Proceedings Paper:

Al-Hamadani, H., Long, H. orcid.org/0000-0003-1673-1193 and Cartmell, M. (2016) Effects of model complexity on torsional dynamic responses of NREL 750 kw wind turbine drivetrain. In: Power Engineering. The International Conference on Power Transmissions ICPT 2016, 27-30 October 2016, Chongqing, China. Taylor & Francis Group , London , pp. 205-212. ISBN 978-1-138-03267-5

https://doi.org/10.1201/9781315386829

This is an Accepted Manuscript of a book chapter published by Routledge in Power Transmissions on 10 November 2016, available online: http://www.routledge.com/9781315386812

Power Transmissions, 2017 Taylor & Francis Group, London, ISBN 978-1-138-03267-5, page 205-212

Effects of Model Complexity on Torsional Dynamic Responses of NREL 750kW Wind Turbine Drivetrain

Haider Al-Hamadani, Hui Long*, Matthew Cartmell

(*corresponding author: email: h.long@sheffield.ac.uk)

Department of Mechanical Engineering

The University of Sheffield

Sheffield, United Kingdom

ABSTRACT:

Purely torsional multibody dynamic models with different levels of complexity have been developed to simulate the 750kW wind turbine (WT) drivetrain, designed by National Renewable Energy Laboratory (NREL). The developed models use a combined mathematical approach and Simscape in MATLAB/Simulink, by which the fixed-speed and variable-speed generator models are tested. The required parameters for building the multistage gearbox dynamic models are obtained by using 3D CAD models. Eigenfrequency analysis of the WT drivetrain is performed. The natural frequencies and corresponding mode shapes are determined for the two generator models respectively. It has found that increasing the model complexity enables the prediction of more complex mode shapes and an accurate description of the torsional vibration of the drivetrain. It has also found that increasing the gear mesh stiffness results in a wider frequency range and increases frequencies of the gearbox components but has no impact on lowest natural frequencies of the WT drivetrain.

1 INTRODUCTION

The wind power industry has seen a rapid development in the last decade, especially in Europe (Smolders et al. 2010) and the United State (Daniels et al. 2004). At the end of 2003, about 35,000 MW of wind energy capacity had been installed globally (Daniels et al. 2004) and around 11 GW of capacity had been installed in Europe alone by the end of 2013. However, the wind industry faces serious challenges of premature failures of WT gearboxes occurring much earlier than the designed life. The gear and bearing failures in WT gearboxes may be attributed to the excessive loads under various operating conditions (Grujicic et al. 2014). Gearbox failures result in significant component replacements which subsequently increase the cost for wind energy (Grujicic et al. 2014). Replacing WT gearboxes needs heavy duty equipment such as cranes, which is generally expensive to hire and almost impossible to operate in adverse weather conditions for offshore wind farms (Tavner et al. 2008). A good understanding of how and why WT gearboxes are failing prematurely will improve the gearbox component design and reduce the overall maintenance cost. It is important to understand the dynamic behaviour of WT drivetrain under various operational conditions and how the flexible movement of gearbox components, such as planetary carrier and floating sun shaft, within a WT gearbox, reacts to transient loads. The multibody dynamic models developed in this study include the turbine rotor, the gearbox components and the generator, using both fixed speed and variable speed generator models. These models capture more details of drivetrain dynamic behaviour, such as the torsional deformations and dynamic responses of key components of the WT drivetrain, than the widely used two mass or five mass models. The required parameters for building multistage gearbox dynamic models are obtained by using CAD models and these parameters are validated with available data published in literature. Eigenfrequency and mode shape analysis of the WT drivetrain are performed by using the low speed shaft (LSS) torque as the input and the corresponding angular velocity of the generator as the output. The resulting eigenfrequencies are compared with that determined from torque measurements under different transient load conditions published by NREL. The influences of different levels of modelling complexity on the eigenfrequencies of the WT drivetrain and how that affects the dominant frequencies of drivetrain during two WT operating conditions, normal operation and shutdown, are discussed in detail.

