



Performance characterization and modeling of an oscillating surge wave energy converter

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Abstract Testing wave energy converters in the ocean could be expensive and complex, which necessitates the use of numerical modeling. However, accurately modeling the response of wave energy converters with high-fidelity simulations can be computationally intensive in the design stage where different configurations must be considered. Reduced-order models based on simplified equations of motion can be very useful in the design, optimization, or control of wave energy converters. Given the complex dynamics of wave energy converters, accurate representation, and evaluation of relative contributions by different forces are required. This effort is concerned with a performance charac-

terization of the hydrodynamic response of an oscillating surge wave energy converter that is based on a reduced-order model. A state-space model is used to represent the radiation damping term. Morison's representation of unsteady forces is used to account for the nonlinear damping. Wave tank tests are performed to validate simulations. A free response simulation is used to determine the coefficients of the state-space model. Torque-forced simulations are used to identify the coefficients of the nonlinear damping term for different amplitudes and wave frequencies. The impact of varying these coefficients on the response is investigated. An assessment of the capability of the model in predicting the hydrodynamic response under irregular forcing is performed. The results show that the maximum error is 3% when compared with high-fidelity simulations. It is determined that the nonlinear damping is proportional to the torque amplitude and its effects are more pronounced as the amplitude of the flap oscillations increases.

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1 Introduction

Given that 70% of the earth is covered by water and the high energy density of ocean waves, converting wave energy to a usable form of energy is essential for the diversification of renewable energy resources toward

mitigating climate change. One of the most promising designs for wave energy converters (WEC) is the oscillating surge wave energy converter (OSWEC) [1–3]. It is a single degree of freedom flap that pierces the free surface. It is positioned perpendicular to the wave directions and hinged at its bottom to the seafloor in shallow waters or to a moored platform in deep water. The flap undergoes an oscillatory rotational motion as the wave's crest and trough pass by its free end. In an OSWEC, this motion is then converted to power using a power take-off (PTO) system. Because full-scale testing of such a converter is expensive, its design, performance evaluation, optimization, and control require simulations with an acceptable level of accuracy of its hydrodynamic response. High- and medium-fidelity computational fluid dynamic (CFD) models have been used to simulate the complex hydrodynamic interactions between wave-induced forces and OSWEC motion. High-fidelity numerical simulations based on solving unsteady Reynolds-averaged Navier–Stokes (RANS) equations can provide highly accurate predictions; nonetheless, such simulations are computationally expensive for exploring a large design space. Medium-fidelity numerical simulations that solve unsteady Euler equations can give reasonably accurate results, but still computationally expensive to run many cases for optimization purposes [4]. Furthermore, because of their large number of degrees of freedom, CFD simulations are not amenable for use in implementing modern control approaches that are necessary for enhancing the performance of such wave energy converters to reduce their levelized cost of energy (LCOE).

In contrast to the computationally expensive numerical simulations of the flow field generated by the waves and OSWEC response, low-fidelity models based on a simplified equation of motion can be very useful in the design, optimization, or control of OSWEC [5]. One model is the Cummins' equation [6], which is based on Newton's second law and accounts for the wave excitation forces moving the OSWEC and forces produced by the OSWEC motion. Ringwood et al. [7] noted that although effectively linear in nature, the Cummins equation is not particularly convenient for control/estimation purposes, nor for computationally efficient motion simulation and performance assessment, due to the presence of the convolution term representing the wave radiation damping. As such, a common practice among WEC control-oriented modeling is to

compute an approximation for the convolution operator in terms of a finite-dimensional linear system.

