δx/δt) in torque, in angular acceleration of the drive train twist angle, in rotational speed of the drive train twist are obtained.

The differential of (6) in time will result in equation (7) and a system of two equations of type (8,9) are set up for two time steps δt. This system can be solved for the two unknowns.

This Method is denominated "differential" for easy recognition.

$$\frac{\partial T_{rot}}{\partial t}\bigg|_{range} = \left[\frac{\partial^2 w_{rotor}\big|_{high}}{\partial t^2} - \frac{\partial^2 w_{high}}{\partial t^2}\right]\Bigg|_{range} \cdot Inertia + \left[\frac{\partial w_{rotor}\big|_{high}}{\partial t} - \frac{\partial w_{high}}{\partial t}\right]\Bigg|_{range} \cdot Damping + \left[\frac{\partial \varphi_{rotor}\big|_{high}}{\partial t} - \frac{\partial \varphi_{high}}{\partial t}\right]\Bigg|_{range} \cdot Stiffness \qquad (7)$$

For reasons of simplification this is:

$$K = \frac{\partial^2 B}{\partial t^2} \cdot Inertia + \frac{\partial B}{\partial t} \cdot Damping + B \cdot Stiffness$$

$$A = \frac{\partial^2 B}{\partial t^2} \cdot Inertia + \frac{\partial B}{\partial t} \cdot Damping \qquad (8,9)$$

$$Damping = \frac{A_2 \cdot \frac{\partial^2 B_1}{\partial t^2} - A_1 \cdot \frac{\partial^2 B_2}{\partial t^2}}{-\frac{\partial^2 B_2}{\partial t^2} \cdot \frac{\partial B_1}{\partial t} + \frac{\partial^2 B_1}{\partial t^2} \cdot \frac{\partial B_2}{\partial t}}$$

$$Inertia = \frac{A_1 - Damping \cdot \frac{\partial B_1}{\partial t}}{\frac{\partial^2 B_1}{\partial t^2}} \qquad (10,11)$$

C) The Constant Method

This approach can be used when the stiffness is known but not the ratio of inertia to damping.

This Method applies a system of two equations (12,13) based on equation (6), but instead of solving sample by sample (as in Method A) time steps δt are used as described in the Method B, i.e. average time elapsed for one complete rotation of the rotor. For each of the time steps δt the min-max-ranges of the changes (Δx) in torque, in acceleration of the rotor azimuth angle, in rotational speed of the drive train and in the rotor azimuth angle are obtained. This method is denominated “constant” for easy recognition.

$$\left(\left(T_{rot}\right)\Big|_{range}\right)_{period\_i} = \left(\left(\Delta\frac{\partial w}{\partial t}\right)\Big|_{range} \cdot Inertia + \left[\Delta w\right]_{range} \cdot Damping + \left(\Delta\varphi\right)\Big|_{range} \cdot Stiffness\right)_{period\_i}$$

(12, 13)

$$\left(\left(T_{rot}\right)\Big|_{real\_sample}\right)_{period\_i+1} = \left(\left(\Delta\frac{\partial w}{\partial t}\right)\Big|_{range} \cdot Inertia + \left[\Delta w\right]_{range} \cdot Damping + \left(\Delta\varphi\right)\Big|_{range} \cdot Stiffness\right)_{period\_i+1}$$

Solving for Inertia and Damping

The introduced methods have been applied to solve for inertia and damping. For all methods the same signal treatment was applied to the data (filtering). Due to the computational effort the processing has been limited to 66 files covering the wind speed range from 5 m/s bin to 10 m/s (Figure 4-14).

As solving for inertia and damping is an automatic process applied on any data, some of the data can include zero or non possible solutions. Figure 4-15 shows the initial sets of solutions for inertia and damping for one single 10-min-time series in red colour. The remaining set of solutions avoiding implausible values is shown in blue colour. For each of the applied methods the set of solutions is reduced considerably after exclusion of implausible data.

After exclusion of implausible solutions the above described methods A, B, C deliver frequency distributions of solutions for each processed 10-min-data set (Figure 4-16, Figure 4-17 and Figure 4-18). For each such set of solutions the median value is determined and plotted versus power (Figure 4-19, Figure 4-20 and Figure 4-21).

Capture matrix

Windturbine:	**Protest**
Wind speed bin size (x-axis):	**1 m/s**
Turbulence bin size (y-axis) :	**2%**

V(m/s)	0	1.5	2.5	3.5	4.5	5.5	6.5	7.5	8.5	9.5	10.5	11.5	12.5	13.5	14.5	15.5	16.5	17.5	18.5	19.5	20.5	21.5	22.5	23.5	>24.5
I(%)	1.5	2.5	3.5	4.5	5.5	6.5	7.5	8.5	9.5	10.5	11.5	12.5	13.5	14.5	15.5	16.5	17.5	18.5	19.5	20.5	21.5	22.5	23.5	24.5	V out
0- <3																									
3- 5						2	7	4	15	1															
5- 7						1	4	3	3																
7- 9						2		2	2																
9- 11					1	2	1	2																	
11- 13					1	2	2	2																	
13- 15					2	1																			
15- 17					1	1	1																		
17- 19					1																				
19- 21																									
21- 23																									
23- 25																									
25- 27																									
27- 29																									
>29																									
	0	0	0	0	6	11	15	13	20	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0

Datasets :	**66**
Mean Turbulence	**3.20**

Figure 4-14: *Capture matrix with the data used for the evaluation of the inertia, period from 2008-12-11 21-40-00 to 2008-12-12 08-20-00, including the data set 2008-12-13 22-10-00*

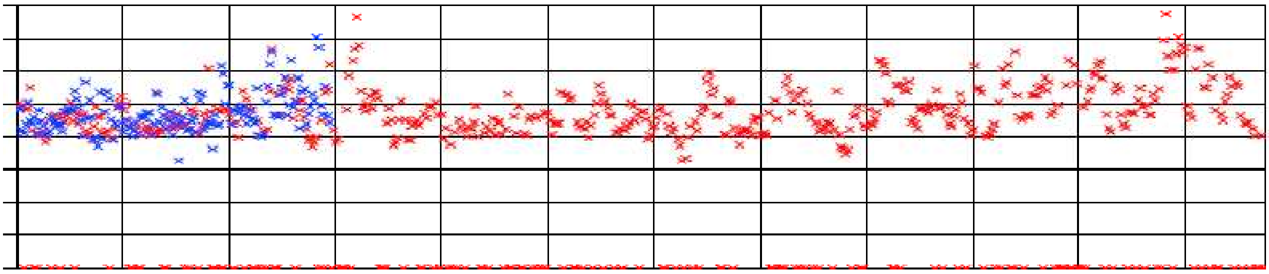


Figure 4-15: *Set of solutions for one data file before (red) and after exclusion (blue) of implausible values*

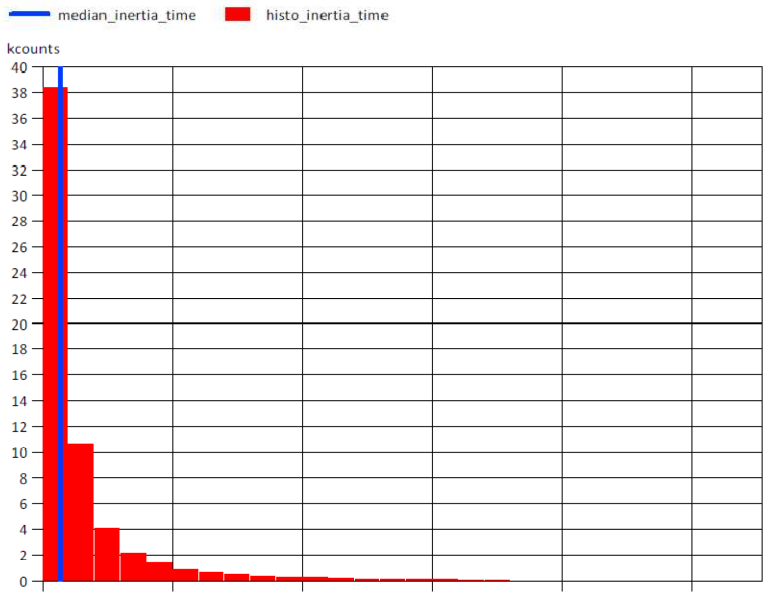


Figure 4-16: *Set of solutions for one 10-min-time series for "time" method A*

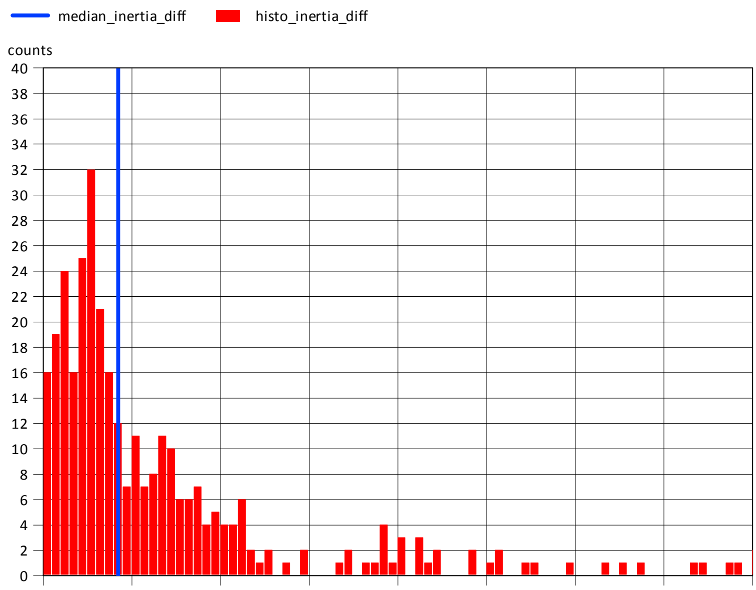


Figure 4-17: *Set of solutions for one 10-min-time series for "differential" method B*

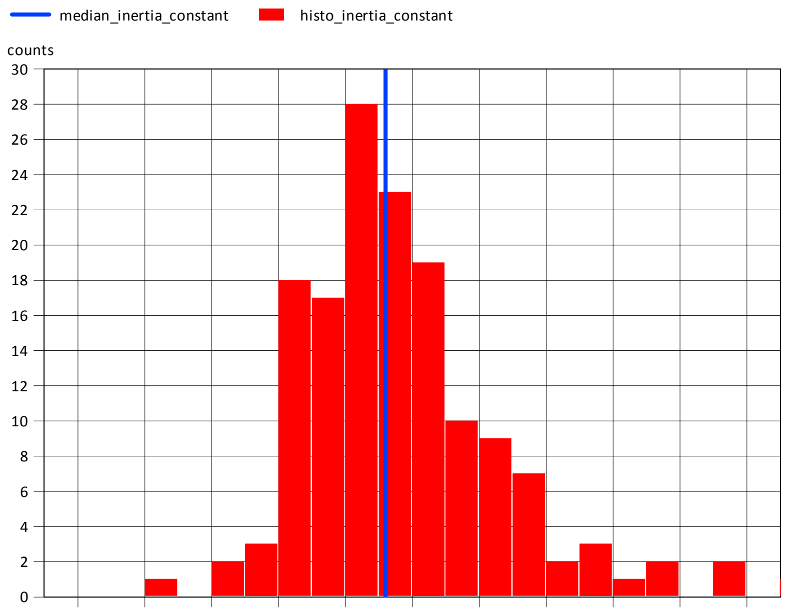


Figure 4-18: *Set of solutions for one 10-min-time series for “constant” method C*

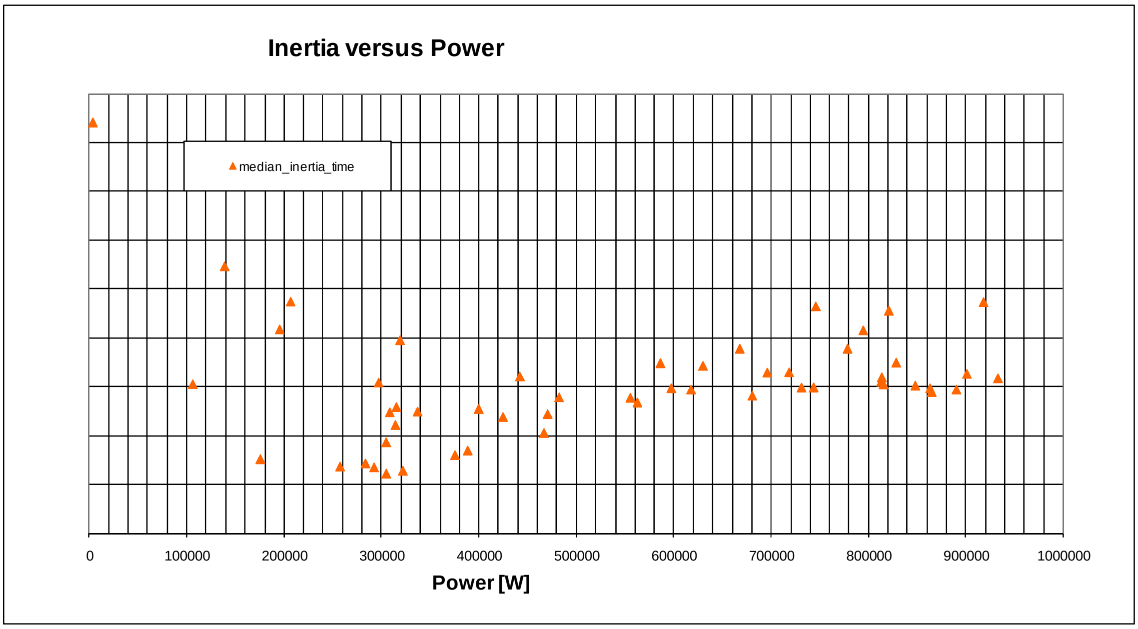


Figure 4-19: *Median values of all sets of solutions obtained for “time” method A*

