METHODOLOGY AND RESULTS OF LOADS ANALYSIS OF WIND TURBINES WITH ADVANCED AEROELASTIC MULTI-BODY SIMULATION

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Summary

Multi-body simulation (MBS) codes have been used in the past years for specific analyses of wind turbine subsystems like drive trains, but have also been used for aero-elastic problems. This paper presents a comprehensive high-fidelity aeroelastic MBS methodology to perform integrated loads analysis according to guidelines with a parameterized wind turbine model. The implementation in MBS of flexible bodies in different levels of detail is described, representing linear and non-linear static and dynamic behaviour. Exemplary loads analysis results with models of varying level of detail are presented and differences in load predictions are compared to other modelling approaches. The results yield that for an advanced MBS wind turbine model with nonlinear flexible body representations, lower blade tip deflections and bending moments are observed compared to linear flexible body models. Different drive train torsional models are also investigated. The 2nd and 3rd eigenfrequencies implying the drive train torsion are influenced by applying multi-torsional drive train models.

1. Introduction

Design evaluations of wind turbines and the mandatory aeroelastic simulations for the load determination are generally performed using codes that represent the structural behaviour of the wind turbines (WT) by only few modal degrees of freedom. Recent research indicates that these codes are not able to capture all possibly relevant effects. The presented multi-body simulation (MBS) modelling methodology for simulation of WT offers unique aeroelastic capabilities and great flexibility to use different degrees of modelling depths adjusted to the specific problem posed.
In MBS systems the parts or bodies of the structure are connected using joints with different types of force elements acting from the inertial system on the bodies (e.g. aerodynamics on the rotor, hydrodynamics on the support structure) and between bodies (e.g. spring-damper elements). In MBS systems the bodies are generally considered rigid, as the relative deflection of the bodies is small in comparison to the rigid body motion. The inclusion of flexible bodies into the MBS environment to account for larger deflections is also possible and is described in the next chapter.

Rotor aerodynamics in this study are calculated with a blade-element-momentum approach (BEM) with common empirical corrections. A potential flow based model is included to model the influence of the tower on the passing blades [2].
Advanced drive train and gearbox models can be included to account for important coupling and resonance effects. The control system that has significant influence on the wind turbine dynamics is included in the proposed model as well as an electric generator /converter model.
The WT model and the load simulation methodology are validated using onshore design load cases according to the Offshore Code Comparison Collaboration (OC3) Phase 1 [6]. Starting with a relatively simple model used in the OC3 benchmark, the model complexity is increased in two steps for a more detailed study. First, a model with similar degrees of freedom (DOF) like in the commercial WT design code FLEX5 is set-up and an IEC DLC 1.3 ultimate load case [4] analysis is performed. In the second stage, an advanced WT model with more complex formulations for blades and tower is developed and the same IEC load calculations are performed. The MBS model with advanced flexible blade and tower representations features more than 600 DOF and shows effects on the system's predicted loads, eigenfrequencies and damage equivalent loads. These effects and its implications are discussed. The results enable the identification of the relevant structural DOF that have a significant effect on the loads and ultimately lead to a specifically optimized MBS WT model.
In addition, the FLEX5 MBS wind turbine model with single and multi-torsional representations for the drive train and rigid and flexible bedplate models are analysed by comparing frequency domain results and load calculations according to the guideline for certification of WT by Germanischer Lloyd [1].

In summary, this paper first describes methods for flexible blade and tower modelling in MBS systems. Then a description of MBS drivetrain modelling methodologies, including results, is given, followed by definitions of three investigated models of increasing level of detail. Finally results of the OC3 benchmark exercise and the IEC DLC 1.3 load case simulations are presented, finished by a conclusion. For the presented study the non-linear multi-purpose MBS code SIMPACK [7] is used.

2. Flexible Blade and Tower Modelling

2.1 Blade
To account for flexibility in the representation of the blades in the MBS code, three general approaches have been investigated:

- Lumped mass approach
- Unreduced finite element implementation
- Modal reduced FE (beam, shell, 3D brick elements) representation

In this study, the latter method is selected using Euler-Bernoulli beam elements. The blade model is reduced by the Craig-Bampton (C-B) method and is capable of considering not only bending in flap- and edgewise direction, but also torsional and tensional stiffness. Relevant coupling effects due to offsets of aerodynamic center, shear center and center of gravity to the elastic center are considered. This approach leads to a linear model.