Solving the Cummins equation in the time domain is complicated and computationally demanding [8]. Different approximations have been proposed to reduce these complications. One approach is Prony's method that approximates the impulse response function in the convolution integral by a sum of complex exponential functions [9], which may not be suitable for arrays [8]. Another approach is to approximate the convolution integral by a state-space model [10]. Yu and Falnes [11] followed this approach on a vertical cylinder and replaced the convolution term by a state-space model based on the least squares curve fitting of the radiation impulse response function with a linear sub-systems. They validated the prediction of the model with experimental data. Their results showed that such an approach is accurate and convenient for time domain simulations and can be applied to ships and other marine structures as well as WECs. Taghipour et al. [12] reviewed different approaches for replacing the convolution term with a state-space model, specifically the realization theory and frequency domain regression approaches. They determined that both methods are accurate proving that state-space approximations reduce the computational time significantly in comparison to solving the convolution term. Kristiansen et al. [13] developed a high-order state-space model to replace the convolution term using the singular value decomposition (SVD) method and then discussed a procedure for model reduction to generate a low-order state-space model that can be utilized with other systems. They validated the accuracy of the low-order state-space model with some experimental data. Their approach emphasized the efficiency of utilizing the state-space model over solving the convolution integral directly. A different alternative was proposed by Sutulo and Soares [14] when they investigated replacing the total radiation force by a state-space model instead of solely replacing the convolution term. Their results showed that a low-order transfer function could simulate the response of marine structures with reasonable accuracy and high efficiency. More approaches and different applications for replacing the radiation torque by a state-space model can be found in [15–21]. Most of the studies relied on the potential flow with assuming linear wave theory, wherein waves are considered monochromatic, propagate linearly, have small amplitudes, and the response of the system is linear with the wave amplitude [22]. There is a

need to develop models that make use of higher fidelity simulations to consider effects from large amplitudes WEC oscillations.

As aforementioned, adopting linear wave theory and assuming monochromatic waves is helpful to solve the equation of motion in time domain. However, in real sea-states, the waves are not monochromatic. Moreover, the added mass moment of inertia and radiation damping coefficients are frequency-dependent, which requires evaluating their values at each frequency of the WEC oscillation. Furthermore, at high amplitudes, vortices and flow separation become more prominent, which emphasizes the possible dominance of the nonlinear damping effects over the linear radiation damping. This violates the assumptions of the linear wave theory and highlights the need to include effects of flow separation through a nonlinear damping term in the equation of motion. As such, optimizing the performance of an OSWEC in waves cannot rely on small-amplitude linear wave theory.

In this paper, a performance characterization of the hydrodynamic response of an OSWEC that is based on a reduced-order model is considered to assess the relative importance of the linear and nonlinear damping, and the applicability of using a model derived under monochromatic forcing to simulate the response under irregular forcing. Toward these objectives, a reduced-order time domain model is developed to simulate and predict the response of an OSWEC under different regular and irregular forcing conditions. Free response simulations are used to determine the coefficients of the state-space model. Torque-forced simulations are used to identify the coefficients of the nonlinear damping term for different amplitudes and wave frequencies. Wave tank tests are performed to validate the simulations and the reduced-order model. Sensitivity analysis is performed to determine the impact of variations in values of the added mass moment of inertia and the nonlinear drag coefficients on the accuracy of the predicted response. Variations of the nonlinear drag coefficient with the forcing amplitude are quantified. Response predictions under regular and irregular forcing are validated through analysis and comparison with predictions from high-fidelity simulations.

The paper is organized as follows: The numerical simulations used to generate the data for analysis and experimental setup for their validation are detailed in Sect. 2. Section 3 provides a detailed explanation of the reduced-order model, the approach to identify the coef-

ficients of all terms, and the validation of the reduced-order model under regular and irregular forcing. In Sect. 4, scaling of the linear and nonlinear damping terms of the reduced-order model is presented to quantify their relative contributions. The conclusions are presented in Sect. 5.

2 Numerical simulations

2.1 Governing equations and computational domain

The numerical simulations were performed using the commercial computational fluid dynamics (CFD) software solver ANSYS FLUENT. Here, FLUENT is used to solve the unsteady Reynolds-averaged Navier–Stokes (RANS) equations written as

$$\frac{\partial U_i}{\partial x_i} = 0 \quad (1)$$

$$\frac{\partial U_i}{\partial t} + U_j \frac{\partial U_i}{\partial x_j} = -\frac{1}{\rho} \frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j} \left(\nu \frac{\partial U_i}{\partial x_j} - \overline{u_i u_j} \right) + g_i \quad (2)$$

where u_i represents the velocity vector, U_i is the time-averaged velocity vector, $\overline{u_i u_j}$ is the Reynolds stress term that models the effects of turbulent fluctuations, ρ is the density, P is the time-averaged pressure, ν is the kinematic viscosity, and g_i is the gravitational acceleration vector.

