On the damage detection of a laboratory scale model of a tripod supporting structure by vibration-based methods

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Abstract.

The paper presents a comparison of two vibration-based methods for the damage detection of a laboratory scale model of a tripod. Tripods are a part of the supporting structures for offshore wind turbines. The tripod model structure allows the investigation of the propagation of a circumferential representative crack in one of the cylindrical upper braces of the tripod itself. The first damage detection method addresses the use of acceleration signals in a genuine experimental modal analysis (i.e. input-output modal analysis) while the second one is based on operational modal analysis (i.e. output only modal analysis). The progressive damage is monitored by the calculation of the modal parameters and following their deviations. Both methods were performed on the undamaged and damaged structure for different support conditions and excitations (shaker, hammer, in water basin under wave excitation). The results suggest that both the methods can be considered useful tools for damage detection in dry and in-water conditions for offshore support structures. The presented technique proves to be effective for detecting and assessing the presence of representative cracks.

Keywords: offshore wind turbine, Support structures, Damage detection, Experimental modal analysis

1. Introduction

In the last few years, the use of renewable energy sources has strongly increased, with the aim of reducing the oil/carbon utilization and improving the life quality in terms of environmental and air pollution. Nowadays, the offshore wind energy can be considered a very important renewable energy source. In many scenarios, it is foreseen as *The Future* among all possible renewable energy sources. The offshore wind industry is currently facing the challenge of reducing costs for fabrication (direct) and installation (indirect) of jacket and tripod like support structures. Very important are research activities oriented on the reduction of operational and maintenance costs [1-3], which are much higher compared to the onshore installations. This is the scenario in which the following activity takes place. Offshore support structures installed in harsh sea environment have to withstand extreme and fatigue loads in the form of vibrations from wind and waves, as well as from wind turbines operation. Remote monitoring [3] can optimize the number of inspections and repairs and substantially reduce the related operation and maintenance costs and thus the cost of wind energy. Nowadays, operational and failure data availability allows for the predictive maintenance; thus, the development of condition monitoring systems is a high priority in the research agendas for the development of the future large size turbines [4,5]. Vibration-based damage detection and condition monitoring methods [6] analyze the vibration spectra of the structure [7,8]. Component deterioration affects the stiffness properties and modifies the natural frequencies of the structure [9]. Different types of sensors are used to monitor the support substructure: strain gauges, optical fiber sensors based on strain measurements [10], vibration inclination, displacement and vibration sensors [11]. These authors have already worked on this field, defining a first approach focused on operational modal analysis aspects [12].

The main goal of the presented research is the development of a vibration-based procedure based on modal analysis for the identification of structural failures in a laboratory scale model of an offshore turbine: it has to be considered a first insight in order to develop an automated procedure for damage detection in operational conditions. Further research will be carried out to continue this activity in order to develop an autonomous system based on vibration experimental measurements as well as numerical models for the real time monitoring and diagnostics of the structural integrity of offshore wind turbines. In this context, the autonomous system can strongly reduce the maintenance costs (e.g. inspection costs) as well as reduction of the collapse risk.

Experimental vibration tests have been carried out on a laboratory scale model. The model was developed within a research project aimed to evaluate the feasibility of different types of support structures (i.e. monopile, gravity-based, tripod) for offshore wind turbines to be placed in the Polish Baltic sea. The main goal of the project was to select the best type of structure with respect to the soil conditions. For this purpose, the models were placed on a polypropylene rotary table with a 6 DOFs torque sensor, in order to identify the forces acting on the soil. Regarding the downscaling of the models, it was computed in accordance with the capacity of the wave tank and its maximum wave height achievable. One of the aspects investigated within the mentioned project (and presented in this paper) is the feasibility study of a vibration based method to monitor the structural conditions of the support structure during its operating life. The tripod support model was the only one suitable for this particular aspect of the project, as it is entirely made of metal (aluminum), while the monopile and the gravity based model are made with a combination of wood, foam and metal. It has to be underlined, since there is no elastic similarity between the tripod model and the real support structure, that the following study has to be considered as a feasibility analysis.

A first experimental campaign regarded an experimental modal analysis of the tripod tested in-air, while a second experimental campaign addressed measurements in-water with the tripod model assembled in a towing tank where different operational conditions in terms of wave motion and direction and rotational blade speed can be simulated. The presence of a crack in the support structure (the upper central joint of the tripod support structure happens to be the most critical point for fatigue design) should induce an effect on its natural frequencies and modal damping that remains the same despite the changing of operational conditions due to different kind, intensity and direction of the wave motion on the sea surface. The original aspect of this paper regards the investigation on the consistency of the operational modal analysis technique in the damage detection of offshore wind turbine support structures, considering different kind of possible wave motions. The applicability of operational modal analysis (OMA) to offshore wind turbine structures is based on several assumptions that must be fulfilled both by the structure and the excitation conditions in order to obtain meaningful results. The structure must be linear and time invariant, while the excitation has to be random, broad banded and uncorrelated (white noise assumption). One of the main issues in the applicability of OMA to offshore wind turbines is that an offshore wind turbine (OWT) is not a linear and time invariant system, due to non-linearity effects produced by the aeroelastic effects and hydrodynamic effects/soil interaction. In particular, the rotation of the turbine has a considerable effect on the dynamic behavior of the wind turbine system (tower and nacelle): significant resonant phenomena of the wind turbine system are observed when the structural frequencies agree with the frequency resulting from the mass unbalance of blades and the harmonic frequencies of the rotating speed [13]. Also, the excitation of the structure performed by the wind and wave motion of the sea surface cannot be considered as white noise (large part of the energy is contained in the spectrum below 0.2 Hz) [14]. Furthermore, it has to be underlined that the natural frequencies of the observed object in marine environment might be affected by more factors like changes in the soil stiffness, scour, corrosion, marine growth. The presented approach can be considered a first attempt to tackle this fascinating field. Additional measures as well as numerical models [15,16] have to be used to identify the location of the damage and the corresponding reason.

The paper is organized as follows: Section 2 describes the mechanical system under test; Section 3 is devoted to the acceleration-based measurements and results both regarding the dry condition as well the in-water tests; Section 4 draws the concluding remarks.

2. Object of the investigation

The object of investigation is a laboratory scale model of a tripod type supporting structure of an offshore wind turbine [17,18]. It is made of aluminum cylindrical tubes. The model is about 2 m high and a mass of 30 kg (Figure 1). It comprises of three pile guides fixed to the central column with upper and lower braces. In one of the three upper braces, a flange is placed to interrupt the structural continuity (Figure 1, right). The screws in the flange can be closed with different tightening torques in order to simulate different types of circumferential representative crack in the cylindrical brace. In this study the flange was a practical measure to introduce the brace stiffness reduction. The model used for the test cannot be compared to a real tripod due to the lack of elastic similarity between the real structure and the model itself; also this paper does not address the investigation of a real system with a real circumferential crack behaviour with regard to the bolted flange connection, obtained for example after a fatigue test. Anyway, the excitation provided by shaker and hammer has an adequate frequency content to properly excite the modes of the model used in the test. The present work has to be considered as a feasibility analysis and a first attempt to assess the possibility to use modal analysis in the crack detection of OWT support structures. The ability to reduce the stiffness of the flange and its influence on the member natural frequency was available through five screws present in the flange. Typically, in wind turbine support structures, the strength/ safety analysis are concerned on joints [19,20], where stress concentration results in a fatigue damage occurrence is the most possible. Although damage of theoretically less important elements like braces also can result in catastrophe as the Alexander L. Kielland accident, where a crack developing in a circumferential direction of one brace was the cause of the offshore platform collapse [21]. It was a motivation of location and type of the damage taken into consideration in a tripod model. A representative crack [22] was introduced by means of pretention of the flange screws, namely Top Screw (TS), Right Screw (RS), Left Screw (LS), Right Bottom Screw (RBS), Left Bottom Screw (LBS).

