



Gearbox and Drivetrain Models to Study Dynamic Effects of Modern Wind Turbines

Preprint

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*Presented at the IEEE Energy Conversion Congress and
Exposition
Denver, Colorado
September 15–19, 2013*

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Conference Paper
NREL/CP-5500-58960
October 2013

Contract No. DE-AC36-08GO28308

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Gearbox and Drivetrain Models to Study Dynamic Effects of Modern Wind Turbines

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Abstract—Wind turbine drivetrains consist of components that directly convert kinetic energy from the wind to electrical energy. Therefore, guaranteeing robust and reliable drivetrain designs is important to minimize turbine downtime. Current drivetrain models often lack the ability to model both the impacts of electrical transients as well as wind turbulence and shear in one package. In this work, the capability of the FAST wind turbine computer-aided engineering tool, developed by the National Renewable Energy Laboratory, is enhanced through integration of a dynamic model of the drivetrain. The dynamic drivetrain model is built using Simscape in the MATLAB/Simulink environment and incorporates detailed electrical generator models. This model can be used in the future to test advanced control schemes to extend life of the gearbox.

Index Terms—gears, stress control, variable-speed drives, wind power generation

I. NOMENCLATURE

α_{rot}	turbine rotor angular acceleration
β	gear helical angle
λ_{opt}	optimum tip speed ratio
ω_{rot}	turbine rotor angular speed
c_{eff}	effective drivetrain torsional damping with respect to the low-speed side of the gearbox
f_n	nonzero eigenfrequency of the two-mass model
k_{eff}	effective drivetrain torsional stiffness with respect to the low-speed side of the gearbox
k_{gear}	gear tooth stiffness
k_{mesh}	constant gear meshing stiffness
k_Y	torsional stiffness of each drivetrain shaft, $Y \in \{\text{low-speed shaft (LSS), high-speed shaft (HSS), and } l^{\text{th}} \text{ intermediate shaft, } l = 1, 2\}$
r_b	gear base circle radius
J_{eff}	effective inertia of the generator and the gearbox with respect to the low-speed side of the gearbox

J_X	inertia of the drivetrain components, $X \in \{\text{turbine rotor (rot), planet gears and the carrier (PC), carrier only (C), sun gear (S), generator (gen), and planet gears (P}_m\text{), } m = 1, \dots, M, \text{ and parallel gear components (G}_i\text{), } i = 1, 2, 3, 4\}$
M	number of planet gears
N	overall drivetrain gear ratio
N_j	gear ratio of each gear stage, $j = 1, 2, 3, N_j \geq 1$
Q_{aero}	wind aerodynamic torque
Q_{em}	generator electromagnetic torque
Q_{opp}	rotor opposing torque
R_{rot}	turbine rotor radius
V_w	effective wind speed

II. INTRODUCTION

THE wind energy industry has experienced substantial growth in recent decades, and there has been similar growth in the structural size of and power output from wind turbines. The operating conditions of wind turbines are largely determined by structural loadings such as wind turbulence and, in offshore wind turbines, sea wave excitations. Aeroelastic computer-aided engineering (CAE) tools such as FAST [1], GH Bladed [2], and HAWC2 [3] have been developed to model and simulate the dynamics of wind turbines in response to different wind fields and controllers. These codes simulate the most relevant environmental conditions and output time series of turbine operational load variations.

Because the wind turbine drivetrain consists of components that directly convert rotational kinetic energy from the wind to electrical energy, ensuring the reliability of drivetrain designs is critical to preventing wind turbine downtime. Because of the steadily increasing size of wind turbines, larger forces and torques bring up the influence of the gearbox and other drivetrain flexibilities in the overall turbine dynamic response [4]–[5], often leading to failure in

the drivetrain components. Failure in drivetrain components is currently listed among the more problematic failures during the operational lifetime of a wind turbine. In particular, gearbox-related failures are responsible for more than 20% of the downtime of wind turbines. Although the expected lifetime of gearboxes is usually advertised as 20 years, in practice gearboxes usually need to be replaced every 6 to 8 years [6]–[7].

