Inverse estimation of breaking wave loads on monopile wind turbines^{☆,☆☆}

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Abstract

Breaking waves occur in shallow water and cause large forces on offshore wind turbines, which can determine the design of the support structure. Due to the strong nonlinearity and statistical variability of breaking waves, the corresponding loads used in the design are generally characterized by large uncertainty. This paper presents a validation of an inverse method to estimate breaking wave loads on offshore monopile wind turbines based on wave basin tests. The accelerations measured on a scale model of a monopile wind turbine are passed to a force identification algorithm together with a dynamic model of the structure in order to identify the wave loads. In addition, the member forces in the support structure are estimated. Both the wave loads and member forces are estimated with a reasonable accuracy. The proposed methodology creates opportunities for operational monitoring of offshore wind turbines, hereby providing essential information to improve design guidelines for future wind turbines and allowing for a continuous fatigue assessment of the wind turbine during the operation.

Keywords: offshore wind turbine, wave load identification, design feedback, structural health monitoring, wave basin experiment

1. Introduction

Many existing wind farms and wind farms under development are located in shallow water, characterized by a water depth less than 30 m. For these wind farms, the water depth is such that breaking or near-breaking waves will occur. Breaking waves occur when a wave becomes so steep that it becomes unstable, causing potentially large lateral forces on the offshore wind turbine (OWT), which, in some cases, can determine the design of the support structure [1, 2].

Several models and calculation methods have been developed to assess whether breaking waves will occur and to estimate the corresponding loads for the design of offshore structures. Semi-empirical models have been developed using wave basin experiments on scale models of offshore structures [3, 4]. More advanced methods are for example based on computational fluid dynamics

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(CFD) [5, 6], but are rarely used in practice due to their large computational load. Although a large variety of methods to account for breaking waves in the design is available, the uncertainty on the corresponding loads is generally large, due the strong nonlinearity and statistical variability of breaking waves. Hallowell et al. [1] state that there is a need for in-situ measurements of breaking wave loads and the corresponding structural response of OWTs. Within the framework of the Offshore Wind Turbines at Exposed Sites (OWTES) project, the environmental conditions (wave heights, wind velocity and direction), and associated structural response (strains in the tower) have been simultaneously recorded for 17 months during a measurement campaign on a 2.0 MW OWT at the coast of England. An analysis of the measurement data is provided in [1]. The results provide insight into several aspects of breaking waves, such as the variability in breaking wave conditions and forces. However, the method presented in [1] to collect in-situ data on breaking waves has two shortcomings. First, the direct measurement of strains at the mudline to obtain the overturning moment requires the installation of strain sensors below the water level, which is very challenging and infeasible for existing OWTs. Second, the strain measurements on the OWT give information on the response of the structure to the incoming wave field, but not directly on the wave loads and/or pressures, which are of primary interest in the design process.

This paper proposes the estimation of the loads applied by steep and breaking waves to monopile OWTs based on accelerations measured in the tower of the OWT, i.e. above the water level. The proposed methodology is validated using data obtained from model tests on a monopile OWT performed in a shallow water basin at MARIN, Wageningen, the Netherlands, within the frame of the Wave Impact on Fixed turbines (WiFi JIP) project [7]. In a first step, the accelerations measured in the tower are combined with a dynamic model of the structure to estimate the horizontal wave loads and the corresponding overturning moment with respect to the sea bed. The estimation is performed using a state-of-the-art joint input-state estimation algorithm presented in [8]. In a second step, the system states, that are estimated along with the wave loads, are used to predict the member forces at the base of the wind turbine. The results are validated by comparison to the wave loads and member forces that are obtained from direct force measurements. It is shown that the proposed procedure for load estimation allows to identify the peak loads due to extreme waves acting on OWTs and the corresponding member forces with a reasonable accuracy. The estimated wave loads can be used to improve design guidelines for future OWTs. In addition, the corresponding estimated member forces can be used in the fatigue assessment of instrumented OWTs.

The paper is outlined as follows. Section 2 presents the methodology. Section 3 presents the test structure considered in the study and its instrumentation. Next, Section 4 describes the dynamic system model that is used in the wave load identification. Section 5 briefly recapitulates the applied joint input-state estimation algorithm. Section 6 presents the results of the wave load identification and the member force estimation. Section 7 defines the future work that directly results from the work presented in this paper. Finally, Section 8 concludes the paper.

2. Methodology

The response of the OWT, which consists for example of a set of strains and/or accelerations, is used in combination with a dynamic model of the structure to estimate the wave loads by means of system inversion techniques. This process is illustrated in Fig. 1. Whereas in the design the wave loads are assumed known to calculate the dynamic response of the OWT, system inversion techniques are used here to identify the wave loads from dynamic response measurements and a

dynamic model of the structure. In a second step, the system states, that are estimated along with the wave loads, can be used to estimate the structural response (strains, accelerations, ...) at unmeasured locations.

Fig. 1: Scheme illustrating the proposed methodology.

The data used in the system inversion can consist of strains, displacements, velocities, or accelerations, measured in the tower of the OWT, above the water level. In most cases, acceleration data are preferred, since these can be acquired with high accuracy and at a relatively low cost. The uncertainty in the estimated wave loads depends on the sensor locations. The installation of sensors close to the (expected) force location, i.e. near the wave crest, generally results in low noise amplification and high estimation accuracy [8].

