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LARGE AEROELASTIC MODEL OF A FLOATING OFFSHORE WIND TURBINE: MECHANICAL AND MECHATRONICS DESIGN

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ABSTRACT

This paper deals with the mechatronic design of a large-scale wind turbine model (outdoor scaled prototype) based on the DTU 10MW. This is going to be integrated in the model of a multi-purpose floating structure to be deployed at the Natural Ocean Engineering Laboratory (NOEL) in Reggio Calabria (Italy). The floating wind turbine model is the downscaling of the full-scale structure designed within the EU H2020 Blue Growth Farm project. The structural design of the scaled wind turbine is presented, starting from the aeroelastic and aerodynamic design carried out in a previous work.

INTRODUCTION

The EU H2020 Blue Growth Farm project aims at developing an efficient, cost-competitive, environmentally-friendly and multi-purpose offshore farm based on a modular floating structure. Wave energy converters and a wind turbine, which design will be based on the DTU 10MW reference wind turbine (RWT) [1], are integrated with aquaculture for profitable applications in different open-sea scenarios. The development of the multi-purpose floating structure is going to be supported by experimental data collected on a large-scale model (outdoor-scaled prototype) to be deployed at the Natural Ocean Engineering Laboratory (NOEL), in Reggio Calabria (Italy). This paper deals with the wind turbine scale model design, aimed at reproducing the aeroelastic, structural and functional characteristics of the full-scale wind turbine. All the components of the wind turbine model (i.e. rotor, nacelle and tower) must be designed by scaling the main RWT properties, also in

consideration of wind and sea conditions expected at NOEL, and, at the same time, by properly reproducing the functionalities of a real wind turbine. The aerodynamic design and the scaling strategies at the base of this work are briefly summarized (see [2] for a complete description), whereas, in the present paper, the structural project is presented more in detail.

NOMENCLATURE

RWT	Reference wind turbine
FOWT	Floating offshore wind turbine
U [m/s]	Hub-height horizontal wind speed
g [m/s ²]	Gravity acceleration
L [m]	Significative length
H_s	Significant wave height
U_{10min} [m/s]	10-minutes mean wind speed
λ [-]	Scale factor (ratio between full-scale and prototype quantities)
$Fr = \frac{U}{\sqrt{gL}}$	Froude Number
$Re = \frac{UL}{\nu}$	Reynolds Number
ν [m ² /s]	Kinematic air viscosity
t [m]	Airfoil thickness
c [m]	Chord length
C_L [-]	Lift coefficient
AoA [deg]	Angle of attack

E [N/m ²]	Young's modulus
EJ [N m ²]	Bending stiffness

REFERENCE TURBINE AND SCALING STRATEGIES

The DTU 10MW, reference for this work, was designed in the Light Rotor project framework in 2012 [3], starting from the upscaling of the NREL 5MW [4]. Later on, the Light Rotor project design evolved in the DTU 10 MW [1], which is currently used as reference in different research activities related to wind energy development, ranging from wind farm optimization to offshore wind turbine simulation or also for numerical tools benchmarking and validation [5, 6]. Table 1 reports the main DTU 10 MW specifications in terms of dimensions, masses and operating wind speed.

Table 1. DTU 10 MW wind turbine specifications.

Parameter	Value	Units
Cut in wind speed	4	m/s
Cut out wind speed	25	m/s
Rated wind speed	11.4	m/s
Rotor Diameter	178.3	m
Hub Diameter	5.6	m
Hub Height	119	m
Minimum Rotor Speed	6	rpm
Maximum Rotor Speed	9.6	rpm
Rotor Mass	228	t
Nacelle Mass	446	t
Tower Mass	628.4	t

The wind turbine outdoor scaled prototype is characterized by a scale (1:15) larger than the current standard for wind turbine scale models used for laboratory experiments (around 1:100). For this reason, it represents a challenging project that has to satisfy, at the same time, the requirements of a laboratory aeroelastic scale model and of a real small-scale wind turbine. A performance scaling approach had to be used in order to obtain a mechanical system able to grant the same functionalities of the full-scale wind turbine. The typical design strategies adopted to realize aeroelastic scale models [7, 8] were revised to include also criteria that are generally used in designing real machines and structures. In particular, the interaction with the real-life environment, the safety in the laboratory area and the local wind characteristics had to be considered.