TABLE I
MODELING PROPERTIES OF GRC WIND TURBINE

Configuration, Rating	3 Blades, 750 kW
Control	Variable Speed, Collective Pitch
Gearbox, Overall Ratio	3 Stages, 81.49
Rotor, Hub Diameter	24.1, 0.6 m
Hub Height	54.8 m
Rated Rotor Speed	22.1 RPM
Maximum Rotor C_p	0.43

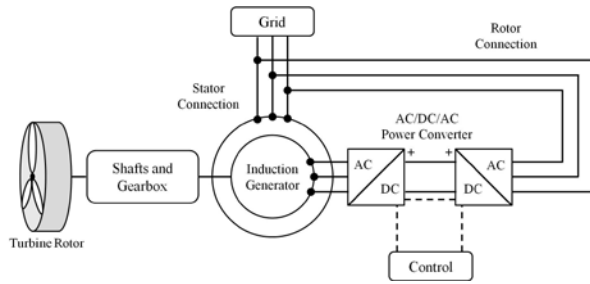


Fig. 1. Schematic of DFIG

Further insights into wind turbine drivetrain dynamics will be helpful in understanding the global dynamic response of a wind turbine as well as in designing and preserving its internal drivetrain components. Thus far, however, in most of the aforementioned codes, the drivetrain model is reduced to a few (mostly two) degrees of freedom, resulting in restricted detail in describing its complex dynamic behavior. Although researchers have developed dynamic models for wind turbine drivetrains with various levels of fidelity [8]–[18], these studies do not provide direct insights on the dynamic interactions between the drivetrain and other components of the entire wind turbine. The most recent study [18] takes a decoupled approach in which the global turbine response was first simulated using an aeroelastic CAE tool (i.e., HAWC2). After the simulation, the resulting loads and motions of the rotor as well as the nacelle were used as inputs to a high-fidelity model of the drivetrain to simulate its internal dynamic behavior. Thus, this approach fails to capture the influence of the drivetrain dynamics onto the overall turbine response.

This study aims to investigate the global dynamic response of a wind turbine drivetrain by enhancing the capability of FAST through integration of a dynamic model of the drivetrain. The drivetrain model is built using Simscape [19]. This tool extends the FAST capability in the MATLAB/Simulink environment, which in the future can

help in design and verification of active control strategies to mitigate the drivetrain loads.

This paper is organized as follows. Section III highlights the wind turbine properties modeled in this study. Section IV describes the various levels of the drivetrain model investigated in this study. Section V discusses the integration of the developed models into the FAST CAE tool. Finally, concluding remarks and a discussion regarding future work are given in Section VI.

TABLE II
PARAMETERS OF THE 750 kW DFIG

Generator	
Line-Line Voltage (RMS)	690 V
Frequency, No. of Pole Pairs	60 Hz, 2
Stator Resistance, Leakage Inductance (pu)	0.016, 0.06
Rotor Resistance, Leakage Inductance, Both Referred to Stator (pu)	0.016, 0.06
Magnetizing Inductance (pu)	2.56
Inertia Constant (s)	2
Converter	
Converter Maximum Power (pu)	0.5
Grid-Side Coupling Inductance, Reactance (pu)	0.15, 0.0015
Nominal DC Bus Voltage	1200 V
DC Bus Capacitor	0.1 F

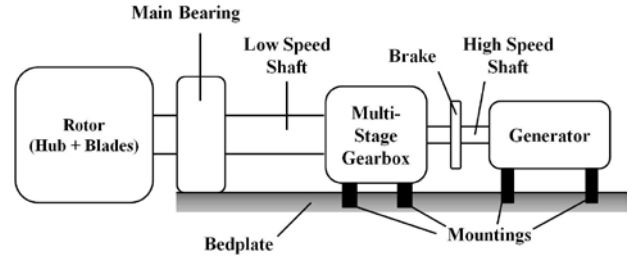


Fig. 2. Modular drivetrain configuration of wind turbine

III. WIND TURBINE DESCRIPTION

The turbine modeled in this study is based on the Gearbox Research Collaborative (GRC) turbine [20] at the National Renewable Energy Laboratory's (NREL's) National Wind Technology Center (NWTC). Table I summarizes the important properties of this turbine.

The GRC wind turbine originally employed two fixed-speed wind turbine generators (WTGs). To model variable-speed operation, the WTG is modeled using a doubly-fed induction generator (DFIG), shown in Fig. 1. Because the drivetrain has much slower dynamics than the mechanical drivetrain, an average model of the AC-DC-AC converter is used in this study, in which the power electronic devices are replaced by controlled current sources. The induction generator model is based on a commercial wind turbine of the same rating available in the market. Table II summarizes the key parameters of the generator and the converter, which are used to build the variable-speed WTG model in the Simscape/ SimPowerSystems environment.