Moreover, five different extents of circumferential representative cracks have been considered, namely "All Screws Open" (ASO), "Full Open 1" (FO1), "Partial Open 2" (PO2), "Partial Open 3" (PO3), "Full Close" (FC). Table 1 lists the tightening torques for each screw in the different configurations being tested. As an example, "Partial Open 3" configuration means that TS has a tightening torque equal to 13.6 [Nm], RS and LS have a tightening torque equal to 27.1 [Nm] and RBS and LBS have a tightening torque equal to 54.2[Nm], which corresponds to the nominal value (NOM). Note that in the ASO configuration, the bolts were still present in the flange but they do not contribute to any bending moment in the flange.

3. Acceleration-based damage detection

Experimental study of the tripod encompassed two main test configurations. The first configuration addresses the genuine experimental modal analysis of the tripod model in dry conditions by using electrodynamic shaker as input excitation and piezoelectric accelerometer in order to measure the acceleration response; the second test configuration regards the operational modal analysis of the tripod incorporated into the entire offshore wind turbine model in the in-water condition in order to reach boundary conditions closer to the real ones.

3.1. Experimental modal analysis in dry conditions

For the experimental modal analysis, the excitation was provided by two electrodynamic shakers that excite the tripod structure in the base of the central column, as shown in Figure 1, left. The two shakers excite the tripod structure along two orthogonal directions. An example of shaker excitation measured by a piezoelectric force sensor is depicted in Figure 2. The response of the tripod was measured by five piezoelectric tri-axial accelerometers. The set of transducers was placed on the particular measurement points and then moved to another until a full coverage of all 76 measurement points (Figure 3) was reached. Measurement points were located near the flange and spread over the structure. Two sections of four measurement points each were defined on the circumference of the brace below the flange and the eight measurement points located above the flange. Each one of the three pile guides and the upper and lower braces were measured in five locations. The tower was instrumented with 12 measurement points. For the in-water test only the measurement points located on the tower (see orange dots in Figure 3) were used due to the practical limitation of moving the sensors. Both excitations and responses have been measured simultaneously to obtain the Inertance, i.e. the Frequency Response Function (FRF) between acceleration and force. The signals were acquired by using sampling frequency and frequency resolution according with the type of model and with the kind of constrain condition. For each of the five different tightening torque configurations (Table 1) the experimental modal parameters have been estimated.

Once the experimental modal tests and analyses have been performed, natural frequencies, modal damping and mode shapes are available for all modes in the frequency band of analysis [23]. The natural frequencies (fn) and modal damping values (ζ) were obtained by averaging the corresponding solution from the Least Square Complex Exponential (LSCE) [24] method and PolyMAX method [25]. In particular, two different modal analysis algorithms have been used in order to increase the robustness of the solution: the LSCE method, which works in the time domain and the frequency domain algorithm PolyMAX.

The results for the EMA are presented in Figure 4 in terms of a FRF-sum, i.e. for each crack configuration the complex sum of the FRF's between the response measurements and the excitation force is plotted. It can be noted that the modes in the frequency range between 0 and 150Hz (circled in red in Figure 4) remain the same for all the configurations (same natural frequency, same modal damping, same mode shape), while the peaks in the frequency range between 180 and 300Hz (green box in Figure 4), referring to mode #6, show a clear variation of natural frequencies and modal damping values for the considered crack configurations. Such a mode shape has different modal damping and natural frequency. This behavior is due to the different extent of representative crack because of the different flange connection stiffness. Figure 5 and Figure 6 present the natural frequencies and modal damping ratio for the different configurations under tests, referring to mode #6. As mentioned, the natural frequency and modal damping values of mode #6 change significantly; In fact, the natural frequency increases from 181 Hz to 295 Hz, with screw configuration that changes from ASO to FC, respectively. This is particularly interesting, because it is expected that the frequency increases while the stiffness of the system increases due to higher tightening torque of screws. The observed change in the natural frequencies indicates that a certain change or damage to the structure did occur, which in this case is the tightening configuration of the flange simulating different crack states. Regarding the damping variation, Figure 6 shows a not monotonic behavior probably due to a variation of the friction mechanisms in the bolt connection. It is interesting to note that the frequency and modal damping variation involves only mode #6; this is due to the particular shape of this mode, which involves mainly the deformation of the brace connection and thus more related to stiffness connection. The confirmation of this mode shift can be obtained considering the values reported in Table 2: in this table, the MAC (Modal Assurance Criterion) values between the 6th mode obtained in FC, ASO, F01, P02, P03 conditions are reported [24]. The MAC value is a dimensionless number resulting from the comparison of two eigenvectors (each one related to a mode shape). Its value goes from a minimum of zero to a maximum of one. A MAC value between 0.9 and 1 indicates that the considered eigenvectors (e.g. the mode shapes) are linearly dependent (e.g. very similar mode shapes), while a MAC value between 0 and 0.1 stands for linearly independent eigenvectors, which means that the considered modes have completely different shapes. In Table 2, the mode shape obtained for the 6th mode in a certain screws tightening configuration is compared with the other configurations. The mentioned table can be considered as a symmetric matrix: the values on the main diagonal are obviously 1, since the mode shape is compared to itself in the same configuration. The high MAC values between all the configurations confirm that, despite mode #6 presents a significant change in terms of frequency (from 295,4 Hz in FC condition to 181,4 Hz in ASO condition), its mode shape remains essentially the same. The results show that the experimental modal analysis can be considered an effective tool for monitoring changes in the natural frequencies of a model scale tripod structure subjected to different artificial crack configurations. It has to be specified that the experimental modal analysis based on shakers and accelerometers has been performed in order to assess the effectiveness of this technique for the damage detection of a component. The experimental modal analysis cannot be considered a suitable candidate for an online monitor of a real WT, since in this technique the input excitation has to be measured; for this reason, the excitation has been usually given by hammers or shakers for which the measure of the input force can be easily performed.

3.2 Non-linear effect verification

It should be recalled that one of the fundamental hypothesis, upon which the experimental modal analysis is based, is the linear dynamic behavior of the structure [24]. Each modal analysis should start with a check of linearity of the structural dynamic behavior. In order to identify and qualify non- linear dynamic behavior different test procedures have been developed: the harmonic detection technique, the Hilbert transform, the damping plot and the direct time stepping method are typical examples of such techniques. Furthermore, comparing frequency response functions obtained using different excitation force levels can be used as a check for non-linearity. If the structure behaves linearly these frequency response functions are independent on the input of force level. If the structure under test shows non-linear behavior, the excitation becomes very important, since the measured frequency response functions will depend on the nature and the level of this excitation signal. So, the study of non-linear effects is a fundamental step in validating the results presented in the previous section. The tripod structure has a representative crack in the upper brace, as explained, since the flange with the screws interrupts the continuity of the structure. However, the structure could show linear behavior in a certain frequency or force range while showing non-linear effects in other frequency or force ranges. Hereafter, the linear behavior will be verified by exciting the structure with different excitation levels and by checking the variation in terms of natural frequency and modal damping ratio.

Figure 7 presents the modal damping ratios for different levels of shaker excitation (0.5, 1, 1.5, 2 voltage excitation) for the configurations ASO and FO1, respectively. It can be noted that the damping values corresponding to mode #1 and 2 significantly change, while the damping values related to mode #3,4,5,6 remain approximately constant. A same trend is visible in Figure 8, which presents the natural frequencies for the ASO and FO1 case, respectively. Therefore, the tripod structure exhibits nonlinear behavior, but only in the low frequency range, where the dynamic behavior is governed by mode #1 and #2. In the medium-high frequency range, nonlinear effects are not present. Therefore, the results presented in the previous subsection address a linear structural behavior as the main hypothesis of modal analysis.

