KAΠE
CRES
CENTRE FOR RENEWABLE
ENERGY SOURCES AND SAVING

Building a competitive Greece
MINISTRY OF DEVELOPMENT

PROcedures for TESTing and measuring wind energy systems (PROTEST)

Template for the specification of loads necessary for designing pitch systems

April 2010

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Grant Agreement no.: **212825**

Project acronym: **PROTEST**

Project title:

PROcedures for TESTing and measuring wind energy systems

Instrument: Collaborative Project

Thematic Priority: **FP7-ENERGY-2007-1-RTD**

Deliverable D4: Template for the specification of loads necessary for designing pitch systems

Date of preparation: April 2010

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WP Leader: CRES

Start date of project: 01.03.2008 Duration: 30 months

Organisation name of lead contractor for this deliverable: CRES

Project co-funded by the European Commission within the Seventh Framework Programme (2007-2013)		
Dissemination level		
PU	Public	x
PP	Restricted to other programme participants (including the Commission Services)	
RE	Restricted to a group specified by the consortium (including the Commission Services)	
CO	Confidential, only for members of the consortium (including the Commission Services)	

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1. Introduction

1.1. PROTEST project

High reliability of wind turbines and their components is one of the pre-requisites for an economic exploitation of wind farms. For offshore wind farms under harsh conditions, the demand for reliable turbines is even more relevant since the costs for repair and replacement are very high. Unfortunately, present day wind turbines still show failure rates between 2 to 5 failures per year that need visits from technicians (derived from i.e. [40], [41], [42]). 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. [43] and discussions with turbine manufacturers, component suppliers, and certification bodies [44] 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. [43] and expert discussions [44] that at present, the procedures for designing rotor blades and towers of wind turbines are much more specific than the procedures for designing other mechanical components such as drive trains, pitch and yaw systems, or main bearings. The design procedures for blades and towers are clearly documented in various standards and technical specifications. The reason for having extensive design standards for blades and towers is that these components are critical for safety: failures may lead to unsafe situations and designing safe turbines did have (and should have) the highest priority in the early days of wind energy. Parallel to the development of design standards, the wind energy community has developed advanced design tools and measurement procedures to determine the global turbine loads acting on the rotor and the tower. At present however, it is no longer acceptable to focus on safety only and neglect the economic losses. Lacking of clear procedures for designing mechanical components and specifying the loads on these components should no longer be the reason for early failures.

In 2007, ECN (NL) together with Suzlon Energy GmbH (DE), DEWI (DE), Germanischer Lloyd (DE), Hansen Transmissions International (BE), University of Stuttgart (USTUT; DE), and CRES (GR) decided to define the **PROTEST** project (**PRO**cedures for **TEST**ing and measuring wind energy systems) within the FP7 framework of the EU. The PROTEST project in fact is a pre-normative project that should result in uniform procedures to better specify and verify the local component loads acting on mechanical systems in wind turbines. The local component loads should be specified at the interfaces of the components. The relationship between global turbine loads acting on the rotor and tower and local component loads action on the interface of components is visualised in Figure 1.1. For gearboxes in common wind turbine architectures the special interfaces and load specification are explained in [1, Annex B].

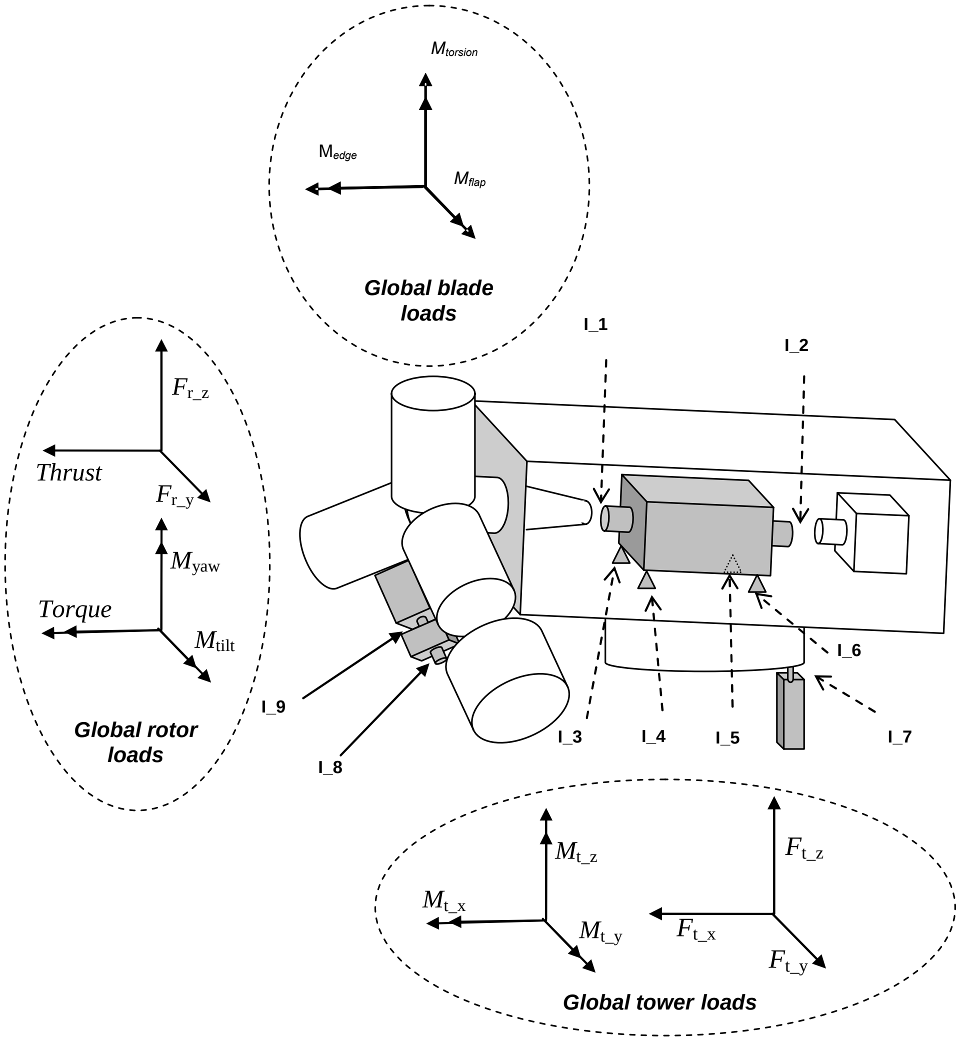

Figure 1.1: Schematic presentation of transforming "global turbine loads" to “local components loads” at nine interfaces, (gearbox, pitch system and yaw system)

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

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

(1) A method to unambiguously specify the interfaces and the loads at the interfaces where the component can be “isolated” from the entire wind turbine structure, and

(2) A recommended practice to assess the actual occurring loads by means of prototype measurements.

The following questions will be answered:

- How should the loads at the interfaces be derived from the global turbine loads?
- Which design load cases should be considered and measured and are relevant for the different components?

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

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

1. <u>State of the art report</u>: An inventory will be made of the present day practice on turbine and component design and testing, including ongoing standardisation work and identification of areas for improvement.
2. <u>Load cases and design drivers</u>: For the selected components, it will be determined which load cases and design driving factors (external, operational or design inherent) should be considered
3. <u>Loads at interfaces</u>: For the selected components, it will be specified how the loads at the design points should be documented with the aim of being a meaningful improvement over the current state-of-the-art (reporting format, time series incl. synchronisation and minimum frequencies, statistics, spectra, time-at-level, etc.)
4. <u>Prototype measurements definition</u>: For each component, a recommended measurement campaign will be defined taking into account the following aspects: load cases, signals (torques, bending moments, forces, motions, accelerations, and decelerations), sensors, measurement frequencies, processing, uncertainties and inherent scatter, reporting.

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

5. <u>Drive train</u>: Suzlon S82 turbine in India with gearbox of Hansen Transmissions.
6. <u>Pitch system</u>: Nordex N80 turbine owned and operated by ECN at flat terrain.
7. <u>Yaw system and complex terrain effects</u>: NM 750 turbine in Greece in complex terrain.

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

8. <u>Evaluation and reporting</u>: 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. <u>Management, Dissemination and Exploitation</u>

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

The project runs from March 2008 until mid 2010.

1.2. Work package 3: Loads at interfaces

This document is prepared within the frame of WP3. The report is aiming to serve as a template for the specification of loads (spectra, figures, time-at-level, displacements, etc.) necessary for designing the pitch system of a wind turbine.

Modern variable speed, pitch controlled wind turbines use the blade pitch angle to limit produced power above rated wind speed and to reduce blade loads around rated. The turbulent nature of the wind, combined with the instable character of a wind turbine, puts high demand on the pitch system. More advanced load reduction strategies, such as individual pitch control (IPC) to reduce wind shear loads on the blades, will impose even higher strain on the pitch system. In some wind turbines the pitch system also serves as one of the two required brakes, which makes it a safety system.