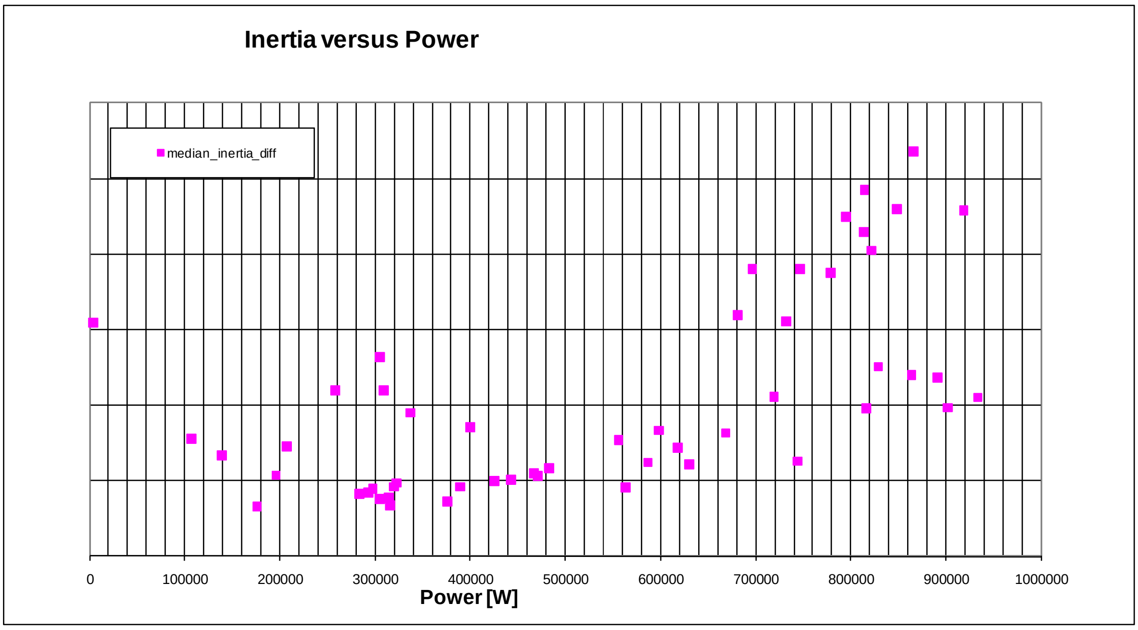


Figure 4-20: *Median values of all sets of solutions obtained for "differential" method B*

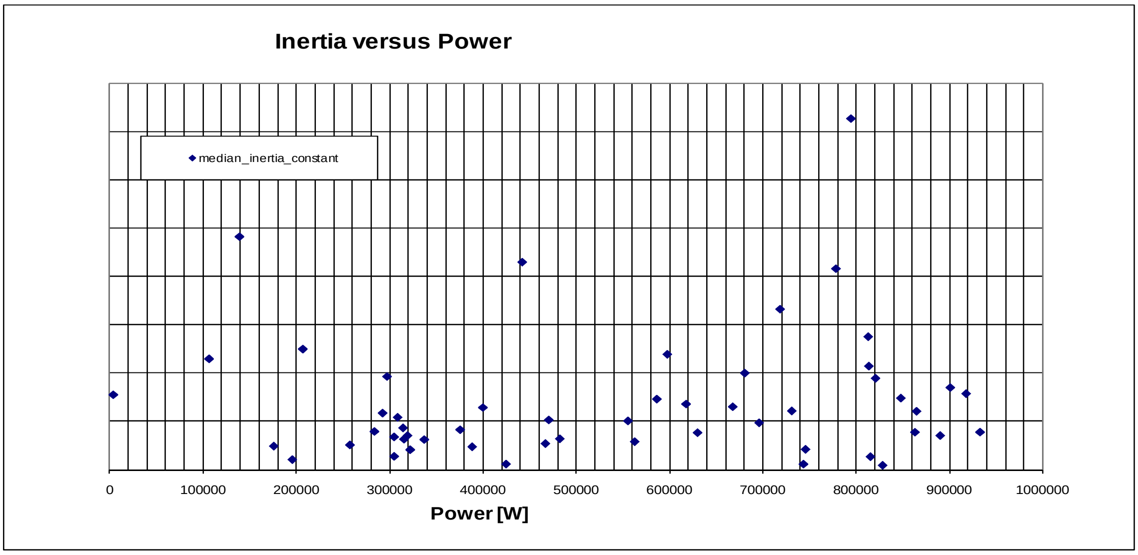


Figure 4-21: *Median values of all sets of solutions obtained for "constant" method C*

Although showing somewhat consistent trends, the results obtained in this case study are subject to rather large variations depending on method and data applied. The main conclusion at this stage is that the process needs further refinement. This refinement shall include assessment of sensor suitability and signal quality, as well as refinement on the post processing and model assumptions. In order to judge on the quality of the derived methods, target values for the overall inertia, damping and stiffness referring to the high speed shaft have to be defined from the models.

Load Validation/Model Validation based on Statistical and Post-Processed Data

Data analysis performed on single time that were recorded in the manual testing campaign revealed abnormal and unexpected results in the Rainflow count of the low speed shaft torque signal. Rainflow counting is one of several state of the art techniques to extract closed load cycles imposed on the component under test. In the presented case the time series taken at rated power operation displayed rather high variations in mechanical torque that could not be detected in the electrical power output (see Figure 4-22).

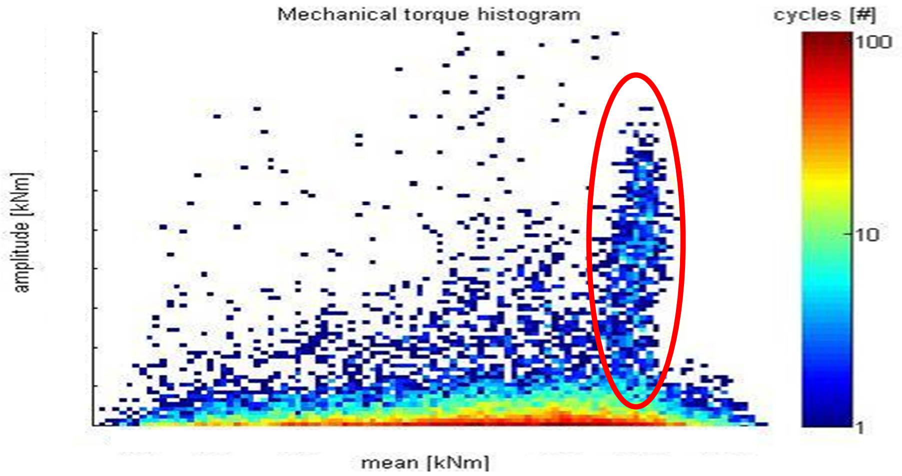


Figure 4-22: *Rain Flow Cycles indicate high variations in mechanical torque around rated torque*

To further investigate this phenomenon in a first step data at rated power operation were visualized, see Figure 4-23.

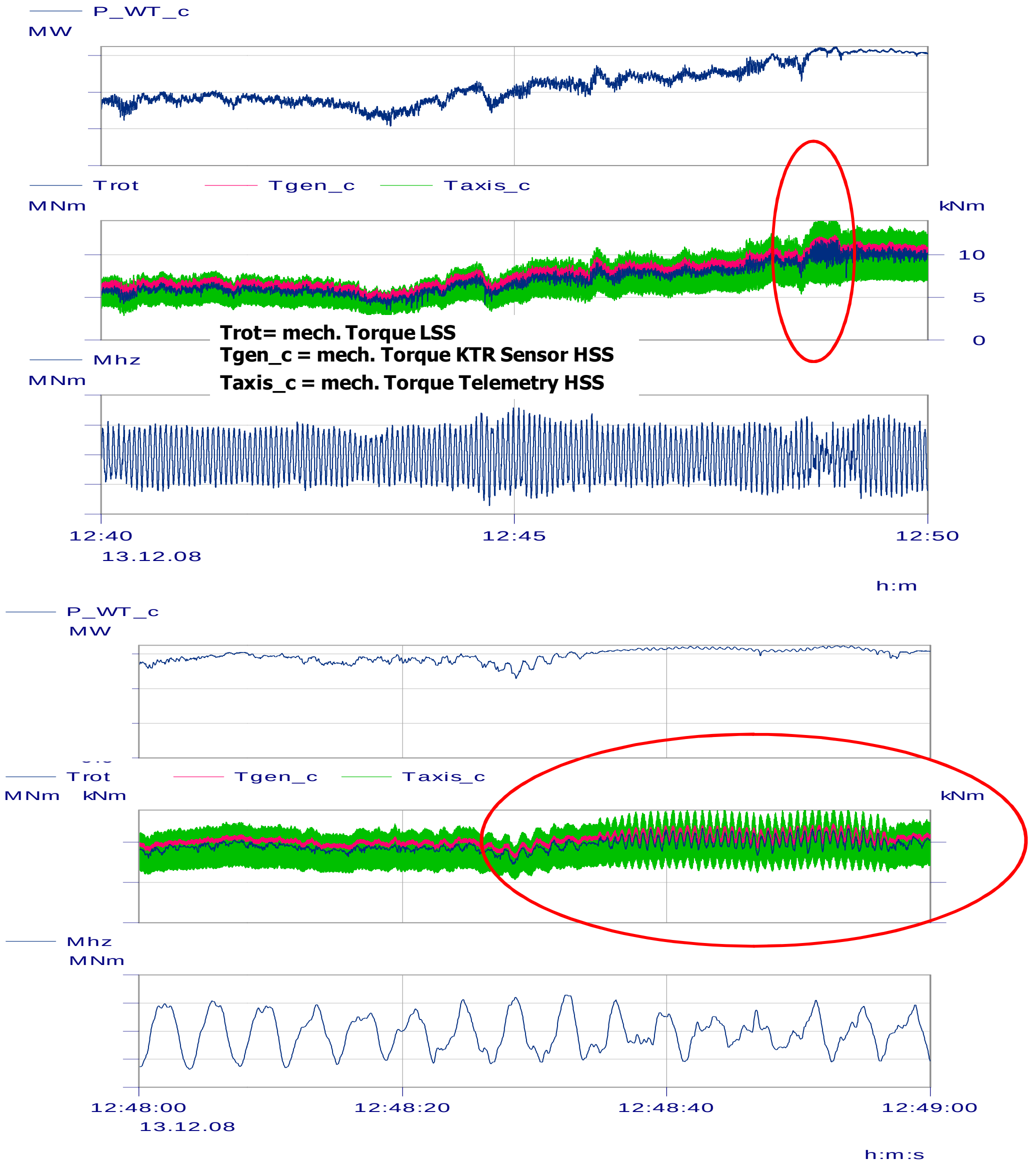


Figure 4-23: *Main shaft torque when reaching rated power.*

In order to further assess the relevance of this phenomenon for the fatigue loading of the turbine drive train data from two monitoring campaigns before and after a change in the wind turbine controller setting were analyzed, see Figure 4-24.

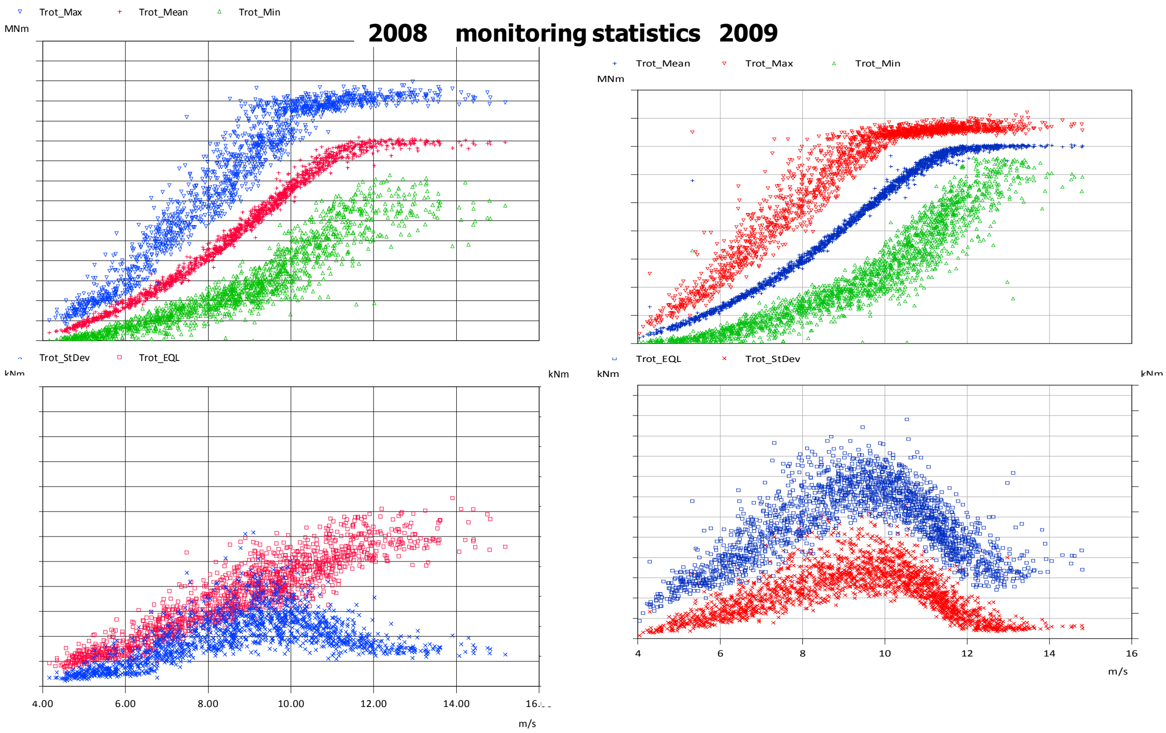


Figure 4-24: *Scatter plots of the main shaft torque before and after a change in the controller of the wind turbine – the new controller in 2009 clearly shows improved behavior at over rated power operation*

The improvement of the drive train loads is clearly visible in the minimized main shaft torque min-max-swing and the decreased damage equivalent load (Trot_EQL in the lower plots). The corresponding effect can also be witnessed in the Rainflow Cycle load spectra and the load duration distribution for the main shaft torque.

In both representations of the drive train torque loading at the main shaft the reduction of loads after the controller change is clearly visible. These statistics establish not only the successful remedy of the drive train oscillation, but also give information on the fatigue relevance of the phenomenon by analysing a multiple of data sets and showing severity of the phenomenon in terms of actual size and frequency throughout the live of the turbine. A further impressive value is produced by deriving the damage equivalent load of the fatigue spectra as given in Figure 4-25. Comparing the damage equivalent load before and after the change reveals a 30% decrease in fatigue loading due to the change.

Such conclusive information calls for such post processing in order to enhance the short term frequency and time series analysis with information on the long term effects of specific loading situations found in time series analysis.

The load duration distributions (LDDs) in Figure 4-26 show distinctly where the problem occurs and give also comprehensive information on the gearbox relevant loading: while the peak frequency value of the load level at rated power operation is significantly reduced considerable frequencies of higher torque loading at over rated power operation can be found.