The computational domain reflected the geometry of the wave tank of the Davidson Laboratory at Stevens Institute of Technology, where model tests were performed, except in the length where only a third of its length is simulated to reduce the computational cost. The domain's width is 5 m, depth is 3 m, and length is 35 m. This domain was discretized into non-overlapping finite volumes in which the discretized governing equations are solved to obtain the velocity field, pressure, and volume of fraction of the flow. The domain was divided to three zones as shown in Fig. 1. The left and right zones were discretized using a structured mesh (hexahedral cells), while the middle zone was discretized using unstructured mesh (tetrahedral cells). The OSWEC was set in the middle zone, and a dynamic mesh was employed to describe its rotation. The middle zone was considered as a deforming zone that changed with time. Smoothing and re-meshing methods were used to keep the quality of the updated

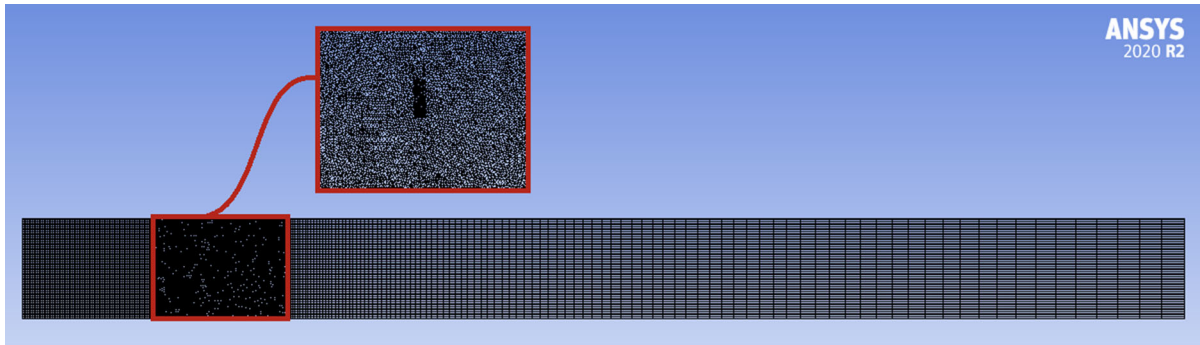


Fig. 1 Zones and meshing of the computational domain

cells, whereas the left and right zones were stationary. As shown in Fig. 1, the cell size was smallest around the OSWEC, larger in the left zone, and largest in the right zone. To ensure a grid-independent solution, simulations were carried out subsequently by refining the grid through increasing the number of cells.

The free surface is modeled using the volume of fluid (VoF) method where a single set of equations are solved for the water and air using their volume fraction represented by γ that is tracked throughout the domain. The equations for each cell are written as [23]:

$$\frac{\partial \gamma}{\partial t} + u_i \frac{\partial \gamma}{\partial x_i} = 0 \quad (3)$$

The values of γ vary between 0 and 1 with a zero value indicating that the cell is empty of the water, while a one value indicates a cell that is full of water. Values between zero and one indicate that the cell contains a free surface.

The computational domain of the simulations was subjected to outlet (downstream), atmospheric (top), bottom, and OSWEC surface boundary conditions. The free surface boundary is set to be pressure outlet to simulate the air at atmospheric pressure (Fig. 2). The sides, left, and bottom boundaries are set to be walls with no slip conditions. A numerical beach is simulated at the outlet boundary to damp the waves and prevent wave reflection. The OSWEC is under static equilibrium when it is set in its upright position.

The governing equations were solved using the coupled pressure-based solver in ANSYS FLUENT. In this solver, the system of momentum and pressure-based continuity equations are solved simultaneously [24]. The Pressure-Implicit with Splitting of Operators (PISO), which is employed in simulations involving large time steps and high degree of mesh distort-

tion [25], was chosen for pressure–velocity coupling. The body-force-weighted scheme was used for pressure discretization to include the body force in the calculations. The compressive scheme was used to obtain the face fluxes for all cells for the volume fraction equation [26]. The realizable k - ϵ model is used to model the turbulence. The second-order upwind scheme was selected for momentum discretization, and the first-order upwind scheme was selected for turbulence model. The solution was considered to converge when all the residuals are less than 10^{-3} for all variables. All simulations were conducted on a workstation with Intel(R) Xeon(R) W-2155 16 cores processor and 64GB of RAM.

2.2 OSWEC dynamics

The hydrodynamic torque T includes contributions from the excitation torque, the radiation torque, and the restoring torque due to buoyancy. The angular acceleration, $\ddot{\theta}$, is then calculated from the equation of motion:

$$I\ddot{\theta} = T \quad (4)$$

where I is the mass moment of inertia of the OSWEC. In this study, simulations and experiments of torque-excited OSWEC are used to develop the reduced-order model.