The model of the structure should accurately represent the dynamic behavior of the OWT in the frequency range which contains the response of the OWT to the impact loading by the breaking wave. For the considered application, it is generally sufficient to consider the low frequency range from 0 to 10 Hz. The dynamic system model can be obtained from a finite element (FE) model of the OWT, or can alternatively be directly identified from experimental vibration data using system identification techniques, see e.g. [9, 10]. The latter approach is followed in this paper.

In force identification, a distinction is made between the localization of the forces and the estimation of the force amplitudes. Previous studies have shown that the impact force of breaking waves is focused close to the wave crest [4]. Wienke and Oumeraci [11] assume a constant horizontal line load q (in N/m) over the area of impact, reaching from $\lambda\eta_b$ to η_b , where η_b is the water surface elevation at the wave crest, and λ is the so-called crest factor (Fig. 2). A commonly cited value for the crest factor given by Goda [12] is $\lambda = 0.4 - 0.5$ for plunging wave breaks. Deltares [13] proposes to replace the constant load distribution [11] by a (symmetric) triangular distribution over the impact area (Fig. 2). In the following, it is assumed that the wave crest height η_b is known for all breaking waves hitting the OWT. The wave elevation can be measured using a wave radar, see for example [1]. In addition, a crest factor $\lambda = 0.5$ and a triangular wave load distribution are assumed. Given the force location and distribution, the force identification boils down to the estimation of the peak line load $q^{\max}$ (Fig. 2), which allows calculating the force amplitude $F = q^{\max}\eta_b(1 - \lambda)/2$ and the corresponding overturning moment $M = F(d + \eta_b(1 + \lambda)/2)$.

It is expected that viscous forces on the monopile are not negligible for the considered case of breaking waves, especially for the time interval right after the wave impact. These viscous forces are not independently accounted for in the force estimation, however, since this would strongly complicate the estimation process. Furthermore, it is expected that during the wave impact the

Fig. 2: Indication of the wave impact area and wave load distribution proposed by Wienke and Oumeraci [11] (green rectangle) and Deltares [13] (red triangle).

inertial wave force component is far more important than the drag component. As such, it is expected that disregarding the viscous forces still enables accurate estimation of the impact load. When applying the proposed methodology for a case where data is obtained from a full-scale experiment on an operational offshore wind turbine, wind loads are additionally acting on the structure. It is assumed here that the structural response is dominated by the wave loading when the design values of breaking wave loads are considered. The wind loads are therefore disregarded in the wave load estimation.

Many algorithms have been proposed in the literature to solve the inverse force identification problem presented in Fig. 1. These algorithms tackle the problem in the frequency domain [14, 15] or alternatively in the time domain [16, 17]. The algorithm used in the present application is a recursive joint input-state estimation algorithm which was proposed by Maes et al. in [8]. The algorithm operates in the time domain and allows to identify the forces applied to a structure at known locations and the corresponding system states in a recursive fashion. Linear structural behavior is assumed. The main advantage of the algorithm is that it can be applied online for continuous monitoring of structures, without requiring any prior information on the time history of the identified forces.

Once the forces and the corresponding dynamic system states are identified, they can be used for so-called response estimation, i.e. to identify the dynamic response of the structure at unmeasured locations. In this case, response estimation is used to identify the member forces at the base of the wind turbine, which can be adopted as a basis for a fatigue estimation of the structure. The concept of fatigue monitoring based on in-situ vibration measurements was introduced by Papadimitriou et al in [18], where the Kalman filter is used for stress estimation at fatigue hotspots. Following this first publication, a number of subsequent approaches is presented in the literature, which mainly differ in the way they perform the stress/strain estimation [8, 19, 16]. A validation of vibration-based strain estimation based on full-scale experiments on an offshore wind turbine can be found in [20, 21].

3. Test structure

The experiments considered in this paper have been performed in a shallow water basin at MARIN, Wageningen, The Netherlands. The basin has a length of 220 m, a width of 15.8 m and a constant water depth, which equals 1.1 m in this case. A piston-type wave generator is installed on the short side of the basin, which can generate both regular and irregular waves with a wave period from 0.5 to 5 s and a significant wave height up to 0.25 m. An absorbing parabolic beach is installed at the opposite side of the basin, in order to minimize wave reflection. A scale model of a flexible monopile is installed at approximately 65 m from the wave generator (Fig. 3). A rigid substructure is installed right next to the flexible model in order to collect information on the incoming wave field.

Fig. 3: (a) The MARIN shallow water basin, (b) the test setup with the flexible monopile (left) and rigid substructure (right) installed on the basin floor, and (c) picture of the flexible monopile (left) and the rigid substructure (right) prior to installation.

The flexible monopile is constructed after a reference design of a 4 MW bottom-fixed offshore monopile wind turbine by Ramboll [22]. The model is scaled according to Froude scaling, using a geometrical scale 1:30.56. Table 1 gives an overview of the scaling factors. The flexible monopile has a height of 4.0 m, which corresponds to a height of 124 m on full scale. The rotor-nacelle assembly is modeled by a mass of 278 tons (full scale), mounted on top of the tower. The monopile is composed of two PVC tubes with different diameter and thickness. The stiffness is tuned by making cuts at several vertical locations in the lower PVC tube, such that the natural frequencies and mode shapes of the of the first two bending modes of the reference design are approximately reproduced. The rigid substructure is made of steel tubes with the same outer diameter as the PVC tubes used for the construction of the flexible monopile.

Quantity	Scale model	Full size	Ratio
Distance	1 m	30.56 m	λ
Time	1 s	5.53 s	$\sqrt{\lambda}$
Frequency	1 Hz	0.18 Hz	$1/\sqrt{\lambda}$
Acceleration	1 m/s ²	1 m/s ²	1

Table 1: Scaling factors according to Froude scaling.