2 DYNAMIC SYSTEM MODELLING

Several studies related to the dynamic modelling of the WT drivetrain have been reported. Some of these studies are focused on analysing the loading on bearings within the WT gearbox under various operation conditions, such as normal operation and shutdown. However, these studies ignored the dynamic behaviour of some key gearbox components (Bruce et al. 2015) (Scott et al. 2012). Three types of modelling approaches have been examined by Peeters (Peeters et al. 2005). The first is the purely torsional multibody model where the gears are modelled as rigid bodies, each with a single degree of freedom (DOF) in the torsional axis, connected to each other by linear springs. Such models are able to investigate torsional loads, loads on gears and bearings, and eigenfrequencies (Girsang et al. 2014) (Mandic et al. 2012) (Shi et al. 2013). The second approach is the 6-DOF rigid multibody modelling with discrete component flexibilities, which produces a more accurate model; however, the complexity of the gearbox modelling has been increased. These models take into account of individual gear stages of the gearbox and the influence of bearings stiffness. The component flexibility is represented by spring-damper systems. Such models facilitate a more detailed description of gear mesh and bearing stiffness. The third approach is the fully flexible multibody model which increases the modelling accuracy of gearbox component flexibilities from the second approach by using finite element modelling. The third approach allows the visualisation of the influence of different subcomponent flexibilities, however it is computationally expensive. Increasing flexibility does not always result in more accurate modelling results. The addition of flexible components to the WT drivetrain model increases the model complexity and affects slightly on eigenfrequencies however much more on the corresponding mode shapes (Peeters et al. 2005). LaCava et al. (LaCava et al. 2013) observed that the influence of increasing the level of gearbox component flexibilities on the bearing and gear loads using seven models of various levels of complexity. Their results concluded that the flexibility of gearbox sub-components, such as housing, carrier and the main shaft, has noticeable influences on the loading of the plant bearings. It has been reported that using a flexible coupling with reasonable stiffness can reduce the torque amplitude of the high speed shaft (Peeters et al. 2006). Increasing model complexity only has a small effect on modelling accuracy while resulting in higher computational costs (LaCava et al. 2012). The different levels of modelling complexity help gearbox designers to improve designs and assess the dynamic behaviour of chosen designs under specific loading conditions (Oyague et al. 2008). The torsional dynamic model for WT gearboxes is one of the common modelling approaches because of its fast solution time with low computational costs. The purely torsional multibody dynamic models developed in this paper are computationally effective to capture the torsional loads and dynamic responses of key WT drivetrain components during free and forced vibrations.

3 DEVLOPMENT OF DYMANIC MODELS

3.1 Drivetrain Modelling

In this study, a multistage gearbox designed by NREL, for use within the complete drivetrain of a 750kW WT, is modelled and studied with various levels of modelling complexity. The gearbox design consists of a planetary stage with a fixed ring gear and three planet gears; this is followed by two parallel gear stages. The overall gearbox ratio is 1:81.49 (Oyague 2009). A simplified schematic of the WT drivetrain is shown in Figure 1. Purely torsional dynamic models with lumped masses, using 2, 5, 9 and 11 DOFs respectively, are developed to represent this drivetrain with a fixed or variable speed generator model. All gearbox components, including planet carrier, gears and shafts as well as the generator and the rotor are modelled, using one torsional DOF for each component in the rotational direction.

The rotor assembly includes three blades and the rotor hub, which is connected to the gearbox through the low speed shaft (LSS). The mass and inertia of the blades (M_b , J_b) can be calculated by the following formulas (Rodríguez et al. 2007):

$$M_b = 2.95\ L_b^{2.13} \qquad (1)$$

$$J_b = 0.212\, M_b\, L_b^2\, n \qquad (2)$$

where L_b is the length of the blade and n is the total number of blades. The total amount of polar moment of inertia of the rotor assembly as one lumped mass (J_{rotor}) can be calculated from the summation of the blades inertia and the inertia of the hub (J_h) (Oyague 2009):