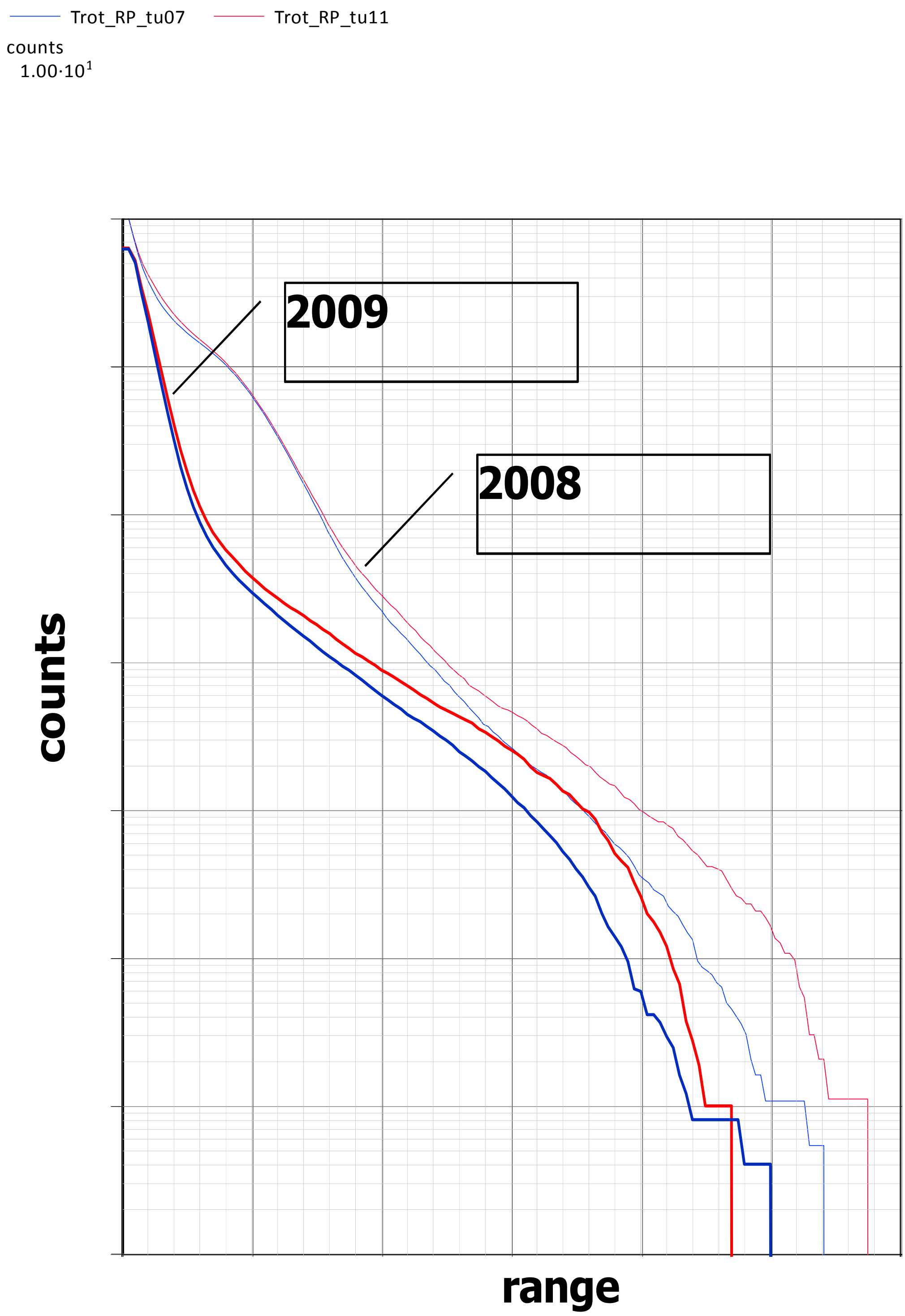


Figure 4-25: *Rainflow Cycle Load Spectra for Torque loads before and after a change in the controller of the wind turbine – red and blue lines are spectra for two different turbulence levels*

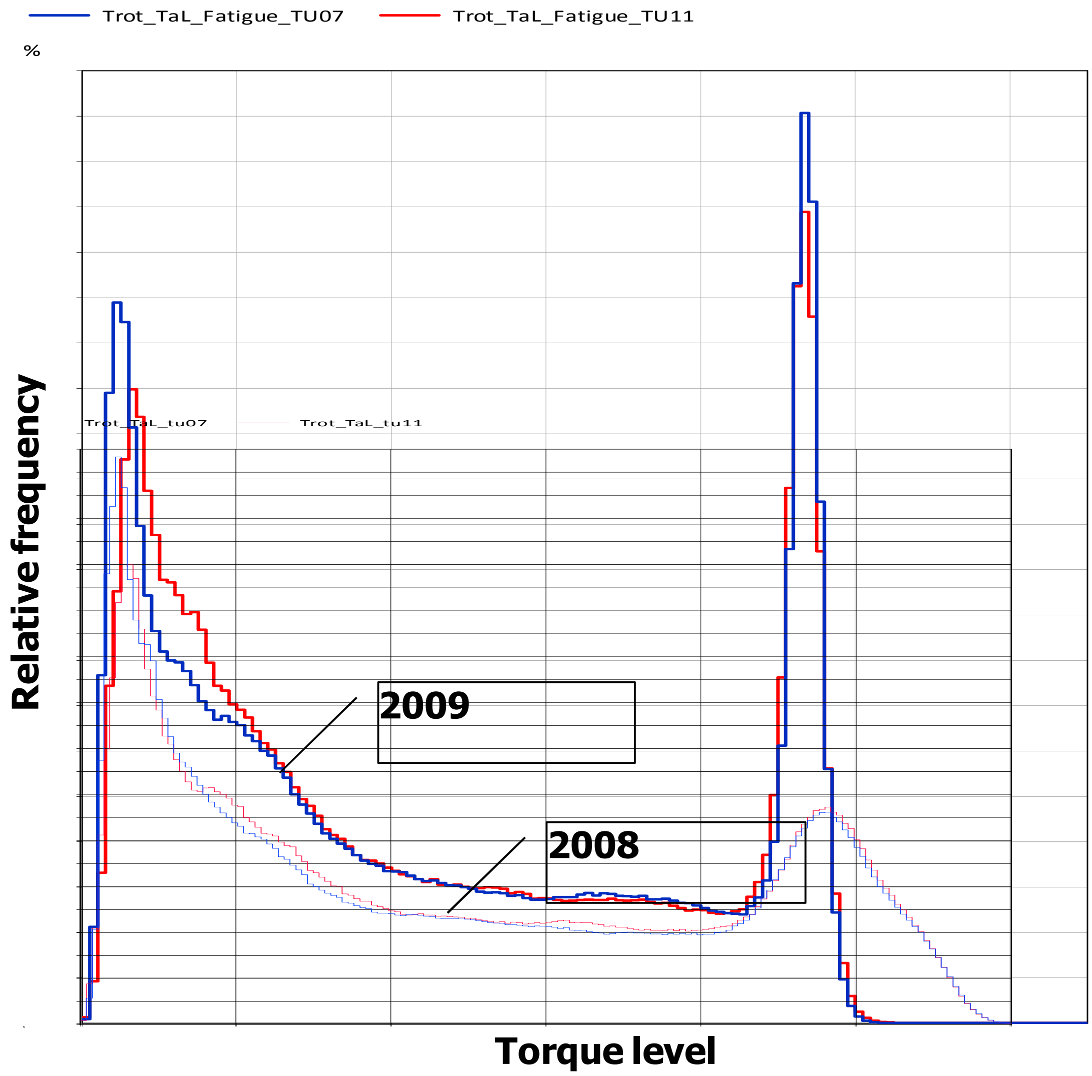


Figure 4-26: *Load duration Distribution (LDD) for Torque loads before and after a change in the controller of the wind turbine – red and blue lines are spectra for two different turbulence levels*

The example impressively shows the importance of such post processing results and as an extension to the preliminary work presented in 2008 [42] demonstrates their use for model and load validation in the component testing for gearboxes. Especially the shape and the maximum load levels reached in the LDD will be of importance to the gearbox manufacturer.

In the model and load validation of the gearbox model computed time series can be processed in the same way as done with the measured load data. If the model is "true" the post processing of its computed time series should come up with results that are consistent with the measured post processing results.

4.6.3 Conclusions

The six-step-approach is tested in a case study on the drive train of the S82 1500kW wind turbine. The results of the two stages of the SIMPACK models are according to the FLEX5 model results in the time and frequency domain. The built models are useful starting points to set up measurements campaigns for load and model parameter validation.

For this case study, sensors and measurement resolution were chosen due to the project task. A load measurement and additional measurements for model parameter (eigenfrequencies, inertia, stiffness and damping) validation have been carried out.

For the data analysis different methods to determine eigenfrequencies, stiffness, damping and inertia have been developed and applied.

For arriving at a simple model corresponding to the measurement signals a single mass rotational oscillating system has been chosen. The equations used have been referred to the high speed shaft side. In order to judge on the quality of the derived methods target values for the overall inertia, damping and stiffness referring to the high speed shaft have to be defined from the models.

The clearly defined target value for stiffness is reproduced.

All statistical methods show similar trends with respect to inertia and damping values varying with power output. At least for inertia it would be expected that there should be a stable value throughout the complete operation range. Further investigations on the applied methods and the measurement and model comparison are needed.

The effect of a major improvement on the drive train loading could be established throughout all available validation tools and approaches like frequency analysis, time series analysis, statistic analysis and further complex post processing analysis.

5. Main results for pitch system measurements and analysis

The main results from the different Work Packages that concern the pitch system are described in this chapter. First the Critical Design Variables and load cases are discussed in section 5.1. This is followed by a description of the loads at the interconnection points. The measurement definitions are given in section 5.3. Finally the results of the measurements and the analysis of the pitch system are described in section 5.4.

5.1 Load cases and Critical Design Variables (CDVs) (WP2)

Though blades and tower are sufficiently covered in the standards, the mechanical components are less well presented. To enable the specification of the Critical Design Variables for the pitch system use is made of the breakdown given in Table 5.1. It should be noted that this is limited to those components or subsystems which are relevant for the structural integrity and therefore should be aimed at when specifying CDVs and design loads.

In general the components mentioned in Table 5.1 are present in a pitch system. However, the exact pitch system architecture should be considered to assess whether this list is still covering the actual design.

Table 5.1: *Breakdown of pitch system*

- Pitch bearing
- Pitch gearbox
- Pitch drives
- Actuator gear

For the he relevant Critical Design Variables (CDVs) for the pitch system, the corresponding design loads are given below in Table 5.2. Beside the specification of the design loads, it is indicated whether the external condition introducing these design loads are covered by an existing design load case (DLC) in IEC-61400-1.

Table 5.2: *Design loads for pitch system*

Name	**Type of load**	**Covered by DLC**
Drive torque for pitching	Ultimate strength and fatigue	Yes, Note 1
Drive torque for positioning	Ultimate strength	Yes, Note 1
Grid loss and gust	Ultimate strength	DLC 1.5
Deformation of hub - blade joint	Constraining forces and moments in pitch bearing	Yes, Note 1
Dynamic oscillating torque in pitch drive train	Fatigue and Hertzian stresses in the gears	Yes, Note 1

Note 1: All typical wind turbine operation modes are covered in the guidelines, so the external condition(s) introducing this design load is captured.

Three new DLCs have been proposed, but these are mainly of importance for the drive train and are therefore discussed in section 4.2.

5.2 Loads at interconnection points (WP3)

Objective of work package 3: “Determination of loads at interfaces” was to determine the procedures for the selected components, among them the pitch system, that describe how the loads at the

interconnection points should be defined, taking into account the load cases specified in work package 2.
To this end, the specification of the interfaces of the selected wind turbine mechanical systems, more specifically for 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 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 is necessary at the interfaces for designing the mechanical components of the pitch system.
The details of the findings were reported within [28]. 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 5-1. The schematic diagram of the pitch system, its components/subsystems and the relevant interfaces are shown in Figure 5-2 (internal interconnection points are indicated as i).

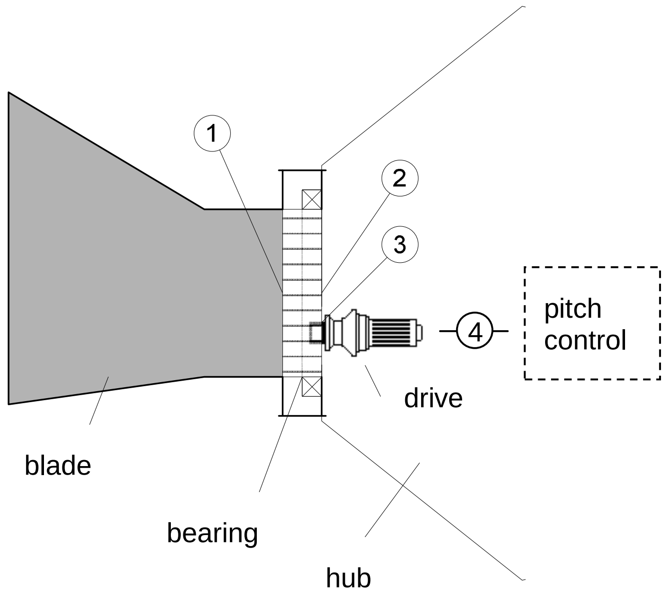


Figure 5-1: *Simplified sketch of pitch system showing interfaces.*

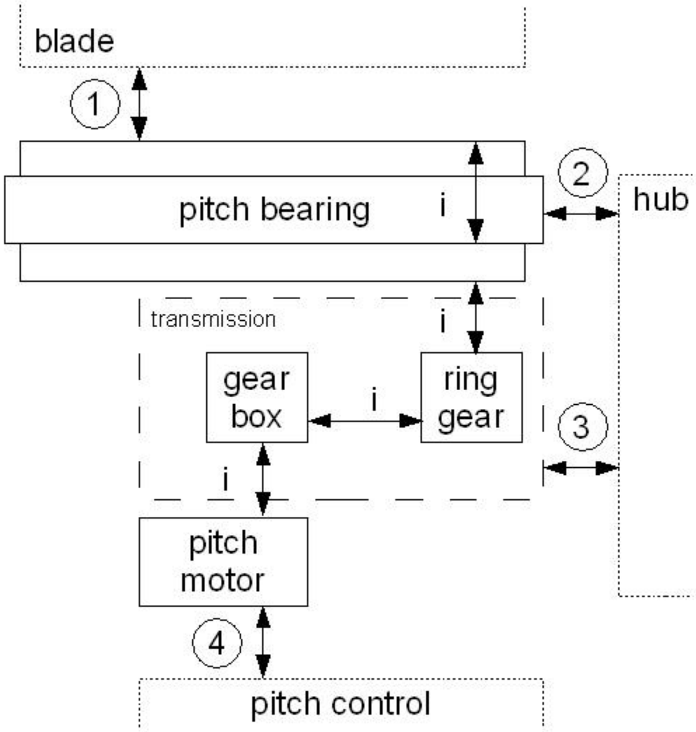


Figure 5-2: *Schematic diagram of the pitch system and its interfaces.*

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 [20]). 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 [28]:

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

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 [28]. 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.