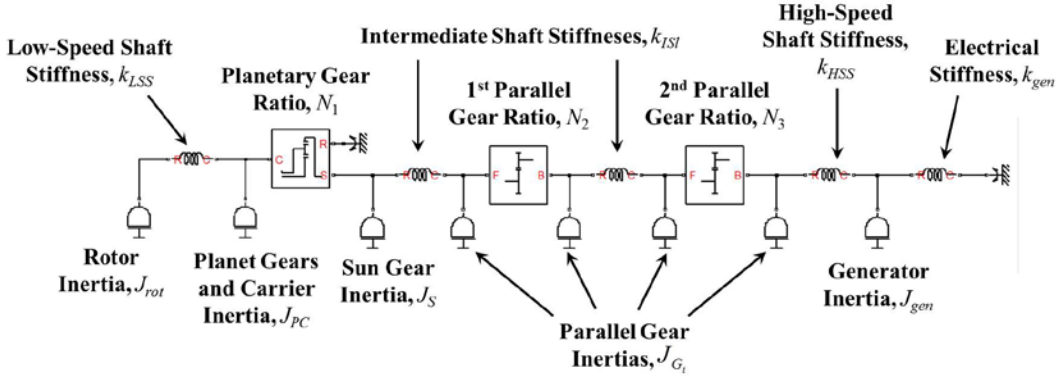


Fig. 3. Five-mass model of wind turbine drivetrain in SimDriveline with fixed-speed induction generator

For wind speeds below 12.5 m/s, which is the rated wind speed of this turbine, the output power of this WTG is controlled to track the maximum power coefficient (C_{Pmax}), while maintaining constant pitch angle at its optimum (-3.5° for this turbine). In variable-speed operation, the rotor speed, ω_{rot} , is proportional to the wind speed, V_w :

$$\omega_{rot} = \frac{\lambda_{opt} V_w}{R_{rot}} \quad (1)$$

where λ_{opt} is the desired optimum tip speed ratio.

IV. DRIVETRAIN MODELING

A. Five-Mass Model

This study focuses on a commonly used modular drivetrain configuration in operating turbines [8]. Fig. 2 shows the building blocks of the configuration. In this turbine, the multistage gearbox consists of a planetary (epicyclic) gear set and two parallel gear sets, with two intermediate shafts.

Fig. 3 shows the five-mass model of the wind turbine drivetrain with fixed-speed generator, developed in the Simscape/SimDriveline environment. A fixed-speed generator has an electrical torsional stiffness between the air gap magnetic field and the generator rotor. This stiffness behaves as a spring to the inertial reference frame of the drivetrain, which provides restoring torque to the rest of the drivetrain. Such stiffness arises because of tight, allowable speed variation in the fixed-speed turbine. Effects of this stiffness are prominent in the transient response of the generator (e.g., during generator start-up or grid-fault events). This stiffness value can be obtained through experiments [21]. For a variable-speed generator, this restoring effect does not exist, and the drivetrain model shown in Fig. 3 will have a free boundary condition on the other side of the generator.

Frequency analysis on this five-mass model reveals eigenfrequencies of the drivetrain, as summarized in Table III. These frequencies imply several possible resonant excitations that will lead to amplified loads in the drivetrain.

B. Two-Mass Model

Fig. 4 illustrates the configuration of the two-mass model commonly used to model the dynamics of drivetrains in wind turbine aeroelastic CAE tools, such as FAST [1]. Inputs into the model are the five parameters: J_{rot} , k_{eff} , c_{eff} , N , and J_{eff}/N^2 . The generator electrical torsional stiffness is generally not required in such codes, because this stiffness is inherent to the generator model used for the analysis. Parameters of the two-mass model can be derived from the five-mass model as follows:

$$J_{eff} = J_{PC} + N_1^2 (J_S + J_{G_1} + N_2^2 (J_{G_2} + J_{G_3} + N_3^2 (J_{G_4} + J_{gen}))) \quad (2)$$

$$\frac{1}{k_{eff}} = \frac{1}{k_{LSS}} + \frac{1}{N_1^2 k_1} + \frac{1}{(N_1 N_2)^2 k_2} + \frac{1}{(N_1 N_2 N_3)^2 k_{HSS}} \quad (3)$$

The effective drivetrain torsional damping, c_{eff} , can be determined experimentally through several braking events [8]. The nonzero eigenfrequency of this two-mass model can be calculated as:

$$f_n = \frac{1}{2\pi} \sqrt{k_{eff} \left(\frac{1}{J_{rot}} + \frac{1}{J_{eff}} \right)} \quad (4)$$

The resulting eigenfrequency using the two-mass model for the GRC drivetrain is 2.96 Hz, which is quite different from the first nonzero eigenfrequency of the five-mass model that is 2.52 Hz. This first eigenfrequency is of importance because it stores the most torsional energy. As exhibited in the following section, the two-mass model has limitations in providing insights on possible resonant excitations of the drivetrain as well as in analyzing the loads experienced by different components of the drivetrain.