$$J_{rotor} = J_b + J_h \qquad (3)$$

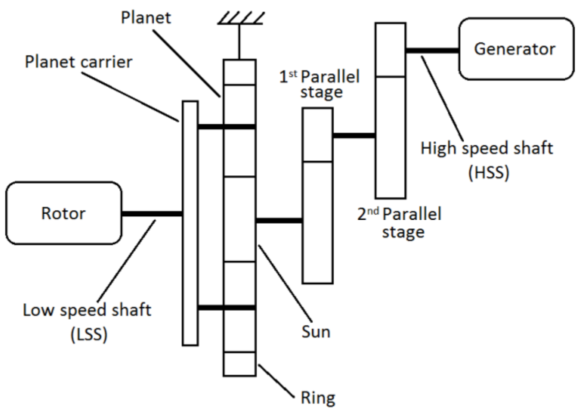

Figure 1. Simplified schematic of wind turbine drivetrain

In some studies, all the shafts of the WT drivetrain are assumed to have constant cross-section area and therefore the complex geometry of the different shafts is simplified to solid or hollow cylinders (Shi et al. 2013) (Shi et al. 2014). In this paper, the complex geometry of all WT drivetrain shafts has been considered and the values of polar moment of inertia of the main shaft and all gearbox shafts are calculated from the CAD models. The generator resistance torque, which acts as a reactive loading to the drivetrain at the generator side, affects the WT drivetrain system when the generator engages or disengages with the electrical grid. In this study, the generator's electrical resistance torque is represented by a torsional spring. For a 750kW WT, the value of the electrical torsional spring stiffness between the generator armature winding and magnetic field of the generator is K_{gen} = 28100 N.m/rad. This value is taken from (Mandic et al. 2012), which has been calculated and validated experimentally for different operation conditions. The shafts stiffness values and the electrical stiffness value used in the developed drivetrain model are shown in Table 1. A CAD model for the NREL 750kW WT gearbox has been created as shown in Figure 2. All the required parameters such as the mass and polar moment inertia for all gearbox components have been calculated as shown in Table 2.

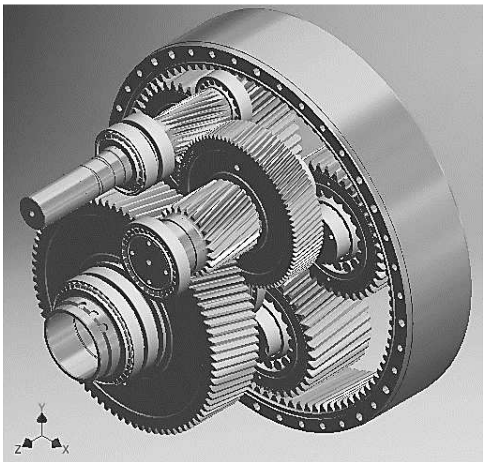

Figure 2. CAD model of the NREL multistage gearbox

The calculated parameters have been validated with available data published in literature (Girsang et al. 2014) (Mandic et al. 2012) (Oyague 2009). Figure 3 shows the complete 750kW WT drivetrain representation of eleven lumped masses of 11 DOFs with consideration of gear mesh stiffness for all gear stages. The gear ratios for each stage of the WT gearbox are shown in Table 3. For the purely torsional model, the gear mesh stiffness is modelled as a linear spring (Peeters et al. 2005). The development of mathematical dynamic model of the drivetrain using five lumped masses is detailed in this paper. The results of eigenfrequencies and modal shapes of the WT drivetrain using two, five, nine and eleven lumped masses are produced, compared and analysed in detail. The influence of damping is neglected in this study.

Table 1. Stiffness calculations of the shafts (N.m/rad)

K_{LSS}	Low speed shaft stiffness	3.69e+07
K_{IS1}	Stiffness of the shaft connecting the sun gear to the 1st parallel gear stage	2.45e+07
K_{IS2}	Stiffness of the shaft connecting the 1st parallel gear stage to 2nd parallel gear	2.70e+08
K_{HSS}	High speed shaft stiffness	2.08e+06
K_{gen}	Electrical torsional stiffness	2.81e+04

Table 2. Inertia calculations of the lumped masses (kg.m^2)

J_{rotor}	Inertia of the rotor	998138
J_{PC}	Inertia of planetary carrier	65.2
J_P	Inertia of planet gear	3.2
J_S	Inertia of sun gear	1.02
J_{G1}	Inertia of gear in 1st parallel stage	31.72
J_{G2}	Inertia of pinion in 1st parallel stage	0.4
J_{G3}	Inertia of gear in 2nd parallel stage	3.42
J_{G4}	Inertia of pinion in 2nd parallel stage	0.08
J_{gen}	Inertia of the generator	24

Table 3. Wind turbine gearbox ratios

N_1	Gear ratio of planetary stage	5.714
N_2	Gear ratio of 1st parallel stage	3.565
N_3	Gear ratio of 2nd parallel stage	4.00