Fig. 4 presents an overview of the instrumentation of the monopiles. The flexible monopile is instrumented with five three-axial accelerometers, distributed over the height of the structure. Six

resistance-type relative wave probes are installed along the perimeter of the flexible monopile to measure the wave elevation. A six-component force frame is used to measure the internal forces and moments at the seabed (basin floor).

Fig. 4: Overview of the measurement setup in the MARIN wave basin.

The rigid substructure is used to collect information the hydrodynamic excitation, (local) wave pressures, and the wave elevation. A single three-axial accelerometer is installed right above the waterline. Five resistance-type relative wave probes are installed along the perimeter of the rigid substructure to measure the wave elevation. Fifteen pressure sensors are installed over the height of the rigid substructure to measure the local wave pressure. Finally, a six-component force frame is used to measure the internal forces and moments at the seabed (basin floor). The external forces applied by the slamming wave (horizontal load and overturning moment) are derived from the measured internal forces. A correction for the inertial forces due to the rigid body motion of the structure is made, based on the measured accelerations [23].

The acceleration data, the data obtained from the wave probes, and the data obtained from the force frames are acquired at a sampling frequency of 200 Hz (model scale), corresponding to a sampling frequency of 36.18 Hz on full scale. The pressure data are acquired at a sampling frequency of 1000 Hz (model scale), corresponding to a sampling frequency of 180.9 Hz on full scale. All data are immediately rescaled to full scale, using the scaling factors presented in Table 1. No additional data manipulations are performed in the data acquisition.

Breaking waves are generated in the wave basin using so-called wave focusing. The wave generator first generates short waves, after which it starts generating long waves. Due to dispersion, each wave component travels at its own speed. The timing between the different components is such

that the wave crests of the different waves meet at a predefined focusing point. In this case, this focusing point is located at the wind turbine. Fig. 5 shows an example of the wave elevation at the side of the flexible monopile. The large wave elevation observed at time $t = 50$ s corresponds to the steep slamming wave obtained by the wave focusing. Fig. 6 shows a snapshot of the wave impact at the moment where the wave is breaking. This picture shows that the actual breaking point for this experiment is located right after the monopile and therefore does not perfectly coincide with the intended focus point.

In total, 27 experiments with breaking waves are considered in the following. Slightly different sets of input parameters of the wave generator have been considered in the different experiments, as to achieve (1) slightly different wave focusing points and (2) different wave heights. For most sets of input parameters, multiple experiments have been performed to investigate repeatability. It was found that the variation in the wave elevation, wave pressure, and therefore the wave impact load observed for different experiments with the same input parameters is very large and often even larger than the variation between experiments with different input parameters. This is most likely due to the strong nonlinearity of breaking waves, making them very sensitive to small variations in the incoming wave field.

Fig. 5: Time history of the wave elevation at the side of the flexible monopile during experiment 1.

Fig. 6: Snapshot of the wave impact on the flexible turbine obtained from a high-speed camera during experiment 1.

4. Dynamic system model

A modally reduced-order state-space model describing the dynamic behavior of the flexible monopile is used in the force estimation. Linear behavior of the structure is assumed. The presented model describes the dynamic behavior of the flexible monopile along the incoming wave direction (x)

and is constructed from the experimentally identified modal characteristics (natural frequencies, mode shapes, and modal damping ratios) corresponding to the first four bending modes of the structure.

The identification of the modal characteristics is based on the free vibration response of the flexible monopile following a hammer impact applied in the x -direction. The accelerations at the five measurement levels (Fig. 4) are processed using the reference-based covariance-driven stochastic subspace identification (SSI-cov/ref) algorithm [9] to extract the modal characteristics. After a spline interpolation using an inter-node distance of 0.01 m along the height of the turbine (full-scale), the identified mode shapes are scaled to unit modal mass. The (lumped) mass matrix used in the mass normalization is constructed from the known mass distribution of the scale model. Table 2 shows the identified natural frequencies and modal damping ratios corresponding to the first four bending modes. Fig. 7 shows the corresponding mode shapes. Note that the modal characteristics of modes 3 and 4 are not representative of the full-scale wind turbine, since only modes 1 and 2 have been tuned to obtain a good agreement between the model and the reference design (see also Section 3).

Mode	Natural frequency		Natural period		Damping ratio
	Model scale	Full scale	Model scale	Full scale	
[–]	[Hz]	[Hz]	[s]	[s]	[%]
1	1.63	0.30	0.61	3.38	0.7
2	6.56	1.19	0.15	0.84	1.6
3	17.04	3.08	0.06	0.32	2.0
4	39.88	7.21	0.03	0.14	2.4

Table 2: Natural frequencies f_j , natural periods T_j , and modal damping ratios ξ_j of the first four bending modes in the x -direction, obtained from the system identification.

Fig. 7: Mode shapes of the first four bending modes in the x -direction, obtained from the system identification. The waterline ($z = 0$ m) is indicated with a blue horizontal line. The circular markers indicate the accelerometer locations.