5.3 Measurement definitions (WP3)

Based on the results for the loads transferred across the interfaces of the pitch system, Table 5.3 presents a summary of the recommended quantities to be measured during an experimental campaign focusing on the pitch system components.

Table 5.3: *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 5.3, 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.

5.4 Measurement and analysis results (WP4, WP6)

This section shows the main results obtained in the second case study on the pitch system of a modern wind turbine. It starts with an overview of the system under test, followed by results from the measurement campaign and the data analysis. The section concludes with the evaluation of the six steps approach.

There are two confidential reports that give more details and additional analysis on the work performed within this work package: the instrumentation report [31] and analysis report [32].

5.4.1 The pitch system

The pitch system under consideration in the present case study is the pitch system installed at the Nordex N80 wind turbines at the ECN Wind turbine Test site Wiegingermeer (EWTW). During power production, the pitch system is used either to actively pitch the blade limiting power above rated wind speed or to keep the blade at a constant (optimal) angle during power production below rated. The pitch system is also used to engage a normal shutdown (running to idling) or an emergency shutdown. For the Nordex N80, active pitching starts at an average wind speed of around 14 m/s. The pitch system discussed here is an *electric driven pitch system* of alternating current type with electronic voltage-speed regulation capable of individually pitching the turbine rotor blades.

The pitch system can be subdivided into several components. A sketch of the pitch system used in the N80 and some of the individual components is depicted in Figure 5-3. For every component, an analysis can be made about their respective critical failure modes or phenomena. The different components and their functions considered here are:

Component	**Function**
• pitch slewing bearing	transfer load blade to hub, rotate blade, reduce friction
• pitch pinion gear	drive bearing rotation, increase torque output gearbox
• pitch gearbox	increase torque output from electric motor
• pitch brake	maintain rotor blade in fixed position
• pitch electric motor	supply torque for blade rotation
• pitch controller/electronics	control pitch motor (not displayed in Figure 5-3)
• pitch encoder	feedback pitch position (not displayed in Figure 5-3)

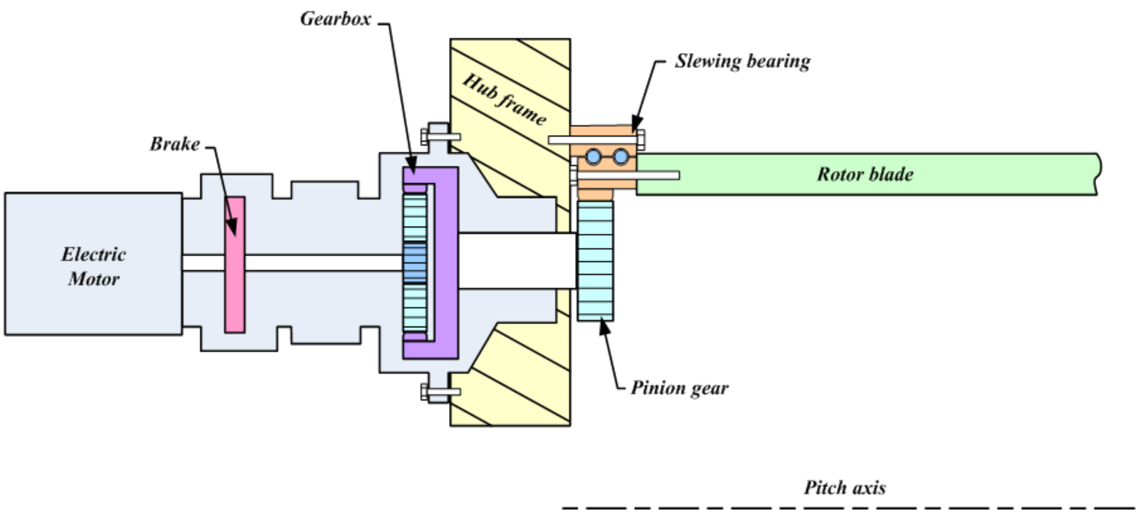


Figure 5-3: *Sketch of pitch system and interface to blade via slewing bearing (cross-sectional view)*

Other components related to the pitch system (e.g. bolt connections, motor housing, hub frame) are not considered in the present case study.

5.4.2 Overview of the measurement campaign

The measurement campaign for this case study on the pitch system of the Nordex N80 test turbine started July 2009 and lasted till March 2010, which gives us more than nine months of data. It was decided to continue the measurements at least until the end of the project in August 2010.

This section presents an overview of some of the relevant measurements on the pitch system. The measurements are shown in relation to wind speed and wind turbine operation.

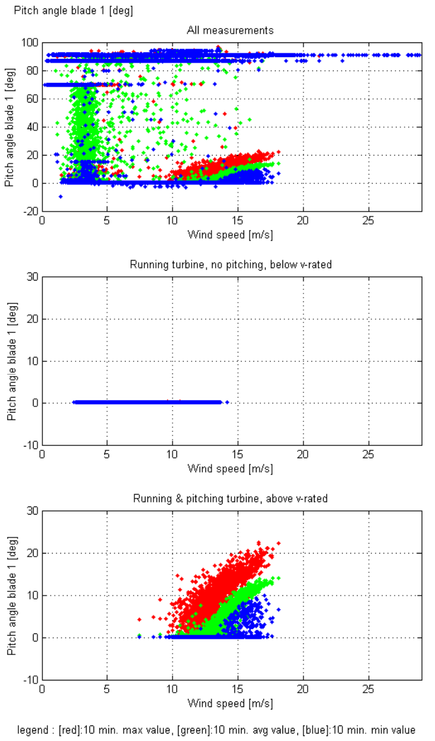


Figure 5-4: *Pitch angle measurements in relation to wind speed and wind turbine operation (r: max, g:mean, b:min)*

5.4.3 Data analysis

As shown in the six steps approach, the first step in the analysis of the pitch system is the identification of critical failure modes or phenomena for the subsystem/component. In a detailed failure mode and effect analysis, the highest risk priority was assigned to reduced performance of the pitch motor or the slewing bearing. The analysis in this case study therefore focuses on the pitch drive train (section 5.4.3.1) and the pitch bearing deformation (section 5.4.3.2).

5.4.3.1 Pitch drive train analysis

This chapter shows the results of the analysis of the pitch drive train using the six steps approach. The combination of the electric motor, brake, gearbox, pinion gear and blade bearing is referred to as the *pitch drive train* (see Figure 5-3). The primary function of the pitch drive train is to pitch the rotor blade by applying a torque. To rotate the blade, the pitch drive train has to overcome the friction torque of the bearing and gears, the moment due to aerodynamic and gravity forces on the rotor blade and the moment due to acceleration of the inertia of the drive train components and rotor blade.

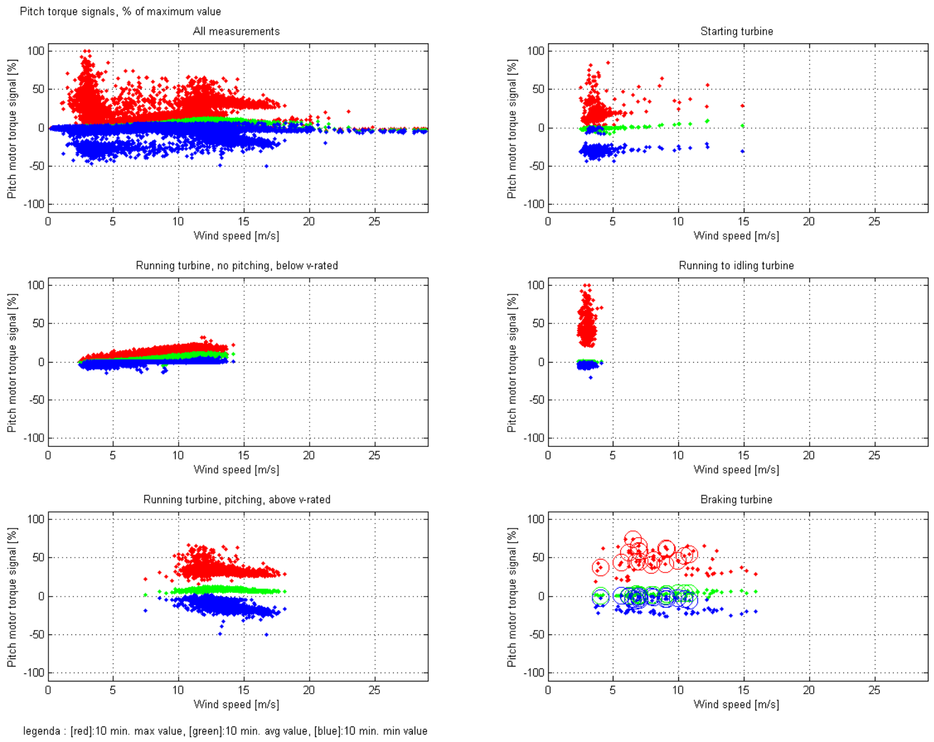


Figure 5-5: *Pitch torque measurements in relation to wind speed and wind turbine operation*

The analysis in WP6 originally started with the analysis of a friction model for the slewing bearing. As a starting point, a simple model (Figure 5-6) was used to determine the friction. However the analysis of the measurement data revealed that unfortunately measuring the blade torsion is not straightforward due to strain gauge calibration with respect to anisotropy of the blade material and misalignment. To further investigate the effects on misalignment and anisotropy of the blade torsion measurements a detailed analysis is performed (see [32]). The analysis concludes that exact calibration of the 45-45 strain gauges mounted in the blade root to measure blade torsion using the own weight of the blade is not possible. However, using a reasonable assumption, an approximate calibration can be obtained.

To be able to proceed the analysis (second iteration) without reliable blade torsion moment measurements, the focus was shifted to the motor torque, since this is the actual load that the motor has to deliver. Friction of the bearing will still be part of the equation, but in the modelling approach it is then assumed that differences in the correlation between the modelled pitch torque and the measured pitch torque are the result of the friction model contribution in the pitch motor torque model. If necessary, the friction model can then be tuned with the friction coefficient to correlate the output of the pitch torque model to the measured pitch torque.

Pitch motor torque

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

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

Blade torsion moment

PHATAS inputs

Figure 5-6: *Simple pitch drive train model*

All external design load cases (DLC) are prescribed in the IEC design requirements as specified by [7]. These DLCs can be used as input for ECN's aeroelastic simulation program PHATAS (Program for Horizontal Axis wind Turbine Analysis and Simulation). From these simulations and initial analysis of the measurements (Figure 5-5), the normal and emergency shutdowns appeared to be critical DLC's. Figure 5-7 shows a normal shutdown operation, which is in fact running to idling at low wind speed, by pitching the blades from 0 to 70 degrees. Figure 5-8 shows the contributions to the pitch motor torque of the different components in the system.

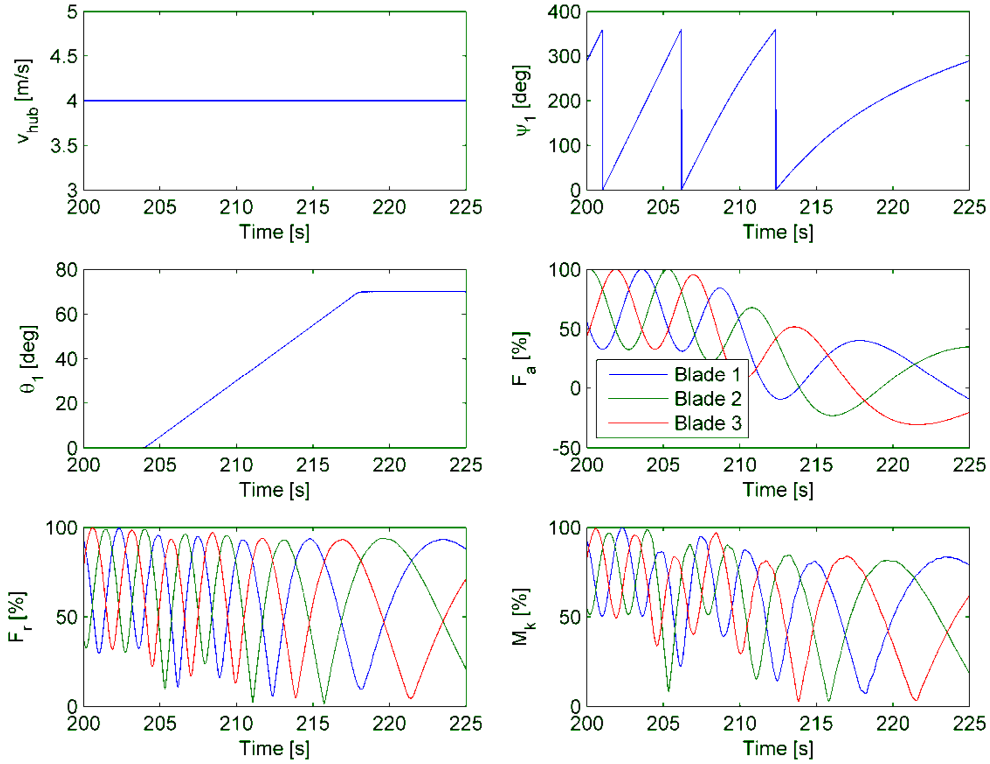


Figure 5-7: *PHATAS output DLC 4.1 for a 25 second interval of the simulated 600 s time series: wind speed, azimuth angle blade 1, pitch angle, axial forces, radial forces and bending moments (anonymous)*

The analysis of time series of measured pitch motor torque data revealed that the maximum pitch motor torque always occurs at the start of a normal shut-down. This corresponds to the initial design consideration, where this starting torque was considered as a critical design parameter.
Figure 5-9 shows that the pitch motor torque model has quite good qualitative similarities for all three rotor blades. The trend in the pitch motor torque during blade rotation follows a similar path. However, quantitatively the model has fewer agreements to the measurement. Specifically the starting torque peak seems to be much lower in the model compared to the measurements. As can be seen in Figure 5-5 the measurement campaign also revealed a large scatter in the maximum pitch torque measurements for the normal shutdown. An effort is made to understand the cause of this scatter by studying the relation between the pitch motor torque and other measured signals. This research into the pitch motor peak torque scatter is elaborated [32], but no clear correlation was found, other than the lagging of the pitch rotation with respect to the pitch setpoint. The peak torque has high frequency dynamics and relatively short time span (see Figure 5-9), as after a certain time the pitch torque seems to be limited. This leads to the conclusion that the pitch system electronics and motor controller most likely play a big role here. It is therefore recommended to add these to the model for pitch drive train analysis if this effect is to be captured.
We also see that during blade rotation, the model fits blades 2 and 3 relatively well, while blade 1 seems to have a clear offset with respect to the model. Other time series showed similar offsets for the measured pitch motor torque of blade 1. Moreover, the offsets of pitch torque for the three blades changed significantly after maintenance was performed on the test wind turbine near end of the measurement campaign (May 2010). These observations call for adding a load independent term in the friction model. Also higher pitch speed has an influence on the pitch torque offset, which can be

included by adding a speed dependent term to the friction model. The order of the torque speed relation could not be established from the measurements; additional test cases (at multiple constant speeds) should be performed.