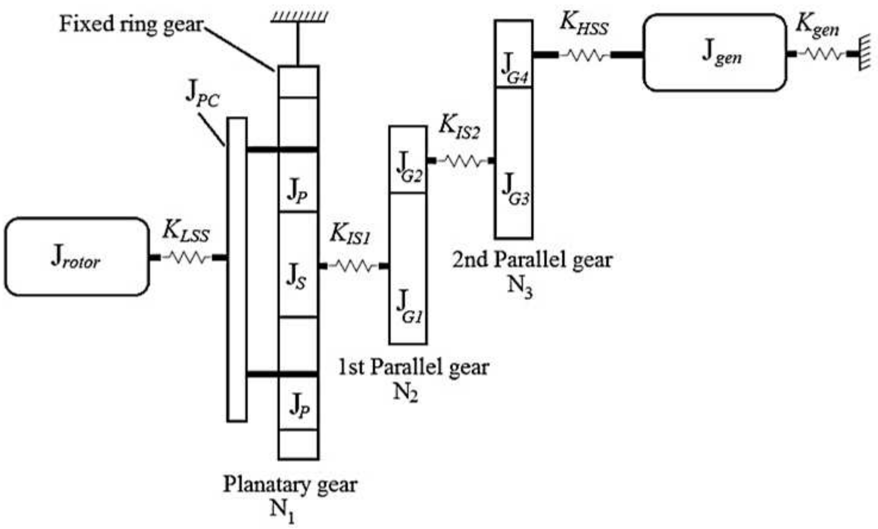

Figure 3. Representation of 750 kW WT drivetrain with eleven masses of 11 DOFs.

3.2 Mathematical Models of Wind Turbine Drivetrain

To illustrate the lumped mass modelling, the five-mass model of 5 DOFs of the WT drivetrain is shown in Figure 4. Each gear stage is combined as one effective mass. The gear mesh stiffness is ignored in the mathematical models. The five effective mass inertias, with respect to the LSS of the drivetrain, are calculated as follows:

$$J_1 = J_{rotor} \tag{4}$$

$$J_2 = J_{pc} + N_1^2 J_s \tag{5}$$

$$J_3 = N_1^2 J_{G1} + (N_1 N_2)^2 J_{G2} \tag{6}$$

$$J_4 = (N_1 N_2)^2 J_{G3} + (N_1 N_2 N_3)^2 J_{G4} \tag{7}$$

$$J_5 = (N_1 N_2 N_3)^2 J_{gen} \tag{8}$$

The stiffnesses of the LSS, all gearbox shafts and the generator resistance torque for a fixed speed generator model are calculated as follows:

$$K_1 = K_{LSS} \tag{9}$$

$$K_2 = N_1^2 K_{IS1} \tag{10}$$

$$K_3 = (N_1 N_2)^2 K_{IS2} \tag{11}$$

$$K_4 = (N_1 N_2\, N_3)^2 K_{HSS} \tag{12}$$

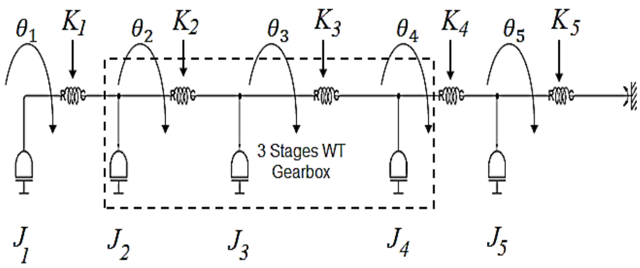

Figure 4. Five mass model representation of the WT drivetrain.

$$K_5 = (N_1 N_2\, N_3)^2 K_{gen} \tag{13}$$

Lagrange's equation is used to form the equations of motion. For free torsional vibration, the equation of motion of five mass model can be shown in matrix form as follows:

$$[J]\{\ddot{\theta}\} + [K]\{\theta\} = 0 \tag{14}$$

$$\theta = [\theta_1 \quad \theta_2 \quad \theta_3 \quad \theta_4 \quad \theta_5\,]^T \tag{15}$$

$$J = \begin{pmatrix} J_1 & 0 & 0 & 0 & 0 \\ 0 & J_2 & 0 & 0 & 0 \\ 0 & 0 & J_3 & 0 & 0 \\ 0 & 0 & 0 & J_4 & 0 \\ 0 & 0 & 0 & 0 & J_5 \end{pmatrix} \tag{16}$$

$$K = \begin{bmatrix} K_1 & -K_1 & 0 & 0 & 0 \\ -K_1 & K_1 + K_2 & -K_2 & 0 & 0 \\ 0 & -K_2 & K_2 + K_3 & -K_3 & 0 \\ 0 & 0 & -K_3 & K_3 + K_4 & -K_4 \\ 0 & 0 & 0 & -K_4 & K_4 + K_5 \end{bmatrix} \tag{17}$$

