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On Initial Design and Modelling of a 10 MW Medium Speed Drivetrain for Offshore Wind Turbines

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Abstract. This paper presents the initial design and modelling of a medium speed drivetrain for the DTU 10 MW offshore wind turbine. Design basis and criteria of the drivetrain are described. Minimum weight and volume of the drivetrain are chosen as the main design targets. A four-point support, two main bearings and two torque arms, drivetrain configuration is selected in this study. One conventional gearbox layout, two planetary stages and one parallel stage, which is commonly used in large offshore wind turbines is adopted. Four gearbox layout options are provided and compared, and one optimal option is finally selected with holistic considerations of volume and weight as well as load sharing performance principles. Then, a high fidelity drivetrain numerical model for the 10 MW reference turbine is established using multi-body system (MBS) approach. In addition, model comparison is conducted, which shows that drivetrain eigenfrequencies in torsional mode obtained from the numerical model agree well with the values in the DTU report. Finally, future work relating to dynamic response analysis, critical limit states check for the drivetrain model is proposed.

1. Introduction

The offshore wind energy industry has been growing fast in the last decade, primarily due to the strong and steady wind resources in offshore fields. Meanwhile, the size of offshore wind turbines has been increasing rapidly over the years, as it is believed that the large size turbine has a great potential to reduce the levelized cost of wind energy, considering with the transportation and installation issues. Many multi-megawatt offshore wind turbines have been appeared on the market, such as the 9.5 MW turbine manufactured by MHI Vestas, 8 MW turbine launched by Siemens Gamesa, and 6.7 MW turbine developed by Goldwind [1].

However, increasing the wind turbine size usually leads to the increase of its weight. To address this challenge, a 10 MW wind turbine concept, with a specialized light-rotor design, was proposed by Technical University of Denmark (DTU) in 2013 [2]. The DTU 10 MW reference wind turbine (RWT) was designed by upscaling the National Renewable Energy Laboratory (NREL) 5 MW offshore wind turbine [3]. In order to carry out the test and validation of the new wind turbine concept, design, modelling and analysis for the entire wind turbine system are impending. Many efforts have been devoted by researchers to demonstrate the concept design or dynamic response analysis of some critical structures in the 10 MW wind turbine. Based on the rotor design of the DTU 10 MW RWT, active trailing edge flaps were integrated into the blades design by Barlas et al.[4], and aeroelastic optimization of the design was conducted by utilizing a multi-disciplinary wind turbine analysis and optimization tool, HawtOpt2. Preliminary



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monopile designs for water depths 20m, 30m, 40m and 50m were established by Velarde et al. [5] to support the DTU 10 MW RWT, and fatigue limit state (FLS) analyses were conducted using the software SIMO-RIFLEX to verify the design. Two spar platforms were designed by Hegseth et al. [6] for the DTU 10 MW RWT, and a semi-analytical frequency domain model was established for efficient design optimization of spar floating wind turbines. Smilden et al. [7] presented multiple kinds of wind turbine control strategies, and investigated the effects of these control strategies on fatigue damage of offshore wind turbine support structures due to varying site conditions, with a case study of the DTU 10 MW monopile wind turbine.

Much less attentions, however, have been paid on the 10 MW drivetrain study. A critical review of the available publications for wind turbine drivetrain research indicates that most studies are based on small scale wind turbines (e.g. 750KW-3MW), with mainly for onshore applications [8, 9, 10]. Meanwhile, limited studies for drivetrains applied in multi-megawatt offshore wind turbines have been published. For instance, Nejad et al. [11] designed a 5 MW gearbox model for the NREL offshore baseline wind turbine. Detailed modelling parameters and technical data of the model were provided for public use, aiming to support the concept studies for large offshore wind turbines. In their another work [12], fatigue damage of the 5 MW drivetrain, supported on a land-based structure and other four types of floating structures: spar, TLP and two semi-submersibles, is compared, which deals with the feasibility of large wind turbine drivetrain designed for land-based turbine, on floating wind turbines.

All of these studied drivetrains, however, are configured with a conventional high speed layout, which can not provide essential reference for the 10 MW RWT drivetrain design, as an innovative drivetrain configuration, medium speed drivetrain, was proposed by DTU for the RWT. The medium speed drivetrain is a compromise between the conventional high speed drivetrain and direct drive drivetrain. On one aspect, it might lower the failure probability compared to the high speed drivetrain. On the other aspect, it could effectively reduce the generator size and weight compared to the direct drive drivetrain. Thus, the medium speed drivetrain concept seems to be a promising option to apply in large scale offshore wind turbines.