Two further modifications are introduced to obtain a non-linear rotor blade model. At first, the geometric stiffening is included by implementing the second order bending terms in the reduction, representing a geometric non-linear model for medium displacements.
For the second modification, the blade model is split into separate C-B reduced flexible bodies, which are each connected by joints where all six degrees of freedom are constrained. This method represents a geometric non-linear blade model for large displacements, typical for wind turbine blades of the multi-MW class. It combines the advantages of the rigid-body and modal reduced FE approach.

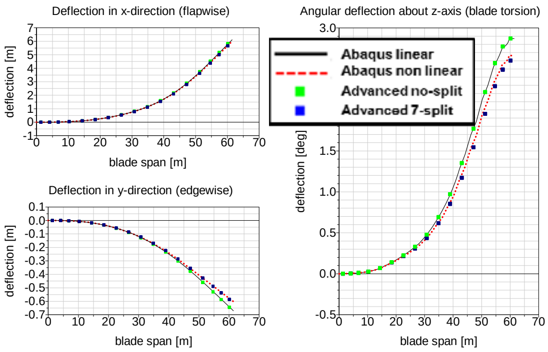

Fig.1: Tip-displacements due to aerodynamic loads

Fig. 1 presents a static comparison between a split (7 parts) and non-split blade of the NREL 5MW baseline turbine. Both models also include geometric stiffening. As a reference the linear and non-linear FE solution calculated with the FE code Abaqus is given. The aerodynamic loading represents rated conditions. It is shown that the non-split blade represents a good approximation of the linear FE solution. The model that is split into 7 substructures provides a good approximation of the non-linear FE solution. These differences between the linear and non-linear blade representation also affect loads and deflections in dynamic load calculations and are discussed in Section 5.3.

2.2 Tower

The tower is first modelled with FE Bernoulli beam elements and then modal reduced analogue to the blades with the Craig-Bampton method. In contrast to the blade, which has no additional mass at the blade-tip, the large tower top mass, consisting of the combined weight of the rotor-nacelle assembly, has to be considered. This mass has significant influence on the calculated eigenmodes of the tower, respectively the reduced model when compared to the unloaded tower.

Fig. 2 presents a comparison of the first four tower eigenfrequencies using different models. The first column shows the first eigenfrequencies of a tower model, where the tower top mass has been included in the FE model prior to C-B reduction.

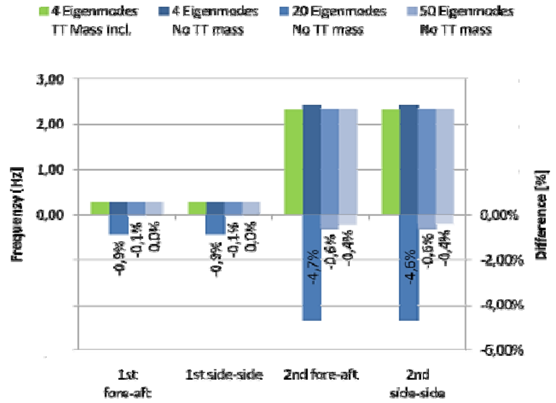

Fig.2: Comparison of flexible tower models

This model serves as reference for the alternative modelling approaches. The next three columns represent models, where the top mass is not included before the reduction, but has been added later in the MBS environment on top of the tower with a lumped mass and a rigid joint. Different numbers of eigenmodes are selected for these three models, ranging from only 4, over 20 up to 50 eigenmodes. When comparing the results with the reference model, the differences for the eigenfrequencies of the second tower fore-aft and side-to-side modes decrease significantly when choosing more eigenmodes; from a significant deviation of 4.7% for 4 modes to only 0.4% for 50 selected eigenmodes. The increase of the number of eigenmodes enables to more accurately describe the loaded tower mode and is therefore recommended by the authors. For the advanced model, a representation of the tower with 50 selected eigenmodes is used. Opposite to the blades, the tower does not require to be split into multiple flexible bodies, because the tower top displacement is small compared to the tower length, so the linear approach seems to be reasonable.

3. Drive Train Modelling

The new 2010 guideline for certification of wind turbines by Germanischer Lloyd [1] demands a more detailed drive train component analysis, i.e. resonance analyses. In industry, these described types of analysis are mainly done using MBS based codes, due to the achievable good accuracy and moderate computational effort. For specific wind turbine design load case analyses, where a detailed drive train model significantly influences the results for certain load cases, the MBS formulation enables to include detailed drive train models into the wind turbine model. With these MBS models integrated loads analyses can be run in a straightforward manner, such as presented below. Part of the presented drive train analyses were carried out in the scope of the EU PROTEST project [3].