3.3 Operational modal analysis in the wave tank

The second part of the experimental campaign was performed in the auxiliary towing tank of the Ship Design and Research Centre in Gdansk (Poland). This tank has dimensions of 55.0 m x 7.0 m x 0.2-3.0 m and it is equipped with an irregular wave generator, capable of generating regular and irregular waves with the aim to reproduce the variety of operational conditions that interest the turbine in its operating life. In particular, the generator is capable to simulate irregular waves up to the 8^{th} degree of the Douglas sea scale (the sea state at model scale was achieved using Froude scaling) and regular waves of a height of up to 0.5 m at a length of up to 7.0 m or of a height of 0.18 m and length of up to 14 m. The tripod was mounted on a thick polypropylene turntable such that the orientation of the structure could be varied with respect to the direction of incoming waves. The wave excitation has a very low frequency content, but the environmental broadband noise at high frequencies is anyway sufficient to excite the modes of the structure. On the top of the tripod, a cylindrical tower section and a three bladed rotor were assembled, as presented in Figure 9. Different operational conditions were simulated, at first with the rotor not in operation, by hitting the tripod model with simulated waves (Figure 11), each time changing the kind of wave motion and the orientation of the damaged brace with respect to the wave motion direction (0, 90 and 180 degrees) as presented in Figure 12. All data acquisitions were performed in two different conditions: first on the intact structure (FC or 0-crack condition) then with the structure affected by a simulated circumferential crack in one of the lateral braces (ASO or 1-crack condition) in order to assess and verify the feasibility of the damage detection based on the mode shift method. At last, all the tests have been repeated with the rotor blades in operating conditions at 120 rpm: as mentioned in Section 1, the presence of the rotational motion of the rotor complicates the analysis, as it introduces harmonic excitation frequencies. For each testing condition, operational data were acquired by bi-axial piezoelectric waterproof accelerometers installed on the tripod model in five points equally spaced along its central column, visible in Figure 9 and Figure 3 (Node 1, 4, 7, 10, 13). Time data signals of the structure's response were then processed for the estimation of the modal parameters of the tripod model. Figure 10 depicts an example of time data signal acquired on the tripod structure during an OMA test session. Irregular JONSWAP [26] spectrum waves and regular waves (non-JONSWAP) were used for the excitation of the structure: overall, the characteristics of wave patterns simulated in the towing tank in terms of frequency and amplitude are reported in Table 3. As an example, the time and frequency domain characteristics of the irregular 1-year storm wave are presented in Figure 13. Figure 14 depicts the power spectral density (PSD) of all the wave motions used in the towing tank to perform the test. It is possible to see that only a narrow frequency band (till 15 Hz) results to be properly excited: this is due to the physical characteristics of the wave motion itself, which cannot take place at higher frequencies.

Before the operational modal analysis test, an experimental modal analysis (EMA) was performed by using the roving hammer method on the intact tripod model in the calm water (no wave motion), in order to individuate the modal parameters of the structure, reported in Table 4, to be used as a reference. In this experimental modal analysis, both the input excitation (force) and the output response (acceleration) were measured simultaneously enabling the estimation of the FRFs, i.e. the ratio between acceleration and force. Figure 15 depicts an example of hammer input force, in terms of PSD.

Overall, eleven different operational modal analysis test conditions were arranged, combining the wave patterns listed in Table 3 with the three possible orientations of the tripod visible in Figure 12. The tests have been carried out firstly with the rotor blades not in operation, then with the rotor in rotation at 120 rpm. For each test condition, data were acquired both on the intact structure (0-crack condition or FC) and then on the damaged structure (ASO or 1-crack condition), in order to compare the results and highlight the differences. Table 5 reports all the configurations arranged for the test: starting from left to right, the first column indicates the wave pattern used, the second regards the tripod orientation and the third indicates the structure condition (0-crack or 1-crack condition). Due to the bad quality of some acquired signals (which were highly affected by noise), the 0-crack data of the 90 and 180 degrees' orientation tests were not accounted. Therefore, for all these cases, the comparison between 0-crack and 1-crack condition was made by using the 0-crack at 0 degrees' orientation as a reference representing the behavior of the intact structure.

Table 6 depicts the comparison between modes estimated by the experimental modal analysis (EMA) with hammer excitation in calm water and modes obtained by OMA with RW2 excitation wave pattern in 0 degrees' position (taken as example) with the rotor not in operation (center column) and rotating conditions at 120 rpm (right column). The three mode sets are referred to the intact structure (flange joint completely closed). Obviously, only some of the natural frequencies estimated by the EMA have been individuated also by the Operational Modal Analysis: this is due to the particular excitation conditions in the towing tank, which do not provide a sufficient level of energy to excite properly all the modes of the structure in the frequency range from 0 to 100 Hz, as visible from the PSD plot in Figure 14. Also, the presence of the rotational motion of the rotor introduces a considerable disturbance on the analysis procedure, due to the presence of the harmonic excitation frequencies. For this reason, the OMA has been processed till about 40 Hz, where the wave motion excitation was enough. Therefore, all the modes of the tripod model above 50 Hz will not be adequately excited by the operational conditions simulated in the towing tank: regarding operational modal analysis indeed, only the modes from the 1st to 8th are effectively observable. The data acquired with the rotor in rotation at 120 rpm have been processed with a harmonic removal filter in order to remove the harmonic excitation frequencies from the acquired signals. As an example, Figure 17 depicts the cross-spectra of the original signal referring to the accelerometer located in Node 1, direction X, and the filtered one. Each one of the five piezoelectric bi-axial accelerometers installed on the structure provides two different time signals, which are correlated to independent directions X and Y, as visible in Figure 3 (the X axis is aligned with the flanged brace of the tripod model). Therefore, a total number of ten output time-signals are available for each measurement session: one or more of these signals have to be chosen as reference, in order to generate the cross-power function that will be used by the polyreference LSCE method to estimate modal parameters from operational data. The reference signal was chosen by comparing the auto-power spectra of all the acquired time signals in terms of clarity and highest signal-to-noise ratio: as a result of this comparison, X and Y signals related to node 10 in Figure 3 have been chosen as reference. Crosspower spectra have been created between each signal and the chosen reference signals (i.e. 10-X and 10-Y) and then the LSCE method has been applied to the experimental data. To evaluate the effect of the simulated crack on the modal parameters of the structure, the cross-spectra computed in the 0-crack (FC) and 1-crack (ASO) conditions have been compared for each test configuration.

Table 7 shows the comparison of the natural frequencies in the five cases of interest: on the left hand side of the table, the natural frequencies of the intact structure (FC or 0-crack condition) are reported, comparing the EMA results obtained with roving hammer excitation in calm water, and OMA results obtained with the RW2 wave pattern excitation (which was taken as reference case for the intact structure): a good correlation can be noticed between the listed frequencies. On the right hand side of the table, the natural frequencies yielded from OMA on the damaged structure (ASO or 1-crack condition) are shown: each column represents a test condition (combination of orientation and wave pattern). The five cases in the table are considered the most representative in terms of results: in all these cases, the frequency shift of the 6^{th} mode (from 29 Hz in the 0-crack condition to 25 Hz in the 1-crack condition) appears to be consistent and clearly observable. As mentioned before, the remaining test configurations did not permit to obtain significant results in terms of quality of the poles selected in the LSCE method. Figure 16 depicts the cross-spectra of the five remarkable cases, overlapped in a single graph.

Thus, the five cases considered significant in terms of results confirm that by using different wave patterns (RW2, IRW1) and wave motion directions (0, 90 or 180 degrees), the same natural frequency decrease is observable in the structure, when passing from the intact to the “damaged” condition with the rotor not in operation. The frequency reduction is in accordance with the reduced stiffness of the tripod structure consequent to the presence of a simulated circumferential crack in one of the lateral braces. As observed with experimental modal analysis in dry conditions (Figure 4), not all the natural frequencies of the tripod are influenced by the presence of the simulated damage: in this particular case only the 6^{th} natural frequency results to be lower in the 1-crack (damaged) condition. This phenomenon is due to the modal shape of the natural frequency mentioned above, which involves the lateral brace of the structure where the simulated crack is present, characterized by a lower stiffness.