The motivation of this study is to provide a baseline medium speed drivetrain model for the DTU 10 MW RWT, that could be used as a reference model for multi-megawatt offshore wind turbines. A four-point support drivetrain configuration equipped with a traditional gearbox layout, consisting of two planetary stages and one parallel stage, is proposed. Drivetrain design basis and criteria are described. Critical drivetrain components-gears, bearing and shafts-are designed based on relevant international design codes. All of these components are designed to withstand fatigue loads and ultimate loads during normal operating conditions of the wind turbine. Four gearbox design options are presented and compared, and the final gearbox design option is selected through a holistic assessment of its weight, volume and load sharing performance. The numerical drivetrain model is established by means of the MBS method. The first order torsional eigenfrequencies, as the significant drivetrain property, are compared between the numerical model and the simplified drivetrain model provide by DTU, indicating a good agreement.

2. Description of the 10 MW reference wind turbine

In this paper, the drivetrain is designed for the DTU 10 MW RWT supported on a bottom-fixed monopile structure in an offshore siting with an IEC class 1A wind climate. The overall specifications of the wind turbine are presented in Table 1. In terms of the drivetrain, only limited properties are provided by DTU, as presented in Table 2, and DTU did not design the drivetrain and establish a detailed model.

Table 1. DTU 10 MW wind turbine specification [2].

Parameter	Value
Wind regime	IEC Class 1A
Type	Upwind/three blades
Cut in, rated, cut out wind speed (m/s)	4, 11.4, 25
Rated power (MW)	10
Rotor, hub diameter (m)	178.3, 5.6
Hub height (m)	119.0
Drivetrain	Medium speed layout
Rotor mass (kg)	227962
Nacelle mass (kg)	446036
Tower mass (kg)	628442

Table 2. 10 MW Drivetrain properties provided by DTU [2].

Parameter	Value
Rated rotor speed (rpm)	9.6
Rated generator speed (rpm)	480
Gearbox ratio	1:50
Electrical generator efficiency (%)	94
Generator inertia about high-Speed shaft (kgm ²)	1500.5
Equivalent drive-shaft torsional-spring constant (Nm/rad)	2317025352
Equivalent drive-shaft torsional-damping constant (Nm/(rad/s))	9240560

3. Wind turbine drivetrain design

3.1. Drivetrain design basis and criteria

The main components of drivetrain include hub, mainshaft, main bearing, gearbox, coupling, generator and mainframe. The geometry of hub, mainshaft, coupling, generator and mainframe are designed by referring to the corresponding components in a 2 MW land-based wind turbine, as presented by Wang et al. [13]. The gearbox, one of the essential parts in the drivetrain, is designed based on the the international standard IEC 61400-4 [14], which documents the design requirements for wind turbine gearboxes. Drivetrain support configuration, serving as a significant interface between the wind turbine and the gearbox, is firstly defined, due to the input loads applied on gearbox are influenced by the interface. According to the description of the common drivetrain configurations in IEC 61400-4 [14] and the study of Torsvik et al. [15], investigating the feasibility of different drivetrain arrangements applied in modern large offshore wind turbines, one four-point support drivetrain configuration is selected in this study. The studies of Jiang et al. [16] and Nejad et al. [17] present that the operational wind conditions are the most important to fatigue life of gearbox. Thus, the drivetrain design loads, implying the ultimate and fatigue design loads, are derived from the normal operating conditions, specializing with the design load cases DLC 1.1 and DLC 1.2 in the international standards IEC 61400-1 [18] and DNVGL-ST-0361 [19].

The design service life of gearbox is defined to be 20 years according to the recommendation in the IEC 61400-4 [14] standard. All gearbox components, gears, bearings and shafts are designed to satisfy the required safety conditions in relevant design codes. According to IEC 61400-4 [14], gears in the wind turbine gearbox are designed based on the ISO 6336 [20, 21, 22] series of standards, by which the gear carrying loads and load capacities are calculated. Gear safety

Table 3. Four gearbox design options.