For free force vibration, the eigenmodes and eigenfrequencies of the WT drivetrain system can be calculated using:

$$\omega^2 = eig\,(J^{-1}K) \qquad (18)$$

Where J, K and θ are the inertia matrix, the stiffness matrix, and the rotational displacement vector respectively. θ_1, θ_2, θ_3, θ_4 and θ_5 are the torsional displacement of the rotor, three gear stages and the generator respectively. By applying the same modelling method, the drivetrain dynamic models of two, nine and eleven masses can also be developed. The comparison analysis of the calculated eigenfrequencies for different drivetrain models, using both fixed and variable generator speed models, is summarized in Table 4. The results of the natural eigenfrequencies are compared with those available in published literature (Girsang et al. 2014) (Mandic et al. 2012) and show good agreements. Exciting the drivetrain system at any of these eigenfrequencies will lead to amplified loads in the WT drivetrain. To validate the models of this study, the experimentally measured values of the 1st and 2nd natural frequencies of two mass model of the 750kW WT drivetrain are listed in Table 4 (Mandic et al. 2012) (Girsang et al. 2014). The analytical results from this study are close to those values.

3.3 Modelling of Wind Turbine Drivetrain by Using MATLAB/Simulink

The WT drivetrain modelled mathematically in the previous section with various levels of complexity is also modelled by using Simscape in MATLAB/Simulink. The configuration of the Simulink model is shown in Figure 5. For the eleven mass model of 11 DOFs, the mesh stiffness between the gears is included, while it is ignored in the previous models developed. The mesh stiffness is assumed to be a linear spring positioned in the plane of the action and acting at the centre of the gear widths and calculated according to ISO6336-1:2006. The calculations of frequency response function (FRF) of different lamped mass models of 2, 5, 9 and 11 DOFs have been performed, using a fixed and variable speed generator model respectively. The calculated eigenfrequencies by mathematical models are compared with that of the Simulink models, showing an excellent agreement. For the variable speed generator, a simple induction generator model is used (Bruce et al. 2015). The generator model is capable of providing generator resistance and controls the rotor speed for two different operational conditions: normal operation and shutdown. The effects of these two operating conditions on the dynamic responses of the drivetrain system are investigated in this paper.

Based on data available in published literature, there are four different values have been used for gear mesh stiffness of the planetary gear stage of the 750kW WT gearbox. These values are summarized in

Table 5. In the following section, the effects of different gear mesh stiffness values on the dynamic responses of the WT drivetrain are compared. Numerical analysis of the torque load calculated from the MATLAB/Simulink models of the drivetrain is also performed by using Fast Fourier Transform (FFT).

Table 4. Comparison of frequency values of different lumped mass models of the drivetrain (Hz)

Mode	Experimental data		2-Mass Model		5-Mass Model		11-Mass Model	
	Constant speed*	Variable speed**	Constant speed	Variable speed	Constant speed	Variable speed	Constant speed	Variable speed
1	0.806	0	0.864	0	0.864	0	0.841	0
2	/	1.76	5.888	2.524	5.889	2.524	5.859	2.44
3	/	/	/	/	312	312	154	154
4	/	/	/	/	402	402	307	307
5	/	/	/	/	1974	1974	353	353
6	/	/	/	/	/	/	748	748
7	/	/	/	/	/	/	1020	1020
8	/	/	/	/	/	/	1530	1530
9	/	/	/	/	/	/	4398	4397

* Mandic et al. (2012)
**Girsang et al. (2014)

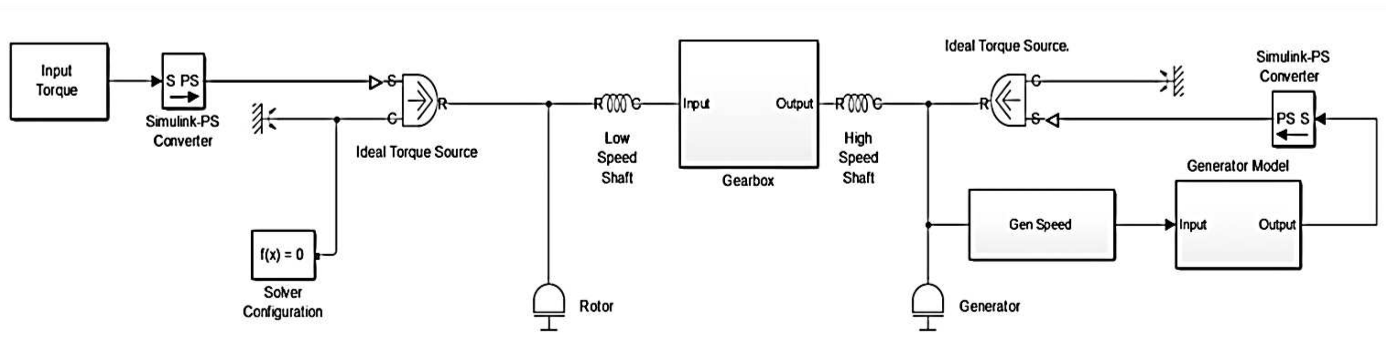

Figure 5. Configuration of MATLAB/Simulink of eleven mass model of the drivetrain

