Development of an Offshore Direct-Drive Wind Turbine Model by Using a Flexible Multibody Simulation

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Abstract

Modern wind turbines are complex, highly coupled systems. The dynamic interaction between various components is especially pronounced for multimegawatt wind turbines. As a result, the design process is generally split into several phases. The first step consists of creating a global aero-elastic model that includes essential dynamics of structural components using the minimum possible number of degrees of freedom (DOFs), with the most important simplifications concerning the drivetrain and rotor nacelle assembly (RNA). This approach has been shown valid for several wind turbine configurations. Nevertheless, with the increasing size of wind turbines, any simplified design approach must be validated. The present work includes the comparison and validation of two modelling approaches for direct-drive offshore wind turbines. An RNA/drivetrain model idealized as a collection of lumped masses and springs is compared to a detailed finite element method (FEM) based model. The comparison between models focuses on the dynamic loads related to the drivetrain system and was performed under several operational conditions to explore the range of validity of the simplified model. Finally, a numerical-based workflow is proposed to assess the validity of simplified models of the RNA/drivetrain in an aero-elastic global wind turbine model.

Figure 1. Haliade™ 150 6-MW. First offshore installation in November 2013 (left). First land-based installation in March 2012 (right)
Source: ©Alstom

The Direct-Drive Wind Turbine

The object of the study is the Alstom Haliade 150 (Fig. 1): a 6-megawatt (MW) direct-drive wind turbine incorporating a permanent magnet generator of about 7 m in diameter intended for offshore installation. Similar to Alstom's land-based products, this wind turbine incorporates the PURE TORQUE™ design. The hub rotor is mounted on a mainframe set through a spread pair of tapered roller bearings, ensuring that nontorque rotor loads are transmitted directly to the structure. Torque is transmitted from the hub to the generator rotor by means of three coupling arms and three elastic couplings that isolate the drivetrain from misalignments and nontorque loads (Fig. 3). The elastic pads are coupled through two hydraulic pipeline systems, thereby balancing the hydraulic pressure between the elastic pads at the same side of each window. This coupling ensures equal torque load sharing between three generator rotor windows and gives a high torsional stiffness, while avoiding constraint forces in the other DOFs. The generator is therefore operating under nearly pure torque load, resulting in a very stable air gap between the rotor and stator generator.

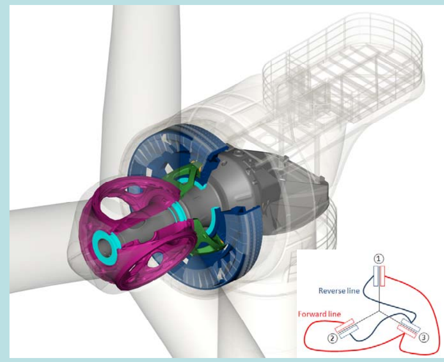

Figure 3. Detailed illustration of the main components of the RNA (left) and the elasto-hydraulic coupling system (right)
Source: ©Alstom

Design Process and Loads Calculation

The structural detailed design of a wind turbine normally relies on an estimation of reaction loads [as defined in the International Electrotechnical Commission (IEC) 61400-1] at the boundaries of their main structural components. That loads set is calculated by a simplified, global wind turbine loads model (*Loads Model*), which is computationally fast and efficient. This model approach has been shown to be successful in several wind turbine designs, and fits within demanded time frames.

Main features of the design are:

- **Aerodynamics** – based on blade element momentum (BEM) theory with wind-turbine-specific corrections
- **Blades, tower, and substructure** – Timoshenko beam-like segmented elements
- **Drivetrain dynamics** – single inertia over a torsional stiffness (1 DOF)
- **RNA** – lumped mass rigidly connected to the tower top section
- **Control and regulation algorithms** – based on dynamic link libraries
- **Structural dynamics** – modal decomposition based on linearization.

Nevertheless, this design contains evident simplifications with respect to the real system, therefore the following limitations should be assessed:

- Possible flexible behavior of the RNA mainframes caused by a large amount of mass located far away from the tower vertical axis
- Possible specific nonlinearities of the drivetrain elastic couplings and the hydraulic system
- Large deflection effects not accounted for because of the linear theory used.