A further complication is added by the peculiar environmental conditions met at the test site. Experiments will be performed in a so-called natural laboratory where, differently from traditional wind tunnel and ocean basin testing activities, it is not possible to control wind and wave characteristics. The design of the experiment and, consequently, of the wind turbine model, had to rely on probabilistic data about the marine and meteorological conditions at the test site, sampled over a period of one year. Moreover, the interdependence of wind and waves found at the NOEL site is not a perfectly scaled version of what is found at target FOWT deployment sites. This imposed to find a compromise in the definition of the scaling parameters to simultaneously match hydrodynamic and aerodynamic-related phenomena.

Considering that Froude scaling is mandatory because the hydrodynamic interaction of the floater is driven by gravity, this criterion was used to scale the thrust force acting on the rotor, the masses and the tower aeroelastic parameters. Other scaling approaches were considered to ensure the correct reproduction of the wind turbine rotor aerodynamics. A specific analysis was required to define the rated wind speed for the wind turbine model. The rated wind speed resulting by Froude-scaling the value defined for the RWT (see Table 1) is very low and not compatible with the wind statistics for the NOEL site. The rated speed for the model wind turbine was fixed at 5 m/s since this value grants an occurrence probability for the wind turbine operating conditions similar to those expected for the full-scale machine [2]. The resulting hub-height mean wind speed is also suitable as a dimensioning value for the structure with respect to ultimate loads. The wind profile and turbulence were not considered since the reproduction of fatigue loads on the wind turbine components was considered of secondary importance.

The rotor aerodynamic design also required a specific strategy since the outdoor scaled prototype rotor has to operate at Reynolds numbers different from those of the RWT. By geometrically scaling the wind turbine blades, a substantially lower power and thrust force, not in line with the full-scale, would be experienced [7]. It was then necessary to adopt a modified scaling approach, called performance scaling [8, 10]. This approach was aimed at reproducing the scaled thrust force at the rated wind speed (i.e. 5 m/s) by modifying the blade geometry. A new airfoil, specifically thought to have high efficiency at low Reynolds numbers, was used in place of those of the DTU 10MW rotor. The complete blade was then defined in terms of chord length, thickness and twist angle as function of the blade station (i.e. radial distance from rotor apex) using the thrust as main target of the optimization procedure. The power and thrust coefficients of the prototype, as resulting from steady-state FAST simulations, are compared to those of the DTU 10MW RWT in Figure 1.

Figure 1. Power (C_p) and thrust (C_t) coefficients for the DTU 10MW (blue) outdoor-scaled prototype (orange) for different pitch angles β and tip-speed ratios (TSR).

The tower design was simplified considering a cylindrical geometry without tapering. The cross sections diameter was then chosen as the best compromise between what is required to match drag force and to reproduce the Froude-scaled bending modes of the full-scale tower. The resulting tower aerodynamic behaviour is, with respect to Reynolds number, sub-critical whereas the full-scale tower behaves as a post-critical body. This implies an increased drag coefficient for the model (see Figure 2) and some differences in the wake dimension and flow characteristics. However, the main fluid-dynamics are similar for the model and full-scale system [12] and the developed drag force, even if different from the target value (see Table 4) is still negligible with respect to other aerodynamic loads like those related to the rotor thrust force.

In the nacelle design, the masses and the general dimensions were reproduced following Froude rules, while all the other features were defined in order to satisfy functionality requirements.

STRUCTURAL DESIGN

As highlighted in the previous section, the outdoor-scaled prototype is both an aeroelastic model of the RWT and a small machine that has to satisfy the strength requirements of a real structure exposed to the external environment. For this reason, the model design was assessed according to the IEC standard. This identifies four classes of wind turbines, among which class IV collects small wind turbines, with a swept rotor area smaller than 200 m². The standard [13] suggests 10 load cases, but only the five reported in Table 2 were considered in this work, discarding the others for not being representative for the current project. The load cases are defined by four parameters: wind speed, rotor-collective blade-pitch angle, rotor RPM and nacelle-yaw angle. The Rated case, with a rated wind speed of 5 m/s, is associated with the maximum thrust force in operating conditions. In the Park case (H in [13]) the rotor blades are fully feathered, and the wind turbine is exposed to the extreme wind for the design. The Full Exposure case was added by the authors to those defined in [13] to enhance the wind turbine design safety. The parameters are the same of the Park case, except for the blade pitch angle, that is set to zero to maximize the blade area projection normal to wind and, consequently, the aerodynamic loads on the parked rotor. For both the Park and Full Exposure the 50-year return period wind speed for the test site (33 m/s) was considered. The resulting design is safer than what is currently required by standards, also considering that the Full Exposure case that is not prescribed, is the most demanding among those of Table 2.