The angular velocity, $\dot{\theta}$, is derived by numerically integrating $\ddot{\theta}$ using fourth-order multi-point Adams–Moulton formulation [27]:

$$\dot{\theta}^{n+1} = \dot{\theta}^n + \frac{\Delta t}{24} (9\ddot{\theta}^{n+1} + 19\ddot{\theta}^n - 5\ddot{\theta}^{n-1} + \ddot{\theta}^{n-2}) \quad (5)$$

This velocity is used to update the dynamic mesh with the new OSWEC position.

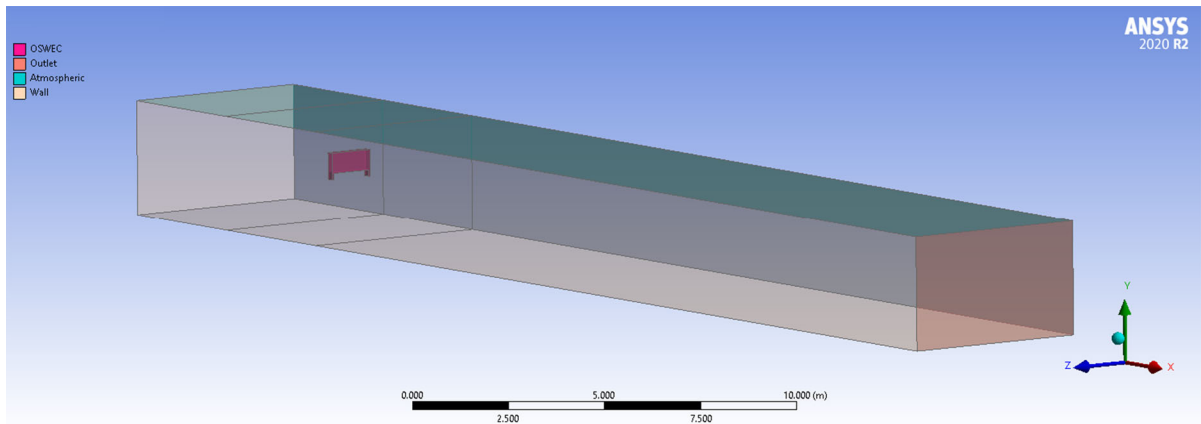


Fig. 2 Boundary conditions of the computational domain

Table 1 Dimensions and mass properties of the OSWEC

Parameter	Measured value
Width $\times$ height $\times$ thickness	1.2 $\times$ 0.7 $\times$ 0.2 m
Mass	35 kg
Mass moment of inertia	10 kg m ²
Center of gravity location from hinge	0.28 m (40%)
Hinge depth from still water level	0.57 m
OSWEC draft	0.6 m (86%)

2.3 Validation of numerical simulations

The experimental OSWEC model consisted of a rectangular aluminum frame (1.2 m wide, 0.7 m high, and 0.2 m thick) filled with foam. To adjust the center of gravity and buoyancy load, the model had an opening at the bottom that is 1.1 m wide, 0.2 m high, and 0.2 m thick (Fig. 4). The dimensions and mass properties of the model are presented in Table 1. All experiments were performed in the wave tank of the Davidson Laboratory at Stevens Institute of Technology. The tank is 100 m long, 5 m wide and 3 m deep, as shown in Fig. 3. The model was placed in the middle of the tank on a base that is 1.4 m high. The base was fixed to the tank floor. The water depth is fixed to 2 m.

Free decay and forced excitation tests were performed for validating the numerical simulations. In the free decay test, an initial displacement was applied on the OSWEC and then released. The angular displacement of the model was measured by a simple setup utilizing a linear variable differential transformer (LVDT)

sensor, a pulley, and a linear spring. As shown in Fig. 5, the pulley was connected to the shaft at the hinge. A string was used to connect this pulley to another pulley at the top of the OSWEC. The LVDT was used to measure the position of this pulley, which is related to the rotation of the OSWEC. The string was kept in tension by connecting it to a linear spring. Calibration was performed to convert the linear displacement from the LVDT sensor to the angular displacement of the OSWEC. All data were recorded with a sampling rate of 100 Hz.