3.1 Torsional Modelling

In a first step, exemplary, the model of the complete wind turbine has been reproduced under SIMPACK, based on the FLEX5 topology (cf. Fig. 4), with a total of 21 degrees of freedom. Aiming at reproducing the dynamic behaviour of the whole turbine and in particular of the drive train, this so-called stage 1 of the model has been extended with a multiple torsional model of the gearbox (Fig. 3). In this more detailed, so-called stage 2 approach, inertia and stiffness of gears and shafts are represented, with a total of 14 supplementary DOFs. Modal analyses and time simulations have been run to validate the models with FLEX5 as benchmark and to estimate the additional value of the new model [3].

Comparison of the outputs of modal analyses have shown that only the 2^{nd} and 3^{rd} eigenfrequencies implying the drive train torsion have been noticeably influenced by the advanced modelling, with deviations of 1,7% and 3,2% (the more detailed model witnessing in that case higher frequencies).

However, the comparison of the amplitudes of the main shaft and generator torques did not show any significant difference during normal production load cases (GL 1.0 and 1.2). The relevant detailing has to be established for each load case. While linear stiffness has been considered in that case, it is also possible to implement non-linear stiffness e.g. for the representation of bearings, gear teeth contact or backlash.

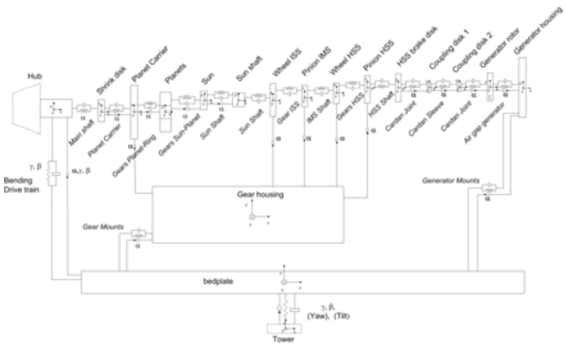

Fig. 3: Stage 2: Advanced torsional model of the drive train

3.2 Flexible bedplate

To investigate the influence of the bedplate flexibility on the drive train loads, a bedplate model has been designed using the optimization tool of the FE-software Permas. After modal reduction (C-B) of the designed model, it can be integrated into the MBS model of the overall turbine. Depending on the phenomena to be investigated (failure cases under particular load cases) and the potential coupling with other modes, the selection of the eigenmodes is to be determined by the modeller as a part of the C-B reduction, case by case.

4. Wind Turbine Model Descriptions

The presented models and results, except the drivetrain models, are computed with the widely used NREL 5MW Baseline turbine [5], to ensure optimal comparability to other design codes' results. For the OC3 validation and the eigenfrequency comparisons, the turbine's tower consists of a 77.6m tower section and a seamlessly connected 30m monopile section cantilevered to the ground. This assembly is modelled as one flexible structure in SIMPACK. For the basic and advanced models, the onshore tower with 87.6m height is utilized. The topologies of the three models are presented in Fig. 4, with the grey boxes indicating the differences.

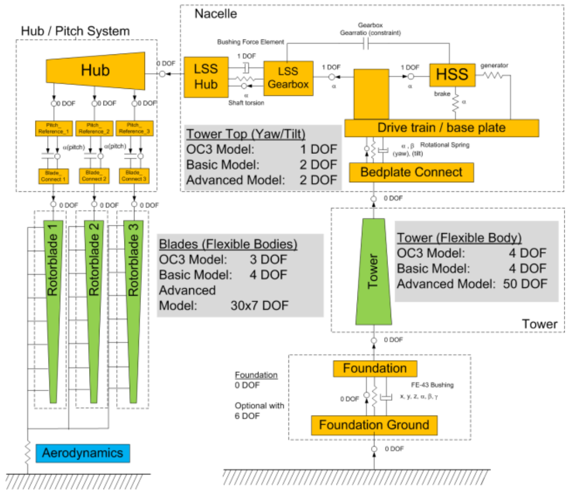

Fig. 4: OC3, basic and advanced model topologies

4.1 OC3 Model

For the OC3 validation model, only the most significant WT DOFs are selected, as described in the OC3 load case definitions [6]. Regarding the representations of flexible bodies in particular, the blades are represented by 2 flapwise and 1 edgewise bending modes and the tower by 2 fore-aft and 2 side-to-side bending modes.

4.2 Basic Model

Compared to the OC3 model, the basic model represents the level of detail of models in FLEX5, where the nacelle tilt and the three 2^{nd} edgewise blade bending modes have been added. This model was chosen to validate against the ultimate loads predicted by FLEX5 simulations of IEC DLC 1.3.