This report starts with the system and component definition following the results from WP2. The relevant standards and results from literature survey are listed. In chapter 4 issues concerning the design and modelling of pitch system components are discussed. This leads to the interface definition and the specification of the loads across these interfaces (chapter 5). Finally a format for the description and presentation of the loads is proposed in chapter 6.

2. Component definition

In order to identify the interfaces necessary for the design of each component, clarification of the system is an essential step. Therefore, in this section the components are defined as identified within WP2 of the PROTEST project [37]. The break down of the pitch system into subsystems/components presented in this section closely follows the results of WP2.

The pitch system allows the relative rotation of the blade with respect to the hub about the axis along the blade to enable adjustment of the pitch for changes in wind conditions. Below rated wind speed, or in case the wind conditions remain constant, the pitch system is employed to maintain the pitch of the blade at a constant angle. Moreover, the loads acting on the blade are transmitted to the hub through the pitch system. The pitch system is also employed as a brake of the wind turbine by using the aerodynamic properties of the blade to this end (which turns it into a critical component with respect to safety), as well as during maintenance operation to position the blade in predefined pitch setting.

Although various alternatives of pitch systems are currently employed, featuring hydraulic or electric actuators, etc., the pitch system in general consists of the following subsystems:

- pitch bearing: which transmits all loads from the blade to the hub and guides the rotation of the blade during pitching (only rotation is allowed). Static and load dependent bearing friction provides partially retention torque.
- pitch transmission systems/components: (rod, gears, etc.) which provide incremental motion or fixes the pitch setting of the blade when the pitch system is inactive.
- pitch driver: which provides the required driving torque to rotate the blade and can provide retention torque to keep the blade in the required pitch position (could be electric or hydraulic).
- pitch brake: which fixes the pitch position of the blade when the pitch system is inactive. This component is not present when passive (friction) or electrical (motor) braking is applied.

Figure 2.1 shows a picture of the above mentioned main components; section 4.2 treats these components and their interfaces in more detail.

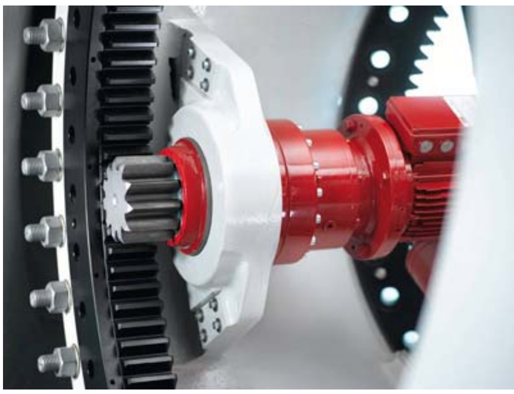

Figure 2.1: Main components of an electrical pitch system [http://www.wind-industry-germany.com/]

3. Survey of standards and relevant literature

Starting from the design guidelines (standards, regulations) for the components/systems a review of the relevant standards is performed in this section with focus on the load definition necessary for designing the pitch system of the wind turbine. Where applicable the reference standard for the design of a component of the system under investigation is mentioned. Additional standards (guidelines, specifications) cover testing and certification procedures are also addressed.

- **IEC 61400-1** [3]: Is the baseline standard covering the design of the whole wind turbine.

  In this document the design load cases for all wind turbine components are defined. It is also specified, which load cases will be taken into account within a fatigue analysis and which for an ultimate (extreme) analysis. Both the Normal Turbulence Model and Normal Wind Profile model are cases treated in fatigue analysis, including start-up, shut down events and parked conditions (for wind speed <0.7 V_{ref}). Wind speed distribution is prescribed.

- **IEC 61400-4** [1] (Currently ISO 81400-4:2005 [2]): Is the baseline standard covering the design of the gearbox of the wind turbine.

  Although this document is not directly addressing the design of the pitch system, it provides useful directions for the design of this wind turbine component. Through this document, the user is directed to IEC 61400-1 [3] for the definition of design load cases. However, a more detailed analysis is described in Annex B (ISO 81400-4:2005 [2]) regarding the load spectra, including both normal loads and transient events. Especially for the transient events, the wind turbine manufacturer is required to provide an estimation of the number of probable occurrences over the life time of the wind turbine, but these numbers are site dependant (e.g. driven by the reliability of the grid). In IEC 61400-4 [1] apart from reference to the IEC 61400-1 [3], it is stated that in the required specification at least the following should be provided:

  (1) a description of the DLC relevant for transmission design,

  (2) the frequency of occurrence,

  (3) the probability of occurrence e.g. abnormal or normal load case,

  (4) the duration of occurrence,

  (5) information on load calculation model including transmission model and

  (6) a reference to DLC, or identification or relevant partial safety factor for loads, with clear information whether these are already included, or need to be added.

  Design loads should be given at the interface (interconnection point). Great emphasis is given on this aspect in the IEC 61400-4 [1] document, including the definition of these points.

The current work will be limited to defining the load cases required according to the abovementioned documents. This is due to the fact that the standards for the design of the subcomponents are beyond the scope of this project. However, the relevant standards will be mentioned, including short descriptions.

- **ISO 6336(-1,-3,-5, -6):** Are the standards covering load capacity of spur & helical gears [4]-[8]. In particular ISO 6336-6 deals with the calculation of service life under variable load [8].

- **DIN 3990:** Is the standard to be followed for the fatigue rating of gears according to GL – Guidelines for the certification of wind turbines [9].

- **DIN 743:** Is the standard for shafts & axles [10]
- **ISO 76 & ISO 281:** Are the standards covering rolling bearings. ISO 76 covers the static loading [11] and ISO 281 the dynamic loading [12].

Additional standards/documents that provide important information for the performance of the scheduled work are the following:

- **IEC WT01:** Is the baseline standard covering the certification procedure of the wind turbine (including components) [13].
- **IEC/TS 61400-13:** Is the baseline standard covering load measurements on components of the wind turbine (field measurements) [14].
- **GL Guidelines for the certification of Wind Turbines** [9]**:** This document includes a description on what procedures to follow (including references to relevant standards) during the design of pitch bearings. Moreover, it is clearly stated in the document that for these components, not only the average values of the fatigue loads are necessary, but also the distribution of the Load Duration Distribution (LDD) should be specified. Additionally, there are directions on how to combine the Design Load Cases for the fatigue analysis.
- **ISO 8579-2:** Acceptance testing for gears – vibration level [15].
- **DIN 45667:** Is the standard covering the load duration distribution procedure [16].
- **GL Guideline for the certification of Condition Monitoring System for Wind Turbines** [17]**:** In this document additional information is provided on the measurements required to monitor the condition of wind turbine components.
- **Specification of manufacturers:** e.g. INA catalogue [18] and Rothe Erde technical document [19] covering slewing rings and FAG publication [20] covering the design of rolling bearing mountings.

NOTE 1: American Equivalents (e.g. ANSI/AGMA, ASTM) of the abovementioned standards are not referenced in the current document.

NOTE 2: Standards that are referenced and should be followed when applying the standards listed in the current document, but which refer to special design methodologies (material & lubrication specifications, etc.) are not referenced in the current document.

4. Modelling of components during wind turbine design

The process of designing a wind turbine can be divided into two stages. One stage involves the determination of loads and estimation of the wind turbine behaviour due to the stochastic wind loading and the other stage involves the detailed analysis of each component/system. During the design of a new wind turbine a loop is necessary for exchanging information between these two stages. In other words, using wind turbine nominal data (e.g. gross dimensions, reference values for cut-in, rated and cut-out wind speed, etc.) and employing assumptions, as the final design specifications are not at hand (at this stage), the first dimensioning loads are estimated through aerodynamic simulations. These are distributed to the designers of the basic wind turbine components, i.e. blades, gearbox, tower, generator, etc. After the initial dimensioning of the components the designer of each component provides details regarding the component for the initial aeroelastic analysis of the wind turbine. The scope of performing an aeroelastic simulation at this stage is to derive as accurately as possible the induced loads on the various wind turbine components, as well as to evaluate the overall behaviour of the wind turbine under the influence of the wind conditions. Multiple loops of the process result in the final load estimations and the detailed design of each component/system.

The outline of this chapter follows the above described process. The first section discusses issues regarding the aeroelastic simulation of the wind turbine for the pitch system. The second section describes the modelling of the components of the pitch system in detail. Conclusions on the modelling of components for the pitch system are in section three.

4.1. Aeroelastic simulation of the wind turbine

A review of state of the art aerodynamic and aeroelastic simulation procedures is given in [21]. In this work, a description of the structural modelling of the wind turbine is given: “The main components of a wind turbine are the blades, the drive train and the tower. They are all modelled as beam structures and typically the structural properties are assumed for each component to continuously vary along the corresponding elastic axis. However, localized properties can be added in the form of concentrated masses, dampers, or springs. The gearbox (if present), the generator, the hub are usually added in this way. Other examples are the flexibility or damping characteristics of the yaw bearing or the pitch mechanism. The involvement of different body motions for each component in combination with the connections where loads and displacements are communicated from one component to the other, calls for a global formulation of the dynamic problem. To this end most works adopt a multi-body approach, which consists of considering each component separately subject to appropriate boundary conditions, which fit the different components into the complete configuration”.