Table 5. Different mesh stiffness values used for planetary stage of 750kW WT gearbox

Source	Mesh stiffness value		Units
	Planet-Sun	Planet- Ring	
Guo et al. (2012)	$16.9\text{x}10^{9}$	$19.2\text{x}10^{9}$	N/m
Link et al. (2011)	-	$4.35\text{x}10^{12}$	Nm/rad
Hong et al. (2014)	$5\text{x}10^{8}$	-	N/m
Bruce et al. (2015)	$13.1\text{x}10^{9}$	$13.1\text{x}10^{9}$	N/m^{2} *

*per unit face width

4 RESULTS AND DISCUSSION

The result comparison of the different models with various DOFs indicates important differences of the dynamic behaviour of the gearbox components during different operational conditions. Figure 6 illustrates the mode shapes of the eleven-mass model using the variable speed generator model. Increasing the model DOF enables the prediction of more complex mode shapes of the drivetrain components and a wider frequency range thus more accurate descriptions of the torsional vibration of the transmission chain. It is impossible to predict the frequencies and the related mode shapes of the WT gearbox by using the two mass model. Figure 7 shows the influence of different gear mesh stiffness values (given in Table 5) on eigenfrequencies, using the MATLAB/Simulink model of eleven mass with the fixed speed generator model. Increasing the gear mesh stiffness value results in a wider frequency range and increases the frequencies of the gearbox components but has no impact on lowest natural frequencies of the drivetrain components, i.e. the LSS, the generator and the HSS. However, when variable speed generator model is used, the order of the LSS and generator frequencies is changed, thus the first three lowest frequencies represent the responses of the generator, LSS and the HSS respectively. The remaining modes represent the gearbox frequencies. Figures 8-11 illustrate the simulation results in time domain and the FFT analysis in frequency domain of the HSS torque during normal operation and shutdown, respectively. In normal operation, the frequency spectrum shows the dominated frequency is 0.84 Hz which is close to the estimated natural frequency of the LSS thus may cause the system resonance and load amplification. During shutdown the most dominated frequency is 2.61 Hz which is very close to the estimated natural frequency of the generator (i.e. 2.44 Hz) which may also contribute to the resonance of the drivetrain system.

5 CONCLUSIONS

The WT drivetrain models with different levels of complexity influence eigenfrequencies and eigenmodes of the WT gearbox, LSS, HSS and the generator. The eleven lumped mass drivetrain model developed by using MATLAB/Simulink captures the torsional loading on all stages of the gearbox. FFT analysis highlights the importance of developing detailed WT drivetrain model. It has found that the increasing gear mesh stiffness value results in a wider frequency range, raises the gearbox frequencies but has no impact on 1st, 2nd and 3rd lowest frequencies of the drivetrain. Moreover, it has no influence on two lowest and dominant frequencies of the drivetrain (i.e. 0.84 and 2.44 Hz) during normal operation and shutdown.

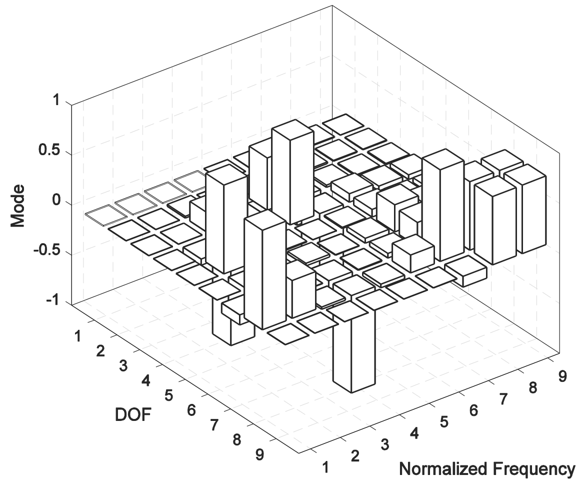

Figure 6. Mode shapes of eleven mass model of drivetrain with variable speed generator model

