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Investigation on pitch system loads by means of an integral multi body simulation approach

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Abstract. In modern horizontal axis wind turbines the rotor blades are adjusted by three individual pitch systems to control power output. The pitch system consists of either a hydraulic or an electrical actuator, the blade bearing, the rotor blade itself and the control. In case of an electrical drive a gearbox is used to transmit the high torques that are required for blade pitch angle adjustment. In this contribution a new integral multi body simulation approach is presented that enables detailed assessment of dynamic pitch system loads. The simulation results presented are compared and evaluated with measurement data of a 2 MW-class reference wind turbine. Major focus of this contribution is on the assessment of non linear tooth contact behaviour incorporating tooth backlash for the single gear stages and the impact on dynamic pitch system loads.

Keywords. wind turbine simulation, pitch system, multi body simulation, gears, backlash

1. Introduction

Wind turbines (WT) are not only growing in rotor size to increase nominal power production but also to assess low wind sites. By increasing rotor diameters, higher loads are introduced to the turbines components. In recent years, advanced control strategies for load alleviation of individual WT components, most likely the blades or the tower, have been introduced to reduce dynamic loads. However, as pitch angle is a mature control value, additional dynamics are added to the pitch system that has to set the pitch angles required. The most popular load reduction approach is individual pitch control (IPC) that adjusts the pitch angles of each blade in reference to local loads acting on the blades’ root individually to mitigate structural loads. To achieve the control objectives, additional pitch angle adjustments per revolution are performed adding extra pitch drive activity. This might induce additional loads on the components of the pitch system. In modern WT the pitch system consists of either a hydraulic or an electrical actuator, the blade bearing, the rotor blade itself and the control. In case of an electrical drive a gearbox is used to transmit the high torques that are needed for pitch angle adjustment. In this paper an integral, hybrid multi body simulation (MBS) modelling approach including turbine control is presented, that enables further examination of the dynamic loads acting on the pitch system and load effects that are introduced by the pitch system itself for different control strategies. Therefore, the paper is organized as follows. In section 2 the principal approach for the study is described. In section 3 the MBS model, the aerodynamic load calculation and the control are presented. In section 4 simulation results are presented and compared with
measurement data. In section 5 the conclusions are drawn and the outlook for future work will be given.

2. Approach

Aim of the investigation is to get a deeper insight into pitch system loads acting on the system and also the ones introduced by the system itself. Therefore, in a new approach, the mechanical pitch system including pitch drive, gearbox and pitch bearing friction model have been integrated into a wind turbine model. The multi body simulation (MBS) framework used for modelling is SIMPACK [1], commonly used for WT system and subsystem analyses at CWD [2, 3, 4]. The model had initially been briefly described in [5, 6]. By introducing the pitch system into the turbine model, the coupled behaviour between pitch system and turbine structure can be incorporated. The model introduced is evaluated by comparison to field measurement data of a 2 MW-class reference WT.

The structural properties of the turbine model are based on the technical data of the reference field turbine. The technical data for the pitch system drive train used in the reference such as dimensions and tooth properties had been extracted from design data provided by the pitch drive supplier. The gearbox of the pitch drive consists of three stages in total. At the first stage, additional measurement devices had been installed to measure torque and rotational speed at the three pitch drive gearboxes each. In this contribution the collected data from those measurements is used to investigate dynamical pitch system loads. Besides this, measurement data of main shaft input torque and speed as well as produced power at the generator were provided. Wind speed conditions were estimated from those values. A WT controller was coupled to the mechanical model and extended by an additional pitch drive control. By applying different control strategies, the impact on pitch system loads due to higher pitch activity can be evaluated. The general modelling approach is described more in detail in section 3.