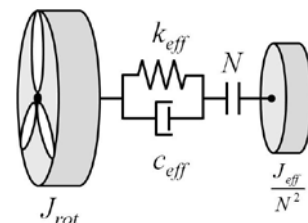


Fig. 4. Two-mass model of wind turbine drivetrain

TABLE III
EIGENFREQUENCIES OF FIVE-MASS MODEL

Drivetrain with Fixed-Speed Generator		
Mode	Description	Frequency (Hz)
1	Low-Speed Shaft Mode	0.86
2	Generator Start-Up Mode	6.05
3	High-Speed Shaft Mode	312
4	1 st Gearbox Mode	402
5	2 nd Gearbox Mode	1960
Drivetrain with Variable-Speed Generator		
Mode	Description	Frequency (Hz)
1	Generator Static Model	0
2	Low-Speed Shaft Mode	2.52
3	High-Speed Shaft Mode	312
4	1 st Gearbox Mode	402
5	2 nd Gearbox Mode	1960

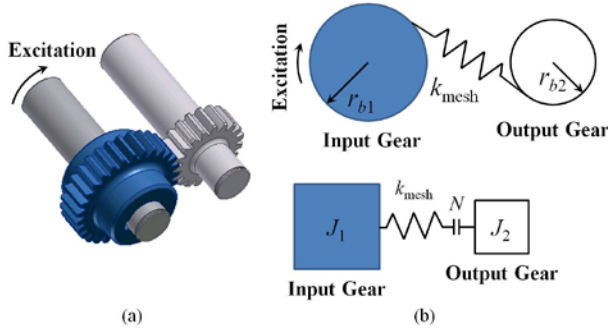


Fig. 5. Parallel gear stage and gear mesh stiffness representations

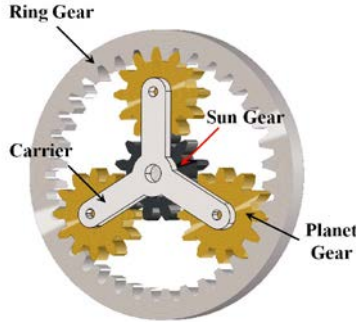


Fig. 6. Planetary gear set with three planet gears.

C. Pure Torsional Model of Gearbox

In the two previously described drivetrain models, the meshing gear is modeled as an ideal static gain for torque and speed transmission. In reality, the gear transmission error, which is defined as the difference between the actual and ideal angular positions of the rotating gear mainly caused by the gear elastic deformation, contributes to the dynamics of the pair meshing gear. This phenomenon contributes to the definition of gear meshing stiffness. Several studies have been performed to investigate the dynamics of the gearbox using flexible multibody models built in various software packages [8]–[18]. These studies, however, focus on the internal dynamics of the gearbox, and the developed model packages are not readily

compatible with available wind turbine aeroelastic CAE tools.

In this study, a purely torsional model of the gearbox with constant meshing stiffness is built in the Simscape/SimDriveline environment. The model development and analysis on both planetary and parallel gear stages will be discussed in this subsection, followed by some remarks on the integration and simulation of this model into the aeroelastic CAE tool of interest (i.e., FAST) in the following section.

1) Parallel Gear Stage

Fig. 5a shows a parallel gear set, which is a torque reducer, commonly employed in wind turbine drivetrains. Fig. 5b represents its flexible equivalent, in which the meshing stiffness acts on the line of action of the meshing gears. This meshing stiffness k_{mesh} , with respect to the input gear, can be represented as [11]:

$$k_{mesh} = k_{gear} (r_{b1} \cos \beta)^2 \quad (5)$$

where the gear tooth stiffness k_{gear} can be determined according to standards [22]–[23]. Because this system can be seen as a two-mass system, the local eigenfrequency of this gear set can be determined using a similar approach as in (4).

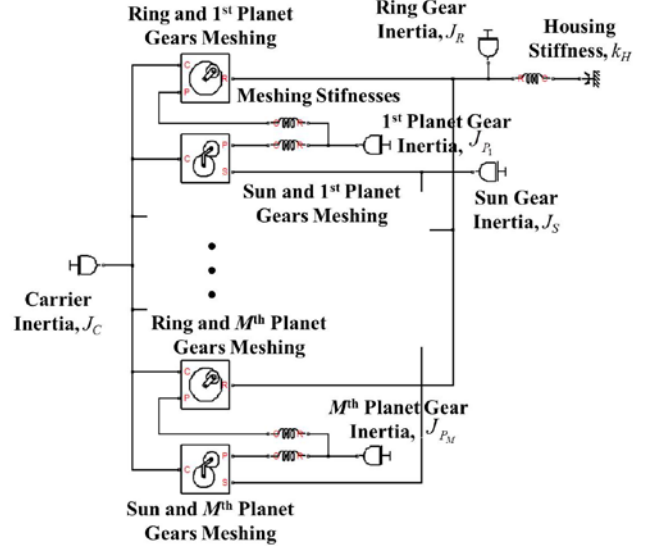


Fig. 7. Torsional model of planetary gear stage with M planet gears

2) Planetary Gear Stage

Fig. 6 shows a planetary gear set with three planet gears and similar configuration to the one installed in the GRC turbine.