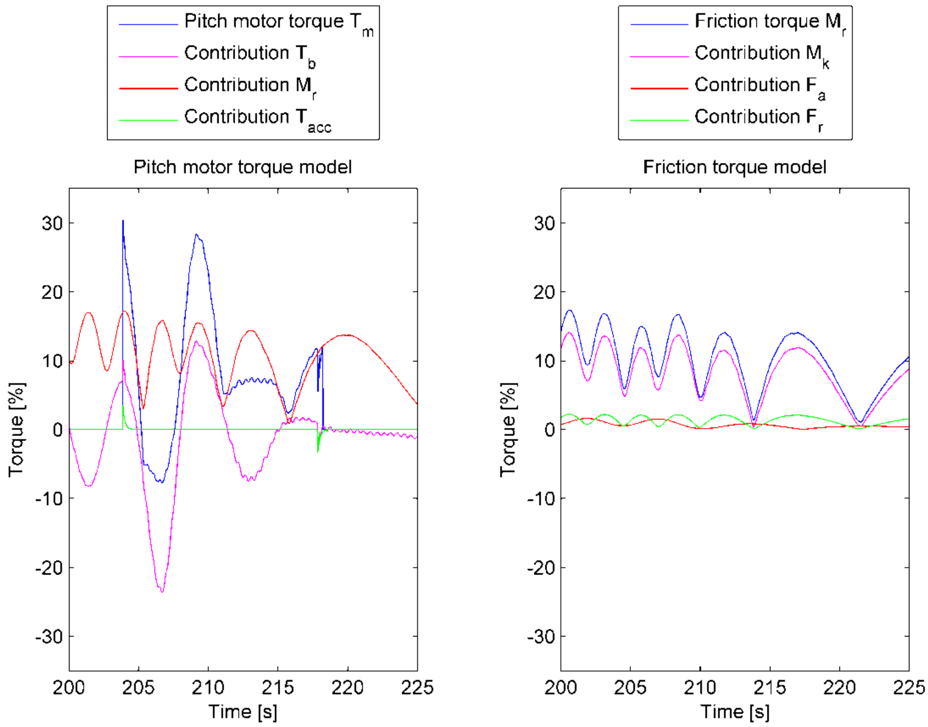


Figure 5-8: *Pitch motor torque model output and input contributions (L) and friction sub-model output and input contributions (R) for blade 2 (anonymous)*

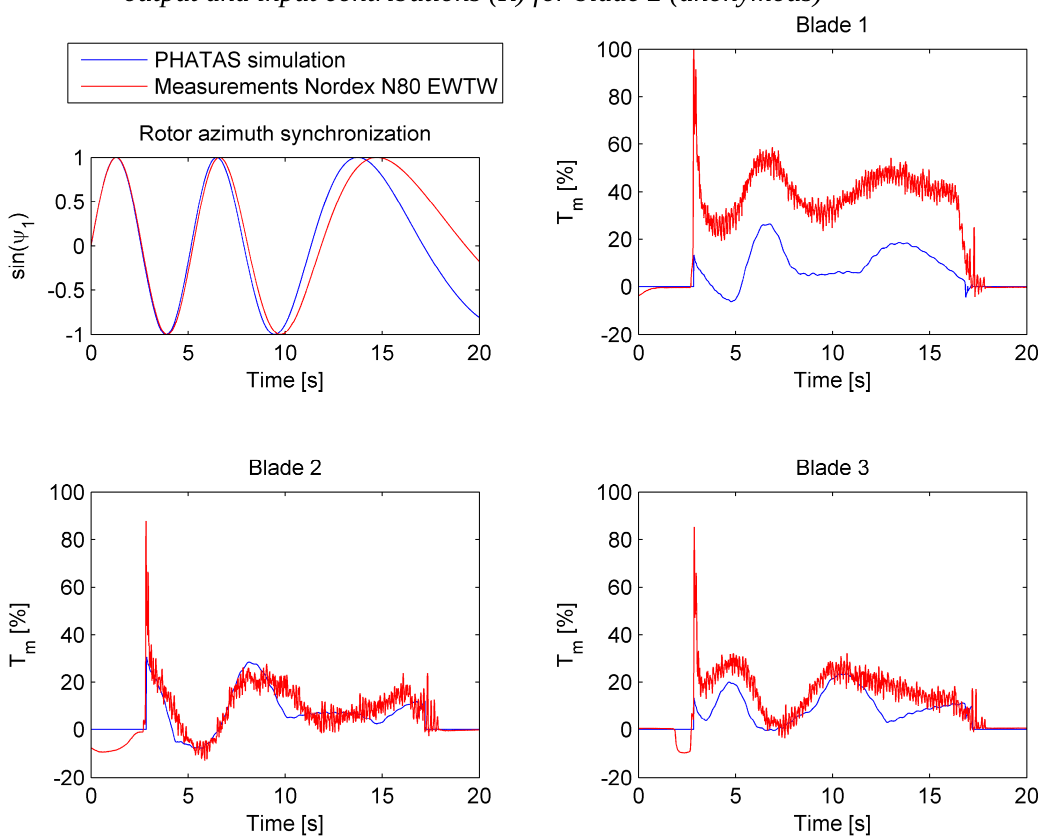


Figure 5-9: *Synchronized pitch motor torque comparison where the red lines indicate measurements and the blue lines modelling results*

5.4.3.2 Pitch bearing deformation

This section describes the setup of a measurement campaign to identify the deformation of the blade root bearing. The blade root bearing transfers the loads of the blade to the hub of the wind turbine. Because these bearings are difficult to replace, it is important to know and predict the failure mechanisms of the bearing. It is assumed that the deformation of the bearing is a mechanism that may lead to excessive wear of the bearing.
Out of practical considerations (known, cheap and commonly used measurement principle), it was decided to use strain measurements to verify the shape and to measure the direction of the assumed ovalisation.

First a model had to be developed for the relation between bearing strain and bearing deformation. This model is fitted on strain measurements. Finally the relation between loading and deformation must be determined using measurements. Once this relationship is established, design load cases can be used to find the deformation for extreme loads.
The initial analytical model is based on an assumed deformed shape, for instance an oval as in Figure 5-10.

- deformations are small
- length of the neutral line is constant
- only (flat and edgewise) bending moments are considered

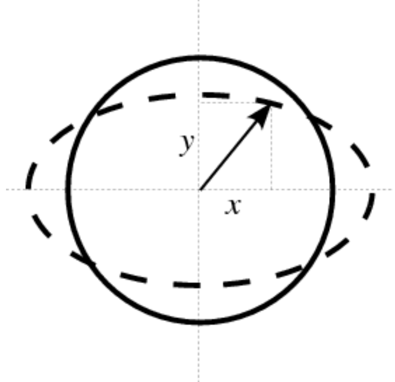


Figure 5-10: *Bearing deformation; oval shape*

Next to the strain measurements, two linear variable displacement transducers (LVDT) were placed measuring the edgewise and flatwise deformations across the bearing. These are used to establish and verify the correct magnitude of the ovalisation.

Figure 5-11 shows the locations of the strain sensors and the locations of the LVDTs. These locations should allow verification of the global model, but should also show whether the model of ovalisation is valid for small sensor spacing.

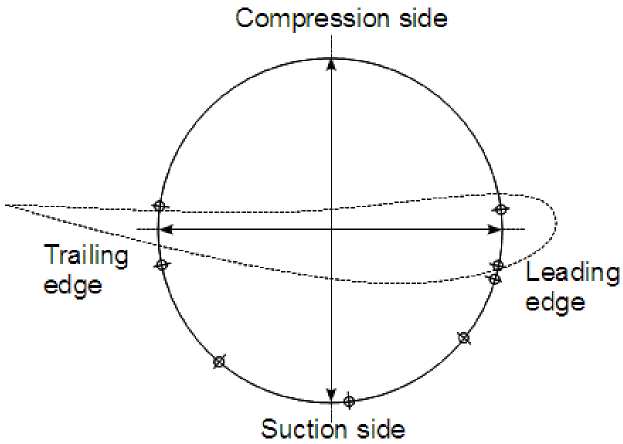


Figure 5-11: *Sensor locations: 8 strain sensors and 2 LVDTs*

Table 5.4: *Final measurement campaign specification for pitch bearing deformation*

Measured value	Sensor type	Locations	Frequency
Distance	LVDT	Leading edge – trailing edge and Compression side – suction side	128 Hz
Strain	Strain gauge	Relative to leading edge (0°) at -6.1, 13.2, 17.2, 39.1, 84.37, 129.6, 168.0 and 187.0°	128 Hz

The detailed time history was used to establish whether it was possible to correlate the measured strains and deformations with the measured blade bending moments. Correlation between strains and measured loads is good for single datasets or datasets covering similar wind conditions (see Figure 5-12).

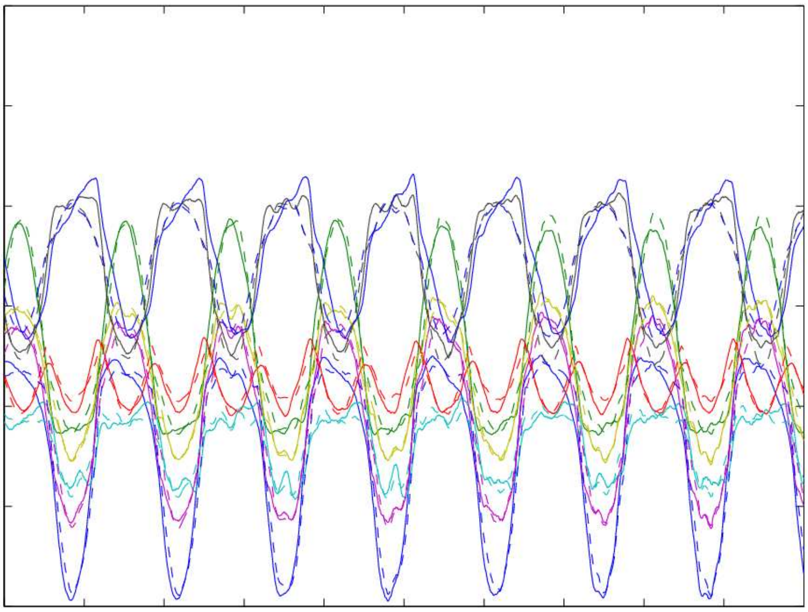


Figure 5-12: *Measured data (solid lines) and data reconstructed on the basis of correlation (dashed lines) match well if the average wind speed does not change much*

It is more difficult to find a single correlation that covers a large range of wind speeds. This could indicate that the behaviour can only locally be approximated with a linear relation or that the strains are also influenced by other loads, such as in-plane forces on the bearing or the spanwise force on the blade.

Model identification is performed by fitting the model on the measurements. There is a definite offset in the data that needs to be addressed. The data was calibrated using idling conditions where moments in flat and edgewise directions are both nearly zero. In addition, the algorithm was allowed to correct any constant offsets. However, the assumed model does not match well with the data (Figure 5-13).

Because the model does not fit the data, it needs to be adjusted. Three variations were examined that are based on assumptions of the shape of the deformation:

- the shape is equal to the deformation of a ring under point force loading
- the shape is equal to the deformation of a ring under a distributed load
- the shape is a sum of circle and a sine with two periods around the edge of the circle
- two of the above shapes superimposed

All these models did not result in a satisfying fit between model and measurements. It was therefore decided to move to finite element (FE) analysis. An FE model does require a lot more information, as the hub, the blade and the loads together cause the deformations in the bearing. 3D structural models of all three need to be made and combined, which is described in [32].

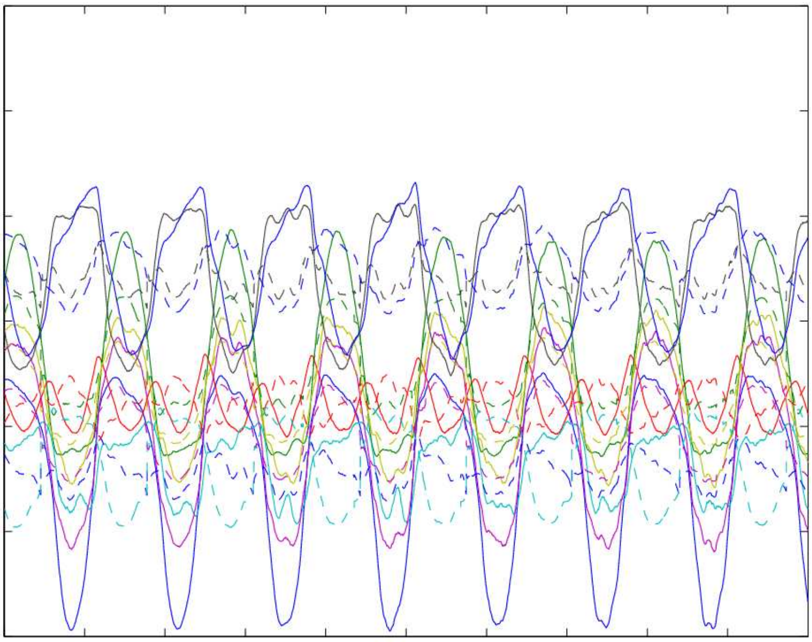


Figure 5-13: *Measured data (solid lines) and the fitted model based on ovalisation (dashed lines) do not match well*

Figure 5-14 shows the equivalent strain of the top-left bearing (deformations are scaled up by factor 3100). It shows that the strain may be significant along the radial direction of the surface of the bearing. This seems to be due to the out-of-plane bending of the bearing. Because the tangential strain varies along the surface of the bearing, the location of the strain sensor must be accurately known to match modelled and measured strains.