Parameter	A	B	C	D
First stage	1:6000 (3p)	1:5.280 (4p)	1:4.423 (5p)	1:3.316 (6p)
Second stage	1:5.348 (3p)	1:5.160 (3p)	1:5.192 (3p)	1:5.625 (3p)
Third stage	1:1.556	1:1.826	1:2.179	1:2.680
Total dry weight ($\times 1000kg$)	65.66	60.59	60.43	57.16
Maximum gear outer diameter (m)	3.878	3.396	3.098	3.068

factors, in terms of gear tooth pitting and bending failure modes, recommended by IEC 61400-4 [14] are used to guide the gear design. Meanwhile, bearings in this study are designed according to the standard ISO 281 [23]. The bearings' types and arrangements are defined by referring to the description in IEC 61400-4 [14] and the study relating to the 5 MW offshore wind turbine gearbox design, conducted by Nejad et al. [11]. The required 20 years of working life is served as the design criteria of bearings.

3.2. Gearbox design options

The most common gearbox applying in multi-megawatt wind turbines consists of two planetary stage and one parallel stage, and this configuration is used for the 10 MW RWT drivetrain design in this study, by considering the feasibility of the application. Schmidt et al. [24] concludes that conventional two stage planetary gearboxes are facing design restrictions limiting their transmission ratio to about 40. As a 1:50 gearbox ratio is defined in this case study, two planetary stages + one parallel stage gearbox could be a reasonable configuration. The priority design principle in this study is minimizing the weight and volume of gearbox, due to a heavy gearbox would lead to a hefty transportation and lifting cost, especially for large offshore wind turbines installed in deep ocean fields.

The flowchart of gearbox design is presented in Figure 1. An initial gear ratio for each stage is firstly assigned, and gear size is then calculated based on gear bending and pitting fatigue limits, which followed by the determination of shaft' dimension and bearing' selection. Then, the gearbox concept model is established with initialled parameters of gears, bearings and shafts. The fatigue and ultimate limit states check for the concept model is further carried out, with defined fatigue and extreme design loads, according to design requirements in relevant design codes. When all components are approved to satisfy the design requirements, the potential of the gearbox weight and volume optimization is checked by assessing the safety margin in each stage. A gearbox model with optimized weight and volume can be eventually obtained by taking an iterative design process through reassigning the input of gear ratio in each stage. The entire gearbox design is performed using the professional gearbox design tool KISSsoft 2018 [25].

Table 3 shows four kinds of gearbox design options, and each gearbox scheme is designed according to the flowchart in Figure 1. The priority attention in this comparison is maximum gear outer diameter, because a larger gear outer diameter would lead to a larger nacelle volume such that the nacelle weight would be heavy. Option A and Option B are presented with relatively large gear outer diameter and heavy total dry weight, so that they are not chosen. An almost equivalent gear outer diameter with a bit light total dry weight can be found in option D in contrast with option C, but the loss of the load-sharing performance of the option D with six planets design would outweigh the gain of weight reduction compared to the option C. Therefore, Option C equipped with five planets in the first stage and three planets in the second stage is selected in this case study.