The rotational input is from the carrier of the planetary gear stage, which provides rotational motion through the planet gears, and finally to the sun gear. The ring gear is modeled to have flexible coupling with the gear housing. Flexibility between the meshing planet and ring gears and between the meshing planet and sun gears can be modeled

similar to that of a parallel gear set using (5), as shown in Fig. 5b.

Fig. 7 shows a torsional model of a planetary gear set built in the Simscape/SimDriveline environment. This model can be adapted for any M equispaced-planet gear set. To validate this model, comparisons with published frequency analysis [11] of planetary gear sets with various numbers of planet gears were performed. The results of the comparisons show good agreement and are summarized in Table IV. Fig. 8 shows the frequency response function (FRF) of the planetary gear set in Table IV that has three planet gears. The FRF was produced by inputting torque at the carrier and taking the rotational speed of the sun gear as the output. It is important to note that the planetary gear sets used for the analysis in Table IV and Fig. 8 are different from the planetary gear set used in the GRC wind turbine drivetrain.

Table V summarizes the frequency analysis results of the GRC wind turbine drivetrain using the torsional model of the gearbox and variable-speed generator model. Exciting any of these eigenfrequencies will lead to amplified loads in the drivetrain.

V. MODEL INTEGRATION

Fig. 9 illustrates the proposed strategy to integrate the described drivetrain models into the two-mass model inherent inside the FAST CAE tool. For simplicity, the flexible modes of the other turbine components modeled inside FAST, such as those of the blades and tower, are not depicted in the schematic diagram in Fig. 9.

In FAST, the two-mass drivetrain model is reduced to a single-mass model consisting of solely the rotor and the rigid shaft (as shown in the bottom part of Fig. 9). This is done by deactivating the flexibility of the drivetrain (simulating rigid transmission) and setting the gear ratio and the generator inertia to unity and zero, respectively. The rotor equation of motion can be expressed as:

$$J_{rot} \alpha_{rot} = Q_{aero} - Q_{opp} \quad (6)$$

TABLE IV
EIGENFREQUENCIES OF PLANETARY GEAR STAGE

3-Planet Gears		
Mode	SimDriveline Model	Peeters [11]
1	2.273 kHz	2.217 kHz
2	6.340 kHz	6.159 kHz
3	11.296 kHz	11.205 kHz
4-Planet Gears		
1	2.207 kHz	2.138 kHz
2	6.911 kHz	6.688 kHz
3	12.699 kHz	12.577 kHz
5-Planet Gears		
1	2.153 kHz	2.059 kHz
2	7.403 kHz	7.105 kHz
3	13.980 kHz	13.810 kHz

TABLE V
EIGENFREQUENCIES OF GRC WIND TURBINE DRIVETRAIN
WITH TORSIONAL GEARBOX MODEL

Mode	Frequency (Hz)
1	0
2	2.44
3	154
4	307
5	353
6	748
7	1020
8	1530

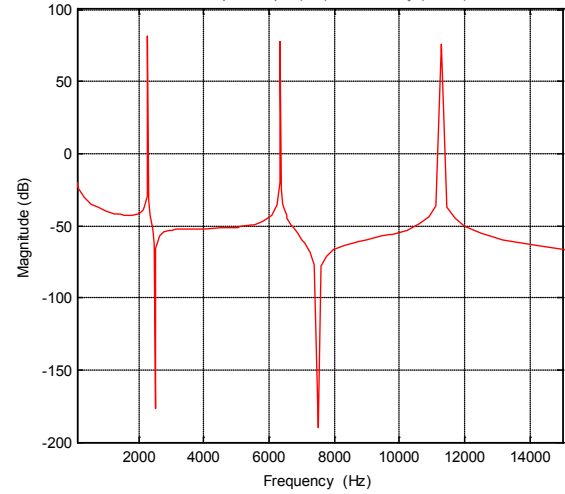


Fig. 8. Frequency response function of three-planet planetary gear stage for the gearbox presented in [6]

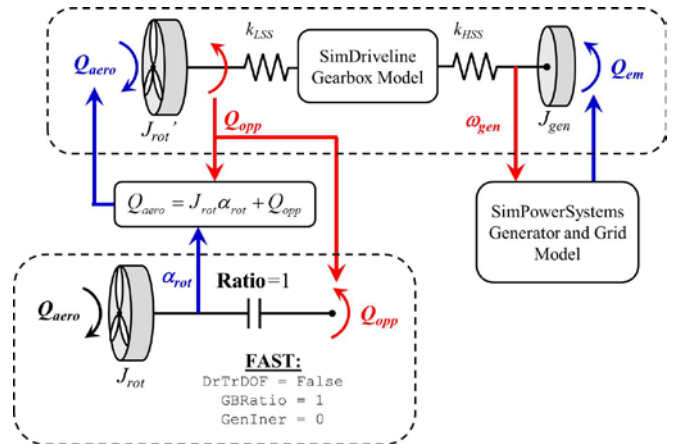


Fig. 9. Proposed schematic of integrating the Simscape drivetrain model into the FAST aeroelastic CAE tool