For all systems/components, the 3D structure is reduced to fit in the aeroelastic simulation. For example, the 3D structure of a multi-layered composite material blade is modelled as a beam with varying cross sectional properties along the span.

The specific information required for the pitch system to perform an aeroelastic simulation is:

- Blade inertia about the pitch axis
- Blade mass
- Blade centre of gravity
- Pitch bearing friction
- Pitch actuator inertia about the pitch axis (in case of electromechanical actuators)
- Pitch actuator model (time delay, spring-damper)

- Parameters of the controller, including pitching (angular) rate and acceleration limits

As an output of the model, sophisticated aeroelastic simulation tools provide the following information for the pitch system:

- Blade root forces and moments
- Pitch rotation (angle, speed, acceleration)
- Pitch torque

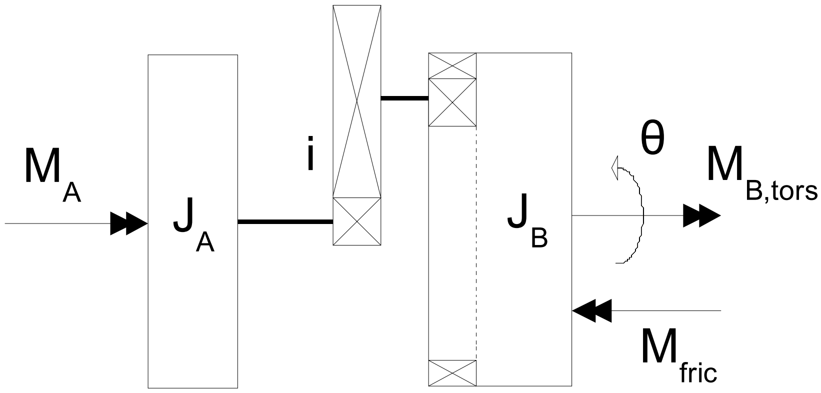

Figure 4.1: Basic pitch system model (sign conventions), only consisting of actuator *A* and blade *B* and a single transmission ratio *i*

For a more elaborate analysis of the pitch system, more detailed knowledge of the system response is necessary. For example, the damping of hydraulic actuators could be incorporated in the model, as well as the elasticity of the pitch system. Additionally, solutions could be provided depending on where you define the input and output of the pitch system. In other words, one can obtain a solution when modeling the input *to* the wind turbine system *of* the pitch system and another solution when modeling the input *of* the wind turbine system *to* the pitch system. These can form different load cases specifically designed for the pitch system.

Although the simulation tools are available, as also pointed out in [23], the following limitations in current simulation approaches are identified:

1) Aeroelastic simulation of the entire wind turbine to extract the entire set of load cases is time consuming. Discussions with aeroelastic analysts indicated a time frame of one week is needed for the extraction of the complete set of load cases described in the IEC 61400-1, however, using a simple model of the drive train. When using a more detailed model for the drive train, as in [24], the simulation time of an “emergency stop” transient load case increases drastically.
2) Although the transformation of the details of the 3D structure of the blades and the tower into a 1D structure, which is usually used during the aeroelastic simulation, is well documented and straight forward, the documentation of transforming the 3D structure/system of the pitch mechanisms into suitable information to be used in aero-elastic simulations of the wind turbine is not publicly available. The reasons behind this are twofold: on the one hand the sophisticated aeroelastic simulation tools employ a very simplistic modelling of the pitch system, while when more sophisticated simulation tools for the pitch system are used the aeroelastic input part of the structure is oversimplified, due to cost and time required for each of the two analysis types. On the other hand, as

pointed out in [23], the data transfer between parties, especially regarding the gearbox and bearings could be problematic.

3) Although the simulation runs are available and explicitly defined, the proportion of each load case in the entire life of the wind turbine is not. For some of the critical load cases for the pitch system such as starts and stops, emergency shutdown etc., there is no clear description of how many are expected during the life of the wind turbine, since these depend strongly on the site and on controller settings. Only GL – Regulation for the certification of Wind Turbines [9] provides some guidance on that aspect. In other words, aero-elastic simulations provide the load time series that should be taken into account, but the description of the life spectra, which is formed of these time series, is lacking.

4.2. Wind turbine component modelling

Taking the loads provided by the aeroelastic simulation tools as input, a detailed analysis of the wind turbine components is performed by the designer of the component/system. To this end, finite element methods are usually employed with varying modelling detail reaching up to the modelling of each rolling element or gear-tooth (depending on each case). In cases where the detailed analysis is performed for the design of components it involves proprietary information by the manufacturer/designer (e.g. Leaflet of SKF for pitch/yaw bearings [25]).

Much effort is put into verifying the modelling tools, in terms of assessing whether the model predicts the behaviour of the system accurately, e.g. [24] dealing with the drive train, [26, 30, 32] addressing the modelling of pitch bearing of a wind turbine. But still no matter how detailed the analysis is performed, no matter how accurately the response of the system is predicted, if the estimation for the loads during the operational life of the system is not accurate, the estimation of the operating life will be subject to uncertainties.

In this section, some design requirements will be discussed for the main components pitch system. The intention is not to cover all aspects of the component design, but to provide a basis for the selection of important signals to be specified at the interfaces and measurement campaign.

4.2.1 Pitch bearing

The pitch bearing enables the pitch rotation of the blade and transfers the blade loads to the hub. The intermittent rotation of the blade and the high moment on the bearing complicates the design of a pitch bearing. This section looks at design requirements for a pitch bearing from theory and practice to derive the definition of the interfaces.

Design requirements (theory and standards)

In a review on the available standards for rolling element bearing design [29], the following design considerations are identified:

- fatigue life (rolling contact fatigue)
- static load capacity
- surface and core hardness
- lubrication
- friction torque
- miscellaneous
    - external bolting
    - integral seals
    - cages/separators/gear

The first two, dealing with the bearing loading, and the calculation of friction torque will be described here in more detail.

A fatigue life calculation for rolling element bearings

There are several standardised methods for bearing life calculation, as summarised in [29]. The methods are based on the same basics, the Lundberg-Palmgren theory. This theory relates rolling contact fatigue life to the basic dynamic axial load rating C_a and the equivalent axial load P_{ea} as:

$$L_{10} = \left(\frac{C_a}{P_{ea}} \right)^p$$, with p = 3 for ball and p = 10/3 for roller bearings

The industrial practise is to specify bearing life according to a 10% probability of failure (L_{10}).

The difference between the three methods listed below is mainly the correction factors for special requirements/conditions/applications.

1) American National Standard method: American National Standard Institute / American Bearing Manufacturers Association (ANSI/ABMN) Standard 9 for ball bearings and Standard 11 for roller bearings. This method specifies correction factors for reliability (other than 90%), material (surface hardening), lubrication and other conditions (flexible support, oscillating motion).

2) International Standard method: the standard for design of rolling element bearings developed by the International Organization for Standardization (ISO 281 [12]) also contains a method for calculating bearing fatigue life. This method uses two correction factors. The first one specifies reliability. All other corrections (mainly dealing with lubrication) are combined in a single integrated life adjustment factor a_{ISO}. Surface hardening is not considered.

3) Stress life method: This method also has only one extra adjustment factor a_{SL} apart from the reliability factor, which is derived using the fatigue limit stress based on the Von Mises stress criterion. All the corrections that influence fatigue are converted to stresses and compared to the stress criterion to determine the possibility and probability of fatigue failure.

The pitch bearing of a wind turbine blade can be classified as a bearing that is in oscillatory rotation, which means that no full turns are made, but the bearing rotates forth and back over a certain angle θ. The critical amplitude of oscillation (θ_{crit}) is defined as the angle of rotation for which the raceway portion stressed by one rolling element touches, but does not overlap, the raceway stressed by adjacent elements. For very small oscillations, the dither amplitude (θ_{dith}) is defined as the width of the contact footprint of the rolling element divided by the radius of the rolling path.

Thus three regions of oscillation are defined:

$\theta_{crit} < \theta < 180°$ The contact stresses of the individual rolling element overlap.

$\theta_{dith} < \theta < \theta_{crit}$ The contact stresses of the individual rolling element do not overlap; each element has its own discrete stressed volume, which must be combined statistically to calculate the fatigue life. The bearing should be rotated over an angle greater than the critical as often as possible to redistribute the lubricant.

$\theta < \theta_{dith}$ As fretting corrosion most likely will occur, operation at these small oscillations should be avoided.

For θ > 180°, the behaviour is similar to normal rotation.

Bearings that undergo oscillating rotation are prone to false brinneling (see also [34]), which is surface degradation due to the local lack of lubrication. For bearings under these conditions, the Lundberg-Palmgren fatigue life calculation is modified [29].