In the forced excitation experiments, a periodic torque was applied at the top of the OSWEC using a motor-driven setup, as shown in Fig. 6. The motor was connected to a 1:6 gearbox, which is connected to a driving pulley. In turn, the pulley was connected to an arch fixed on top of the OSWEC using a belt. The pulley-belt transmission was exploited to amplify the output torque from the motor. The oscillating excitation torque is the motor torque amplified by the gearbox ratio and the pulley-belt transmission. A torque sensor was placed between the gearbox and the small pulley to measure the output torque from the motor. Calibration was carried out by fitting the motor encoder data and the rotation of the OSWEC to determine the amplification by the pulley-belt transmission. The rotation of the OSWEC corresponding to the excitation torque was recorded using the same LVDT sensor setup and the same sampling rate as in the free decay test.

The validation is based on comparing numerical simulations and experimental measurements of the free decay and forced OSWEC responses. Additional val-

Fig. 3 Schematic for the Davidson Laboratory high speed towing tank

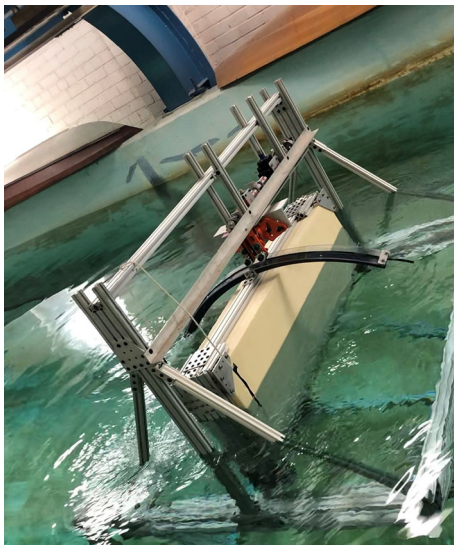
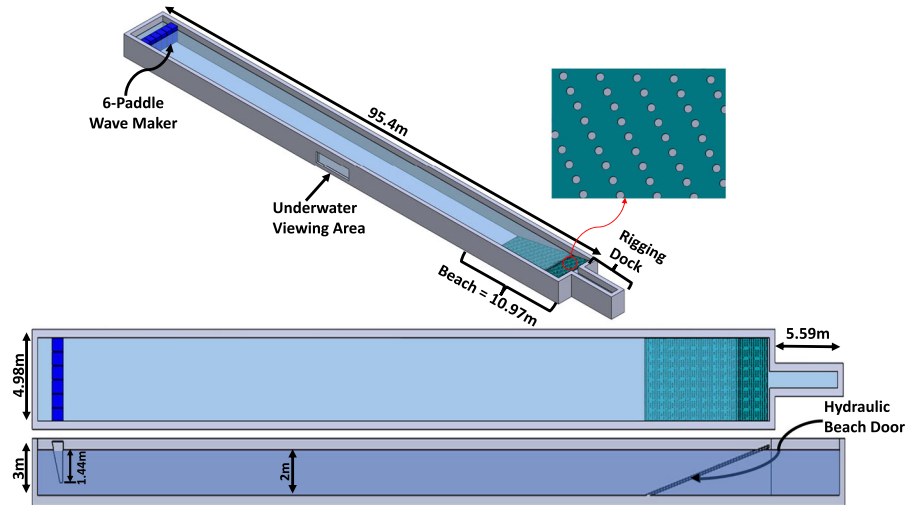


Fig. 4 OSWEC as set in the wave tank

validation is based on comparing values of the natural frequency (ω_n), damping ratio (ζ), and added mass moment of inertia (I_a) from the simulations and experiments. Figure 7 compares a plot of the OSWEC rotation obtained from a RANS simulation against experimentally measured rotation from the free decay test. The plots show a high level of agreement especially during the first four cycles. The subsequent small discrepancies are most likely due to the difference in length between the numerical domain and the wave tank. A comparison of the experimentally measured and simulated values of natural frequency, damping ratio, added