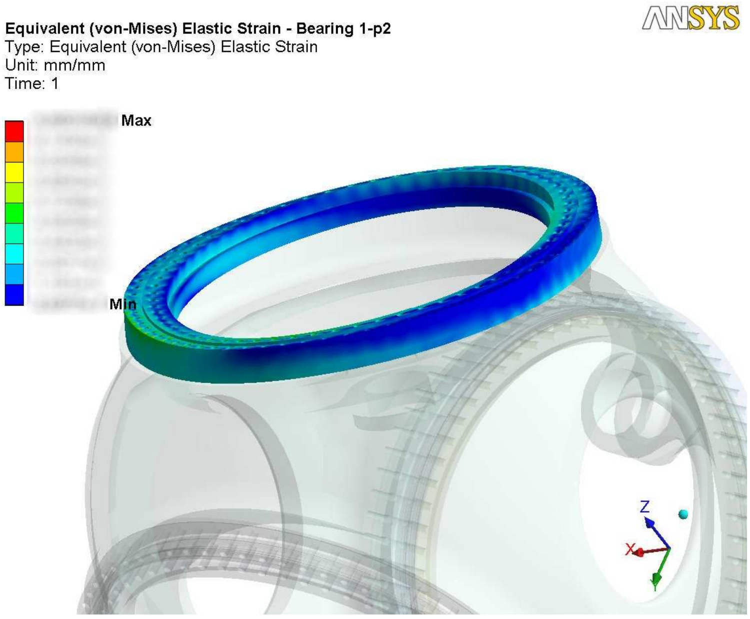


Figure 5-14: *Out of plane bending contributes to strains in the bearing*

Figure 5-15 shows the changes in the diameter of the bearing as calculated with the FEM analysis and as measured with the LVDT sensors. Though in the right order of magnitude, the model does not capture the full behaviour and is a bit too stiff. Figure 5-16 shows the same results but for the high load case, here the difference in behaviour is significant.

The FEM models indicate that the deformation of the bearing is mostly caused due to deformations of the hub itself. It also indicates that the deformation results in a shape that is difficult to capture and match based on the sensors that we could place in the hub. It would be better to measure these deformations of the rotor in a more controlled environment first and then examine the differences when the rotor is installed on a turbine.

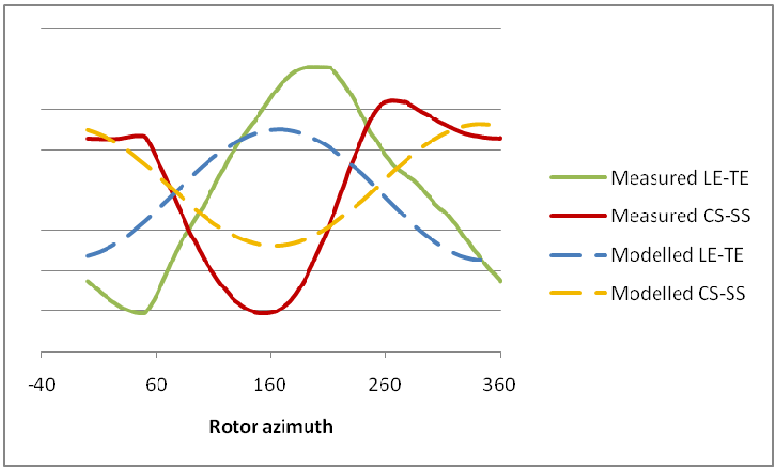


Figure 5-15: *Change in bearing diameter, modelled vs measured, idling*

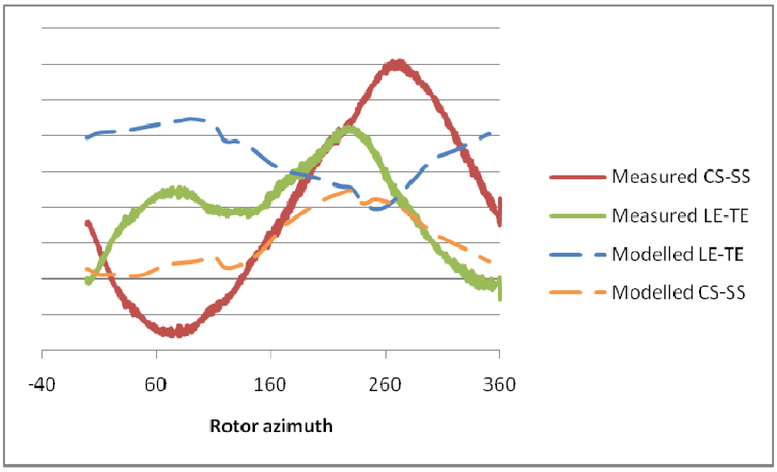


Figure 5-16: *Change in bearing diameter, modelled vs measured, high load*

5.4.4 Evaluation of the six steps approach

In WP4 of the PROTEST project a six steps approach is suggested. The purpose of this approach is to use the designed model as input for specification of the measurement campaign required for validation of that model.

During the case study applied to the pitch drive train, the six steps approach has been applied and feedback of the measurement campaign is taken into account to make adjustments to the model and therefore the measurement campaign itself:

1. The initial model was set-up to determine friction at the blade-bearing interface. This model relied on blade torsion as a modeling input parameter. The measurement set-up for blade

torsion proved to be very difficult to calibrate and hence the results would be unreliable. A new model was set-up to calculate the required pitch motor torque, which is closely related to friction at the blade-bearing interface according to the failure mode and effect analysis.

2. Analysis of measurement data for the pitch motor torque revealed that the normal shut-down is one of the critical load cases for the design of a suitable pitch motor (DLC 4.1). The measurement data shows a large scatter in the recorded pitch motor starting torques.

The feedback loops after step 6 have been followed to change both the model and the measurement campaign. The steps followed in the six steps approach are marked in red in Figure 5-17.

The six steps approach worked well for the analysis of the pitch drive train. The "Adjust model?" block and feedback loop from step four to step two was not used and considered redundant. This is due to the fact that a model is usually set up with the requirements on in- and output already in mind, as it is part of the purpose of modelling in the first place. The loop will therefore only be used in the rare cases that running the DLC's on the model shows unexpected results, but this also implies that the modeller did not fully understand the system and the model of it.

Because there is no real model available for the deformation of the pitch bearing, the six steps approach for designing a measurement campaign described in WP4 is adapted to suit the needs of this problem. That means that the selection and running of DLCs is postponed, because running the DLCs is pointless unless your model is verified and that a redesign of the measurement campaign ought to be only necessary when your model design is changed.

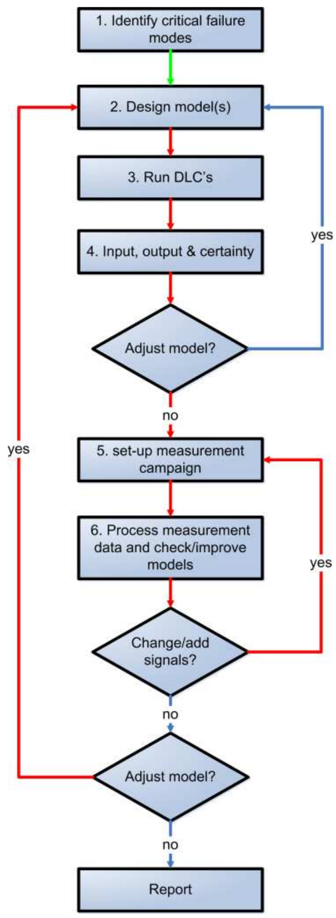


Figure 5-17: *Steps followed (red) in six steps approach of pitch drive train analysis*

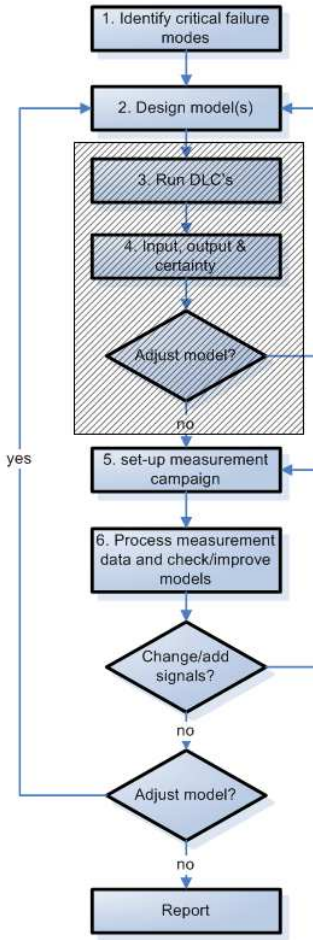


Figure 5-18: *Because there is no model to start off with, the process is adapted; DLCs are postponed until a suitable model is found*

6. Main results for yaw system measurements and analysis

The main results from the different Work Packages that concern the yaw system are described in this chapter. First the Critical Design Variables and load cases are discussed in section 6.1. This is followed by a description of the loads at the interconnection points. The measurement definitions are given in section 6.3. Finally the results of the measurements and the analysis of the yaw system are described in section 6.4.

6.1 Load cases and Critical Design Variables (CDVs) (WP2)

Though blades and tower are sufficiently covered in the standards, the mechanical components are less well presented. To enable the specification of the Critical Design Variables for the yaw system use is made of the breakdown given in Table 6.1. It should be noted that this is limited to those components or subsystems which are relevant for the structural integrity and therefore should be aimed at when specifying CDVs and design loads.

In general the components mentioned in Table 6.1 are present in a yaw system. However, the exact yaw system architecture should be considered to assess whether this list is still covering the actual design.

Table 6.1: *Breakdown of yaw system*

- Yaw bearing
- Yaw brake
- Yaw drives
- Yaw gear

For the relevant Critical Design Variables (CDV's) for the yaw system, the corresponding design loads are given below in Table 6.2. Beside the specification of the design loads, it is indicated whether the external condition introducing these design loads are covered by an existing design load case (DLC) in IEC-61400-1.

Table 6.2: *Design loads for yaw system*

Name	**Type of load**	**Covered by DLC**
Drive torque for yawing	Ultimate strength and fatigue; Wear of yaw brake and yaw gear	Yes, Note 1
Drive torque for braking (position holding)	Ultimate strength	Yes, Note 1
Deformation of the tower top flange or nacelle main frame	Constraining forces and moments on yaw bearing	Yes, Note 1

Note 1: All typical wind turbine operation modes are covered in the guidelines, so the external condition(s) introducing this design load is captured.

Three new DLCs have been proposed, but these are mainly of importance for the drive train and are therefore discussed in section 4.2.

6.2 Loads at interconnection points (WP3)

The main objective of work package 3: "Determination of loads at interfaces" was to determine the procedures for the selected components, among them the yaw system, that describe how the loads at the interconnection points should be defined, taking into account the load cases specified in work package 2 and discussed in section 6.1 of the present document.

To this end, the specification of the interfaces of the selected wind turbine mechanical systems, more specifically for 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 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 [29]. 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 6-1.

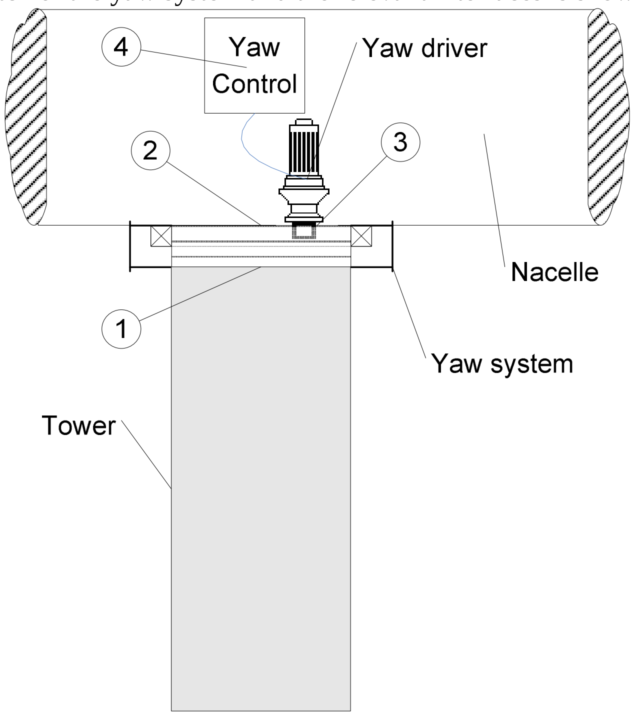


Figure 6-1: *Simplified sketch of yaw system showing interfaces.*

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:

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

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 [29], where the interested reader is referred to.

6.3 Measurement definitions (WP3)

Based on the results for the loads transferred across the interfaces of the yaw system, Table 6.3 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.

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

Table 6.3: *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		..

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.

6.4 Measurement and analysis results (WP4, WP7)

Based on the determined loads to be transferred across the interfaces of the yaw system, as described within 6.3 of the present document, a measurement campaign was designed and carried out within work packages 4 and 7 of the PROTEST project, respectively. As a result of the research conducted, for the yaw system and its components, **in addition to** the quantities specified within IEC/TS 61400-13 [21] as mandatory, and since 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, the quantities classified into load quantities and operational parameters shown in Table 6.4 should be considered.

Table 6.4: *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

The measurement techniques involved are reported within [43] and basically recommend the use of strain gauges for the measurement of bearing bending moment, torsion, axial and radial forces, with a methodology closely following that used for measuring the tower loading. In addition to that, alternative measurement techniques for the gear torque are recommended, either by direct measurement of the torque or indirect estimation through the power consumption of the yaw driver.
Based on these recommendations a measurement campaign was carried out on the NM44/750 wind turbine installed at CRES wind farm in Lavrion, Greece. Among the signals monitored some were monitored for the first time on a wind turbine, such as the shear and axial forces on the tower top. These are to be used for the direct estimation of the yaw bearing loads as well as incorporated within the estimation of frictional moment to determine the actual loads on the yaw motor.
For assessing whether these measurements are required or are redundant Figure 6-2 presents some characteristic statistics of load magnitudes measured during the yaw campaign. One could argue that for example the shear loads on the tower top might be estimated through the bending moments on the tower top and the tower bottom, if the wind loading acting on the tower is neglected or additionally estimated. The graph on the right of Figure 6-2 presents the 10-minute mean value of the difference of the total bending moment between tower top and tower bottom, versus the 10-minute mean value of the shear force measured on tower top. The correlation is strong.