The results obtained from the tests with the three-bladed rotor rotating at 120 rpm are reported in Table 8. On the left side of the table, the results of EMA are compared with the results of OMA performed on the structure in calm water with the operating rotor ; on the right side of the table, the natural frequencies obtained from the OMA on the damaged structure are reported for the five cases of interest. The combinations of wave motion and tripod orientation are the same considered for the rotor not in operation, except for the 90 deg-IRW1 configuration, which was not available and substituted by the 90 deg-IRW3 configuration. Even in these conditions, despite the complication introduced by the presence of the rotational motion of the rotor (which produces an excitation on the structure that is considerably higher in frequency with respect to the excitation of the wave motion), the shift of the 6^{th} mode from about 29 to 25 Hz remains visible. The cross-spectra of the five cases considered for the test with the presence of the rotational motion of the rotor are overlapped in Figure 18. The positive results obtained show that a significant and effective phenomenon occurs when a structural damage affects the tripod: in some cases this phenomenon is sufficiently clear and observable to define a criterion that could be used for an in-operation damage detection system. The presence of the rotational motion of the rotor creates a complication that is avoidable using suitable tools to remove the harmonic excitation frequencies from the acquired signals. Further tests could be done considering different positions of accelerometers: this could permit the discovery of the configuration that highlights in the best way the variations in the modal parameters of the structure when a damage is present.

4 Concluding remarks

In this work, the effectiveness of the experimental modal analysis and operational modal analysis is studied with the aim of investigating the structural integrity of a tripod supporting structure of an offshore wind turbine. In particular, several experimental tests have been carried out on a laboratory scale model of the tripod type supporting structure of an offshore wind turbine. In one of the three upper braces of the tripod, a flange is placed to interrupt the structure continuity in order to simulate a crack. Experimental tests address the tripod structure in dry conditions and incorporated into the entire offshore wind turbine model in the in-water condition under wave excitation.

The major pros of the modal analysis approach consist in the effectiveness of the method, which gives consistent results either with EMA and with OMA; and its applicability in different operational conditions. On the other side, the cons are the low frequency content available for the excitation of the structure in operational conditions (the simulated wave motion is characterized by a frequency content below 2 Hz, which cannot provide a suitable level of energy at higher frequencies, with a consequent difficulty in the post-processing of the acquired data); and the working conditions of accelerometers in an hypothetic real application: they would be installed on the structure several meters below the sea surface, at a considerable distance from the acquisition equipment (placed on the upper part of the turbine) with consequent need of signal amplification, hard to provide considering the particular ambient conditions.

However, it must be remarked that the first natural frequencies of the real-dimensions tripod structure would be considerably lower than what was observed in the laboratory model: in a real application therefore, they could be better excited by the wave motion.

Therefore, the genuine experimental modal analysis and the operational modal analysis can be an effective investigative tool in the identification of the propagation of cracks in structures that require high maintenance costs as offshore wind turbines, due to environmental conditions and the necessary presence of skilled people.

The presented investigation was not observing the dynamics of the rotary table with high damping values of the thick polypropylene rotary plate. To transfer the obtained results to the full scale structure a more in-depth analysis of the sea bottom model and experimental characteristics of the used support should be accounted for.

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TABLES

Table 1. Screws and tightening torque configuration. NOM represents the nominal tightening torque value (54.2 Nm).

	All screws open (ASO) *[Nm]*	*Full Open 1 (FO1)* [Nm]	*Partial Open 2 (PO2)* [Nm]	*Partial Open 3 (PO3)* [Nm]	*Full Close (FC)* [Nm]
Top Screw (TS)	*0*	*0*	*0*	*13.6*	*NOM*
Right Screw (RS)	*0*	*0*	*13.6*	*27.1*	*NOM*
Left Screw (LS)	*0*	*0*	*13.6*	*27.1*	*NOM*
Right Bottom Screw (RBS)	*0*	*NOM*	*NOM*	*NOM*	*NOM*
Left Bottom Screw (LBS)	*0*	*NOM*	*NOM*	*NOM*	*NOM*

Table 2. MAC values table between the 6th mode obtained in FC, ASO, FO1; PO2, PO3 conditions. The MAC value goes from a minimum of 0 (linearly independent eigenvectors, completely different mode shapes) to a maximum of 1 (linearly dependent eigenvectors, identical mode shapes).

	FC [295.5 Hz]	**ASO[181.5 Hz]**	**FO1[221.3 Hz]**	**PO2[246.2 Hz]**	**PO3[285.5 Hz]**
FC [295.5 Hz]	1	0.89	0.80	0.98	0.91
ASO[181.5 Hz]	0.89	1	0.88	0.93	0.98
FO1[221.3 Hz]	0.80	0.88	1	0.95	0.96
PO2[246.2 Hz]	0.98	0.93	0.95	1	0.95
PO3[285.5 Hz]	0.91	0.98	0.96	0.95	1

Table 3. Wave patterns simulated to reproduce operational conditions in the towing tank

Wave pattern	Frequency [Hz]	Amplitude [m]
Regular wave 1 (RW1)	0.5	0.2
Regular wave 2 (RW2)	1	0.1
Irregular wave 1 (IRW1)	0.5÷2	-
Irregular wave 3 (IRW3)	10 years storm	
White noise wave (WN wave)	50 years storm	

Table 4. Natural frequencies determined by EMA with hammer excitation in calm water

Mode	Frequency [Hz]	Damping[%]
1	2.85	0.25
2	8.46	0.71
3	15.06	0.95
4	20.68	1.09
5	21.67	0.94
6	30.70	1.14
7	39.63	0.79
8	43.11	1.43
9	46.78	1.61
10	49.42	2.06
11	53.43	1.55
12	60.67	2.58
13	71.48	0.86
14	78.99	1.15
15	96.25	1.52
16	104.30	1.82
17	109.91	1.87
18	119.30	1.67

Table 5. Test configurations used to perform the OMA in the towing tank; each configuration has been tested with the rotor not in operation and rotating at 120 rpm.

Wave pattern	Tripod orientation	Structure condition
RW2	0 deg	0-crack (FC)/1-crack (ASO)
IRW1	0 deg	0-crack (FC)/1-crack (ASO)
RW1	90 deg	1-crack (ASO)
RW2	90 deg	1-crack (ASO)
IRW1	90 deg	1-crack (ASO)
IRW3	90 deg	1-crack (ASO)
WN wave	90 deg	1-crack (ASO)
IRW1	180 deg	1-crack (ASO)
IRW3	180 deg	1-crack (ASO)
RW2	180 deg	1-crack (ASO)
WN wave	180 deg	1-crack (ASO)

Table 6. Comparison between modal parameters obtained from EMA with hammer excitation in calm water (left column) and modal parameters obtained from OMA with RW2 excitation in 0 degrees position with the rotor not in operation (center column) and rotating at 120 rpm (right column)

EMA of intact structure			OMA-0 deg-0 crack-RW2			OMA-0 deg-0 crack-RW2 rotating at 120 rpm		
Mode	**Frequency [Hz]**	**Damping [%]**	**Mode**	**Frequency [Hz]**	**Damping [%]**	**Mode**	**Frequency [Hz]**	**Damping [%]**
1	2.86	0.57	**1**	2,89	0.26	**1**	2.80	3.22
2	8.46	0.71	**2**	7.00	0.30	**2**	9.00	2.48
3	15.06	0.95						
4	20.68	1.09	**4**	20.31	1.17	**4**	19.95	2.1
5	21.67	0.94	**5**	21.03	0.87			
6	30.70	1.14	**6**	29.63	2.23	**6**	29.00	0.2
7	39.63	0.79				**7**	41.40	0.14
8	43.11	1.43	**8**	41.95	2.14	**8**	42.63	0.3

Table 7. List of the natural frequencies related to the 0-crack (FC) and 1-crack (ASO) conditions with rotor not in operation, in five cases reported as an example.