B static load capacity

The static load capacity can be defined for:

1) maximum deformation

   The maximum load a bearing can take when it is not rotating (or oscillating), based on the maximum Hertz stress of the material for a permanent deformation at the contact point of 0.0001D (diameter of the rolling element).

2) maximum rolling element load

   For a pitch bearing, this load is a function of the applied blade root moments, radial and axial forces (see also [19]).

3) maximum contact stress

   This is the maximum Hertz stress acting at the rolling element and raceway contact. This is a function of the maximum load and the contact area.

C bearing friction

As being part of the load on the pitch drive train, bearing friction should be considered when designing a pitch system. The bearing friction moment consists of static friction (also referred to as starting friction) and dynamic friction, which is velocity dependent.

The static friction moment is dictated by the loading of the bearing. A general bearing friction model (as given in e.g. [31]) consists of several friction coefficients μ that depend on the type and design of the bearing (e.g. type and number of rolling element, clearance, seals etc.):

$$M_s = \mu_k \cdot M_k + \mu_r \cdot F_r \cdot D + \mu_a \cdot F_a \cdot D + M_0$$

with D the bearing diameter, $M_k = \sqrt{{M_f}^2 + {M_l}^2}$ the Kipp moment, $F_r = \sqrt{{F_f}^2 + {F_l}^2}$ the radial force and F_a the axial force on the bearing. Also a load independent moment M_0 can be present due to seals and lubrication.

In [29] and [34] estimates of the friction coefficients for large bearings as applied in wind turbines are given. The determination of the coefficients is part of PROTEST WP6.

Several models for the dynamic friction can be found in literature, e.g. Coulomb, viscous and stick-slip. A common way to describe the dynamic effect is to define velocity dependent coefficients. The velocity effect is estimated to be small for pitch bearings during normal production, due to the relative low rotational speeds. For special cases like an emergency stop with high pitch speed, it could influence loading of the drive train. The dynamic friction will be further investigated in WP6.

summary

The following items are important input to the pitch bearing design (calculations):

- bearing loading (moments and forces)
- bearing operation (rotation)
- lubrication (contamination)
- deformation (in relation to flexible support)
- geometry and assembly bolted connection (see [26, 32])

Design requirements (manufacturer)

For the detailed design of the pitch bearing the following information is requested (e.g. Rothe Erde GmbH KD 100 Questionnaire [19]):

- bearing diameter
- axis of rotation
- bearing under compression, tension or compression and tension[*] (for the pitch system: compression & tension)
- gears (depends on application; according to the specifications of the pitch drive train, the bearing can have internal, external or no gear teeth)
- movement type (for pitch system: intermittent rotation)
- applied loads

  These can be divided in Maximum Working Load, Maximum Test Load & Extreme Load. For the pitch bearing:

  - axial force
  - radial forces
  - bending moments
- collective loads with respective time percentages

  It should be made possible to distinguish between operating hours of the equipment and the actual rotating or slewing time. The various loads must be taken into account in the form of load spectra and percentages of time. For service life calculations one should distinguish between slewing angle under load and without load.
- circumferential forces (tangential) to be transmitted by the gear
  - normal
  - maximum
- speed of rotation or number of movements and angle per time unit together with relating collective loads
  - normal
  - maximum
- pinion data for checking meshing geometry of gears (if applicable)
- condensed stiffness data of the outer ring mounting structure (axial stiffness, bending stiffness, etc.)
- condensed stiffness data for the inner ring mounting structure (axial stiffness, bending stiffness, etc.)
- other operating conditions
  - operating temperatures
  - temperature differences between the outer and inner ring

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[*] For bearings it is important to know if the system will be under tension (e.g. to drive a suspended – hanging system-mass), under compression (e.g. to drive masses on top of the bearing) or alternating under compression & tension (as in the case of the blade pitch bearing)

4.2.2 Pitch transmission

The pitch transmission depends on the selection of the pitch drive (hydraulic/electric, collective/individual[1] pitch actuator). Figure 4.2 shows example layouts for both types of (individual) pitch actuators.

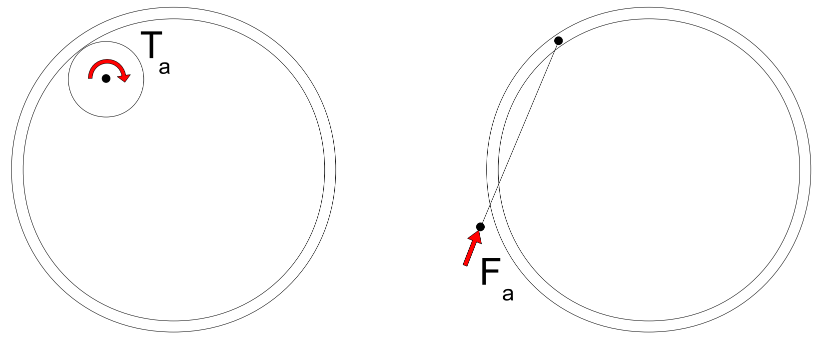

Figure 4.2: Setup of the last transmission stage of an electric (pinion - ring gear) and a hydraulic (pushrod) pitch system

Important aspects for the design of the pitch transmission system are:

- minimum and maximum of acceleration, speed and position
- transmission ratio
- torque, speed & power
- inertia of the components
- torsion stiffness of the components
- friction
- gear free play (loading of gear teeth, control delay)

For specific design requirements for the pitch gears (pinion & bearing ring gear, gearbox) and shafts IEC 61400-4 [1] and ISO 6336 [4-8] should be followed. From Figure 4.1 (and as in [31]), a basic relation for the meshing torque at the pitch bearing can be found, not considering drive train elasticity and gear free play etc:

$$M_{mesh} = M_{B,tors} - sign(\dot{\theta}) M_{fric} + J_B \cdot \ddot{\theta}_B = i \cdot M_A + i^2 \cdot J_A \cdot \ddot{\theta}_B$$

4.2.3 Pitch drive

The primary design requirement for a pitch drive is to set the pitch angle of the blade by producing the required torque (and speed) for rotation. Two types of pitch drives are applied in industry, based on the electric and the hydraulic principle. Both types lead to different requirements, which are discussed in this section.

Most drives will have their own controller, which can be a speed or position loop (or both). This issue will be discussed in section 4.2.4; for now a speed setpoint will be assumed.

[1] The wind turbine can have individual pitched blades, while having collective pitch control.

Using Figure 4.1, the load on the pitch drive can be expressed (at the blade root) as:

$$i \cdot M_A = M_{B,tors} - sign(\dot{\theta})M_{fric} + \left(J_B + i^2 \cdot J_A\right) \cdot \ddot{\theta}_B$$

Several details are not considered in this formulation, such as drive torque limits, elasticity in the drive train and gear mesh free play. These will be investigated in WP6 of the PROTEST project.

<u>electric drive</u>

An electric pitch motor is usually coupled to a speed reduction gearbox. The total transmission ratio (including the ring gear at the bearing) is high, in the order of 1000. For the design of the electric pitch drive, this means that the external torque on the motor (due to aerodynamic moment, drive train and bearing friction and blade inertia) is relatively low, while speed and acceleration are high.

The electric system of the drive consists of the following components and design criteria:

1) motor
    - peak load
    - revolutions and number of start ups (fatigue life)
    - thermal load -> temperature
    - (brake torque, voltage&current)
2) wiring
    - peak current
    - thermal load -> temperature
3) batteries
    - starting current (number of occurrence)
    - capacity

The load duration distribution (LDD) of the RMS value of the motor torque is representative for the thermal load on the motor [31].

For the design/selection of an electric pitch drive, electrical characteristics (such as voltage, current and frequency), size and mounting should also be considered.

<u>hydraulic drive</u>

As pitch speed is the most important requirement for the system, the design of a hydraulic pitch drive depends on the configuration of the transmission. Assuming a direct pushrod setup as shown in Figure 4.2, both rotation and torque (low transmission ratio) of the drive are important factors; not only for the actuator (required flow), but also for the control.

As indicated in [33], control of a hydraulic servo system for blade pitch is complex. The pushrod setup has a nonlinear relation between rod and pitch motion and the speed of a hydraulic actuator is proportional to the compressible fluid flow, which depends on the pressure (pump) and buffer in the system. Also the blade dynamics should be taken into account.

The hydraulic system consists of the following components:

1) actuator
    - peak load (pressure, flow/translation)
    - stroke

- o stiffness (intrinsic: type of fluid; external: control)
- o sealing

2) piping
   - o pressure drop
   - o sealing
3) pressurized fluid buffer
   - o capacity
4) pump
   - o nominal load (pressure, flow)
   - o operating hours
5) hydraulic fluid
   - o compressibility
   - o degradation (contamination, temperature)

Most important signals are the torque and rotation (speed, position). These must be specified such that both the blade and actuator dynamics are captured.