The experimentally identified modal characteristics are used to construct a state-space model, which takes the following form:

$$\mathbf{x}_{[k+1]} = \mathbf{A}\mathbf{x}_{[k]} + \mathbf{B}\mathbf{p}_{[k]} + \mathbf{w}_{[k]} \quad (1)$$

$$\mathbf{d}_{[k]} = \mathbf{G}\mathbf{x}_{[k]} + \mathbf{J}\mathbf{p}_{[k]} + \mathbf{v}_{[k]} \quad (2)$$

where $\mathbf{x}_{[k]} \in \mathbb{R}^{n_s}$ is the state vector, $\mathbf{d}_{[k]} \in \mathbb{R}^{n_d}$ is the output vector, and $\mathbf{p}_{[k]} \in \mathbb{R}^{n_p}$ is the unknown input vector representing the wave loads, with n_s the number of system states, n_d the number of outputs, and n_p the number of inputs. The process noise $\mathbf{w}_{[k]} \in \mathbb{R}^{n_s}$ accounts for modeling errors and unknown forces acting on the structure besides the identified wave loads $\mathbf{p}_{[k]}$. The measurement noise $\mathbf{v}_{[k]} \in \mathbb{R}^{n_d}$ accounts for modeling errors, unknown forces, and measurement errors. For the presented application, a reduced-order discrete-time state-space model is constructed from the experimental modal characteristics. The model includes the four modes listed in Table 2 ($n_m = 4$). The state vector $\mathbf{x}_{[k]}$ includes the modal displacements $z_{j[k]}$ and velocities $\dot{z}_{j[k]}$ corresponding to every mode j ($j = 1, \dots, n_m$) at discrete times $t_k = k\Delta t$, with Δt the sampling time step. As such, the number of states n_s equals 2 times the number of modes included in the model n_m . The time discretization of the system model is performed under a first-order hold assumption on the input [24] and applying a sampling frequency of 36.18 Hz (full-scale), resulting in a time step $\Delta t = 0.028$ s.

The state-feedback matrix $\mathbf{A}$ and state-input matrix $\mathbf{B}$ in Eq. (1) are given by:

$$\mathbf{A} = \exp(\mathbf{A}_c \Delta t) \quad (3)$$

$$\mathbf{B} = \frac{1}{\Delta t} \mathbf{A}_c^{-2} (\mathbf{A} - \mathbf{I}_{n_s})^2 \begin{bmatrix} \mathbf{0} \\ \mathbf{\Phi}^T \mathbf{S}_p \end{bmatrix} \quad (4)$$

where $\mathbf{I}_\alpha \in \mathbb{R}^{\alpha \times \alpha}$ denotes an identity matrix, $\mathbf{\Phi} \in \mathbb{R}^{n_{\text{dof}}}$ is the matrix containing as columns the identified mode shapes ϕ_j after spline interpolation and mass normalization, and $\mathbf{S}_p \in \mathbb{R}^{n_{\text{dof}} \times n_p}$ is a matrix describing the spatial distribution of the different inputs. Note that in this case, where a model is obtained directly from a system identification, n_{dof} represents the number of points along the height of the structure that is assumed in the spline interpolation of the mode shapes. The spatial distribution matrix/vector $\mathbf{S}_p$ ($n_p = 1$) is obtained by calculating the equivalent (horizontal) nodal forces at the area of impact that correspond to the assumed triangular force distribution, for $q^{\max} = 1$ (see also Fig. 8). As such, the identified force $\mathbf{p}_{[k]}$ corresponds to the peak value $q_{[k]}^{\max}$ of the distributed load, which can then be used to compute the horizontal force and the corresponding overturning moment (see also Section 2). Finally, $\mathbf{A}_c$ is the state-feedback matrix of the continuous system, given by:

$$\mathbf{A}_c = \begin{bmatrix} \mathbf{0} & \mathbf{I}_{n_m} \\ -\mathbf{\Omega}^2 & -\mathbf{\Gamma} \end{bmatrix} \quad (5)$$

with $\mathbf{\Omega} \in \mathbb{R}^{n_m \times n_m}$ a diagonal matrix containing the identified natural frequencies ω_j [rad/s] on its diagonal and $\mathbf{\Gamma} \in \mathbb{R}^{n_m \times n_m}$ another diagonal matrix containing the terms $2\xi_j \omega_j$ on its diagonal, where ξ_j is the modal damping ratio identified for mode j .

The state-output matrix $\mathbf{G}$ and the direct feedthrough matrix $\mathbf{J}$ in Eq. (2) are given by:

$$\mathbf{G} = [\mathbf{S}_d \mathbf{\Phi} - \mathbf{S}_a \mathbf{\Phi} \mathbf{\Omega}^2 \quad \mathbf{S}_v \mathbf{\Phi} - \mathbf{S}_a \mathbf{\Phi} \mathbf{\Gamma}] \quad (6)$$

$$\mathbf{J} = \mathbf{S}_a \mathbf{\Phi} \mathbf{\Phi}^T \mathbf{S}_p + \mathbf{G} \left(\frac{1}{\Delta t} \mathbf{A}_c^{-2} (\mathbf{A} - \mathbf{I}_{n_s}) - \mathbf{A}_c^{-1} \right) \begin{bmatrix} \mathbf{0} \\ \mathbf{\Phi}^T \mathbf{S}_p \end{bmatrix} \quad (7)$$

Fig. 8: Illustration of the conversion from the assumed triangular load distribution (left) to equivalent nodal forces that compose the vector $\mathbf{S}_p$ (right).

where $\mathbf{S}_a \in \mathbb{R}^{n_d \times n_{dof}}$, $\mathbf{S}_v \in \mathbb{R}^{n_d \times n_{dof}}$, and $\mathbf{S}_d \in \mathbb{R}^{n_d \times n_{dof}}$ are selection matrices indicating the degrees of freedom corresponding to the acceleration, velocity and displacement or strain measurements, respectively.