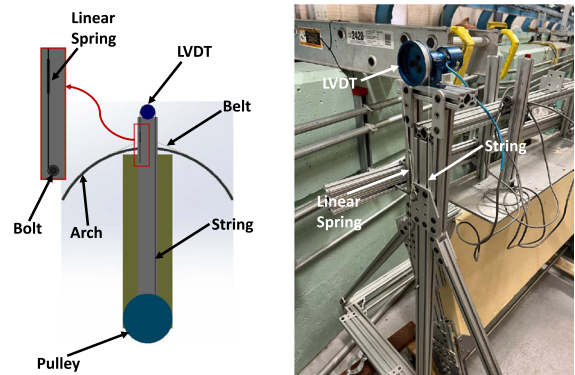


Fig. 5 Schematic for the LVDT sensor on the OSWEC in the wave tank

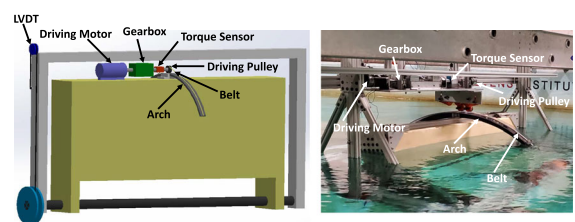


Fig. 6 Schematic for the motor setup on top of the OSWEC in the wave tank

mass moment of inertia, and radiation damping coefficient is presented in Table 2. In the free decay response, the difference in the amplitudes of the peaks is an indicator to a difference in the damping in the system. When the predicted damping ratio from CFD was compared to the experimental one, an error of 1.41% was found. This error percentage is reasonable and acceptable. The

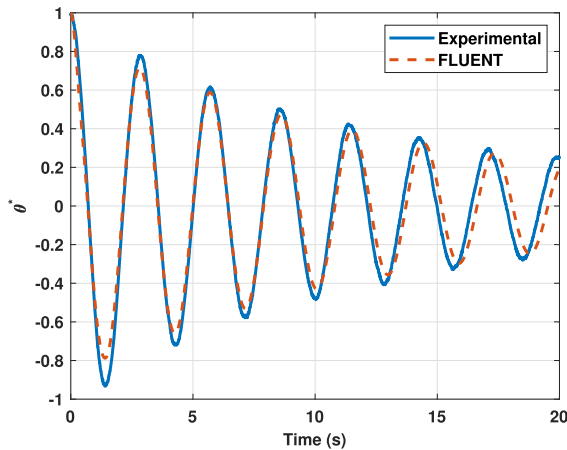


Fig. 7 Comparison of time series from free decay test and high-fidelity simulation

difference in the phase here could be considered as an error in the prediction of the damping period, which is also related to the damping ratio, or could be due to a slight variation in the stiffness. The errors in all determined values are less than 2% indicating a high level of agreement.

Further validation is carried out by comparing amplitudes of OSWEC rotation under regular torque forcing. A series of experiments and simulations covering different values of excitation frequencies and excitation torque amplitudes were performed. As an example, we compare in Fig. 8 the OSWEC response as determined from the RANS numerical simulation with the one measured in the wave tank for an excitation frequency of 1.73 rad/s, 2.09 rad/s, and 2.33 rad/s with excitation torque amplitude of 62 N m, 23 N m, and 20 N m, respectively. The plots show less than 3% difference in the steady-state amplitude of the response validating the performance of the numerical simulations in predicting the response amplitude. The results show that FLUENT can predict the response of the OSWEC under different conditions with high accuracy, which concludes the reliability of using numerical simulations to develop the reduced-order model.

3 Reduced-order model

3.1 Governing equation

The response of the oscillating surge wave energy converter is determined from the hydrodynamic moments

by applying Newton's second law written as:

$$I\ddot{\theta} = T_{\text{exc}} + T_{\text{rad}} + T_{\text{res}} + T_{\text{visc}} \quad (6)$$

where I is the mass moment of inertia, $\ddot{\theta}$ is the angular acceleration, T_{exc} is the excitation torque, T_{rad} is the radiation torque, T_{res} is the restoring torque, and T_{visc} is a damping torque due to viscous effects, including flow separation. The radiation torque, which accounts for the change in the angular momentum of the fluid due to the flap motion, is written as the sum

$$T_{\text{rad}} = -I_a(\infty)\ddot{\theta} - \int_0^t h_{\text{rad}}(t - \tau)\dot{\theta}(\tau) d\tau \quad (7)$$

where $I_a(\infty)$ is the constant positive infinite-frequency added mass moment of inertia, $\dot{\theta}$ is the angular velocity, and $h_{\text{rad}}(t)$ is the impulse response function. The convolution integral represents energy loss (damping) associated with radiated waves generated by the flap motion. The restoring torque is generated by the net buoyancy moment, which is proportional to the position of the flap (θ) and is written as:

$$T_{\text{res}} = -k\theta \quad (8)$$

where k is the stiffness coefficient. The viscous damping torque is represented by a nonlinear term similar to that of Morison's equation and written as:

$$T_{\text{visc}} = -C_D\dot{\theta}|\dot{\theta}| \quad (9)$$

where C_D is the nonlinear damping coefficient. Substituting Eqs. 7, 8, and 9 in Eq. 6 and rearranging the terms yields

$$(I + I_a(\infty))\ddot{\theta} + \int_0^t h_{\text{rad}}(t - \tau)\dot{\theta}(\tau) d\tau + C_D\dot{\theta}|\dot{\theta}| + k\theta = T_{\text{exc}} \quad (10)$$