3. MBS wind turbine model and control

![Figure 1: MBS models of the pitch system and the wind turbine model](image)

(a) Geometric representation of the pitch drive model

(b) Topology of the integral MBS model
Wind turbine model
The mechanical wind turbine model including the mechanical pitch system model is modelled in Simpack. The model topology includes flexible blades, flexible tower and a reduced drive train model as well as the three pitch system models. Thereby, the pitch system for rotor blade 1 is modelled incorporating a detailed pitch drive model, while for rotor blade 2 and 3 the mechanical pitch drive models were reduced to their total inertia. The blades of the turbine were modelled flexible in 8 modal degrees of freedom (DOF) incorporating the first torsional blade mode. The tower was modelled with 4 modal DOF, taking into account the first and second fore-aft and side-side tower modes, respectively. The aerodynamic loads acting on the three rotor blades are determined using the Aerodyn v13 code provided by NREL [7] and implemented as a routine into Simpack at CWD. The rotor blades are coupled to the outer ring of the rotor blade bearings with 0 DOF each. The outer ring is coupled to the inner ring with one rotational DOF. The inner ring is coupled to the hub by 0 DOF. The rotor drive train is represented by its overall inertia and coupled to the mainframe structure with a single rotational DOF. The complete model topology is shown in figure 1b.

Pitch system model
To investigate pitch drive dynamics, for rotor blade 1 a detailed pitch drive model was integrated. Figure 1a shows the geometrical representation of the model, the topology is shown in figure 2. The pitch system’s drive train consisting of motor and gearbox was modelled in torsional DOF. The gearbox consists of a bevel gear stage and two planetary gear stages; the pitch torque is transferred to the blade by an additional gear stage at the blade bearing. Blade bearings’ inertia as well as motor inertia were taken into account.

For the gearbox shafts, the torsional stiffness of the individual components were considered. For the planet carriers - because of the complex geometry - torsional stiffness characteristics were derived from finite element (FE) models and integrated into the MBS model. Each tooth contact was modelled by Simpack’s analytical tooth contact modelling element taking into consideration linear tooth flank deformation as well as non linear tooth meshing stiffness characteristics. Furthermore, tooth backlash for each tooth system was taken into account. Backlash values were taken from the suppliers design data.

Figure 2: Topology of the mechanical pitch drive model of rotor blade 1
Blade bearing friction model

Besides the blade bearings inertia, the inner friction moment of the bearing has been taken into consideration. It is of high interest for pitch drive design as it acts load dependent and has major impact on the pitch drives loads. In wind industry, blade bearing friction is used to be modelled by a static model based on Coulomb’s law of friction [8, 9, 10, 11]. It can in general be described according to equation 1.

\[ M_{fric} = \mu_b M_b + \mu_r F_r d_b + \mu_a F_a d_b + M_0 \]  

Thereby, \( \mu_b, \mu_r, \mu_a \) are the friction coefficients of the load components; \( M_b \) the resultant bending moment, \( F_r \) the total radial force, \( F_a \) the total axial force that act at the blade root on the bearing ring, connected to the blade. \( d_b \) is the bearing diameter and \( M_0 \) accounts for load independent friction moments resulting from e.g cage and sealing friction.

Supplier companies provide modified equations fitted to their bearings that can be used for design purposes, see e.g. [12, 13]. Equation 2 shows the equation provided by Rothe Erde, often used in literature.

\[ M_{fric} = \mu/2(4.4M_b + F_a d_b + 2.2F_r d_b \cdot 1.73) \]  

This equation does not account for load independent and other losses, so that accuracy is said to be \( \pm 25\% \). In the study presented, this model is chosen to model blade bearing friction. As the friction moment always acts against the rotational direction, the model described in equation 2 was extended by the signum-function, see equation 3.

\[ M_{fric,mbs} = M_{fric} \cdot \text{sign}(\dot{\beta}) \cdot g(\dot{\beta}) \]  

Furthermore, as the equation is not well defined for \( \dot{\beta} \rightarrow 0 \), an additional function \( g(\dot{\beta}) \) is introduced to stabilize the equation around \( \dot{\beta} = 0 \) according to the scheme provided in 4.

\[ g(\dot{\beta}) = \begin{cases} 
1, & \dot{\beta} < -m \\
\frac{1}{m} |\dot{\beta}|, & -m \leq \dot{\beta} \leq m \\
1, & \dot{\beta} > m 
\end{cases} \]  

Thereby the constant value \( m \) is chosen to be close to zero.