The system model used in the present analysis is based on the experimental modal characteristics of the OWT. Alternatively, a model can be obtained from a detailed finite element (FE) model of the structure. Models based on first principles are more generally applicable than models that are directly obtained from experimental vibration data, as they allow for a reconstruction of forces and response at any location in the structure. In this case, however, a high fidelity model of the structure could not be obtained from a FE-based approach, since the stiffness distribution of the flexible OWT is significantly affected by the horizontal cuts (Section 3) and therefore not accurately known.

5. Joint input-state estimation algorithm

The wave load estimation is performed by means of the joint input-state estimation algorithm introduced in [8]. This section gives a brief summary of the algorithm.

The joint input-state estimation algorithm makes use of a discrete-time combined deterministic-stochastic state-space description of the structure, given by equations (1) and (2). The system matrices $\mathbf{A}$, $\mathbf{B}$, $\mathbf{G}$, and $\mathbf{J}$ are assumed known. The noise processes $\mathbf{w}_{[k]}$ and $\mathbf{v}_{[k]}$ are assumed to be zero mean and white, with known covariance matrices $\mathbf{Q}$, $\mathbf{R}$, and $\mathbf{S}$, defined by:

$$\mathbb{E} \left\{ \begin{pmatrix} \mathbf{w}_{[k]} \\ \mathbf{v}_{[k]} \end{pmatrix} \begin{pmatrix} \mathbf{w}_{[l]}^T & \mathbf{v}_{[l]}^T \end{pmatrix} \right\} = \begin{bmatrix} \mathbf{Q} & \mathbf{S} \\ \mathbf{S}^T & \mathbf{R} \end{bmatrix} \delta_{[k-l]} \quad (8)$$

with $\mathbf{R} > 0$, $\begin{bmatrix} \mathbf{Q} & \mathbf{S} \\ \mathbf{S}^T & \mathbf{R} \end{bmatrix} \geq 0$, and $\delta_{[k]} = 1$ for $k = 0$ and 0 otherwise. $\mathbb{E}\{\cdot\}$ is the expectation operator.

Joint input-state estimation consists of estimating the forces $\mathbf{p}_{[k]}$ and states $\mathbf{x}_{[k]}$ from a set of response measurements $\mathbf{d}_{[k]}$. A force estimate $\hat{\mathbf{p}}_{[k|l]}$ is defined as an estimate of $\mathbf{p}_{[k]}$ given the output sequence $\mathbf{d}_{[n]}$, with $n = 0, 1, \dots, l$. A state estimate $\hat{\mathbf{x}}_{[k|l]}$ is defined similarly. The force and state estimates are computed recursively in three steps:

$$\hat{\mathbf{p}}_{[k|k]} = \mathbf{M}_{[k]} (\mathbf{d}_{[k]} - \mathbf{G}\hat{\mathbf{x}}_{[k|k-1]}) \quad (9)$$

$$\hat{\mathbf{x}}_{[k|k]} = \hat{\mathbf{x}}_{[k|k-1]} + \mathbf{K}_{[k]} (\mathbf{d}_{[k]} - \mathbf{G}\hat{\mathbf{x}}_{[k|k-1]} - \mathbf{J}\hat{\mathbf{p}}_{[k|k]}) \quad (10)$$

$$\hat{\mathbf{x}}_{[k+1|k]} = \mathbf{A}\hat{\mathbf{x}}_{[k|k]} + \mathbf{B}\hat{\mathbf{p}}_{[k|k]} \quad (11)$$

The first step in Eq. (9), referred to as the “input estimation step”, yields a filtered estimate of the unknown input vector $\mathbf{p}_{[k]}$, given the measured output $\mathbf{d}_{[k]}$ up to time step k . The second step in Eq. (10), referred to as the “measurement update”, yields a filtered estimate of the state vector $\mathbf{x}_{[k]}$. The third step in Eq. (11), referred to as the “time update”, yields a one step ahead prediction of the state vector $\mathbf{x}_{[k+1]}$. The gain matrices $\mathbf{M}_{[k]} \in \mathbb{R}^{n_p \times n_d}$ and $\mathbf{K}_{[k]} \in \mathbb{R}^{n_s \times n_d}$ are determined such that both the input estimates $\hat{\mathbf{p}}_{[k|k]}$ and the state estimates $\hat{\mathbf{x}}_{[k|k]}$ are minimum variance and unbiased (MVU) [25], i.e. the uncertainty on the force and state estimates is minimized, and the error on the estimated forces $\hat{\mathbf{p}}_{[k|k]}$ and states $\hat{\mathbf{x}}_{[k|k]}$ does not depend on the actual forces $\mathbf{p}_{[k]}$. The gain matrices depend on the noise covariance matrices $\mathbf{Q}$, $\mathbf{R}$, and $\mathbf{S}$, on the forces to be estimated, as well as on the sensor configuration.

The algorithm is initialized using an initial state vector $\hat{\mathbf{x}}_{[0|-1]}$ and its error covariance matrix $\mathbf{P}_{\mathbf{x}[0|-1]} (= \mathbb{E} \{(\mathbf{x}_{[0]} - \hat{\mathbf{x}}_{[0|-1]})(\mathbf{x}_{[0]} - \hat{\mathbf{x}}_{[0|-1]})^T\})$. It was demonstrated in [8] that the filtering algorithm converges towards its steady state within a very short time period (several seconds). Once the steady-state is reached, the output of the algorithm is no longer determined by the initial conditions, i.e. $\hat{\mathbf{x}}_{[0|-1]}$ and $\mathbf{P}_{\mathbf{x}[0|-1]}$. The choice of the initial conditions is therefore not crucial for a successful application of the algorithm.