FC (0-crack)			ASO (1-crack) –rotor not in operation				
Mode	**Frequencies EMA [Hz]**	**Frequencies OMA [Hz]**	**0 deg RW2 [Hz]**	**0 deg IRW1 [Hz]**	**90 deg RW2 [Hz]**	**90 deg IRW1 [Hz]**	**180 deg IRW1 [Hz]**
1	2.86	2.89	2.70	2.71	2.72	2.85	2.70
2	8.46	7.01	-	-	-	-	-
3	15.06	-	-	-	-	-	-
4	20.68	20.31	20.28	20.12	20.74	20.32	20.06
5	21.67	21.03	-	-	-	-	-
6	30.70	29.63	25.50	25.47	25.64	25.00	25.72
7	39.63	-	-	-	-	-	-

Table 8. List of the natural frequencies related to the 0-crack (FC) and 1-crack (ASO) conditions with rotor rotating at 120 rpm, in five cases reported as an example.

FC (0-crack)			ASO (1-crack) rotating at 120 rpm				
Mode	**Frequencies EMA [Hz]**	**Frequencies OMA [Hz] rotating at 120 rpm**	**0 deg RW2 [Hz]**	**0 deg IRW1 [Hz]**	**90 deg RW2 [Hz]**	**90 deg IRW3 [Hz]**	**180 deg IRW1 [Hz]**
1	2.86	2.80	2.66	2.66	2.61	2.58	2.65
2	8.46	9.00	8.62	8.96	8.99	9.40	8.55
3	15.06	-	-	-	-	-	-
4	20.68	19.95	18.98	20.56	19.00	19.44	20.56
5	21.67	-	-	-	-	-	-
6	30.70	29.00	24.99	25.35	25.00	25.41	25.40
7	39.63	41.40	-	-	-	-	-

FIGURES

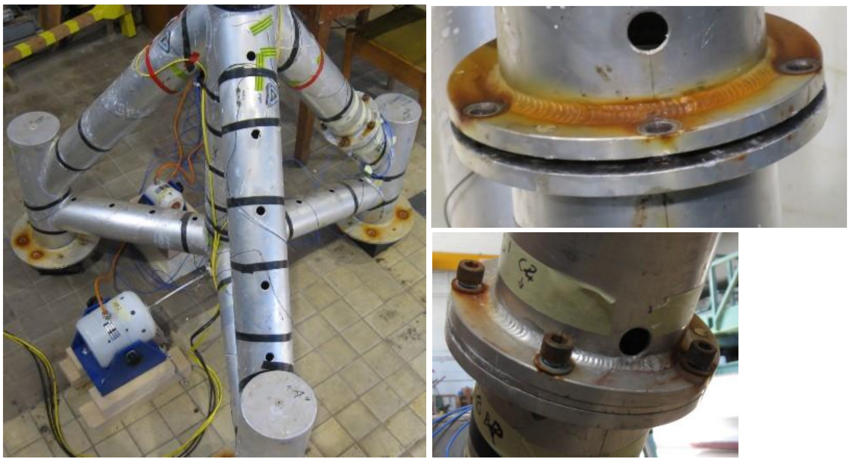

Figure 1. Laboratory scale model of the tripod supporting structure (left) and detail of the flange with model of the circumferential crack (lower right) and total failure (upper right) configurations of screwed flanges.

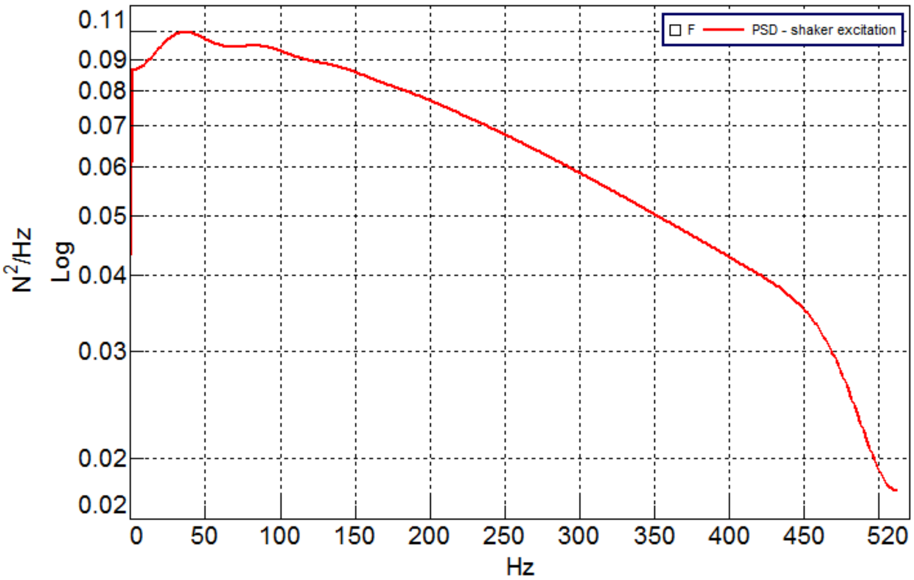

Figure 2. PSD (Power Spectral Density) of shaker excitation measured by a piezoelectric force sensor mounted on the stinger's tip of the shaker. The excitation frequency range is acceptable till 500Hz.

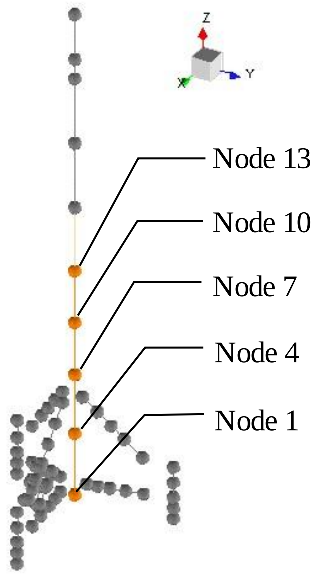

Figure 3. Wireframe model of the tripod: the grey and orange dots represent the positions of the measurement points used for the experimental modal analysis. In the operational modal analysis, only five measurement points were used, corresponding to the orange dots (Node 1,4,7,10,13).