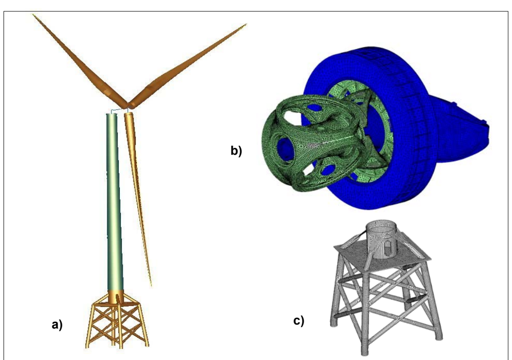

Figure 2. Schematic representation of: a) the Loads Model, b) RNA elements mesh, and c) substructure mesh of the Accurate Model
Source: ©Alstom

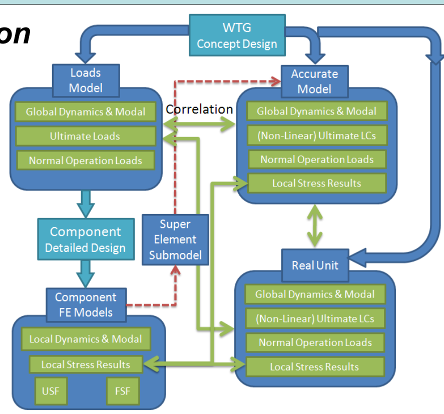

Figure 4. Block diagram showing the work flow between the Loads Model, Accurate Model, and Experimental Data

Loads Validation

The usual model validation approach, also used for the present study, is an experimental one in terms of real unit measurements. Nevertheless, using a more accurate numerical model allows for the investigation of situations that are more difficult to explore in an experimental environment. For example, a *High Fidelity Model* is carried out by using a finite element multibody approach and includes the following main differences with respect to the *Loads Model*:

- **Blades and tower** – nonlinear effects considered by using Timoshenko beams
- **RNA structural components** – the mainframes, hub, coupling arms, and generator modeled by geometrically detailed FEM models and synthesized to a dynamic super element using the Craig-Bampton condensation technique
- **Bearings** – specific stiffness matrices are located at exact geometric positions of the hub-front frame and generator rotor-stator bearings interfaces
- **Drivetrain elastic couplings** – the hydraulic circuit interaction is addressed through Lagrange multipliers between spring elements
- **Substructure** – a geometrically detailed shell-type finite element is condensed to a super element.

The *High Fidelity Model* obtained is above 2,000 DOFs, and the problem is solved in the time domain with an implicit solver, thereby updating the tangent matrix at each iteration step and accounting for the associated nonlinearities.

Results

The initial results have been calculated for the *High Fidelity Model*, which are basically the outcome of validating the model against the *Loads Model*. Such a validation is normally done by a simple "run-up" in which a perfectly laminar wind slowly increases in speed from 3 m/s up to 25 m/s. This load case allows for the comparison of the global behavior of the main wind turbine parameters (rotor speed, torque, and blade pitch system activity) to the parameters expected from the *Loads Model*. As shown in Figure 5, both models match almost perfectly, with slight differences in the pitch angular position at maximum wind speed.

At the same time, such a simple case like this helps provide a first look at the loads in the main positions. Figure 6 compares the spectral activity of the tower top torsional loads. Both models present a similar activity dominated by the blade passing frequency. Nevertheless, when the pitch system is activated, a detailed representation of the RNA elements allows for the detection of higher background activity at the other frequencies. Differences that were discovered between the models are of second order and therefore lie within the expected levels.

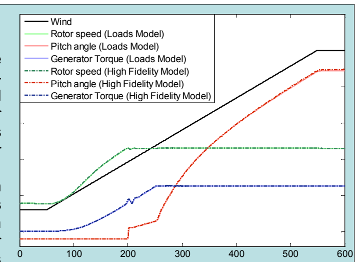

Figure 5. Global parameters comparison

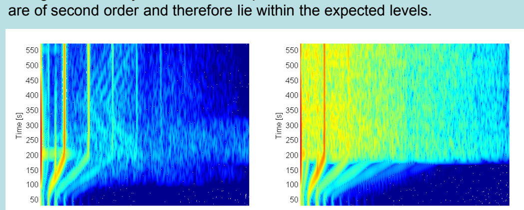

Figure 6. Tower top torsional loads spectrogram: Loads Model (left) versus the High Fidelity Model (right)

Future Work

Future work includes:

- Performing a detailed load path validation to further characterize the performance of the elasto-hydraulic system
- Analyzing the influence of local details on global dynamic activity
- Performing measurement load cases for extreme loading events (emergency stops, one blade stuck in pitch)
- Conducting model-to-test comparisons of vibration and loading
- Performing detailed analysis of stress effects on components.

Finally, the experimental validation will be performed against both models, including global dynamics, loads, and local responses.

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