Table 2. The evaluated design load cases.

Case Name	U [m/s]	Ω [rpm]	Collective Pitch [°]	Yaw Angle [°]
Rated	5	101.9	0	0
Rated Yaw	5	101.9	0	30
Park	33	0	90	0
Full Exposure	33	0	0	0
Cut-Out	10.96	109.47	22.67	0

Tower

The wind turbine tower must be scaled in agreement to Froude law and, as described in [2], the preliminary scaling was carried out having as main target parameters the stiffness, mass and the first two bending modes (fore-aft and side-side). Responding to these constraints, the first tower design consisted of an aluminum tube dimensioned to have the proper flexural stiffness EJ. In a second instance a steel tube was preferred. The characteristics of the two solutions are reported in Table 3 where are also compared to those resulting from scaling the RWT tower properties. The steel solution was considered preferable because it allows to simplify the design of the connections between the tower and the floater and between the tower and the nacelle. Moreover, it was easier to find a steel commercial tube suitable for the application. In both cases, it is possible to note that, in order to match the EJ and then the tower bending modes, it was necessary to define a prototype external diameter smaller than target. On the other hand, even by properly scaling the tower geometry, there would be a strong discrepancy in terms of Reynolds number between prototype and full-scale. As shown in Figure 2, both the considered scale model tower designs are in the subcritical region whereas the real tower is in the supercritical Reynolds Number range. The differences between these two Reynolds conditions can be considered negligible for the definition of the aerodynamic behaviour of the tower and for its interaction with the rotor [12]. Table 4 reports the tower drag force developed by the two options for the model tower and the target. Finally, both the solutions are lighter than required, thus the correct mass and mass distribution will be obtained adding local masses along the tower height.

Figure 2. Drag coefficient of a smooth circular cylinder as a function of Reynolds Number.

Table 3. Tower characteristics.

	Scaled DTU 10MW	Prototype sol1	Prototype sol2
Tower Mass [kg]	186	72*	133*
E [N/m ²]	2.10E11	7.50E10	2.10E11
Material	steel	aluminium	steel
EJ [N m ²]	3.5E5 - 2.3E6	2.30E6	1.8E6
Frequency [Hz]	1.09	0.97	1
External diameter [m]	0.37-0.55	0.28	0.18

*tube mass, without considering added masses

Table 4. Drag force for the tower: prototype and target.

	Diameter [m]	Re	Drag force [N]
DTU 10MW	6.9	5.4E6	10.5
Steel	0.18	6.0E4	25.5
Aluminum	0.28	9.3E4	36.3

A FEM model of the scaled towers was realized to check the bending frequencies which values are reported in Table 3 and, as visible, are very close to target. The prototypes behaviour was also checked in terms of mode shapes: the comparison with those of the DTU 10MW RWT is reported in Figure 3 where a good agreement is observed.

Figure 3. Comparison between the scale model and full-scale first fore-aft mode shape from the FEM model.

The preliminary tower design was verified also from a structural point of view considering the Full Exposure condition as dimensioning case. Among the conditions of Table 2, this corresponds to the maximum rotor thrust force and is therefore the most critical from a structural point of view. As it is possible to see in Table 5, the identified solutions do not highlight any critical behavior both in terms of stresses and of maximum tower-top displacement.

Table 5. Loads, stress and tower top-displacement in Full Exposure condition.

	Thrust [N]	Moment [Nm]	Stress [MPa]	Max Displacement [mm]
Steel	1471	331	133	133
Aluminum	1471	331	53	103

Once the main tower geometry was defined, a more detailed design of the connections was carried out. The anchoring to the floating structure is ensured by a system of screws that allows levelling. Figure 4 shows the base flange obtained by welding a cylindrical element to a reinforced ribbed plate. The tower is in turn welded to the flange. A similar solution is adopted at tower-top, Figure 5, but in this case three elements are fixed to the flange: a) a slew ring bearing by means of which the nacelle can rotate around the pole axis, b) an electric motor that commands the aforementioned rotation and c) a stationary brake. The slew ring bearing has cogs on which two gears mesh. One is fitted directly to the shaft of the electric moto-reducer which steers the nacelle with respect to the wind whereas the other one is fixed to an idle shaft having a disc fixed on one end. On such disc an electromagnetic stationary brake which is normally close, acts. This brake maintains the position of the nacelle during operations and preventing the yaw rotation when the turbine is parked. All the overmentioned elements were chosen in consideration of the static expected loads on the structure. When the model will be completed, a final FEM analysis will be performed, and the final design will be refined in consideration of the estimated floater dynamics.