4.2.4 Pitch control

For variable speed pitch controlled wind turbines, the produced power and rotor speed are limited above rated by adjusting the blade pitch. Primary inputs to this control are the generator torque, rotor speed and blade pitch angle. The calculated setpoint is fed to the pitch driver. Figure 4.3 shows a typical basic wind turbine pitch control loop.

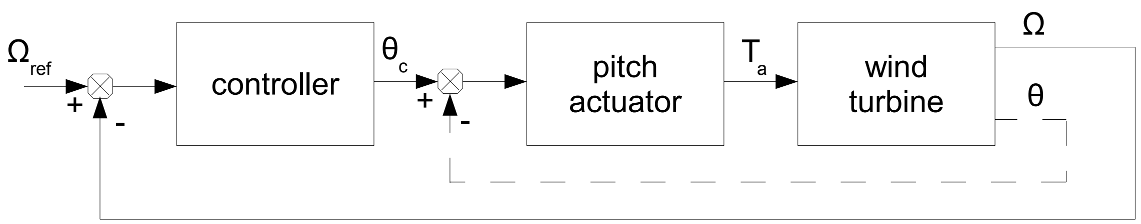

Figure 4.3: Basic wind turbine pitch control scheme

The wind turbine pitch control design is based on the following pitch system specifications:

- control setpoint (e.g. pitch angle, pitch rate)
- sample time
- overall actuator delay
- speed limit
- acceleration limit
- death band (hysteresis) pitch activity

As the control interface to the pitch system (section 5.1) is specified at the driver input, the transmission (cabling, slip ring) of the controller pitch setpoint and the power to the driver is not considered here.

As mentioned before, most currently applied pitch systems use either pitch angle or pitch speed as setpoint. The inner loop motor control then consists of position and/or speed feedback control (see Figure 4.3). As the control setpoint, constraints and actuator delay depend on this loop, a specification (type of control, control parameters and constraints) of this internal controller should be available to the wind turbine control engineer.

4.2.5 Pitch brake

If a separate mechanical pitch brake is present, consider the following aspects:

- Brake force
- Wear of brake pad and disc

4.3. Conclusions on modelling

A lot of effort is currently being put into the complete aeroelastic simulation of the wind turbine, including detailed information on all systems of the turbine. Ongoing work is performed especially for the drive train components in this field, e.g. within the project UPWIND WP 1B2 (contract no. SES6-019945). In order to improve the estimation of the operating life of the component, not only the time series of each loading case needs to be measured (estimated), but also the contribution of the various load cases to the expected life of the component. Load cases that are benign to the other components of the wind turbine, that is the blades and the tower, and which are usually ignored, can play an important role in the design (and operation) of components involving bearings and gears. For example, the load case "Wind turbine parked (standing still or idling)" is estimated, however, no indication is given on the duration and spread in time of this condition, which plays an important role for the yaw and main shaft bearings. In other words, it is different if the machine is parked for a long period of time, e.g. 600 hours continuously during one year, or if these 600 hours are spread over the year. This observation also holds in case of the pitch system. During the load case "1.2: Power production ($v_{hub} < v_{rated}$)" for example, the blade is loaded while the pitch is maintained at optimum angle (no rotation). The duration and spread in time of this condition is essential for bearing fatigue life calculation.

Moreover, a realistic approach should be given for the transient load cases. To be able to determine the number of occurrence of start and stops for instance, it does not suffice to define the cut-in and cut-out wind speeds. The normal starts and stops that the wind turbine will endure during its service life should also be included, irrespective of whether these are due to wind conditions (gust, direction change), decisions of authorities in the power supply lines for the interconnected wind turbines, or simply due to a weak grid. The GL-regulations for the certification of wind turbines [9] could provide a starting point in this aspect, which should be verified with experimental data or other statistics.

Understanding of the design requirements for the individual components is essential to be able to specify the definition of the interfaces. In case of the pitch system, the key component is the pitch bearing, which transfers the blade loads to the hub and enables pitch rotation of the blade to limit power and rotor speed (and reduce loads). The intermittent/oscillating rotation is an important aspect of the design. A theoretical assessment of both the LDD of the mean pitch angle (combining wind distribution, turbulence and control action) as well as a rain-flow count (amplitude of oscillations) should be validated with measurements.

The controller (on the input side of the system) usually provides pitch angle or speed setpoint with a given sample time. The overall pitch actuator delay is essential for model and control design validation.

The configuration of the pitch system depends on the selection of the other components (transmission, driver). The requirements of these specific components should be used to define the internal interfaces.

5. Load definition at the interfaces

5.1. Definition of interfaces

According to IEC 61400-4 the first step is to define the interfaces (interconnection points) for the design. It should be noted that the definition of interfaces follows the definition of interfaces of IEC 61400-4 [1], adequately modified to cover the needs of the pitch system addressed in the PROTEST project. That is, the interfaces are defined as: a defined boundary of the specific system that is either a physical mount to another wind turbine subcomponent or a path of exchange such as control signals, hydraulic fluid, or lubricant. Additionally, instead of the word "interfaces" the phrase "interconnection point" is used herein to connect with the potentials of simulation tools used in the wind energy sector, which provide the output data (displacements, forces, moments, etc.), on "nodes" of the modelled components of the wind turbine. For example the pitch system during simulation is the node connecting the blade root to the hub of the wind turbine and is given certain freedoms and constraints.

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 is shown in Figure 5.2 (internal interconnection points are indicated as i).

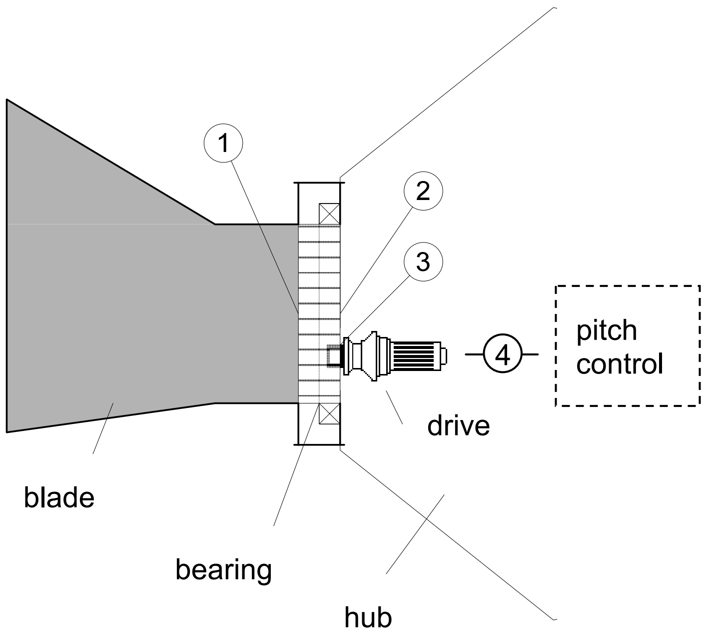

Figure 5.1: Simplified sketch of pitch system showing its main components

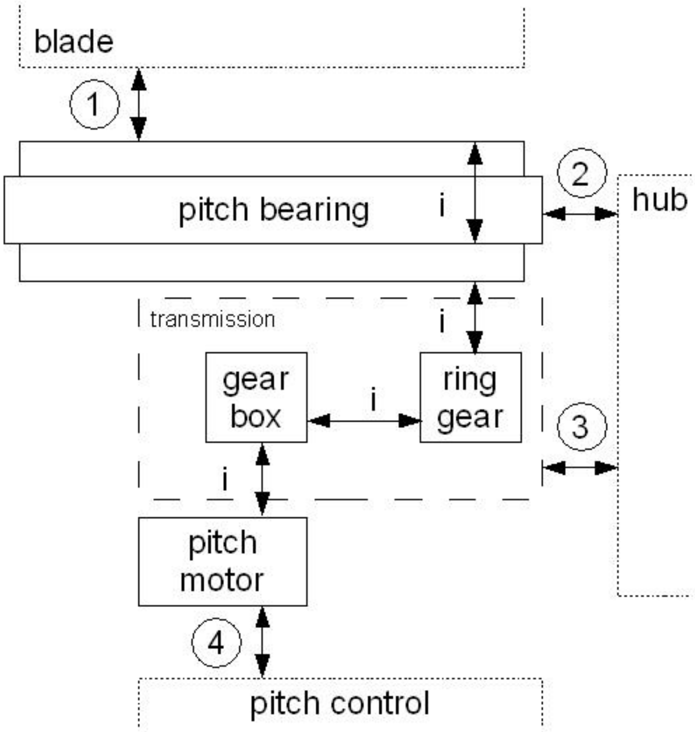

Figure 5.2: Schematic diagram of the pitch system and its interfaces

5.2. Definition of loads transferred across the interfaces

A clear definition of the loads, motions and processes that are transferred across the above defined interconnection points should be provided.

5.2.1 Design load cases

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. 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.

A large number of wind flow conditions (speed, turbulence, shear etc.) and operating states (start up, normal operation, (emergency) shutdown, standstill etc.) can be identified, resulting in a large number of DLC's. According to IEC 61400-4 [1], all DLC's should be analysed, so that the component designer can select the critical ones.