6. Results

This section presents the results of the wave load identification. A two-step approach is followed. In Section 6.1, the wave loads are identified from the measured accelerations on the flexible monopile using the joint input-state estimation algorithm presented in Section 5. In Section 6.2, the system states that are estimated along with the wave loads are used in the estimation of the member forces at the base of the flexible monopile. For both steps, the proposed methodology is first applied for a single experiment in order to illustrate the different steps and intermediate results. Next, the estimation is repeated for 27 experiments to investigate repeatability.

6.1. Wave load identification

The joint input-state estimation algorithm presented in Section 5 assumes the force location/distribution to be known. As mentioned in Section 2, a triangular load distribution over the area of impact is assumed (Fig. 2), with crest factor $\lambda = 0.5$. Fig. 9 shows a three dimensional representation of the pressure time history at different heights z of the slamming zone. Projection of the pressure curves to the Pressure-Time coordinate plane shows very sharp peaks corresponding to the wave impact. The peaks at different locations are slightly delayed because the slamming wave does not hit all points of the slamming zone simultaneously. The largest observed delay equals 0.1 s. Projection of the pressure curves to the Position-Pressure coordinate plane reveals the peak pressure distribution over the slamming zone, which is in reasonable agreement with the assumed triangular load distribution. The position of the peak load, $z = (1 + \lambda)\eta_b/2$, is assumed to coincide with the highest wave elevation observed by the relative wave probes (Fig 5). Furthermore, it is observed from Fig. 9 that the pressure time history of the lower points in the slamming zone does not show a sharp peak. For these points, a slow variation in pressure corresponding to the wave rising (Fig. 6) is observed. This wave rising exerts a distributed load at the lower part of the monopile, which is not accounted for by the triangular load.

The load estimation is performed based on the acceleration measured right above the water level (accelerometer 2, Fig. 4). This choice is clarified in the following. Only the acceleration in

Fig. 9: Time history of the pressure measured by the pressure sensors in the slamming zone during experiment 1. The sensor position corresponds to its z -coordinate. The peak pressures are indicated by blue dots for the different sensors. The gray curves represent a projection of the data to the coordinate planes. The red curve represents a fit of the peak pressure data using the triangular distribution proposed by Deltares in [13], with $\lambda = 0.5$.

the incoming wave direction (x) is considered. Fig. 10 shows the time history and the narrow band frequency spectrum of the measured acceleration $2x$ for experiment 1. The acceleration level significantly increases due to the wave impact at time $t = 50$ s, hereafter showing a clear decay in free vibration. It is clear from the frequency spectrum in Fig. 10b that the wave impact excites the four bending modes of the structure.

From the conditions for instantaneous system inversion presented in [26], it is found that for the considered case, where a single force is estimated ($n_p = 1$), the output vector $\mathbf{d}_{[k]}$ used in the joint input-state estimation should contain at least one acceleration and one displacement. The acceleration is required to ensure a direct feedthrough from the estimated force to the measured response, enabling input estimation without any delay [26]. The displacement is required to stabilize the estimation. Without displacement, the joint input-state estimation becomes unstable, leading to (1) unbounded force and state estimates, and (2) computational problems, caused by divergence of the gain matrices $\mathbf{M}_{[k]}$ and $\mathbf{K}_{[k]}$. Since no displacements were measured during the considered experiments, the displacement of the flexible monopile is obtained by double (numerical) integration of the measured acceleration $2x$. The integration of the acceleration signal is performed in the frequency domain. The displacement signal obtained by integration is passed through a third order Chebyshev type I highpass filter with a cutoff frequency of 0.1 Hz and a filter ripple of 0.01 dB. The aim of this filter is to remove the frequency components from the signals which are contaminated by measurement noise on the acceleration data. It has to be kept in mind that, although the displacements obtained by integration lead to a stabilization of the joint input-state estimation algorithm, i.e. the gain matrices $\mathbf{M}_{[k]}$ and $\mathbf{K}_{[k]}$ stabilize to a finite value, they do not add low frequency information to the estimation. As a result, the force and

Fig. 10: (a) Time history and (b) amplitude of the narrow band frequency spectrum of acceleration $2x$ due to wave impact loading during experiment 1. The natural frequencies of the flexible monopile are indicated in (b) by a vertical dashed line.

state estimates obtained from the system inversion are characterized by spurious low frequency components that originate from low frequency noise on the acceleration data. For this application, these spurious low frequency components have been removed by passing both the estimated force $\hat{p}_{[k|k]}$ and the estimated states $\hat{\mathbf{x}}_{[k|k]}$ through a fifth order Chebyshev type I highpass filter with a cutoff frequency of 0.1 Hz and a filter ripple of 0.01 dB. For applications where it is essential to retrieve accurate low frequency information from the force and state estimation, one should avoid this highpass filtering step by performing displacement and/or strain measurements in addition to the acceleration measurements.

The noise covariance matrices $\mathbf{Q}$ and $\mathbf{S}$ assumed for the application of the joint input-state estimation algorithm are both assigned a zero value. This implies that the estimated point load accounts for every excitation acting on the structure, since this assumption imposes the process noise $\mathbf{w}_{[k]}$ to be zero [8]. The matrix $\mathbf{R}$ accounts for sensor noise and is assigned a diagonal matrix with the measurement noise variance on its diagonal, assuming a variance $10^{-6} \text{ (m/s}^2\text{)}^2$ for the noise on the acceleration data and 10^{-12} m^2 for the noise on the displacement data. The correlation between the noise on the measured acceleration data and the computed displacement data is disregarded.