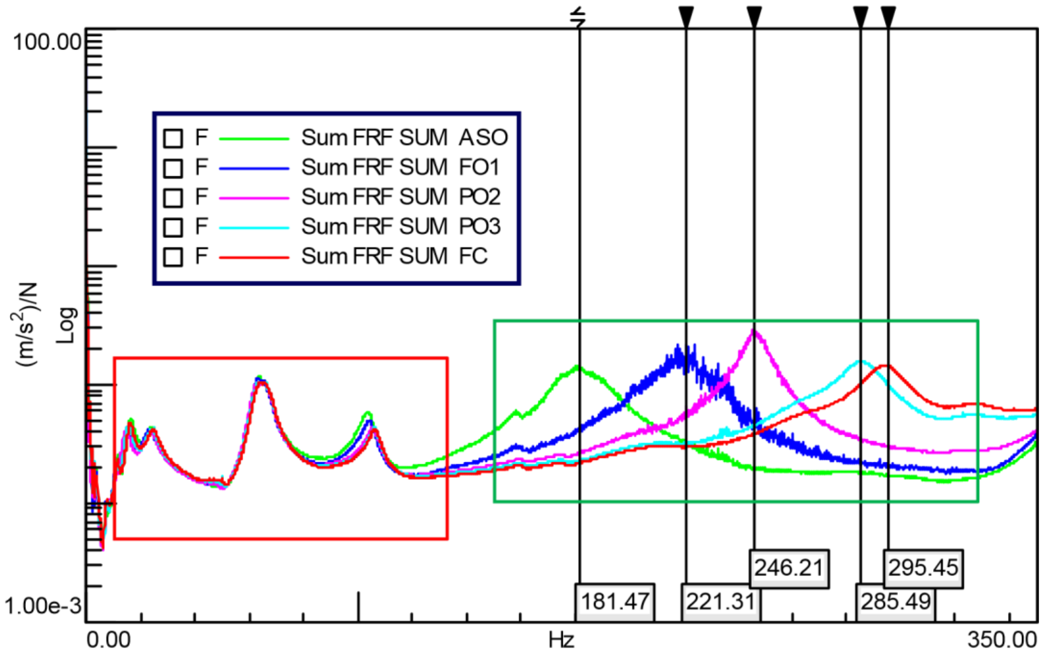

Figure 4. FRFs SUM. In green line, the FRF for the ASO condition; in blue line, the FRF for FO1 condition; in magenta line the FRF for the PO2 condition; in cyan line the FRF for the PO3 condition and in red line the FRF for the FC condition. The FRFs SUM are presented in the frequency range from 0 to 350 Hz.

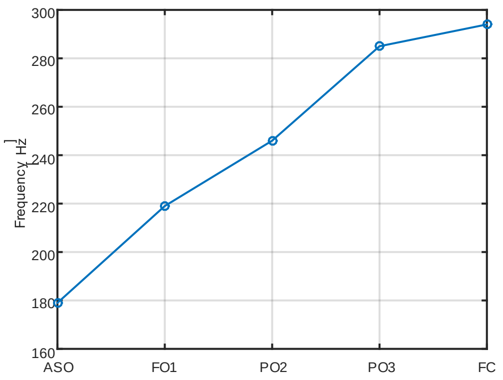

Figure 5. Natural frequencies for the sixth mode for the different kinds of tightening torque in the flange

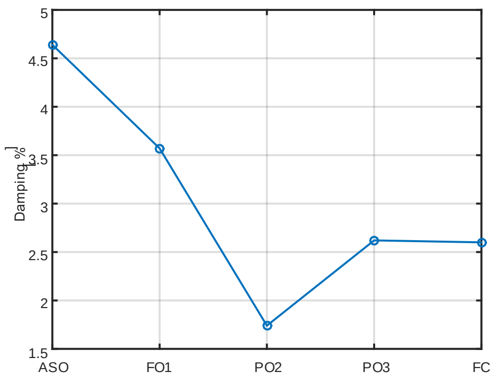

Figure 6. Modal damping for the sixth mode for the different kinds of tightening torque in the flange.

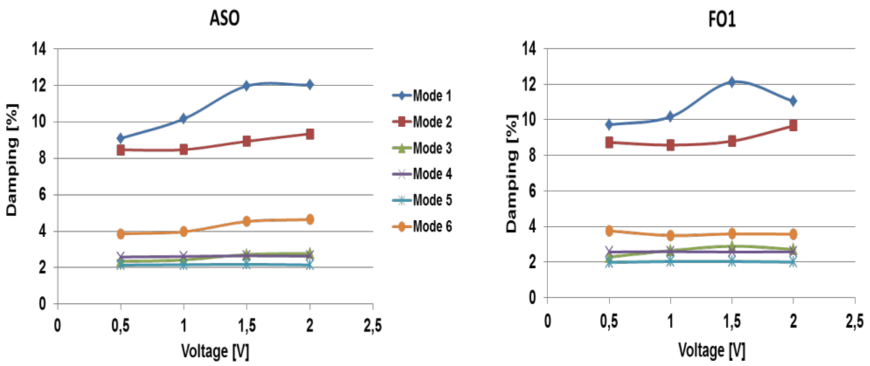

Figure 7. Modal damping variation for the different modes in the ASO and FO1 configuration as a function of the shaker excitation level

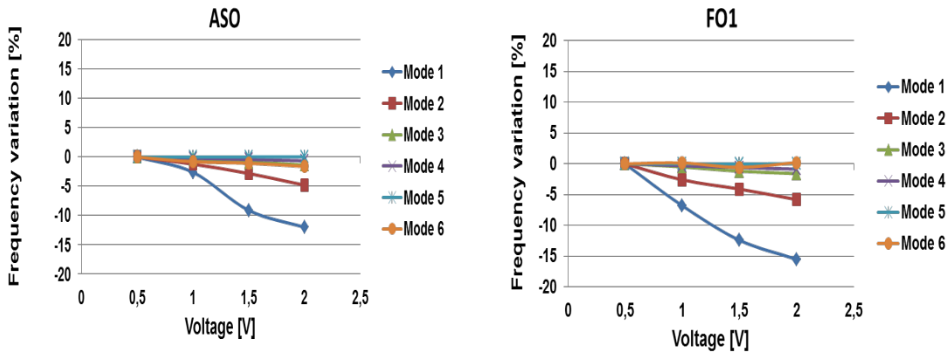

Figure 8. Frequency variation for the different modes in the ASO and FO1 configuration as a function of the shaker excitation level

Figure 9. Assembled test setup in the towing tank mounted on the rotary plate table.

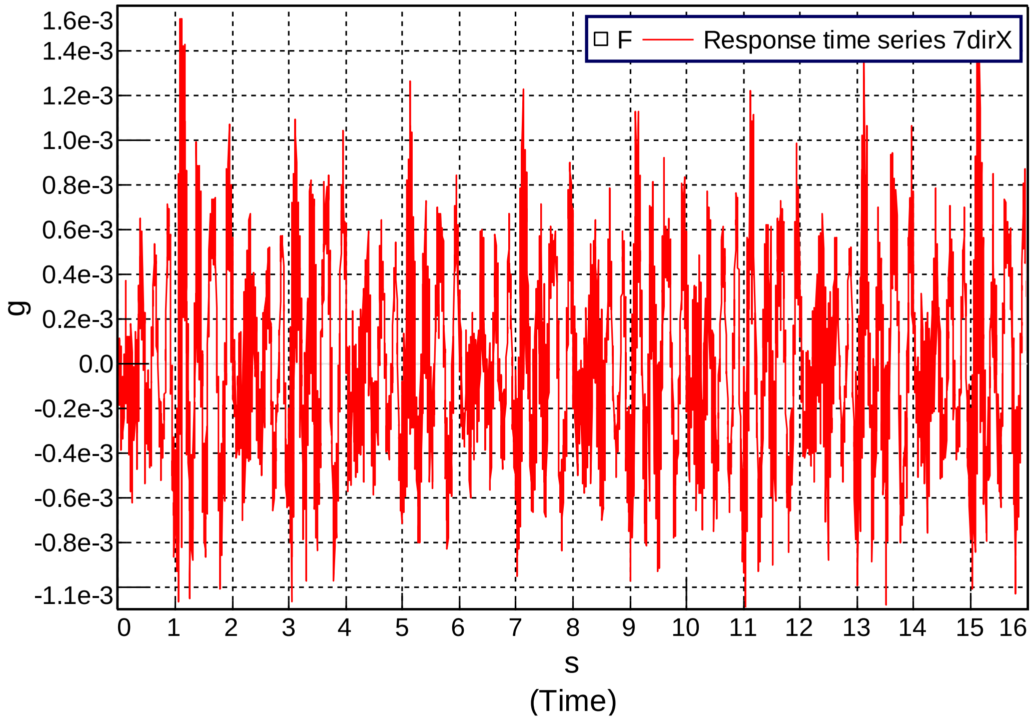

Figure 10. Example of an output time data signal coming from the accelerometer mounted in Node 7, direction X.