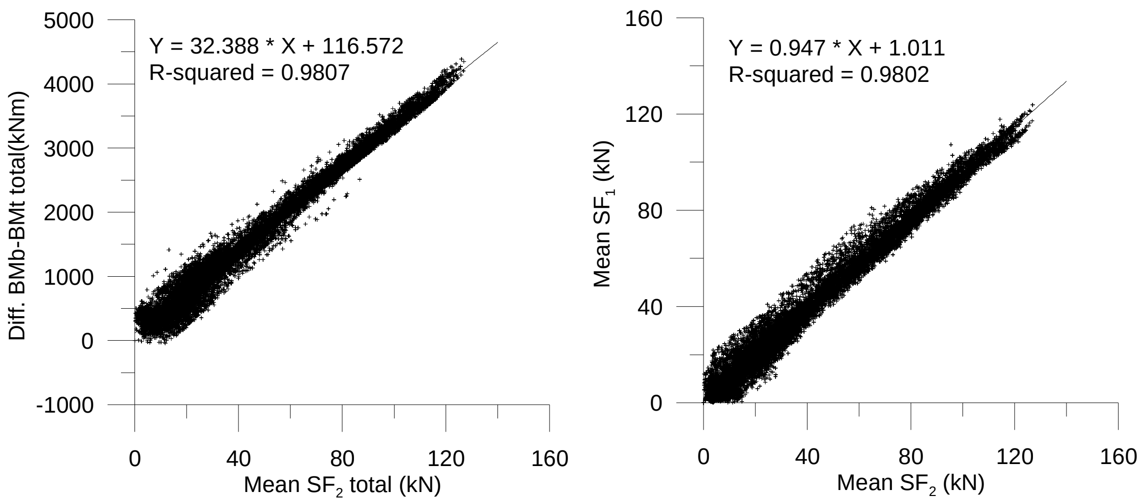


Figure 6-2: *Load statistics and correlations within the yaw measurement campaign*

Yet the actual value obtained through the bending moments for the shear force on the tower top is 9% lower than what was measured (no correction for the wind loading on the tower is made) on the

10-min statistics range. The distance between the tower top bending strain gauge bridge and that of the tower bottom is 35.576m.

In the right of Figure 6-2 two measurements of the shear force at the tower top, but on different sections are compared. The measurement denoted SF_1 is closer to the tower top than that denoted SF_2, while the distance between them is 2.8m. The correlation between them should be perfect, albeit some deviations are seen. This is due to the fact that some stress concentration effects were observed for the first bridge (at about 1.8m below the yaw ring) and therefore, in a second campaign the measurement positions were moved lower within the tower to avoid these effects. The same trend was also observed during the measurement of the axial force at the tower top. Therefore for this load magnitude a second bridge was installed in the second campaign, approximately 2.8m lower within the tower than the original bridge.

Regarding the kinematic statistics necessary for designing the yaw system actuator analysis has been conducted to determine the duration of yaw, time of start between yaw motions, etc. An indicative plot is shown in Figure 6-3 where the yaw duration is presented against the 10-min wind speed only for normal operation cases. The same parameter is plotted only for the transient cases (starts, stops and generator changes) in Figure 6-4. Binned with respect to 10-min mean wind speed, the mean value of yaw duration and the respective standard deviation is shown in Figure 6-5.

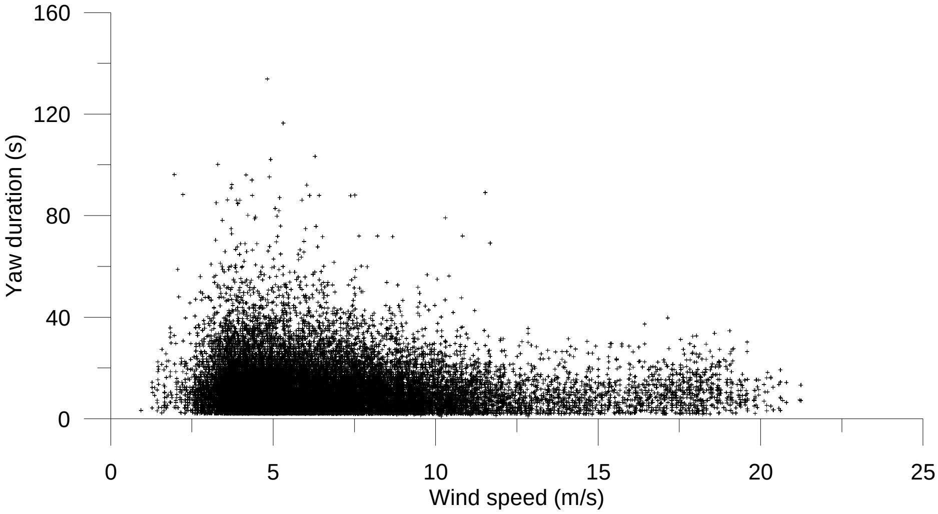


Figure 6-3: *Yaw duration with respect to wind speed (normal operation cases)*

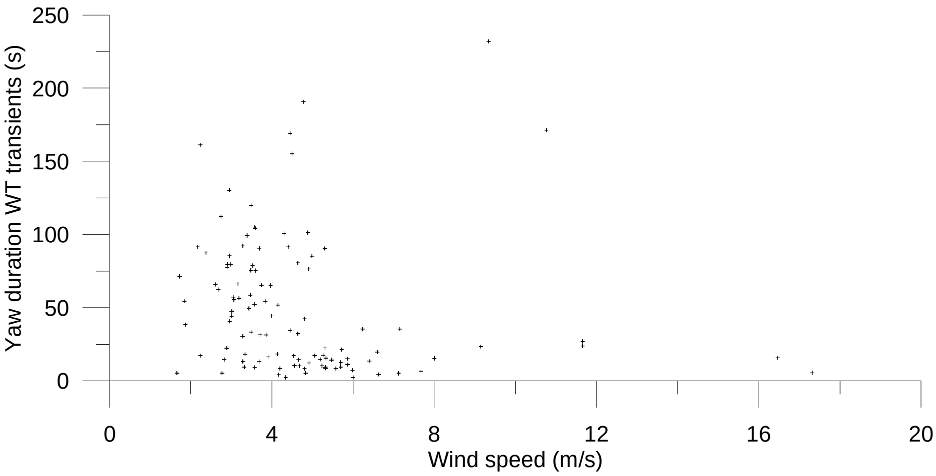


Figure 6-4: *Yaw duration with respect to wind speed (transients)*

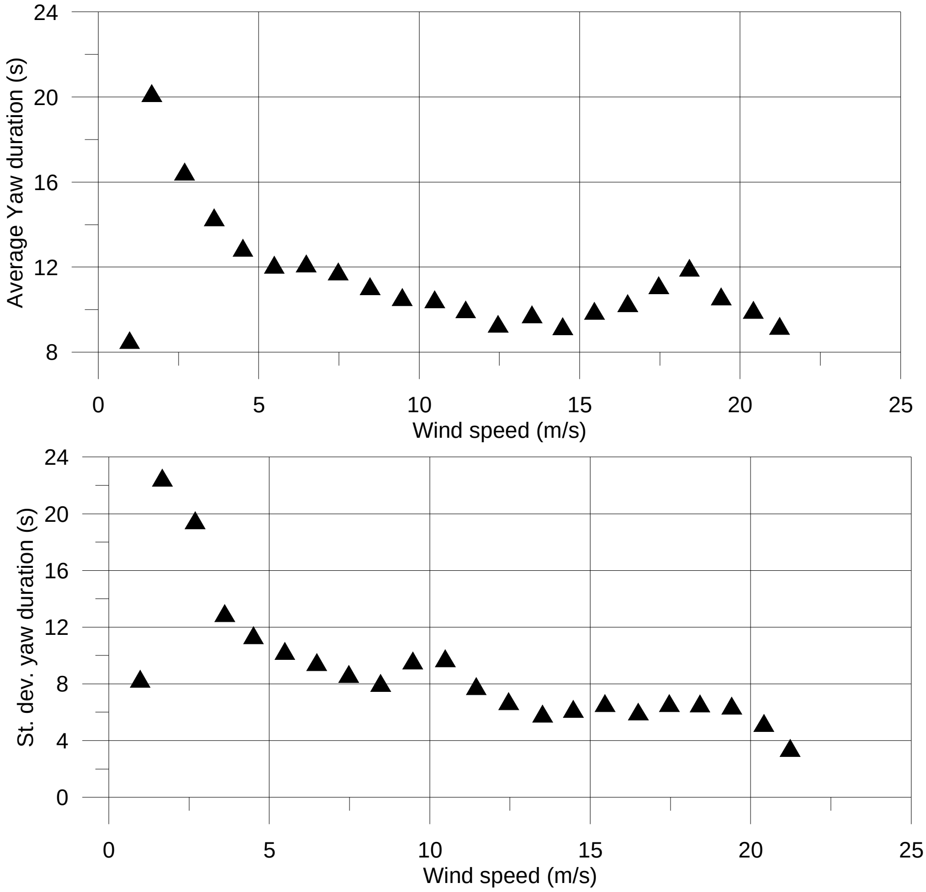


Figure 6-5: *Yaw duration statistics binned with respect to wind speed (complete data base)*

Statistics of the yaw duration with respect to the wind turbine power output are shown in Figure 6-6.

Statics of the time between yaw starts (from the time the yaw motor stops until the next start) with respect to the 10-min mean wind speed is presented in Figure 6-7 with an average time between yaw starts for the complete data base 120 s, while the average yaw duration for the complete data base is about 12 s, corresponding to a yaw motion of about 5°. The same magnitude is presented with respect to the wind turbine power in Figure 6-8.

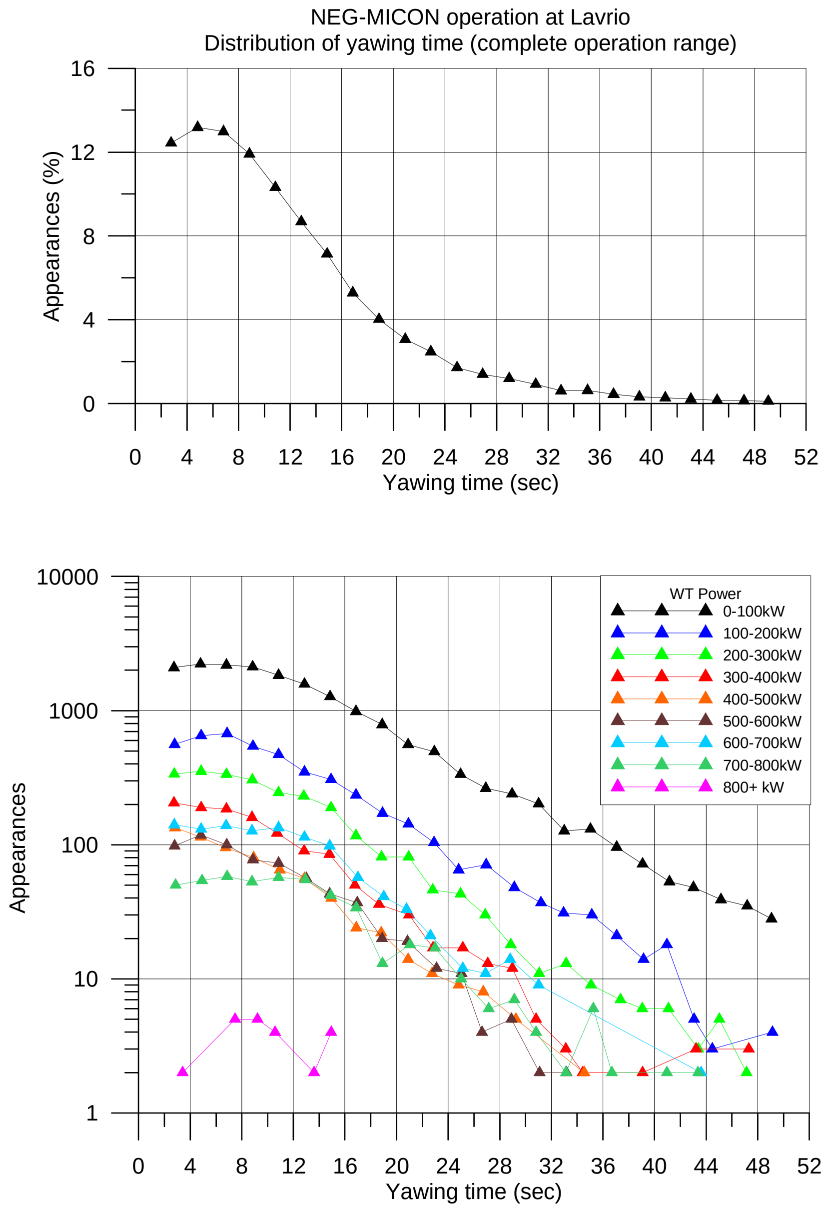


Figure 6-6: *Yaw duration with respect to WT power (complete operation range)*

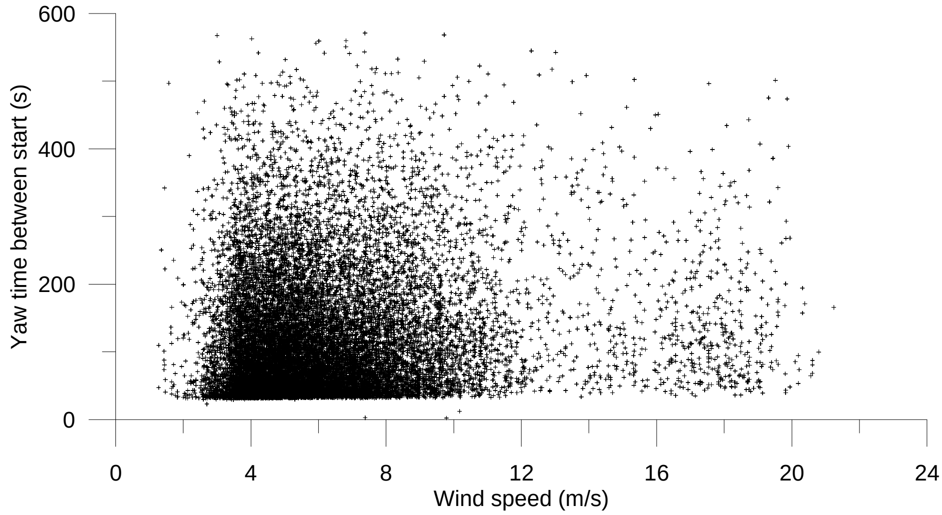
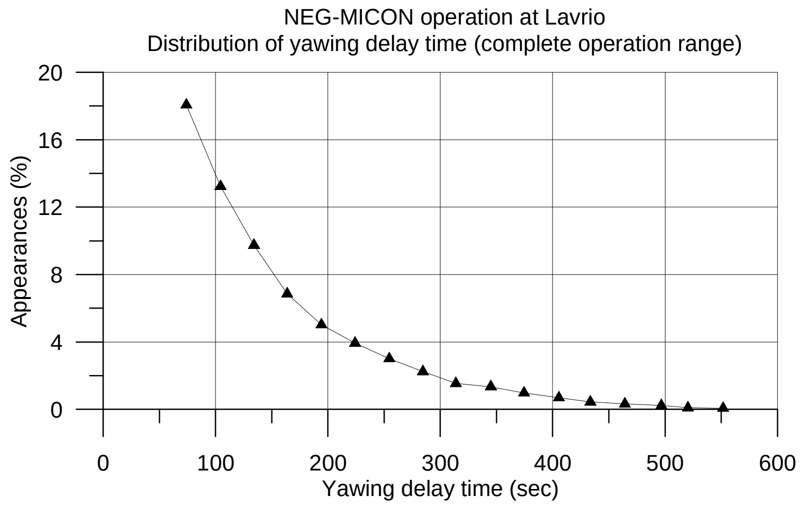