A discussion on wind turbine DLC's can be found in the WP2 report [37]. A selection of critical load cases for the pitch system can be found in the WP4 (draft) report []. The critical load cases for the pitch system are being analysed in WP6 of the PROTEST project.

5.2.2 Load at interfaces

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 [14]). 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.

Three possible routes are identified to determine the complete blade root loading (forces and moments at interface 1). All three paths have their own advantages and disadvantages.

1) measure all moments and forces

   This method requires four measurements (blade torsion moment, radial forces and axial force) that are not specified in IEC 61400-13 and are therefore not commonly installed. Also experience with these measurements on a wind turbine blade is lacking. However, if properly installed and calibrated, it will provide complete blade root loading based on measurements only.

2) measure moments and calculate forces using these measured moments, wind turbine operating point and blade design data (mass distribution and airfoil layout).

   This approach is described in more detail below (see textbox). It only requires additional torsion moment measurement, but is based on both measurements and (blade) design data. For now also quasi static situation is assumed.

3) measure moments and use a validated aeroelastic tool to calculate the blade root loading

   This is the commonly used approach. It also requires only additional torsion measurements. The blade moment measurements are used to validate the wind turbine model and the blade root loading is then calculated with aeroelastic simulations. However, the simulation results are as good and as detailed as the validated model. Moreover, this approach requires complete wind turbine design data (to be able to create the model) and a detailed measurement/description of the excitation (e.g. wind field). Direct comparison of simulated time series and (other) measurements on the wind turbine is difficult, due to differences between the real operating conditions and simulation.

> For the second approach, a new transformation from global to local blade loads is developed, which is described here in short (see appendix A for full text). The method derives the complete loading in the blade root (forces and moments) from the blade moment measurements by using the mass distribution and aerodynamic load distribution of the blade.
>
> In a quasi static situation, the sum of the aerodynamic, gravity and centripetal loads should be equal to the reaction force at the blade root. The gravity and centripetal load can be calculated from blade mass distribution, combined with azimuth angle and rotational speed. Subtracting the calculated moments at the measurement position due to these loads from the measured moments gives the moment due to aerodynamics. This is combined with the distribution to derive the equivalent aerodynamic force at the aerodynamic center of the blade.
>
> As all forces on the blade are now known, the blade root loads can be calculated (and easily transformed to hub center loads etc.).
>
> The method will be tested in WP6. The effect of the assumptions will also be investigated (some of the assumptions can be overcome by using more measurements of operational data).

It might be possible to include the pitch system (bearings) within a verification full scale blade test. In this case however, care should be taken to apply safety factors relevant for the testing of the bearing and not the blade material. Such a test could be used to verify the stiffness of the bearing under bending and axial loading.

Additionally, motion of the pitch system (while the system is maintaining blade position) could be measured on an operating wind turbine with vibration sensors positioned at the blade part of the pitch system, measuring possible small torsion vibrations (rotation and acceleration).

The measurement of pitch torque (load) depends on the pitch system. If the pitch system is electric then the torque of the pitch drive should be measured. In case of a hydraulic pitch system, then the axial force on the rod should be measured. Torque measurements on both sides of the pitch system (the blade and an electric pitch motor) will be investigated within WP6 of the PROTEST project.

Pitch position and acceleration are also measured during conventional load measurement campaigns. Excitation of torsion vibrations of the blade during pitching of the blade should be investigated within WP6 of the PROTEST project. This could also be part of an investigation using a detailed model of the blade, however, with the torsion response of the blade verified through testing.

For the electrical system, drive voltage and current are important signals. To verify control design assumptions, the pitch actuator delay should be determined from the pitch setpoint and resulting rotation.

The deformation (ovalisation) of the pitch bearing case could affect the loads of the pitch system [30, 34]. This deformation can be measured by either displacement measurements or tangential strains of the bearing case. Both measurements will be performed and analysed in WP6 of the PROTEST project.

A summary of the recommended measurements during an experimental campaign specifically designated on the pitch system is presented in the following tables. The “loading” at the interfaces is separated in loads (force and moment), kinematics (translation and rotation) and dynamics (combination of both).

Table 5.1 shows the loading at the external interfaces (section 5.1) and Table 5.2 shows loading of interest at some internal interconnection points between components of the pitch system (see Figure 5.2 for the definition of the interfaces of the pitch system). Finally some additional measurements are suggested to obtain more knowledge on the system and validate design calculations and models.

Table 5.1: Definition of loads at interfaces of the pitch system (external)

Interface	Loading	Synchronicity	Analysis
1) Blade & PS (bearing)	Loads: blade root forces (axial, radial shear) and moments (bending, torsion) Kinematics: (measured at 2) Dynamics: (measured at 2)	WTOD[2] blade pitch angle & pitch speed in 2)	extreme loads mean loads fatigue loads (LDDs)
2) Hub & PS (bearing)	Loads: (measured at 1) Kinematics: pitch angle & pitch speed Dynamics: acceleration on hub in two perpendicular directions	WTOD with loads in 1)	time at level of pitch angle (LDD) oscillation of pitch angle (rain-flow)
3) Hub & PS (transmission & driver)	Loads (driver): reaction torque/force of pitch driver on hub Loads (transmission): reaction torque (or force at torque arms) on hub	WTOD	
4) Controller & PS (driver)	Loads: driver voltage & current / pressure & flow Kinematics: control setpoint (pitch angle/speed)	WTOD with loads in 1)	thermal load (LDD of RMS value)

Additional analysis:

- bearing friction torque (from measured blade root and actuator moments)
- relation between pitch bearing loading and friction torque (determination of friction coefficients)
- correlation between pitch bearing loading and deformation
- gear loading (time duration distribution of meshing torque)
- pitch system time delay (from pitch control setpoint to blade pitch angle/speed)

[2] 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.

Table 5.2: Definition of loads at interconnection points between components (internal)

Interface	Loading	Synchronicity	Analysis
Bearing outer ring & bearing inner ring	Kinematics: clearance (at the four quarters on the bearing)	blade pitch angle & blade root forces and moments	
In case of an electric pitch actuator:			
Driver pinion & ring gear	Kinematics: relative angle of rotation[3]	blade pitch angle	
Gearbox & driver pinion	Loads: driving torque[3]		
Motor & transmission	Loads: driving torque[3] Kinematics: rotational speed[3]	blade pitch angle	peak load
In case of a hydraulic pitch actuator:			
Motor & transmission	Loads: force in driving rod[3] Kinematics: speed and position (nonlinear transmission) [3]	blade pitch angle	

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 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 (see section 4.2.1). 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.

[3] 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.

6. Description and presentation of loads

6.1. Description of loads

Irrespective of whether the loads at the interconnection points are the result of a simulation or a measurement, these come in the form of a time series. According to IEC 61400-4, the output of the simulation runs for each Design Load Case (DLC) should include the following information:

- Description of the DLC relevant for component design (but the relevancy should be decided by the component designer, therefore, all prescribed DLCs should be provided)
- Frequency of occurrence
- Probability of occurrence e.g. abnormal or normal load case
- Duration of occurrence
- Information on load calculation model including transmission model
- Reference to DLC, or identification or relevant partial safety factor for loads, with clear information whether these are already included, or need to be added

IEC 61400-4 also describes that the loads should be documented including:

1) Time series presentation
2) Rain-flow count tables including information on:
    - Which design load cases (DLCs) have been considered
    - The frequency of occurrence for each DLC considered
    - Information on safety factors already applied or to be applied
3) Load Duration Distribution (LDD) expressed as time at level
    - Which design load cases (DLCs) have been considered
    - The frequency of occurrence for each DLC considered
    - Information on safety factors already applied or to be applied
    - Nominal torque
    - Nominal rotational speed

However, it should be mentioned that the relevant standards for the component analysis (e.g. ISO 6336-6) use the Palmgren-Miner cumulative damage calculation principle, which is based on number of cycles at load level (torque level).

It is noted that a major difference from how the loads are presented up to now is the Load Duration Distribution. For the analysis of the time series measured during a wind turbine load measurement campaign, IEC/TS 61400-13 [39] prescribes that analysis includes the estimation of rain-flow matrices (through application of the rain-flow counting method), the definition of the load spectra (through combination of the rain-flow matrices) and the calculation of equivalent loads at a given frequency (usually at 1Hz).

The loads should be measured along with pitch rotation, due to the influence of oscillating rotation on for instance bearing fatigue life.

6.2. *Proposal for the presentation of load measurements*

IEC/TS 61400-13 [39] 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 pitch system.

In addition to the loads described in the IEC/TS 61400-13, i.e. the bending moments on the blade, the torsion will also be measured on the blade. The calculation of the forces and transformation of the loads to the blade root can be done using the procedures described in 5.2.2.