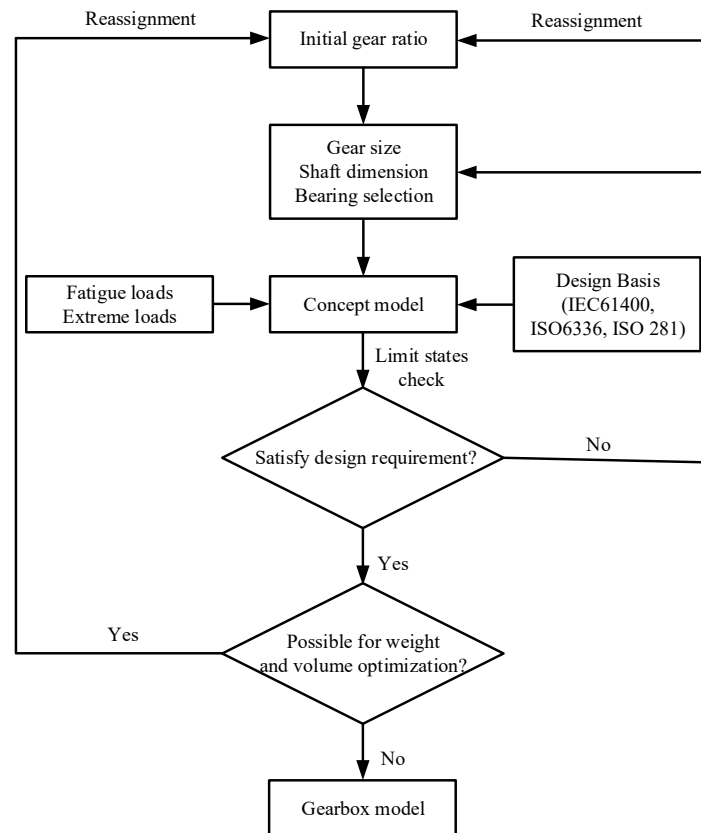


Figure 1. Flowchart of gearbox design

3.3. Drivetrain specifications

Figure 2 presents the 10 MW drivetrain schematic layout. It can be seen from the layout that a two-main-bearing support is adopted in this concept model, where the downwind bearing is designed by bounding two identical bearings together to withstand huge non-torque loads transmitted from the hub center. Moreover, each planet in the gearbox first stage is supported by four bearings in order to resist the huge loads input from the planet carrier. The general drivetrain specifications are listed in Table 4.

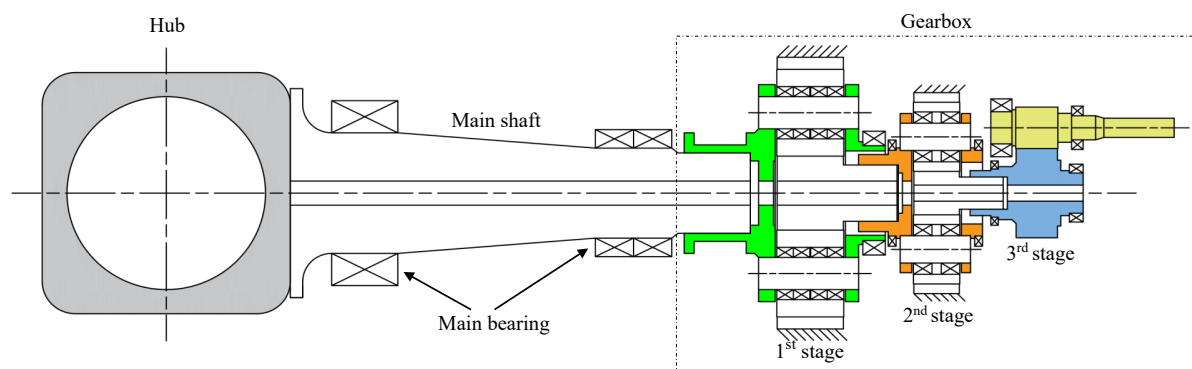


Figure 2. 10 MW reference wind turbine drivetrain schematic layout

Table 4. 10 MW reference wind turbine drivetrain specifications.

Parameter	Value
Drivetrain type	4-point supports
Gearbox type	Two planetary + one parallel
First stage gear ratio	1:4.423
Second stage gear ratio	1:5.192
Third stage gear ratio	1:2.179
Total gear ratio	1:50.039
Designed power (kw)	10000
Rated input shaft speed (rpm)	9.6
Rated generator shaft speed (rpm)	480.374
Total gearbox dry weight ($\times 1000kg$)	60.43
Maximum gearbox outer diameter (m)	3.098
Service life (year)	20

4. Numerical modelling of the wind turbine drivetrain

The MBS approach is applied for the dynamic modelling of the drivetrain. The basic modelling elements in MBS model are “body”, “joint” and “force element”. The MBS model are composed of various rigid and flexible bodies, and the connections between the bodies are modelled with joints or force elements. Dynamic behaviors of mechanical or electromechanical system can be described or predicted by conducting the MBS simulation that has been widely used in the fields of vehicles, robots, and wind turbines, etc.

In this study, the numerical model of the 10 MW medium speed drivetrain is established using a MBS simulation software, SIMPACK[26], as shown in Figure 3. The numerical simulation tool has been widely used and verified for modelling and dynamic analysis of wind turbine drivetrains, as demonstrated in various studies (e.g. Refs. [27, 28, 29, 30]). Topological diagram of the drivetrain model, illustrating the connection relationships between different components and degree of freedoms (DOFs) of each component, is presented in Figure 4. Many studies (e.g. Refs. [31, 32, 33]) have demonstrated the significant effects of flexible body modelling on dynamic response of drivetrain. In order to accurately describe the natural properties of the drivetrain model, main shaft, planet carriers and transmission shafts in the gearbox are modelled flexible bodies, while other components are considered with rigid bodies in this study. The gear tooth contact is modelled by the force element FE225 in SIMPACK, in which the gear meshing stiffness is calculated based on the standard ISO 6336-1 [34]. The bearings are modelled with linear diagonal stiffness matrix, in which the stiffness elements are calculated by using a drivetrain design software, Romax [35].