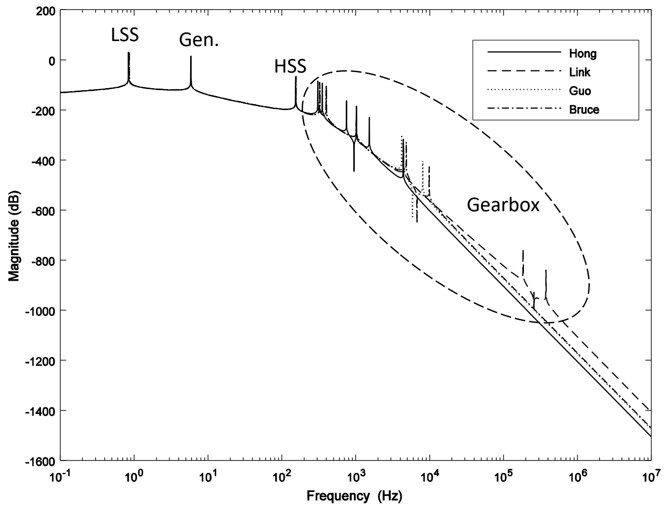

Figure 7. FRF of the drivetrain using fixed speed generator model and variable gear mesh stiffness values

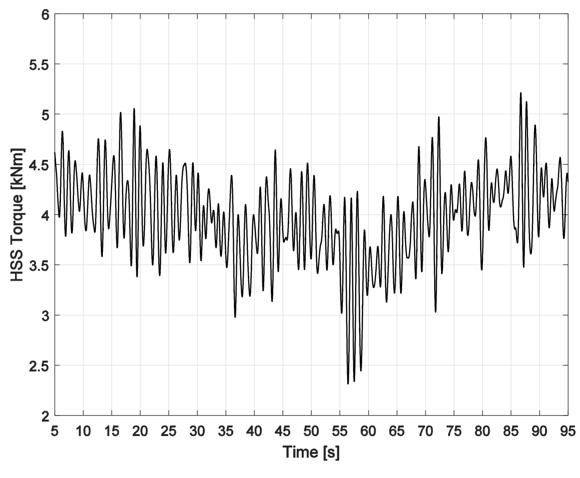

Figure 8. HSS torque - normal operation

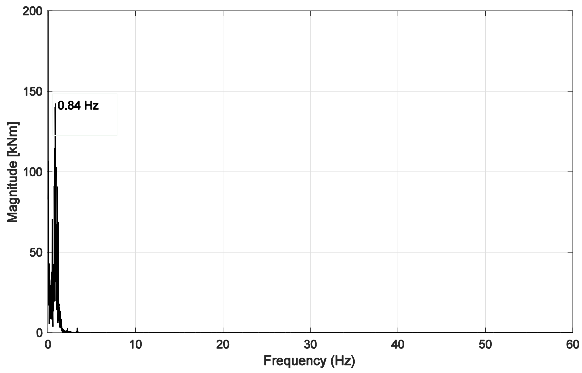

Figure 9. FFT analysis of HSS torque - normal operation

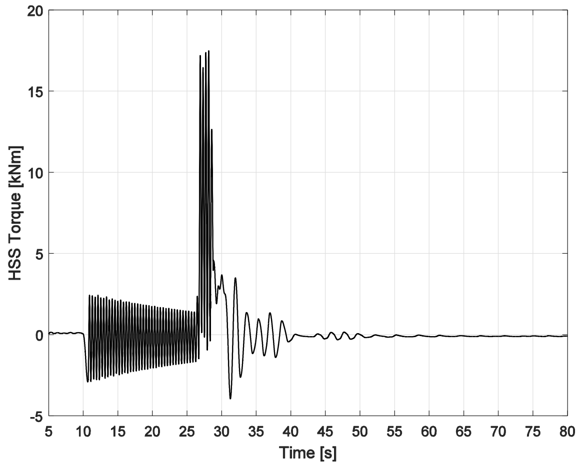

Figure 10. HSS torque - shutdown

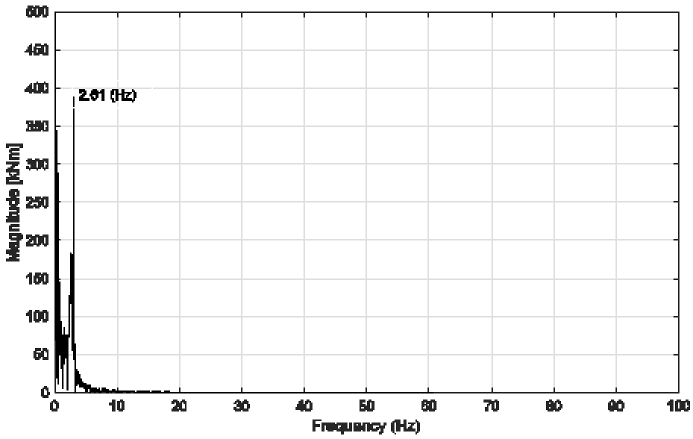

Figure 11. FFT analysis of HSS torque – shutdown

REFERENCES

Bruce, T., Long, H. & Dwyer-Joyce, R.S., 2015. Dynamic modelling of wind turbine gearbox bearing loading during transient events. IET Renewable Power Generation, (February), pp.1–10.

Daniels, L., Johnson, S. & Slaymaker, W., 2004. Harvest the wind: A wind energy handbook for Illinois.

Girsang, I.P., Dhupia, J.S. & Pao, L.Y., 2014. Gearbox and Drivetrain Models to Study Dynamic Effects of Modern Wind Turbines Eduard Muljadi Mohit Singh. , 9994(c), pp.874–881.

Grujicic, M. et al., 2014. Computer-aided engineering analysis of tooth-bending fatigue-based failure in horizontal-axis

Power Transmissions, 2017 Taylor & Francis Group, London, ISBN 978-1-138-03267-5, page 205-212

wind-turbine gearboxes. International Journal of Structural Integrity, 5(1), pp.60–82.

Guo, Y., Keller, J. & Parker, R., 2012. Dynamic analysis of wind turbine planetary gears using an extended harmonic balance approach. International Conference on Noise and Vibration, (June).

Hong, L., Dhupia, J.S. & Sheng, S., 2014. An explanation of frequency features enabling detection of faults in equally-spaced planetary gearbox. Mechanism and Machine Theory, 73, pp.169–183.

LaCava, W. et al., 2012. Determining wind turbine gearbox model complexity using measurement validation and cost comparison, Wind Energy Association annual conference, (April).

LaCava, W. et al., 2013. Three-dimensional bearing load share behaviour in the planetary stage of a wind turbine gearbox. IET Renewable Power Generation, 7(4), pp.359–369.