Figure 6-7: *Time between yaw starts with respect to wind speed (complete data base)*



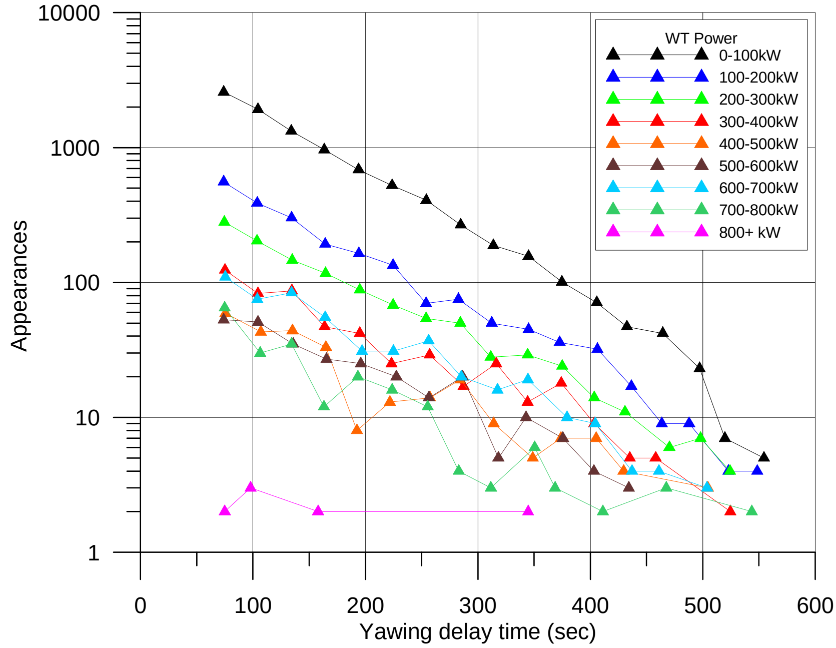


Figure 6-8: *Time between yaw starts with respect to WT power (complete operation range)*

With respect to the time series, as an example results of the analysis of a time-series corresponding to yaw motion, while the wind turbine is running up for start at cut-in wind speed are presented in following figures. Specifically, in Figure 6-9 the HSS speed is shown, along with the nacelle position and the yaw actuator power during the activity, for reference. From this figure one will notice that the yaw motion is not smooth containing halt periods and periods of constant yaw speed. This is also depicted on the yaw actuator power. The corresponding yaw bearing load magnitudes for the same yaw activity are shown in Figure 6-10, presenting similar fluctuations with the yaw actuator power during the first part of the activity (yaw start-50s), which are smoothed out on the second part (50s-end of yaw).
Using these values estimation of the bearing friction, as well as the yaw torque during the activity can be performed, as shown in Figure 6-11.

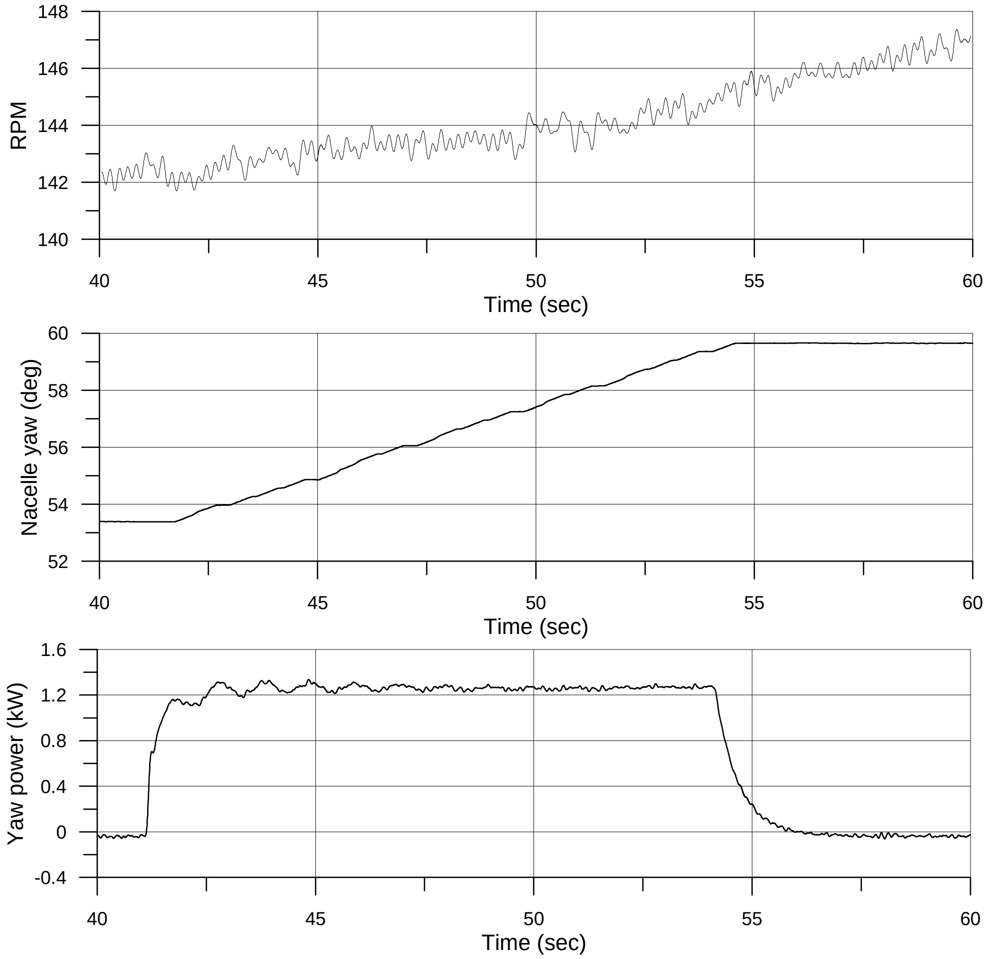


Figure 6-9: *Example time series of yaw movement analysis*

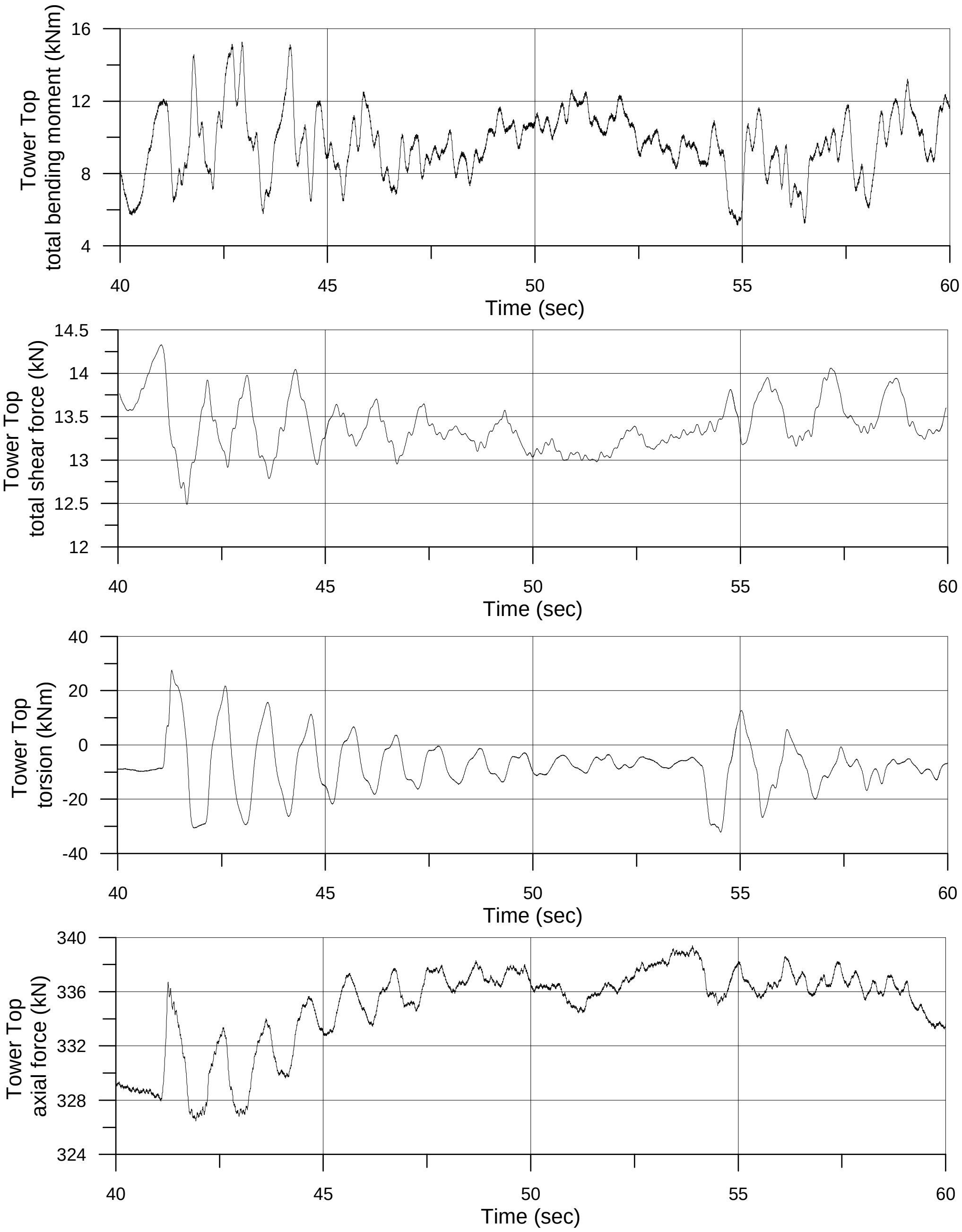


Figure 6-10: *Yaw bearing load magnitudes*

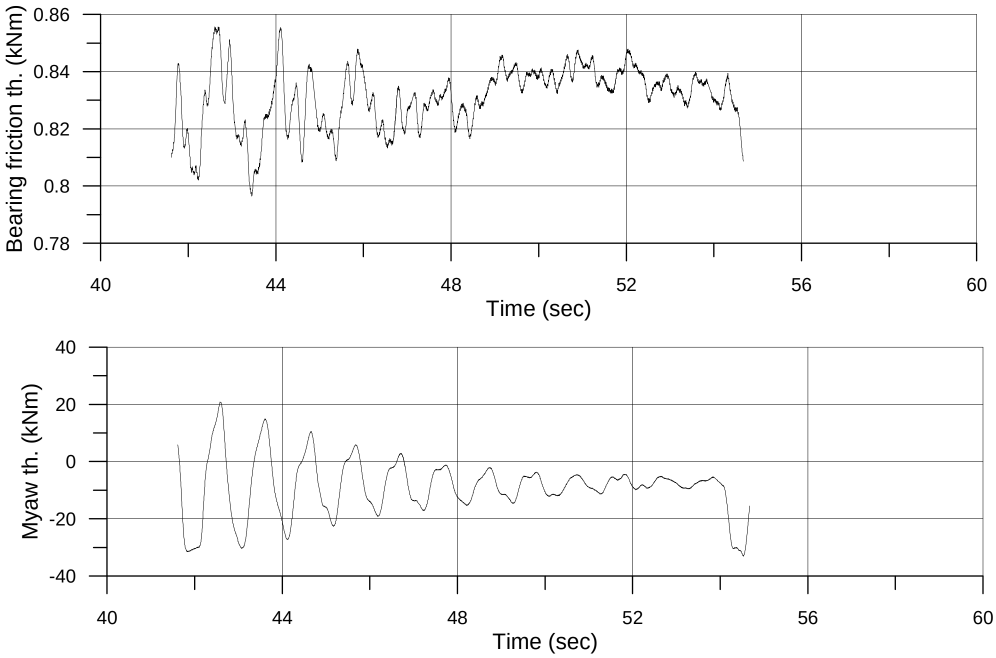


Figure 6-11: *Estimated yaw load magnitudes*

7. Recommendations for Standardisation

The implementation of the project results into practical certification procedures will be discussed in the following for the pitch and yaw system for the single components of these systems. Please note that the recommendations that concern the drive train can be found in section 4.2 (describing three new DLCs for the drive train) and in appendix A.

7.1 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.

7.1.1 Blade Bearing

7.1.1.1 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.

7.1.1.2 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 through 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.

7.1.2 Blade Pitch Drive

7.1.2.1 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 7.1.1.1), 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 7.1.2.2

7.1.2.2 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).

7.1.2.3 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.

7.2 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.

7.2.1 Yaw Bearing

7.2.1.1 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 7.2.1.2).

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

As already described in section 7.1.1.2 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 7.1.1.2 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.

7.2.2 Yaw Drive and Yaw Gear

7.2.2.1 Yaw Gear

For the fatigue proof of strength of the yaw gear the load dependent bearing friction torque (see 7.1.1.1), 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 7.2.2.2

7.2.2.2 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 7.1.2.1).

7.2.2.3 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.

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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 [23]) that are relevant for the drive train.