The convolution term represents a memory effect and its computation requires storing all past information. A more efficient representation of the radiation damping, in terms of computational cost, is obtained by replacing the convolution term with a two-equation state-space model. The first is a subsystem of equations that define the derivatives of the state variables as a weighted sum of the state variables and the system input $\dot{\theta}$. This subsystem consists of a set of n coupled first-order linear differential equations with constant coefficients written as:

$$\dot{X}_{\text{rad}}(t) = A_{\text{rad}}X_{\text{rad}}(t) + B_{\text{rad}}\dot{\theta} \quad (11)$$

where $X_{\text{rad}}(t)$ is the state vector, A_{rad} is an $n \times n$ square matrix of constant coefficients, and B_{rad} is an $n \times 1$

Table 2 Comparison between numerical and experimental measurements for the hydrodynamic coefficients from the free decay test

	Experiment	RANS simulation	Error (%)
Natural frequency (ω_n)	2.215 rad/s	2.221 rad/s	0.29
Damping ratio (ζ)	0.0395	0.0400	1.41
Added mass moment of inertia coefficient (I_a)	49.10 kg m ²	48.75 kg m ²	0.70
Radiation damping coefficient (C_{rad})	10.34 Nm s	10.45 Nm s	1.1

matrix of coefficients that weigh the input. When integrated, this set of differential equations yields the state variables

$$X_{\text{rad}}(t) = \left[e^{A_{\text{rad}}t} X_0 + \int_0^t e^{A_{\text{rad}}(t-\tau)} B_{\text{rad}} \dot{\theta}(\tau) d\tau \right] \quad (12)$$

The second equation of the state-space model relates the output (Y_{rad}), which in this case is the radiation damping represented by the convolution term, to the state variables. It is written as

$$Y_{\text{rad}}(t) = \int_0^t h_{\text{rad}}(t-\tau) \dot{\theta}(\tau) d\tau = C_{\text{rad}} X_{\text{rad}}(t) \quad (13)$$

where C_{rad} is a $1 \times n$ matrix. Substituting Eq. 12 into Eq. 13 yields

$$Y_{\text{rad}}(t) = C_{\text{rad}} e^{A_{\text{rad}}t} X_0 + \int_0^t C_{\text{rad}} \left[e^{A_{\text{rad}}(t-\tau)} B_{\text{rad}} \dot{\theta}(\tau) d\tau \right] \quad (14)$$

Assuming a causal input signal, we set $\dot{\theta}(t) = 0$ at $t = 0$. Therefore, the system has no output and the initial state vector denoted by X_0 is equal to zero, yielding Eq. 14 to:

$$Y_{\text{rad}}(t) = \int_0^t C_{\text{rad}} \left[e^{A_{\text{rad}}(t-\tau)} B_{\text{rad}} \dot{\theta}(\tau) d\tau \right] \quad (15)$$

Comparing Eqs. 13 and 15, the state-space model for a causal impulse response function is represented by:

$$h_{\text{rad}}(t) = C_{\text{rad}} e^{A_{\text{rad}}t} B_{\text{rad}} \quad (16)$$

The coefficients in Eq. 16, A_{rad} , B_{rad} and C_{rad} , will be determined by comparing the state-space approximation for the impulse response function to the true impulse response of the OSWEC using experimental free decay response.

The equation of motion using the state-space representation is then written as:

$$(I + I_a(\infty))\ddot{\theta} + C_{\text{rad}} X_{\text{rad}}(t) + C_D \dot{\theta} |\dot{\theta}| + k\theta = T_{\text{exc}} \quad (17)$$