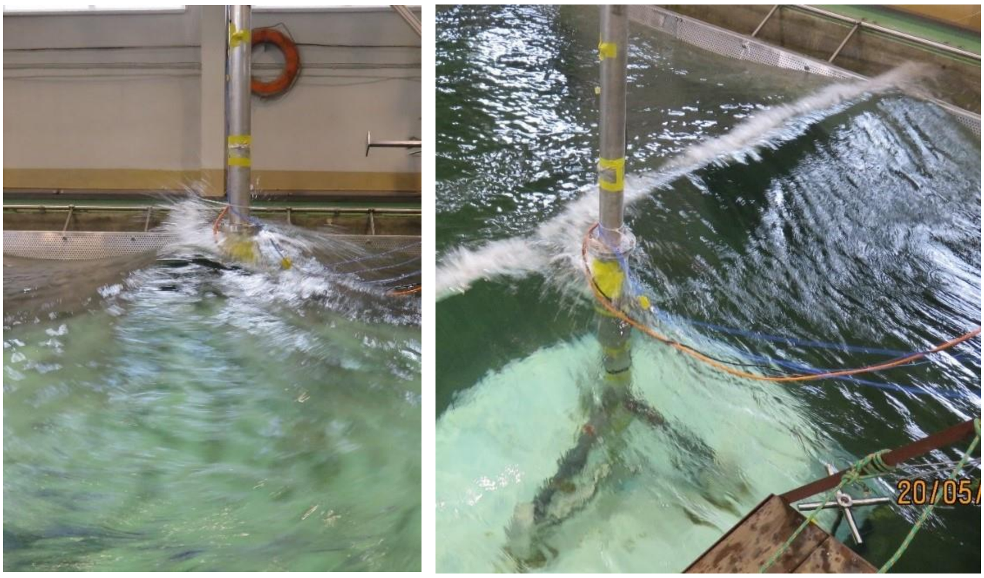

Figure 11. Output only measurement with the unknown force excitation from waves side view (left) and isometric view (right).

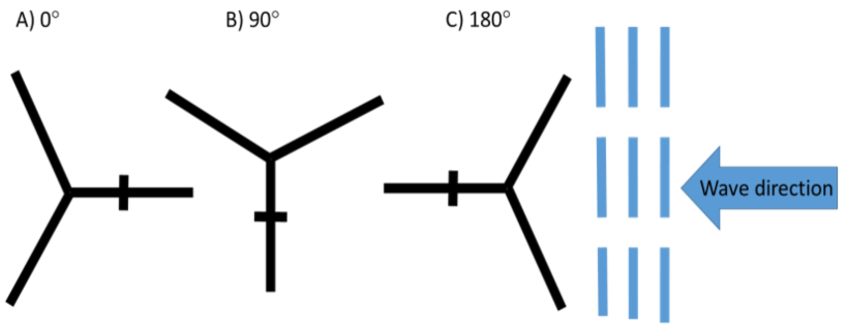

Figure 12. Top view of the three cases of exposure of the structure to the wave direction. A) 0 degrees of exposure: the cracked brace is parallel to the wave direction; B) 90 degrees of exposure: the cracked brace is orthogonal to the wave direction; C) 180 degrees of exposure: the cracked brace is parallel to the eave direction but behind the structure

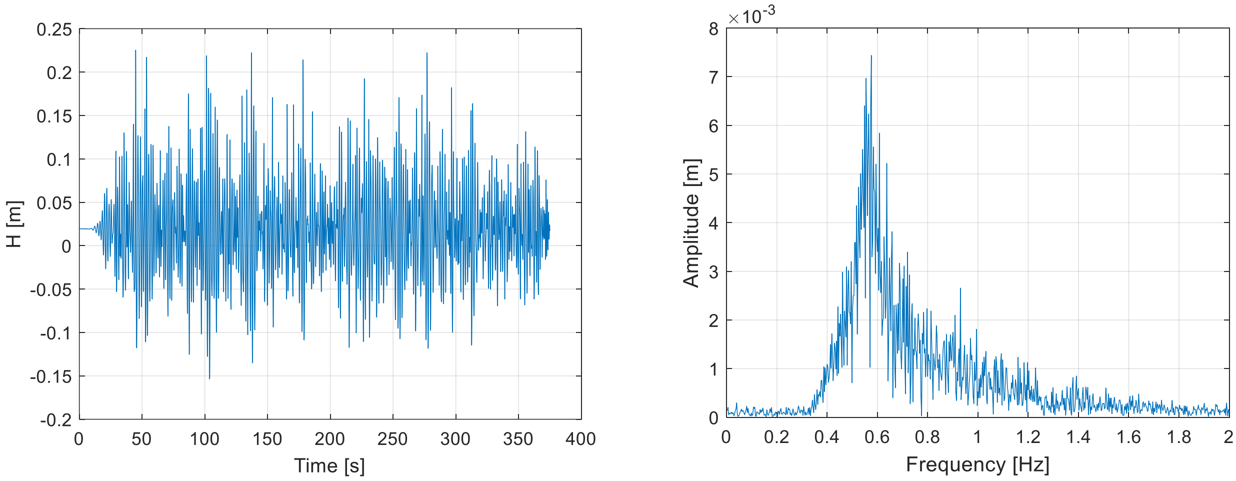

Figure 13. Characteristics of the irregular 1-year storm wave excitation used as input force to perform OMA analysis in the towing tank: time series (left) and spectrum (right).

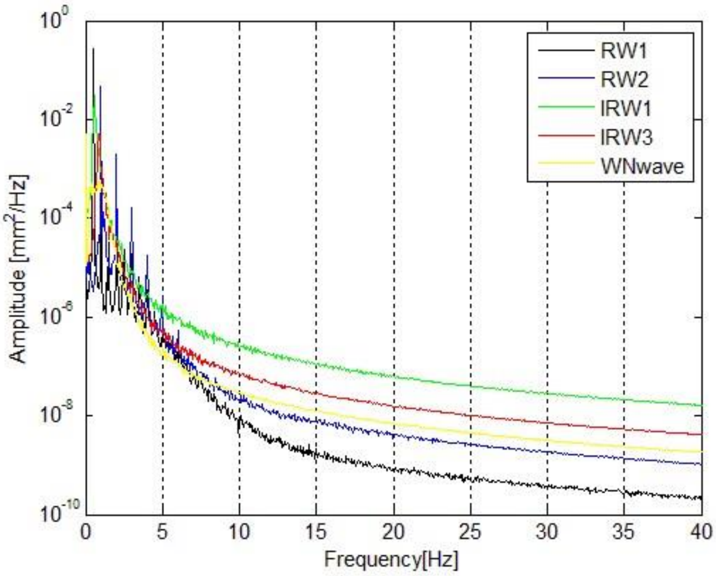

Figure 14. Power spectral density (PSD) of the wave motion used to excite the structure: in black line the RW1 motion; in blue line the RW2 motion; in green line the IRW1 motion; in red line the IRW3 motion and in yellow line the WNwave motion.