5. Model comparison

Compared to the drivetrain model described by DTU using a simplified method, a detailed drivetrain is designed and its numerical model is established in this study. To assess the rationality of the drivetrain modelling parameters, such as stiffness, moment of inertia of drivetrain components, comparison of these two drivetrain models should be conducted. Specifically, the assessment could be performed by comparing the first eigenfrequency obtained from the detailed drivetrain model with the corresponding value from the the simplified model, as proposed in the standard DNVGL-ST-0361 [19].

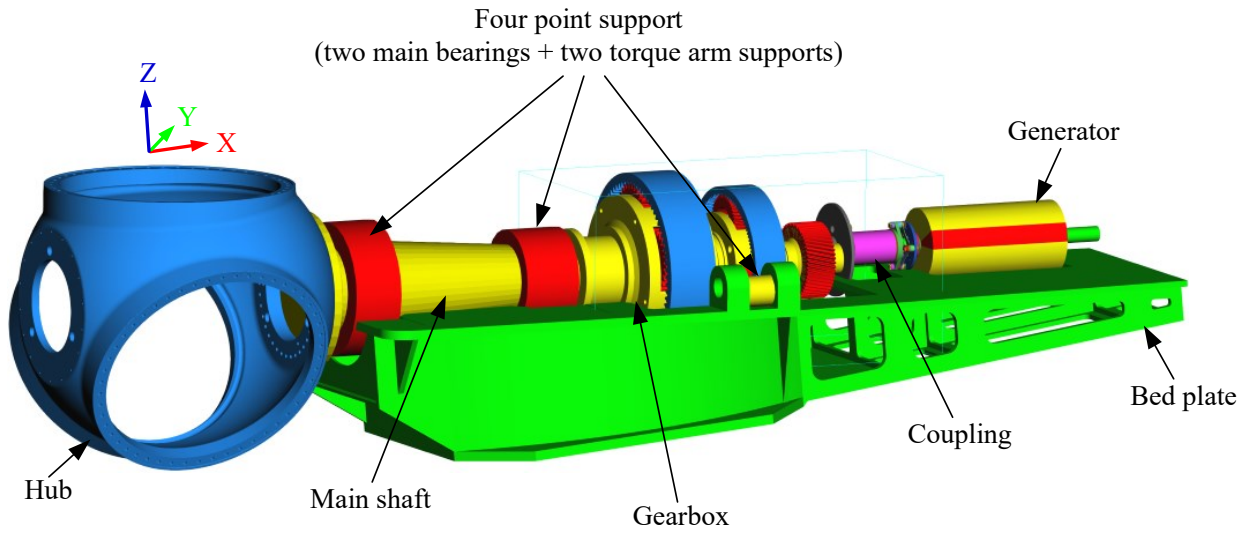


Figure 3. 10 MW drivetrain MBS model

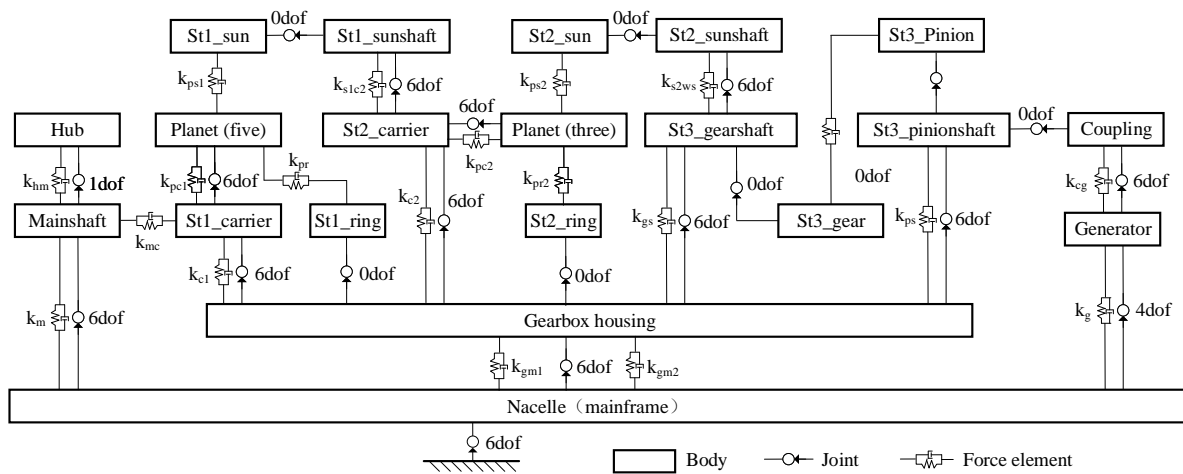


Figure 4. Topological graph of the 10MW wind turbine drivetrain model

In terms of the simplified model provided by DTU, the drivetrain is represented by a two-mass equivalent mechanical model, in which gearbox is regarded as a transmission ratio. A detailed description for the modelling approach are given by Oyague [8] and Nejad et al. [36]. The eigenfrequency in torsional mode can be calculated by the following equations, with fundamental drivetrain parameters (e.g. mass, stiffness and moment of inertia) given by DTU.

When the rotor and generator are free, the natural frequency ($F_{free-free}$) of this two-mass model is calculated by:

$$f_n = \frac{1}{2\pi} \sqrt{\frac{k_{teq}(J_r + n^2 J_g)}{J_r n^2 J_g}} \quad (1)$$

When the rotor is free and generator is fixed, the natural frequency ($F_{free-fixed}$) of this two-mass model is calculated by:

Table 5. Natural frequency for the shaft torsion mode.

Shaft torsion mode	Simulation frequency [Hz]	Reference frequency [Hz]
F _{free-free}	3.889	4.003
F _{free-fixed}	0.611	0.612

$$f_n = \frac{1}{2\pi} \sqrt{\frac{k_{teq}}{J_r}} \quad (2)$$

where J_r : rotor inertia (including shaft, hub and blades); J_g : generator inertia about the low speed shaft; k_{teq} : equivalent drive-shaft torsional stiffness; n : gearbox transmission ratio.

The k_{teq} can be calculated by the equation:

$$k_{teq} = \frac{k_{tr} n^2 k_{tg}}{k_{tr} + n^2 k_{tg}} \quad (3)$$

where K_{tr} : torsional stiffness of main shaft; k_{tg} : torsional stiffness of generator.

Defining these parameters in the SIMPACK numerical model to match the ones given by DTU. Then the drivetrain eigenfrequencies in torsional mode could be obtained by performing modal analysis to the numerical model. The eigenfrequencies obtained from the detailed drivetrain model and the corresponding values from the DTU wind energy report [2] are summarized in Table 5.