The loads will be provided in the usual layout followed for the presentation of loads according to IEC 61400-13. However, also LDD analysis will be performed (at least for the blade root moments). The time variation of the blade root forces is expected to be similar to the blade bending moments usually measured according to IEC 61400-13. The torsion moment however, should have a higher sampling frequency to capture the torsion dynamics of the blade. The first blade torsion frequency of a typical 2MW wind turbine is in the order of 10Hz (as opposed to 1Hz 1st flap bending mode), which sets the required sampling frequency to 80 Hz.

Additional presentation of loads with respect to the azimuth angle of the rotor, as well as the pitch angle of the blade will be performed. This will cover the loads measured at the blade.

The intermittent/oscillating behaviour is essential for pitch bearing design and life time. Time at level of the mean pitch angle, as well as rain-flow count (amplitude of oscillation) should be used to validate the design calculations (e.g. bearing fatigue life).

Capture matrices for both normal power production and transient events (including parked conditions, grid failures, etc.) should be linked to the pitch operation and angle. The same holds for the sample record measurement and analysis. Since it is anticipated that the pitch operation (pitching) sometimes will not cover the whole 10min file, special treatment of these captured files is foreseen. In these cases, a more extensive analysis should be performed, covering more than one data set (10min file). It is also likely that during normal operating conditions the blade pitch position will mostly be in the lower 30° of the range and not cover the full 90°. However, IEC 61400-13 does prescribe cases that cover the higher pitch angle range (e.g. start up, emergency shutdown).

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A. Calculation and transformation of blade root loads

The only required blade load measurements, as specified in the IEC 61400-13 standard on Measurement of mechanical loads [39], are blade bending moments. These bending moments are usually measured with strain gauges positioned at a certain distance from the blade root. The measurements performed on the N80 and described in the measurement report [35] are an example of this approach.

For pitch bearing design, a full definition of the blade root loading is essential. Fatigue life and static friction for instance, are both a function of bending moments and radial and axial forces. To derive a full definition of the loading (forces and moments) in the blade root, the following issues are encountered:

- blade root forces are not measured -> not required for certification, difficult for glass fiber composite material

- blade moments are not measured in blade root -> transformation over distance not possible without forces

This means that the measured blade loads cannot be used to derive the blade root moments and are difficult to transform to any other part of the wind turbine. Moreover, no measured blade forces are available.

The approach until now has been to derive rotor collective forces from tower measurements [35]. Downsides of this method are that only collective rotor loads are available and secondary relations are inevitable. For instance, the axial thrust (combined blade forces aligned along the rotor axis) is found from the measured tower bending moment, compensated for the tilting moment of the nacelle weight on the tower top.

This section proposes a method to derive the complete loading in the blade root (forces and moment) from the blade moment measurements by using the mass distribution and aerodynamic load distribution of the blade. As a start, only the span-wise offsets of forces are taken into account, but this can be extended to a more general formulation. The method will be tested in WP6 of the PROTEST project.

A.1 Method intro

A new transformation from global to local blade loads is developed, which derives the complete loading in the blade root (forces and moments) from the blade moment measurements by using the mass distribution and aerodynamic load distribution of the blade.

The global loads on the blade are the result of:

1) aerodynamics
2) gravity

A ‘virtual’ part due to the blade rotation:

3) centripetal (constraint rotation) and Coriolis (in plane movement in rotating system) effect

Finally, a part from yaw motion:

4) gyroscopic effect due to the conservation of angular momentum (out of plane rotation of rotating system)

These loads are counteracted by the blade root loads, where the rotor is attached to the pitch bearing and pitch drive.

The net blade load will result in acceleration of the blade:

1) rigid body motion (rotor acceleration & pitch motion)
2) structural dynamics (blade deformation)

In a quasi static situation, the sum of the aerodynamic, gravity and centripetal loads should be equal to the reaction load at the blade root. The gravity and centripetal load can be calculated from blade mass distribution, combined with azimuth angle and rotational speed. Subtracting the calculated moments at the measurement position due to these loads from the measured moments gives the moment due to aerodynamics. This is combined with the distribution to derive the equivalent aerodynamic force at the aerodynamic center of the blade.

As all forces on the blade are now known, the blade root loads can be calculated (and easily transformed to hub center loads etc.).

The method is derived using the following assumptions,

- quasi static situation (no rotor, pitch etc. accelerations)
- neglect yaw motion (no gyroscopic effect)
- only span-wise offset of loads
- simplified aerodynamics

Most of the assumptions are not required when using more measurements of operational data (e.g. rotor acceleration, tower top acceleration, yaw speed and pitch drive torque). This is likely to improve the method and will also be investigated in WP6.

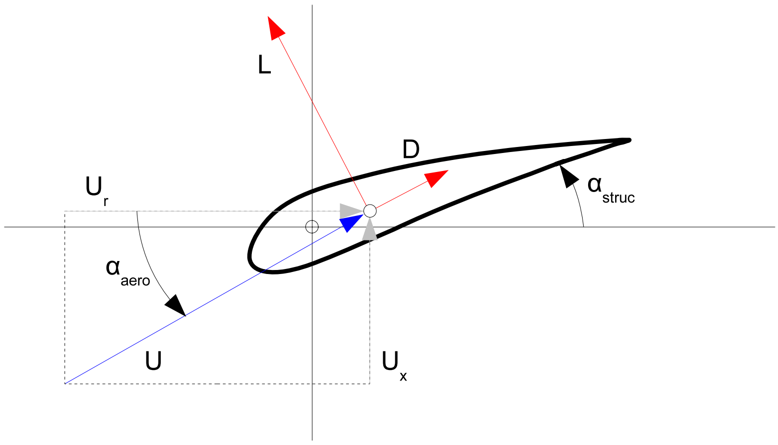
A.2 Calculation of blade aerodynamic center

The aerodynamic load distribution is calculated from the aerodynamic blade design, at each operating point of the wind turbine. This is used to derive the position of the aerodynamic centre, which is key to the method.

This section outlines the procedure to find this aerodynamic centre. First some background is discussed, followed by the practical approach for use with measurements.

<u>background</u>

The aerodynamic layout of a wind turbine blade is built up from several airfoils with different aerodynamic properties. These are defined by the lift and drag of the shape, specified as coefficients c_L and c_D, as function of angle of attack (see Figure A.1). For a certain operating point of the wind turbine, defined by wind speed V_w, rotational speed Ω, azimuth angle ψ and pitch angle θ, the angle of attack α at each blade element can be calculated as shown below. In this case, exponential vertical shear (with coefficient α_{shear}) is assumed.



$$U_x = V_w \cdot \left(\frac{\left(h_{hub} + \cos(\psi) \cdot R_e\right)}{h_{hub}} \right)^{a_{shear}}$$

$$U_r = \Omega \cdot R_e$$

$$\alpha_{aero} = \tan\left(\frac{U_x}{U_r}\right)$$

$$\alpha_{struc} = \alpha_{twist} + \theta$$

$$\alpha = \alpha_{aero} - \alpha_{struc}$$

Figure A.1: Flow around a wind turbine blade airfoil

Using the theoretical c_L and c_D tables, the lift and drag forces at each element can be calculated:

$$L = \tfrac{1}{2} \rho \cdot c_L \cdot c \cdot U^2$$

$$D = \tfrac{1}{2} \rho \cdot c_D \cdot c \cdot U^2$$

with chord c and air density ρ.

Lift and drag forces are defined along the resulting inflow angle, and have to be rotated over (α_{aero}-α_{twist}) to be transformed to the blade coordinate system.

With this aerodynamic load distribution, the aerodynamic centre on the blade can be calculated as:

$$x_L = \frac{\sum_{e=1}^{N_e} L_e \cdot r_e}{\sum_{e=1}^{N_e} L_e}$$

Figure A.2 shows an example of the distribution of the aerodynamic load over the blade for increasing wind speed. Figure A.3 shows of the resulting aerodynamic centre as function of wind speed.