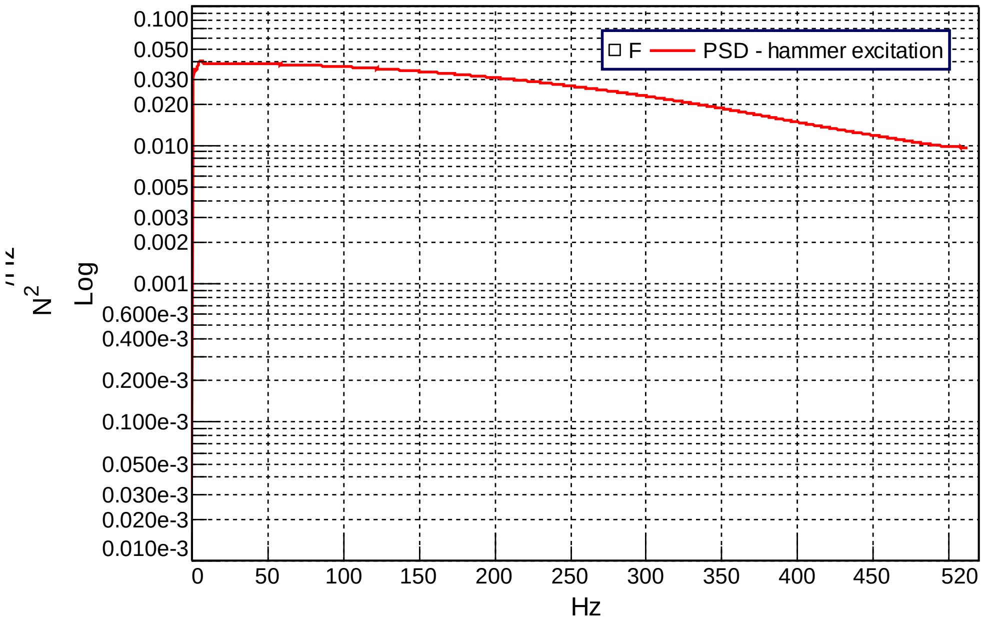

Figure 15.Example of PSD of hammer input force used to perform experimental modal analysis in the calm water. The excited frequency range is acceptable till 500Hz.

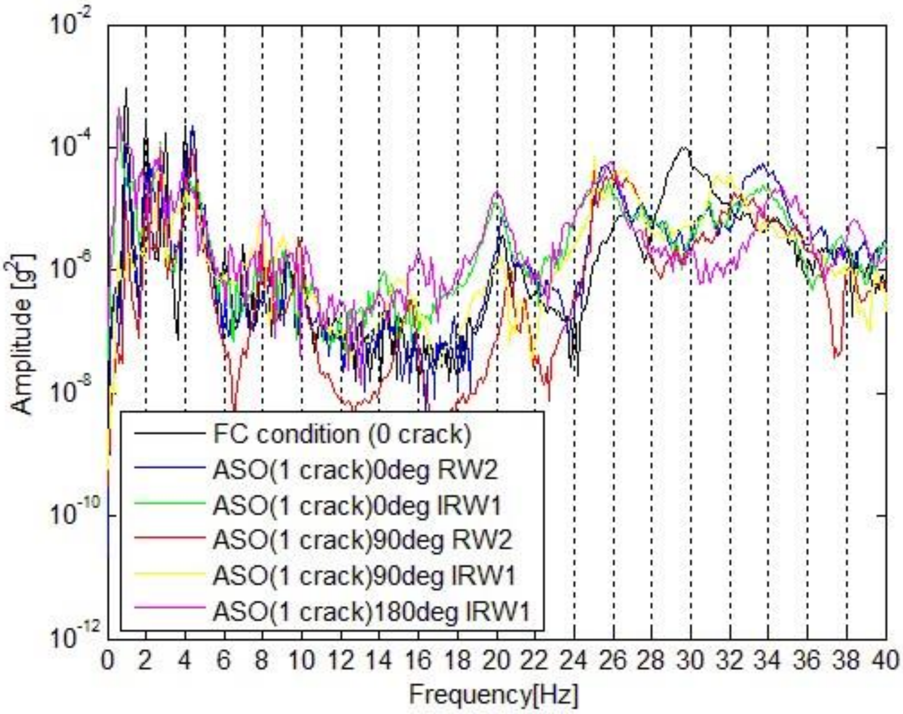

Figure 16. Cross-power spectra of the five cases reported in Table 7.

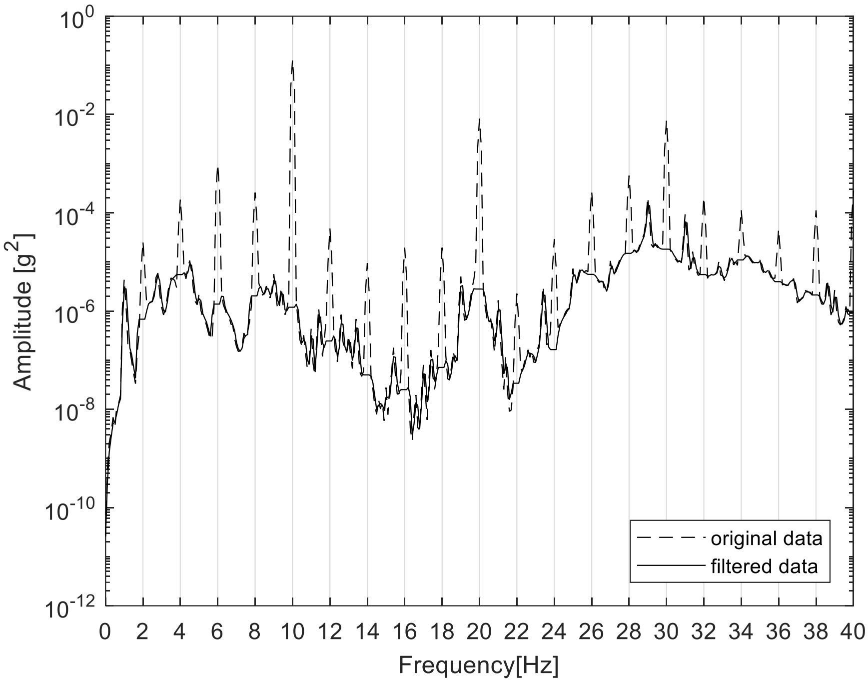

Figure 17. Cross-power spectra of the acceleration measured in Node 1, Direction X in case of original data and application of the harmonic filter in case of rotor rotating at 120 rpm.

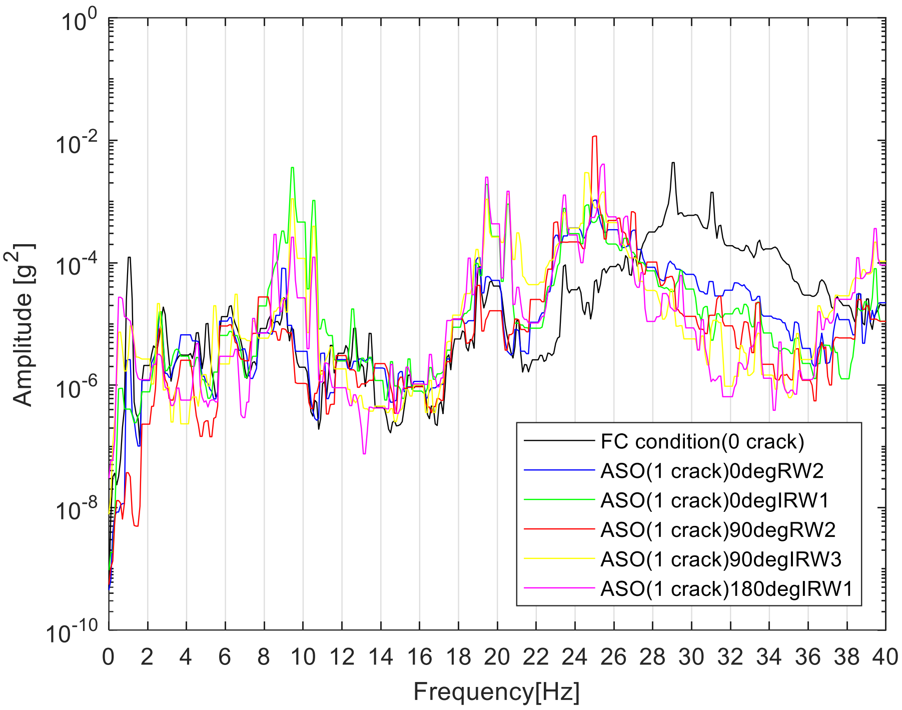

Figure 18. Crosspower spectra of the five cases reported in Table 8.