Figure 4. Tower-base connection flange. **Figure 5.** Tower-top to nacelle connection flange.

Blades

After having carried out the blades aerodynamic design, their structural project must be developed choosing the

appropriate composite material, ply stack and thickness of the single ply (see Figure 6 for the 3D blade shape). The most common fibers used in composite materials are E-glass, Aramid, Carbon and Polyethylene. Since the turbine will be subjected to a marine environment, aramid fibers, like Kevlar, are not a viable option. Polyethylene has good strength and stiffness properties, but its low melting temperature and tendency to creep makes it an undesirable choice for an outdoor prototype in warm climate. Carbon offers a better stiffness and strength than E-glass but a considerably higher cost, thus E-glass was chosen for the final design.

Figure 6. The external geometry of the blade subdivided into 40 blade stations.

The blades structure was realized defining the thicknesses and fibers direction of the lamina in each ply of the external shell and shear webs to achieve the target mass, frequency and strength. A symmetrical and balanced stacking sequence (see Figure 7) was chosen for the both the external shell and shear webs to minimize axial-shear and bending-membrane couplings. Unidirectional plies were preferred over multi-axial for the easier modelling and characterization of the resulting mechanical properties. Four orientations for the fibers were considered: 0° , 90° , 45° , -45° . The 0° fibers were used at the extremities of the laminate, to resist efficiently to bending moments; the 90° fibers were used to resist shear and the $\pm 45^\circ$ fibers to counteract torsion. The stacking sequence is such that two adjacent unidirectional plies can be replaced during the manufacturing phase by bidirectional plies, if necessary. Each blade element has a constant number of layers, as well as the same stacking sequence, for ease of manufacturing.

Figure 7. An example of the ply stack of the external shell. The first and last layer are aligned in the 0° direction.

Starting from the root and moving to the tip, the blade thickness monotonically decreases, keeping the same stacking sequence and removing one layer for each orientation, to meet mass requirements. When just a layer per orientation is remaining, the plies are removed from the middle outwards. In the definition of the plies sequence it is important to take into account the main expected loads acting on the blades: in this first structural design stage only static forces were considered. Once the dynamic excitation due to the floating platform will be defined, the preliminary project will be refined in consideration of dynamic forces. The largest portion of aerodynamic loads is generated in the central region of the rotor. Moreover, the blades are fixed at the hub in correspondence of root sections that are then subjected to the maximum bending moment and the highest stresses. It is important to notice that the thickness of the root region does not influence the resistance of the tip region, since the tip is not constrained. This allows to dimension the root region for ultimate loads and the tip region to reach the target mass and frequency. For what concerns the shear webs (Figure 8), a constant thickness was chosen; the webs are straight, with no twist to limit manufacturing complexity.

Figure 8. The 3D geometry of the shear webs.

The aerodynamic loads acting on the blades were computed from rigid steady-state simulations in FAST and applied in Abaqus to a reference point for each blade station; the reference point was then kinematically coupled with the external surface of the blade. Figure 9 and Figure 10 show the tangential and normal aerodynamic loads per unit length on the blades for each load case reported in Table 2. The normal force is positive when pointing in the downwind direction, the tangential force in the direction of the peripheral velocity of the blade. Gravitational and centrifugal loads were included in Abaqus, as well as volume forces. In order to evaluate the stress state of the blades, the Tsai-Hill criterion was used. This was applied in Abaqus considering IEC and GL (Germanisch-Lloyd) standards that, for the considered application, prescribe a global safety factor equal to 2.43. All the load cases summarized in Table 2 resulted in a large safety factor, whereas the Full Exposure case required a safety factor slightly smaller than the one suggested by the IEC. As already mentioned, this case was not requested by the standard, being the combination of the worst possible fault conditions and extreme wind and was considered in this work to enhance the design safety.

Figure 9. Forces per unit length normal to the rotor plane for all design cases.

Figure 10. Forces per unit length tangential to the rotor plane for all design cases.