It is shown in Table 5 that the simulation eigenfrequencies, for both F_{free-free} and F_{free-fixed}, in the detailed drivetrain model agree well with the reference values in the simplified model, indicating the rationality of the modelling parameters used in the MBS drivetrain model.

6. Concluding remark and future work

This paper deals with the initial design and modelling of a 10 MW medium speed drivetrain for DTU offshore wind turbine supported on a bottom-fixed monopile structure. Drivetrain design basis and criteria are introduced. A four-point support drivetrain configuration equipped with a traditional gearbox layout, consisting of two planetary stages and one parallel stage, is applied. Four gearbox design options are presented and compared, by which the final design option is determined based on the gearbox weight and volume as well as load-sharing behavior principles. Further, a numerical drivetrain model is established using MBS simulation package SIMPACK. Finally, the detailed drivetrain designed in this study is compared with the simplified one provided by DTU. The results indicate the first order eigenfrequencies, representing critical drivetrain properties, in the two drivetrain models match well.

It is noted that this paper only presents the design basis and approach for the 10 MW drivetrain, with focus on the descriptions of motivation, design principles and basic process. The aim of this work is to devote certain efforts for the development of large scale offshore wind turbines by providing a 10 MW reference drivetrain model. Detailed design and modelling parameters and technology data of the 10 MW reference wind turbine drivetrain are provided, in authors' another work [37], for public use, where resonance analysis, long-term fatigue damage analysis of the drivetrain model are conducted.

To capture more steady and strong wind resources, the 10 MW wind turbine is potentially installed in deep ocean field. As the drivetrain is designed based on a bottom-fixed monopile

structure support, it is of interest to evaluate the dynamic behaviors of the drivetrain model supported on different types of floating structures. In addition, the design loads used in this study are only obtained from the normal operating conditions. It is necessary to check the drivetrain model by ultimate limit state (ULS) and accidental limit state (ALS) considering extreme and fault load cases.

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