Link, H. et al., 2011. Gearbox Reliability Collabora*tive Project Report : Findings from Phase 1 and Phase 2 Testing.*

Mandic, G. et al., 2012. Active torque control for gearbox load reduction in a variable-speed wind turbine. IEEE Transactions on Industry Applications, 48(6), pp.2424–2432.

Oyague, F., 2009. Gearbox Modeling and Load Simulation of a Baseline 750-kW Wind Turbine Using State-of-the- Art Simulation Codes , (February).

Oyague, F., Gorman, D. & Sheng, S., 2008. Progressive dynamical drive train modeling as part of NREL gearbox reliability collaborative. Windpower 2008 Conference , (July).

Peeters, J., Vandepitte, D. & Sas, P., 2006. Structural analysis of a wind turbine and its drive train using the flexible multibody simulation technique. Proceedings of ISMA, pp.3665–3680.

Peeters, J.L.M., Vandepitte, D. & Sas, P., 2005. Analysis of internal drive train dynamics in a wind turbine. Wind Energy, 9(1-2), pp.141–161.

Rodríguez, a G.G., Rodríguez, a G. & Payán, M.B., 2007. Estimating Wind Turbines Mechanical Constants. International Conference on Renewable Energies and Power Quality, pp.9–11.

Scott, K., Infield, D. & Barltrop, N., 2012. Effects of Extreme and Transient Loads on Wind Turbine Drive Trains. , pp.1–24.

Shi, W. et al., 2014. Dynamic analysis of three-dimensional drivetrain system of wind turbine. International Journal of Precision Engineering and Manufacturing, 15(7), pp.1351–1357.

Shi, W. et al., 2013. Dynamic modeling and analysis of a wind turbine drivetrain using the torsional dynamic model. International Journal of Precision Engineering and Manufacturing, 14(1), pp.153–159.

Smolders, K. et al., 2010. European Wind Energy Conference (EWEC) Reliability Analysis and Prediction of Wind Turbine Gearboxes.

Tavner, P. et al., 2008. Reliability of different wind turbine concepts with relevance to offshore application. Wind Energy Conference, (April).