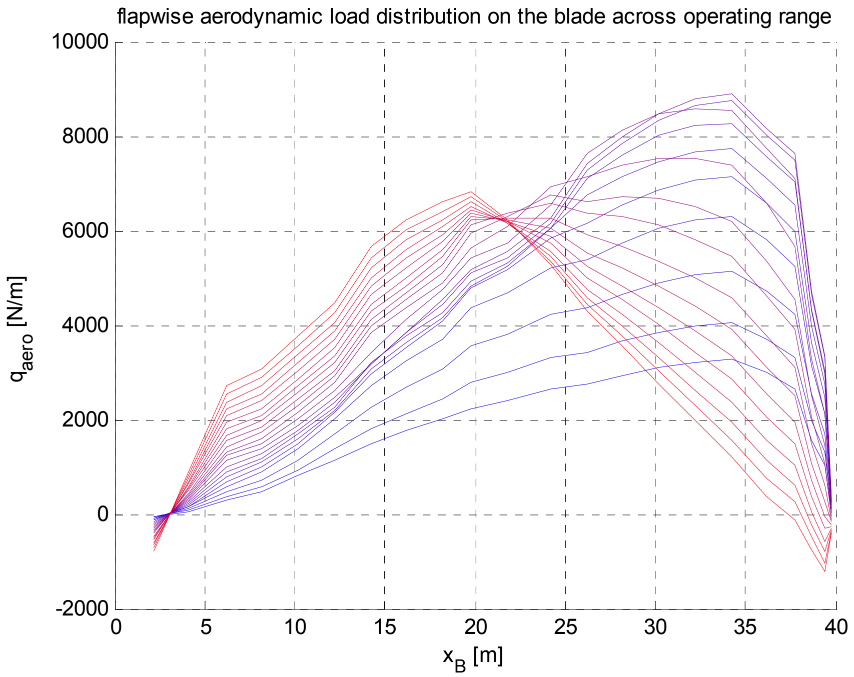


Figure A.2: The aerodynamic load distribution for increasing wind speed (blue->red)

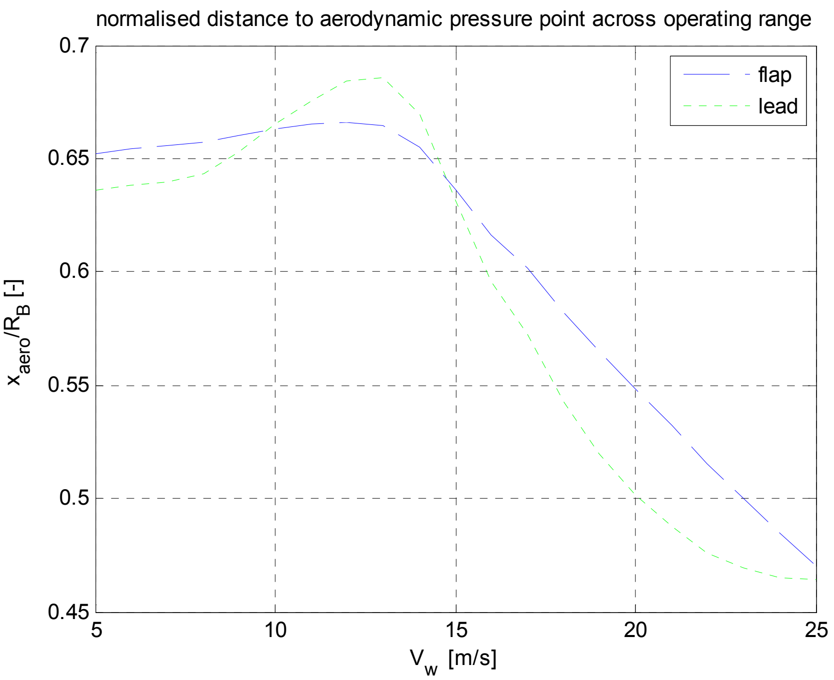


Figure A.3: The aerodynamic centre as function of wind speed

practical approach

It is inefficient to recalculate the aerodynamic centre each sample time, so a lookup table is created containing the aerodynamic centre for a number of points on the whole operating range. The aerodynamic centre for the actual operating point at the current time step is then obtained through interpolation.

A.3 Global to local load transformation

Annex A of internal memo [36], describes the transformation of global to local loads (change of reference frame) anywhere on the wind turbine. Transformation matrices are defined based on the yaw, tilt, azimuth, cone and pitch angle of the wind turbine. The transformation matrix from nacelle to the rotating hub coordinates (Figure A.4) is shown below as an example.

$$\vec{F}_{\mathrm{h}} = \begin{pmatrix} 1 & 0 & 0 \\ 0 & \cos\psi & \sin\psi \\ 0 & -\sin\psi & \cos\psi \end{pmatrix} \cdot \vec{F}_{\mathrm{n}} = T_{n2h} \cdot \vec{F}_{\mathrm{n}}$$

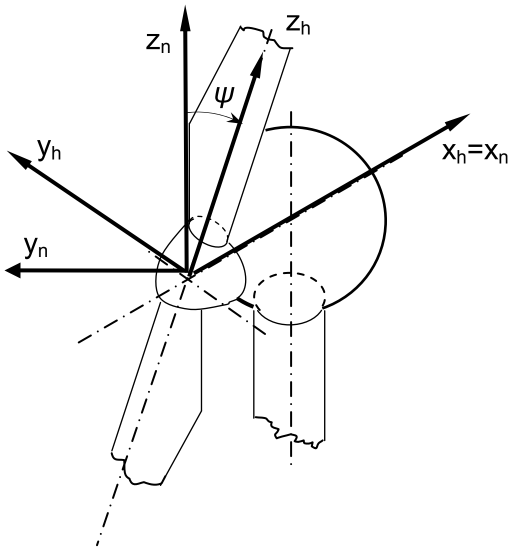


Figure A.4: The nacelle ($\cdot_{\mathrm{n}}$) and hub ($\cdot_{\mathrm{h}}$) coordinate system

One has to bear in mind that this transformation only involves the rotation of the coordinate system; no translation is performed (i.e. the point of application of the force vector does not change).

The moment *M* about the point A due to force *F* in point B (at distance r_{AB}) is the vector cross product:

$$\vec{M}^{A} = \vec{r}_{AB} \times \vec{F}^{B}$$

For now, only a spanwise offset is assumed. To calculate the contribution of the local blade load L^b to the blade root load L^{br}, the following transformation has to be performed:

$$\vec{L}_i^b = \begin{pmatrix} \vec{F}_i^b \\ \vec{M}_i^b \end{pmatrix}$$

$$L^{br} = \begin{pmatrix} 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & -d & 0 & 1 & 0 & 0 \\ d & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \end{pmatrix} \cdot L^b\Big|_d = T_L\Big|_d \cdot L^b\Big|_d$$

with *d* the spanwise distance from point of application to blade root.

A.4 Local blade loads

MASS

The force on mass *m* due to gravity can be written as:

$$F_G = g \cdot m$$

with the gravitational acceleration g = 9.81 m/s^2.

The distributed load on the blade can be represented by one equivalent force acting in the centre of gravity (cog). The distance from the blade root to this cog is calculated as:

$$x_{cog} = \frac{\sum_{e=1}^{N_e} g \cdot m_e \cdot r_e}{\sum_{e=1}^{N_e} g \cdot m_e}$$

The gravity force is a global load, acting at the blade centre of gravity (cog) in the inertial reference frame. The transformation to the blade coordinate system is therefore:

$$\vec{F}_G^b = T_{g2b} \cdot \vec{F}_G^g = T_{bf2b} \cdot T_{h2bf} \cdot T_{n2h} \cdot T_{tt2n} \cdot T_{g2tt} \cdot \vec{F}_G^g$$

CTP

Although assuming quasi-static situation, constant rotation still requires an inward acceleration. The centripetal force of mass *m* rotating with constant velocity *ω* at radius *r* is defined as:

$$F_C = m \cdot r \cdot \omega^2$$

The centripetal force works in the plane of rotation, and acts along the z-axis of the hub coordinate system (see Figure A.4). The transformation to the local blade coordinate system is therefore:

$$\vec{F}_C^b = T_{h2b} \cdot \vec{F}_C^h = T_{bf2b} \cdot T_{h2bf} \cdot \vec{F}_C^h$$

The centripetal load is a function of the radius; therefore its equivalent point of application is different than the cog, but is calculated similarly.

AERO

The moment due to aerodynamic load is obtained as being the remains of the measured moment minus the calculated moments at the measurement position due to gravity and centripetal forces:

$$\vec{M}_A^m = \vec{M}_M^m - \left(\vec{M}_G^m + \vec{M}_C^m\right)$$

This moment is caused by the lift and drag force at distance x_{aero}, as defined previously in this appendix. The aerodynamic contribution to the torsion moment is neglected for now. The forces are calculated as:

$$\vec{F}_A^{\,b} = \begin{pmatrix} \dfrac{\vec{M}_A^{\,b}(2)}{x_{aero}(1)} \\ \dfrac{\vec{M}_A^{\,b}(1)}{-x_{aero}(2)} \\ 0 \end{pmatrix}, \vec{M}_A^{\,b} = \begin{pmatrix} 0 \\ 0 \\ 0 \end{pmatrix}$$

$$\vec{L}_A^b = \begin{pmatrix} \vec{F}_A^b \\ \vec{M}_A^{\,b} \end{pmatrix}$$

BLADE ROOT LOAD

To finally calculate the contribution of the local blade loads to the blade root load, the following transformation has to be performed for each component *i* at point of attachment x_i:

$$\vec{L}_i^{br} = T_L\big|_{x_i} \cdot \vec{L}_i^b\big|_{x_i}$$

The total blade root load is the sum of the components:

$$\vec{L}_T^{br} = \vec{L}_G^{br} + \vec{L}_C^{br} + \vec{L}_A^{br}$$

As only spanwise offset is assumed for the point of attachments, the measured torsion moment directly transfers to the blade root as:

$$\vec{L}_T^{br}(6) = \vec{M}_M^{\,m}(6)$$