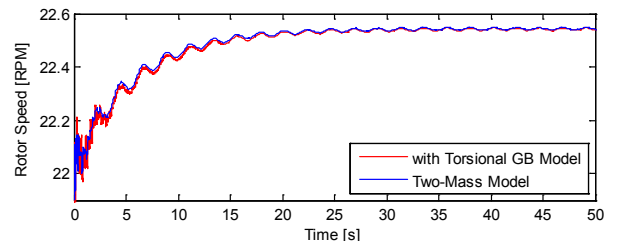


Fig. 10. Rotor speed response

FAST internally calculates the input aerodynamic torque Q_{aero} from the defined wind profile, but does not provide this torque as an output. However, as the rotor acceleration α_{rot} is an available FAST output, the aerodynamic torque Q_{aero} can be reconstructed using (6) to be one of the inputs to the external drivetrain model. In this process, the rotor inertia J_{rot} is assumed constant and replicated in the external drivetrain model. The rotor inertia is connected to the flexible low-speed shaft, the purely torsional gearbox model, the high-speed shaft, and the generator inertia. The electrical machine and grid model will take the generator speed and provide the generator electromagnetic torque to the drivetrain.

The rotor opposing torque Q_{opp} is required as an input to the FAST drivetrain model as well as to calculate the aerodynamic torque Q_{aero} . In SimDriveline, this rotor-opposing torque can be retrieved by utilizing the torque sensor element behind the built rotor body. In general, torque, velocity, and angular-position sensor elements can be placed flexibly within the Simscape drivetrain model to monitor the response of the drivetrain under various load conditions.

In the remainder of this section, simulation results showing the effectiveness of the torsional model of the gearbox under different transient load cases are presented. Simulations using the FAST wind turbine aeroelastic code were conducted in the Simulink environment. In the simulations, all available wind turbine flexible modes in FAST—including that of the blades, tower, and drivetrain—were activated. No damping is defined within the drivetrain model to highlight the transient response of the drivetrain. Aerodynamic damping computed within FAST is the only source of damping that stabilizes the overall drivetrain response. The results are compared with those using an undamped two-mass model of the drivetrain inherent in FAST.

A. Transient Response Caused By Wind Excitation

This simulation was performed under a constant wind speed of 9 m/s (below the rated wind speed), and the turbine speed was initialized to be 22.1 RPM. The start of simulation effectively imparts a large step input to the system that can excite all of the drivetrain modes, especially during the transient period.

Fig. 10 shows the turbine rotor speed using the integrated drivetrain model as well as the inherent FAST two-mass model. The rotor speed steadily increases to reach the optimal tip-speed ratio. Both models are in good agreement in the speed response of the turbine. Fig. 11 is the transient response comparison of the rotor torque between two-mass model and the inclusion of gearbox model. This figure reveals the importance of including the detail model of the gearbox.

Fig. 12 highlights the distinction between the two models in predicting the steady-state load response of the

drivetrain. There are some frequency components coming from other parts of the turbine structure. The frequency of 0.4 Hz comes from the tower fore aft mode, whereas the frequency of 1.1 Hz and its harmonics come from the blade pass frequency (3P). The response of the drivetrain model with the purely torsional gearbox is particularly high at two times the blade pass frequency (i.e., 6P) of 2.25 Hz because it is quite close to the estimated first eigenfrequency of 2.44 Hz. Thus, it predicts amplification of load caused by resonance at the wind speed of 9 m/s. On the other hand, the two-mass drivetrain model estimates an eigenfrequency of 2.96 Hz, which is at some distance from the harmonics of the blade pass frequency, and hence predicts much less resonance.

B. Transient Response Resulting From Grid Excitation

Another transient load can arise from excitations on the grid. One example of grid excitation is simulated in this subsection to predict loads on the gearbox. A voltage drop for 0.15 second, from 100% to 90% and back to 100% of the nominal root mean square voltage, is simulated after the turbine has reached steady state.