Figure 11 compares the estimated load to the actual wave impact load, which has been obtained from the force frame underneath the rigid substructures (see also Section 3). The estimated load history generally agrees well with the measured load, except right after the impact, where spurious oscillations occur. These oscillations are also clearly observed from the frequency spectrum in Fig. 11b at the antiresonance frequency of about 0.44 Hz. For this antiresonance frequency, the contribution of higher modes which have been omitted in the dynamic system model is large [27]. Reducing the non-physical force components near antiresonance therefore requires a quasi-static correction for the disregarded modes. In this case where the stiffness distribution of the structure is unknown, such quasi-static correction cannot be made. In addition to the errors introduced by disregarding the higher modes, it was found that the estimated impact force near the antiresonance frequency is sensitive to the errors in the modal characteristics and the assumed load distribution. Finally, it is noticed that the spurious oscillations for most experiments are much smaller than those observed in Fig. 11 for experiment 1.

The overturning moment of the wave impact load with respect to the sea bed is obtained by multiplication of the estimated wave load with its lever arm $d + \eta_b(1 + \lambda)/2$, which equals

Fig. 11: (a) Time history and (b) amplitude of the narrow band frequency spectrum of the measured (blue dotted line) and estimated (red solid line) wave impact load for experiment 1. The natural frequencies of the flexible monopile are indicated in (b) by a vertical dashed line, the antiresonance frequencies of the reduced-order model by vertical dotted lines.

36.52 m for experiment 1. Figure 12 compares the estimated overturning moment to the overturning moment obtained from the force frame underneath the rigid substructure. Except for the previously mentioned high frequency oscillations, the estimated and measured overturning moments agree very well and a good estimate of the peak moment is obtained.

Fig. 12: (a) Time history and (b) amplitude of the narrow band frequency spectrum of the measured (blue dotted line) and estimated (red solid line) overturning moment with respect to the sea bed for experiment 1. The natural frequencies of the flexible monopile are indicated in (b) by a vertical dashed line, the antiresonance frequencies of the reduced-order model by vertical dotted lines.

The estimation errors observed for the wave load and the corresponding overturning moment are attributed to several sources of uncertainty, which are discussed hereafter.

First, errors in the assumed dynamic behavior as represented by the system model directly result in force estimation errors. The assumption of linear behavior is not necessarily satisfied. Indeed, for the considered impact loading, the PVC might be exposed to small material nonlinearities. In addition, if the structure were to behave in a linear regime, the linear model assumed in the estimation is still subject to model uncertainty, which is due to errors on the identified modal characteristics and disregarding the contribution of higher order modes.

Secondly, the spatial load distribution changes continuously over time. The assumed triangular distribution only approximates the actual load distribution well during the impact (Fig. 9). How-

ever, this does not ensure that the peak impact load is accurately estimated in all cases. Indeed, the response of the structure which is used to estimate the peak load is not only determined by the load at the same time instant, as would be the case for static loading, but also by the loads at the previous time steps, where the assumed load distribution is incorrect. This results in incorrect load and state estimates for time steps prior to the impact, which affect the force estimate at the moment of impact. This source of uncertainty is expected to cause the smallest error in peak load for collocated acceleration measurements, where the so-called direct feedthrough from the loads to the response is much larger than the contribution of the loads at previous time steps (see also [28]). This explains the choice of the acceleration measurement which coincides with the slamming zone for the present application. Taking into account acceleration signals that are obtained at a significant distance of the slamming zone generally results in larger errors in the peak forces.

Thirdly, as mentioned previously, the wave rising exerts a distributed load below the slamming zone, which is not accounted for in the triangular load distribution. One could try to assume an alternative pressure distribution or to estimate multiple loads with different spatial distributions. In the present validation study, the pressure measurements in the tower could be used to derive an alternative load distribution, which is different for all experiments. This is not a viable approach in field tests, however, and, for this reason, is not adopted here. The estimation of multiple loads with different spatial distributions is found to be infeasible as well. The large simplification of the load makes the results very sensitive to the assumed spatial distributions when using multiple loads, resulting in large estimation errors.

Figures 13 and 14 compare the identified and measured peak wave impact forces and peak overturning moments for the 27 focused wave experiments performed in the wave basin. For both quantities, the estimation errors are within -50% to +50% of the measured wave loads. A negative error corresponds to an underestimation. The peak wave forces are mostly underestimated, whereas no clear trend is observed for the overturning moment. The average (absolute) estimation error equals 27% for the impact force and 19% for the overturning moment, implying that the wave loads are identified with a reasonable accuracy.

6.2. Member force estimation

In a second step, the system states obtained from the joint input-state estimation algorithm are used to estimate the internal forces in the flexible monopile at the sea bed. The procedure used for estimating the member forces is outlined next.

Consider the vector $\mathbf{Q}_b$ containing the six member forces at the sea bed:

$$\mathbf{Q}_b = [V_{bx}, V_{by}, N_{bz}, M_{bx}, M_{by}, T_{bz}]^T \quad (12)$$

with V_{bx} and V_{by} the base shear forces in respectively the x - and y -direction, N_{bz} the axial force, M_{bx} and M_{by} the bending moments in the monopile around respectively the x - and y -axis, and T_{bz} the torsional moment.