A side effect of this load dependent friction model is that, besides torsional rotor blade dynamics, it couples out of plane and in plane rotor blade dynamics into the system.

Wind turbine and pitch drive control

The integral pitch system model is controlled to operate under a variety of load conditions. Therefore, the WT control was modelled in MATLAB/SIMULINK and linked to the SIMPACK solver via co-simulation. The i/o data was transferred with a fixed time step of 0.001 s between the two solvers. The WT main control loop is a standard PI controller for speed control via torque and pitch angle, respectively. It is based on the work presented in [14] and was extended by an additional IPC control scheme. Therefore, an IPC-1P control according to [15] was used to investigate the dynamic loads introduced by additional pitch angle set point activity. Special focus was thrown on the pitch drive control for the three pitch systems each. The pitch motor control was built as a cascade controller to control the position of the blades. The control structure and cascade control is shown in figure 3. Based on the deviation of the current pitch angle \( \beta_i \) and the set value \( \beta_{i,\text{set}} \) which is calculated by the main pitch control, the cascade control
computes the motor torque $M_{\text{Motor},i}$. This is than sent as an input to the mechanical system. In this study, the pitch motor current control had been simplified and was modelled as a first order system with a motor time constant of $0.005 \, \text{s}^{-1}$.

In the scope of this study it was assumed that the pitch drive is always under active control, meaning that no motor brake is used.

### 4. Results

**Gearbox dynamics**

In this section first results of the comparison between simulation results and measurement data are presented. First, only low wind operation is assessed, where pitch angle set point is kept constant at zero. The results always show comparisons for rotor blade 1. For the pitch system, two different model configurations were used: 1) pitch system modelled as described in section 3 and 2) the pitch system of rotor blade 1 was reduced according to the ones at rotor blade 2 and 3.
Figure 4a shows the comparison of pitch motor torque in relation to the rotor blade azimuth position for a low wind operation at a mean wind speed of about 4.5 m/s. In this operational condition, pitch motor torque is applied to keep the blade pitch angle at constant position. As wind forces are relatively small due to low wind speed and reduced rotational speed, loads out of dynamic changing of blades center of gravity (COG) due to blade deflection and rotational rotor blade position have a main impact on pitch motor torque.

Compared to model configuration 2) (grey), model 1) (blue) as well as the measurement (red) show step like increase in pitch motor torque at rotor positions around 20−30° and 200−250°. Figure 4b illustrates this behaviour over time for one rotor revolution. In general, tooth free-play is introduced due to the accumulated tooth backlash of the gearbox. In the operational conditions shown, external forces add in a way, that torque, applied to hold the rotor blade at constant pitch angle, changes sign twice a revolution. Tooth backlash leads to free-play while flank contact in the gear is changing direction. This effect can lead to shock-like loading on the tooth flank, appropriated by the peaks after zero crossing points. In reality, damping out of blade bearing friction might reduce this effect in some cases. But measurement data shows, that these loading conditions persist in lot of cases.

In general, the occurrence of tooth free-play strongly depends on the current loading conditions on the blade. Besides the aerodynamic forces, the dynamical change of the blade’s COG has a major impact on the occurrence. Assuming steady state operation, meaning operation without additional pitch angle adjustment at constant wind speeds, in simulations free-play is seen in states where the amplitudes of periodic gravitational blade loading exceed the ones generated from wind loads. This is illustrated in figure 5.