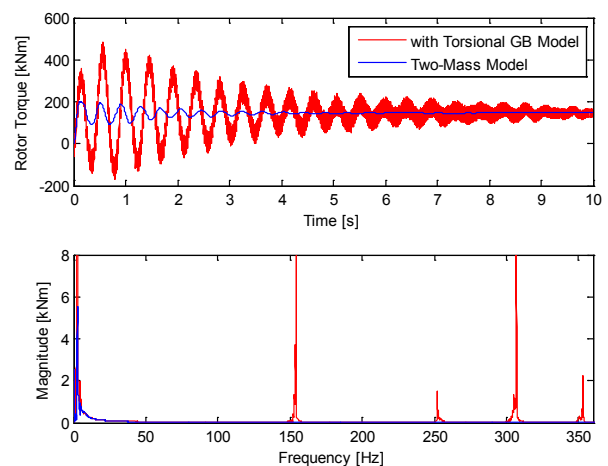


Fig. 11. Transient response comparison of the rotor torque

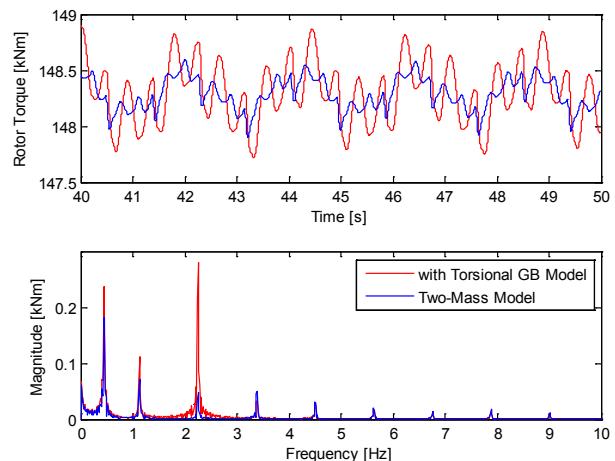


Fig. 12. Steady-state response comparison of the rotor torque

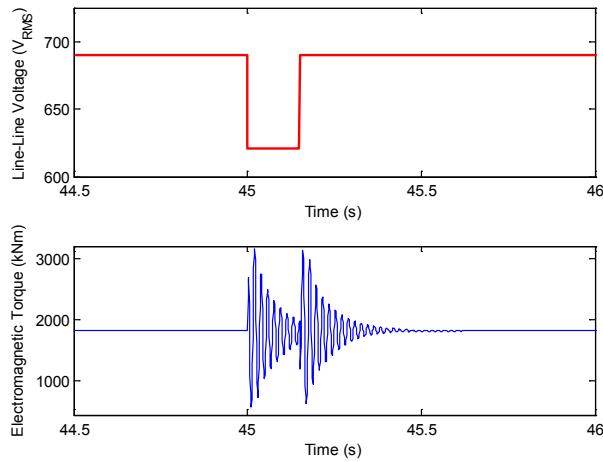


Fig. 13. Electromagnetic torque excitations resulting from a voltage drop on the grid

As shown in Fig. 13, this voltage drop results in harmonic torque excitations onto the drivetrain with frequencies of 50.78 Hz and 56.15 Hz. These frequencies are inherent to the generator characteristic. It is important to note that the frequency component of this torque excitation may cause resonances if the frequency matches any of the drivetrain eigenfrequencies. These resonances cannot be predicted using the standard two-mass model, because the two-mass model can predict only the lowest eigenfrequency of the drivetrain.

Fig. 14 illustrates how this load is transmitted to each stage of the multistage gearbox. The torques acting on the high-speed side of each gear stage (the pinion of each parallel gear stage and the sun gear of the planetary gear stage) are shown in both the time and frequency domains.