For every time step k , the member forces $\mathbf{Q}_b$ are estimated from the contribution of the different modes, using the following equation:

$$\hat{\mathbf{Q}}_{b[k|k]} = \sum_{m=1}^{n_m} \hat{\mathbf{Q}}_{bm[k|k]} = \sum_{m=1}^{n_m} \varphi_{Q_{bm}} \hat{z}_{m[k|k]} \quad (13)$$

Fig. 13: (a) Comparison of the measured (blue diamond) and estimated (red square) peak wave impact force and (b) error on the peak wave impact force relative to the measured value.

Fig. 14: (a) Comparison of the measured (blue diamond) and estimated (red square) peak overturning moment and (b) error on the peak overturning moment relative to the measured value.

where $\hat{z}_{m[k|k]} \in \mathbb{R}$ is the modal displacement corresponding to mode m , which is obtained from the estimated state vector $\hat{\mathbf{x}}_{[k|k]}$. In this case, the vector of modal member forces $\varphi_{Q_{bm}} \in \mathbb{R}^{6 \times 1}$ is determined using the following expression [29], which assumes the mass distribution of the structure to be known:

$$\varphi_{Q_{bm}} = -\omega_m^2 \mathbf{L}^T \mathbf{M} \varphi_m \quad (14)$$

where $\omega_m \in \mathbb{R}$ is the natural frequency corresponding to mode m , $\mathbf{M} \in \mathbb{R}^{n_{\text{dof}} \times n_{\text{dof}}}$ is the mass matrix, $\boldsymbol{\varphi}_m \in \mathbb{R}^{n_{\text{dof}}}$ is the mass-normalized mode shape vector corresponding to mode m , and $\mathbf{L} \in \mathbb{R}^{n_{\text{dof}} \times 6}$ describes the rigid body kinematics of the structure with respect to the base (at the sea bed level) [29], with n_{dof} the number of degrees of freedom assumed in the construction of the mass matrix. The number of modes n_m accounted for in the analysis equals four in this case. For the present application, only the base shear force V_{bx} along the x -axis and bending moment M_{by} around the y -axis are considered.

Figure 15 compares the estimated and measured base shear force V_{bx} . Figure 16 compares the estimated and measured bending moment at the base M_{by} . Both estimated member forces are in very good agreement with the measured forces. The spurious oscillations at a frequency of about 0.44 Hz are also present in the estimated member forces, but less pronounced than for the estimated wave loads. Since the estimation is based on acceleration measurements, the estimated member forces at low frequencies should not be considered. If needed, the low frequency components can be reconstructed by supplementing the acceleration data with strain data [20].

Fig. 15: (a) Time history and (b) amplitude of the narrow band frequency spectrum of the measured (blue dotted line) and estimated (red solid line) base shear force V_{bx} . The natural frequencies of the flexible monopile are indicated in (b) by a vertical dashed line, the antiresonance frequencies of the reduced-order model by vertical dotted lines.

Fig. 16: (a) Time history and (b) amplitude of the narrow band frequency spectrum of the measured (blue dotted line) and estimated (red solid line) bending moment M_{by} . The natural frequencies of the flexible monopile are indicated in (b) by a vertical dashed line, the antiresonance frequencies of the reduced-order model by vertical dotted lines.

Figures 17 and 18 compare the identified and measured peak base shear forces and peak bend-

ing moments at the base for the 27 focused wave experiments performed in the wave basin. For both quantities, the estimation errors are within -40% to +10% of the measured forces. A slight underestimation of the actual member forces is observed for most experiments, which might result from errors in the assumed load distribution (see also Section 6.1). The average (absolute) estimation error equals 20% for the base shear force and 12% for the bending moment.

Fig. 17: (a) Comparison of the measured (blue diamond) and estimated (red square) peak base shear force and (b) error on the peak base shear force relative to the measured value.

Fig. 18: (a) Comparison of the measured (blue diamond) and estimated (red square) peak bending moment at the base and (b) error on the peak bending moment at the base relative to the measured value.

7. Future work

In general, wave load induced fatigue is not only attributed to breaking waves but also to non-breaking waves with frequencies close the first eigenfrequency of the structure [30]. Interestingly, the validity of the Morison equations, used to determine non-breaking wave loads in the design, is questioned for very large wind turbines on so-called XL monopiles, with diameters exceeding 7 m [31]. A methodology that would serve to estimate the wave loads in situ on fatigue driven designs could thus serve to improve future designs and develop a better understanding of wave loading on very large structures. In addition, the effect of for example marine growth on the wave loading can be assessed over time.

In order to transition from the wave-basin to a real-world wind turbine, the method needs to be adapted to cope with simultaneous loads from different sources. In particular wind loads and low-order rotor harmonics will occur in the same frequency band as the wave loads. To correctly assess the wave loads, these different loads need to be accurately separated from each other. One possible additional input is to use the turbine’s operational (SCADA) data to estimate the wind-induced loads [32] in parallel to the wave load estimation.

8. Conclusions

This paper proposes an inverse approach to estimate breaking wave loads acting on offshore monopile wind turbines from in-situ vibration data. A validation of the presented approach is performed using wave basis tests. The accelerations measured in the tower of the wind turbine are passed to a force identification algorithm to identify the wave loads and the corresponding overturning moment with respect to the sea bed. In addition to the load estimation, the member forces at the base of the support structure are estimated. Both the wave loads and the corresponding member forces are estimated with a good accuracy. The estimated quantities provide essential information to improve design guidelines for future offshore wind turbines and allow for a continuous fatigue assessment of the instrumented structures.

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