In regions where blade deflections are relatively small and COG might lay in upwind position relative to the blade center axis, amplitudes of periodic gravitational loads exceed other loading and free-play accounts. For bigger blade deflections the COG of the deflected blade moves downwind and stays behind the blade rotational axis and free-play will not be observed. Than if the blade’s deflection is reduced again due to reduction of aerodynamic loads because of pitch angle adjustment, the COG moves upwind and free-play starts to occur until aerodynamic loads exceed the limits for very high wind speeds again.

In figure 4a it is seen, that the occurrence of free-play is spreading over a wide range of rotor positions, especially in the 200−250° area. To analyse this behaviour, different blade loading conditions were analysed for model configuration 1). Figure 6 shows the impact of different low wind speeds (tl), the impact of different pitch angles (tr), different vertical wind shears (bl) and...
different wind directions \((br)\). It can be seen, that the occurrence of free-play in the \(20 - 30^\circ\) regimes is not impacted as much as the regions beyond \(180^\circ\).

Figure 6: Impact on tooth free-play behaviour for different parameters. \(tl\): wind speeds; \(tr\): pitch angles; \(bl\): vertical shear exponents; \(br\): wind directions

Assessment of bearing friction
For the operational conditions shown so far, bearing friction only accounts as damping on position drifts due to free-play. To assess the accuracy of the blade bearing friction model, operational conditions with pitch angle adjustments were evaluated. Therefore, turbine start-ups were investigated. In figure 7, an initial evaluation (blue) is shown against the measurement data (red) for a start-up at a mean wind speed of \(6.5\) \(m/s\). It is seen, that simulation shows a \(2\)P oscillation leading to underestimated pitch motor torque in the \(0^\circ\) and \(180^\circ\) rotor blade rotational positions. In those areas, bending moments out of blade’s gravitational forces have their minimum and do not add to the reacting moment in the blade friction calculation any more. Compared to the measurements, the blade bearing friction therefore is lower in those rotor positions. A first modification, for which the multiplication factor \(\mu_a\) of the axial force component was increased, shows improved loading behaviour being within \(\pm 25\%\) accuracy for all states (green). However, this single factor modification is not based on physically related...
findings. Introducing $M_0$ to the equation and decreasing $\mu_b$ would have similar effects. Thus, it shows that further investigations on the blade bearing friction model are necessary in a future scope.

![Figure 7: Comparison of bearing friction for run-up state](image)

**Impact of different control strategies**

The wind turbine model presented is controlled by a control scheme capable of using IPC-1P control as described in section 3. Additionally to the comparison of measurement data, the control was used to investigate the occurrence of tooth free-play in relation to different control strategies. Therefore, simulations were carried out that enable the comparison of pitch motor

![Figure 8: Comparison of CPC and IPC operation and the impact on tooth free-play](image)
torque between the standard collective pitch control (CPC) and IPC. Figure 8 shows results for two simulations carried out with a wind field at a mean wind speed of 14 m/s. A time range for multiple rotor revolutions is plotted that shows the additional pitch motor torque applied to fulfil the control objective of IPC. It can be seen that on the one hand, the pitch motor torque always keeps at negative values for CPC. On the other hand, torque switches sign to fulfil the sinusoidal pitch angle adjustments of the rotor blade per rotor revolution for IPC, leading to tooth free-play under high loading. In reality, this might lead to shock loading on the teeth flanks involved.

5. Conclusion and future work

In this contribution, a multi body simulation (MBS) modelling approach was introduced, that allows detailed investigation on pitch drive dynamics in an integral entire system model incorporating the closed-loop interaction between the turbine model and the pitch subsystem model. The pitch system consists of a blade model coupled to a pitch bearing model that integrates a static bearing friction model as proposed in literature. The back geared electrical pitch drive was modelled as a torsional degree of freedom drive train model taking into account the system stiffness characteristics of the single drive shafts as well as non-linear tooth stiffness including tooth backlash for each tooth system so that the model is capable of accounting for tooth free-play. It has been investigated how this effect impacts pitch system dynamics under different load scenarios. Simulations were carried out and compared to measurement results of a corresponding reference field turbine.