4.3 Advanced Model

In this model, the full capabilities of the MBS environment to include detailed representations of flexible bodies are utilized. Each rotorblade is split into 7 flexible sections. For each section, 30 eigenmodes have been chosen to most accurately model the nonlinear behaviour of the blade. The natural damping of each flexible blade segment has been adjusted to provide the correct [5] global damping values of the blade. As mentioned earlier in Section 2.2, the tower is represented by 50 eigenmodes as one single flexible body.

5. Results

5.1 OC3 Model Validation

Results for all OC3 DLCs are well within results from other WT design codes, representing a successful code to code validation. Details and results of this study are published in the International Energy Commission's OC3 final report [6].

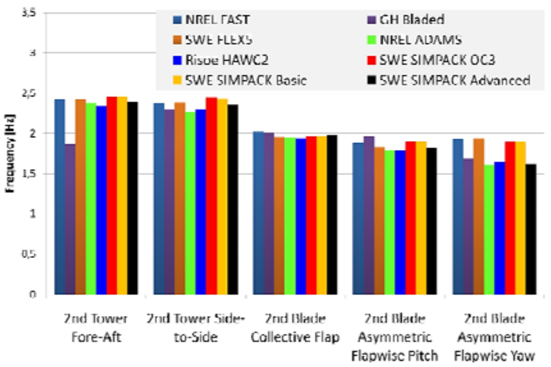

Fig. 5: Eigenfrequencies of 2^{nd} WT modes (OC3 results and SIMPACK OC3, Basic and Advanced model)

5.2 Eigenfrequencies

The eigenfrequencies of the OC3, basic and advanced SIMPACK models were compared with OC3 results (c.f. Fig. 5). For the second modes, significant differences are identified, especially for 2^{nd} asymmetric blade modes: Codes with linear blade and tower models, including the SIMPACK OC3 and Basic model, exhibit higher eigenfrequencies than codes with non-linear blade models. The latter include SIMPACK with the advanced model's non-linear (split) method, NREL ADAMS, implementing a lumped-mass approach and Risoe's HAWC2, using a non-reduced Timoshenko beam approach.

5.3 IEC DLC 1.3 Ultimate Load Comparisons

With the Basic and Advanced SIMPACK WT model, a complete IEC DLC1.3 loads analysis has been performed (2m/s wind speed bins, 6 seeds, 66 10min simulations). According to the static deflection differences between linear and non-linear (split) blade representations, the statistics from IEC DLC1.3 in Fig.6 show higher maximum values for blade deflection for the linear model. This higher deflection also results in higher blade root bending moments for the basic linear model compared to the advanced non-linear (split) model.

5. Conclusions

A methodology for integrated WT simulation in a MBS code has been presented. The MBS approach has been code-to-code validated against OC3 results. Methodologies for detailed drive train, bedplate and advanced blade and tower modelling in MBS are presented. Differences in implementation of flexible bodies, particularly blade and tower, are described. The implementation of multi-torsional drive train models result in differences of the 2^{nd} and 3^{rd} global WT eigenfrequencies implying the drive train torsion.

The discussed MBS methodologies have been applied to develop a basic and advanced WT model. With both models an ultimate loads analysis according to IEC61400-1 DLC1.3 has been performed and results presented. For the advanced MBS model with nonlinear flexible body representations lower blade deflections and bending moments are observed. The results enable the identification of the relevant structural DOF that have a significant effect on the loads and ultimately lead to a specifically optimized MBS WT model.

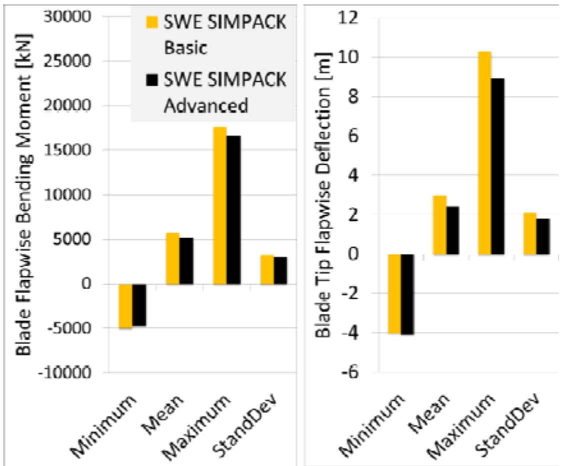

Fig. 6: IEC DLC 1.3 Statistics: Blade tip flapwise deflection (left); flapwise bending moment (right)

Acknowledgements

The research was carried out in the scope of a cooperation with SIMPACK AG and all MBS calculations are performed with the software package SIMPACK.

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