Another significant failure mode to be considered in the blade design is buckling, a behavior often exhibited by thin shells. This is a form of geometric instability, in which the structure responds proportionately to a load pattern until a critical value is reached, after which it shows big deformations and possibly a critical failure. Considering the Abaqus model, the linear eigenvalue analysis methodology was chosen to assess the buckling risk. Following the guidelines of the GL standard, a buckling safety factor of 2.76 was obtained. Considering the five load cases reported in Table 2, each of the sections was loaded with its corresponding sectional loads, as well as all the loads transported from the remaining regions from the tip. The bottom part of each section was kinematically coupled to a reference point and clamped. It is worth mentioning that centrifugal forces were not considered in the present analysis meaning that the results are even safer (centrifugal forces reduce the compressive

load on each section). All buckling eigenvalues exceed the relative safety factor thus the design is considered to be safe.

In addition, the tip deflections were evaluated for each load case, to check for blade-tower clearance. The worst-case deflection is 0.16 m, which is just slightly greater than the hub diameter, and lower than the minimum rotor-tower clearance of 0.47 m. Once the material layout was defined, an eigenvalue analysis in Abaqus allowed to compute the natural frequencies and mode shapes of the blade and to estimate the inertial properties, namely the center of mass and the moments of inertia. The first flap-wise natural frequency is about double the Froude-scaled target, which is 2.36 Hz. Since the blade is very short and has a thin cross-section required by aerodynamic constraints, it is very difficult to simultaneously achieve the correct mass and natural frequencies. The result is considered satisfactory, because the mass is really close to the target value and a stiffer blade is safer since it is subjected to smaller deflections and fatigue loads.

NACELLE PROJECT

The nacelle design was performed in accordance with Froude scaling approach as far as it concerns the overall dimensions and the global mass. On the other hand, the internal components were properly designed as for a real wind turbine in order to satisfy their functionality requirements. In particular, as highlighted in the previous chapter, the nacelle design was realized in order to allow a yaw rotation around the tower axis. For this reason, the nacelle is fixed to the external ring of the tower-top slew ring bearing. This is done by means of welded steel plates forming a box on which another geared slew ring bearing is installed and on which the rotor is fixed (Figure 11). The wind turbine generator is emulated by a motor-reducer unit. A gear is installed on its output shaft and meshes with the one fixed to the slew ring bearing. Motors and transmissions were chosen in consideration of the estimated loads and the requested performances.

Figure 11. Nacelle design.

A slip ring is positioned along the bearing axis and fixed to the nacelle, as shown in Figure 11. The slip ring is required to feed power to and exchange control signals with the rotating

instruments housed on-board the rotor (pitch motors, measurements).

Figure 12. Rotor design.

The main rotor hub element is a steel plate on which blade units and drivers for electrical motors are fixed, as shown in Figure 12. The blade units are mounted onto the rotor base plate by means of a custom-made aluminium component on which a motor-reducer with harmonic drive is fixed. The blade is directly installed to the slow shaft of the motor-reducer by means of a flange so that it can perform a complete rotation around the pitch axis.

Figure 13. Nacelle and rotor design.

The rotor hub is completed by a frontal carbon fiber plate that increases the structure stiffness and consolidates the blade units. The nacelle and rotor design will be finalized adding an aerodynamic fairing to shield the electrical components inside. The final design is reported in Figure 13.

CONTROL STRATEGIES

The wind turbine model will be equipped with a control and monitoring system, actuators and sensors in order to ensure autonomous and continuous operation. The outdoor-scale prototype mechatronic setup is based on five actuators: a main shaft motor, to control the rotor angular speed, three dedicated motors, allowing to control in real-time the individual pitch angle of each blade and a yaw actuator. The model will also be equipped with sensors including the main shaft encoder used as primary feedback for the wind turbine power controller. An embedded system is capable to control the actuators and acquire data from sensors simultaneously. The wind turbine control logics is based on the standard control strategy used by modern wind turbines to regulate power production and rotor speed through the machine operating range (variable-speed variable-pitch, VS-VP). In this scheme, the turbine is programmed to operate at variable-speed and fixed-pitch in below-rated winds, to maximize power extraction, and at variable pitch in above-rated winds, to regulate rotor speed and power. The VS-VP control scheme is shown in Figure 14. The control system is characterized by 3 different regions of operation: Region 1, for wind speeds up to cut-in, used for the start-up of the wind turbine; Region 2, for partial-load operations, extends from cut-in to the rated wind speed: in this region the blade pitch is fixed at its minimum and the turbine is regulated at variable speed through the torque controller; Region 3 extends from the rated wind speed to cut-off and the turbine works at full-load: in this region the generator torque (or power according to control goals) is set at the rated value and the turbine rotational speed is regulated by the blade pitch-to-feather PI controller.

Figure 14. VS-VP controller scheme.

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