It was shown, that the effect of tooth free-play is not neglectable and can clearly be observed in the pitch motor torques in simulation results as well as measurement data. Simulation results in general show good fitting with the measurement data. It was shown that free-play occurs in a wide range of wind speeds. From steady state calculations the region could be delimited. The dynamical changing of the blades center of gravity in relation to the current aerodynamic loads acting on the deflected blade were identified as main drivers. In general, this effect can lead to high shock loading on the tooth flanks. Especially below rated wind speed, where no pitch angle adjustment is performed, single flanks in 0° pitch angle position might suffer. Thus, it is favourable to include this circumstance into design considerations.

Furthermore, it was shown, that application of control strategies that lead to higher pitch activity, like individual pitch control, could lead to an increase of free-play appearance. First investigations have been carried out to evaluate the accuracy of the calculated blade bearing friction moment. Dependent on rotor position, the model underestimated those moments. It was shown that single factor fitting on the friction model improved model accuracy, but no general suggestion on the modelling approach can be thrown so far. The friction moment of the blade bearing accounts for the pitch motor torque by up to 60-90%, depending on the rotor position and wind loads. Therefore it is a major design driver for the pitch drive units. Thus, further investigations on blade bearing friction modelling is suggested. Besides the load acting on the bearing, friction moment also might be influenced by temperature, rotational speed, lubricant properties, supporting structure and others. In a future scope, those effects should also be evaluated.

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References
[1] Simpack AG 2016 *Simpack documentation 9.9.1*
[2] Jassmann U, Berroth J, Matzke D, Schelenz R, Reiter M, Jacobs G and Abel D 2014 *Journal of Physics: Conference Series* **524** 012047 ISSN 1742-6596
[3] Flock S, Schelenz R and Jacobs G 2014 *Proceedings of EWEA* ed EWEA, The European Wind Energy Association
[4] Kamper T, Flock S and Schelenz R 2016 *Multibody System Dynamics* **36** 323–337 ISSN 1384-5640
[5] Berroth J, Kamper T, Schelenz R and Jacobs G 05/2014 4th International Conference E/E Systems for Wind Turbines (Bremen)
[6] Berroth J, Schelenz R, Flock S, Kamper T, Bi L, Werkmeister A and Jacobs G 2014 *Simpack User Meeting - Presentations* ed Simpack AG
[7] Moriarty P and Hansen C 2005 Aerodyn theory manual Tech. rep. National Renewable Energy Laboratory, USA Golden, Colorado
[8] Harris T, Rumbarger J H and Butterfield C P Wind turbine design guideline dg03: yaw and pitch rolling bearing life Tech. rep. National Renewable Energy Laboratory (NREL)
[9] Lekou D J, Mouzakis F and Savenije FJ Protest d4: Template for the specification of loads necessary for designing pitch systems
[10] Nielsen J S, van de Pieterman, René P and Sørensen J D 2014 *Wind Energy* **17** 435–449 ISSN 1095-4244
[11] Holierhoek J G, Lekou D J, Hecquet T, Söker H, Ehlers B, Savenije F J, Engels W P, van de Pieterman R P, Ristow M, Kochmann M, Smolders K and Peeters J 2013 *Wind Energy* **16** 827–843 ISSN 1095-4244
[12] Rothe Erde Slewing bearings: Complete delivery range Tech. rep. https://www.thyssenkrupp-rotheerde.com/download/info/tk_GWL_GB_2016_web.pdf
[13] IMO Momentenlager GmbH & Co KG Slewing bearings: product catalogue Tech. Rep. DV 313 D
[14] Wright A D and Fingersh L J Advanced control design for wind turbines Tech. Rep. NREL/TP-500-42437 National Renewable Energy Laboratory
[15] Bossanyi E A 2005 *Wind Energy* **8** 481–485 ISSN 1099-1824