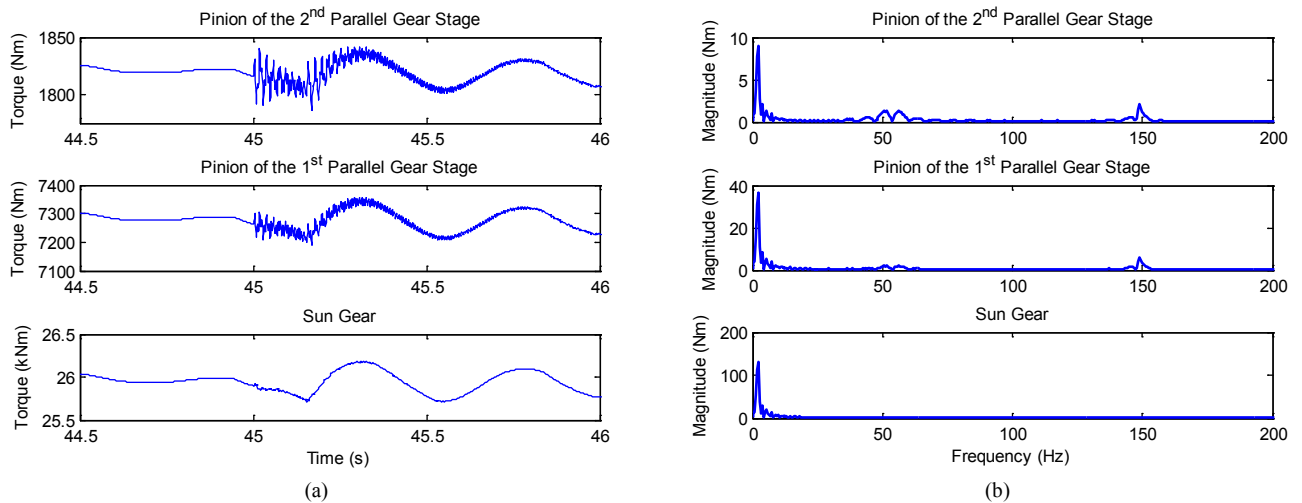


Fig. 14. Transmitted loads onto the gears because of grid excitation in the (a) time domain and (b) frequency domain

The pinion that is directly connected with the generator through the high-speed shaft experiences the largest proportion of high-frequency loads caused by grid excitations compared to other gears. Therefore, it is most prone to failures from fatigue in the event of grid disturbances. Sudden increases in the generator electromagnetic torque excite the two lowest and most dominant modes of the drivetrain (i.e., 2.44 Hz and 154 Hz). This torque also excites the system at its excitation frequencies of 50.78 Hz and 56.15 Hz, but with less dominant effect than at the eigenfrequencies. The transmitted loads on the gear are reduced as the gear gets further from the source of excitation, but the most dominant drivetrain eigenfrequency of 2.44 Hz prevails during the transient regime.

The developed high-fidelity model can also be used in designing drivetrain components to preserve or extend the life of the gearbox. The stiffness of the high-speed shaft, which connects the generator and the gearbox, was varied to investigate its influence on the load transmitted to the gearbox. In practice, this stiffness can be varied by altering the size of the shaft. In simulation analysis, the stiffness was varied to 0.1 and 10 times of the nominal value.

The transmitted loads on the pinion of the second gear stage were evaluated and shown in Fig. 15. A lower stiffness appears to transmit more severe loads onto the pinion, whereas a higher stiffness slightly reduces the loads at the start. A higher stiffness can be achieved by using a shorter and/or larger diameter shaft. The developed model can help in choosing the optimal stiffness value to meet design and cost specifications as well as to maintain a certain range of transmitted loads onto the gear.

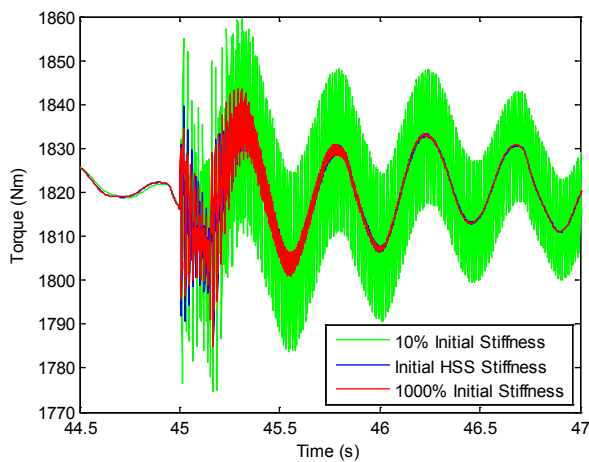


Fig. 15. Loads on pinion of second parallel gear stage under various high-speed shaft stiffness values

VI. CONCLUSIONS AND FUTURE WORK

In this work, the capability of the FAST CAE tool is enhanced through integration of a dynamic model of a drivetrain built using Simscape in the MATLAB/Simulink environment. As described in this study, implementation of the developed drivetrain model enables the software tool to be used in a number of ways. First, the model can be simulated under different wind and grid conditions to yield further insight into the drivetrain dynamics in terms of predicting possible resonant excitations. Second, the tool can be used to simulate and understand transient loads and their couplings across the drivetrain components. Third, the model can be used to design the various flexible components of the drivetrain such that transmitted loads can be reduced on the gearbox.

Ongoing and future work includes extending to a time-varying meshing stiffness in the gearbox model. As even-higher-fidelity models are developed to give more insight into the dynamics of the drivetrain, further control strategies will be explored to attenuate unwanted loads and vibrations in the drivetrain. In the future, we also intend to perform hardware testing on the 2.5-MW or 5-MW dynamometer at the NWTC of NREL.

VII. ACKNOWLEDGMENT

The authors thank Dr. Khanh Nguyen and Dr. Jason Jonkman for fruitful discussions about integrating FAST with the external drivetrain model.

This work was supported by the U.S. Department of Energy under Contract No. DE-AC36-08-GO28308 with the National Renewable Energy Laboratory.

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