3.2 System identification

To identify the state-space matrices A_{rad} , B_{rad} , and C_{rad} in terms of order and values, we follow the “Impulse Response Curve Fitting” approach with the objective of generating an impulse response similar to that of the true system. Such an approximation is performed by minimizing the least-square-error between the true radiation damping impulse response and the one estimated from the state-space model. This error is given by

$$\text{Error} = \sum_1^m g(t) [h_{\text{rad}}(t) - C_{\text{rad}} e^{A_{\text{rad}}t} B_{\text{rad}}]^2 \quad (18)$$

where $g(t)$ is a weight function, $h_{\text{rad}}(t)$ is the true impulse response function, and A_{rad} , B_{rad} , and C_{rad} are the system matrices as defined above. Below, we set the weight function $g(t)$ equal to one. Different representations can be used for the system matrices. Here, we use the companion-form-realization, where the matrices are set in the form

$$A_{\text{rad}} = \begin{bmatrix} 0 & 0 & 0 & \dots & -a_1 \\ 1 & 0 & 0 & \dots & -a_2 \\ 0 & 1 & 0 & \dots & -a_3 \\ \vdots & \vdots & \ddots & & \vdots \\ 0 & 0 & \dots & 1 & -a_n \end{bmatrix} \quad B_{\text{rad}} = \begin{bmatrix} b_1 \\ b_2 \\ \vdots \\ b_n \end{bmatrix}$$

$$C_{\text{rad}} = [0 \quad 0 \quad 0 \quad \dots \quad 0 \quad 1]$$

For a subsystem of order n , the number of unknown parameters to be estimated is $2n$. The true impulse response function, $h_{\text{rad}}(t)$, can be obtained experimentally by measuring the response of the OSWEC when subjected to an impulse input. This input consists of a small initial displacement applied on the OSWEC and yields a free decay signal. The recorded data from the free decay test from wave tank experiments is considered as the true impulse response. Testing second- and fourth-order subsystem representations, it was determined that a second-order subsystem gives a better fit.

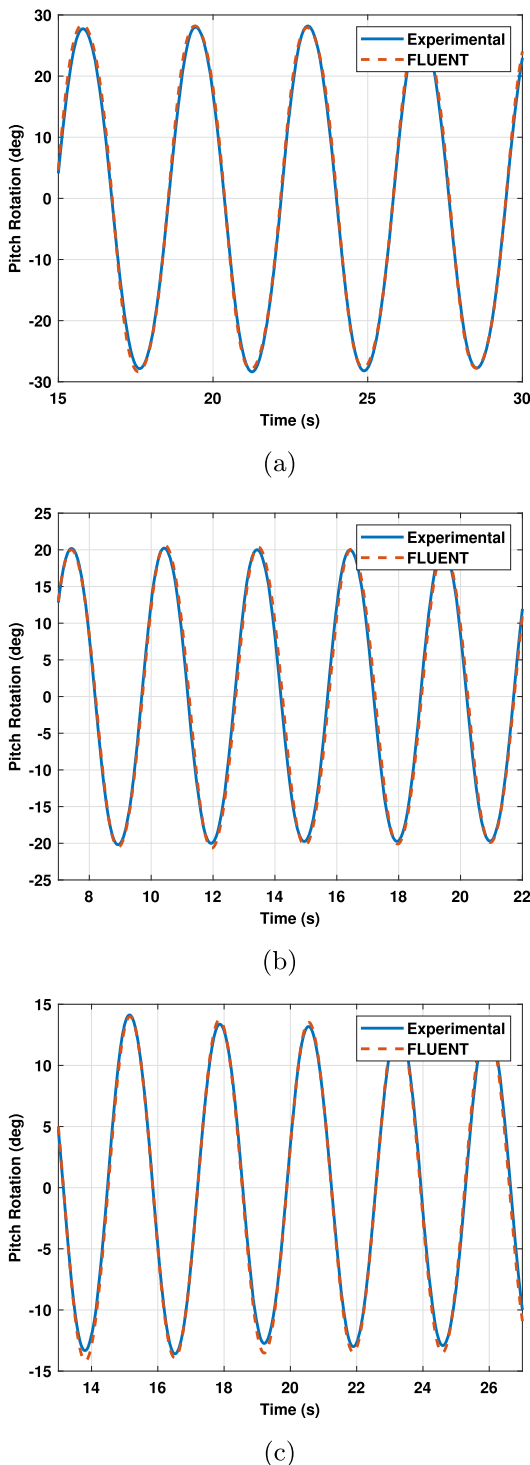


Fig. 8 Comparison of the OSWEC response from the numerical simulation with experiment under regular forcing with **a** 1.73 rad/s excitation frequency and 62 N m excitation torque amplitude **b** 2.09 rad/s excitation frequency and 23 N m excitation torque amplitude **c** 2.33 rad/s excitation frequency and 20 N m excitation torque amplitude with no excitation waves in the tank or numerical simulations

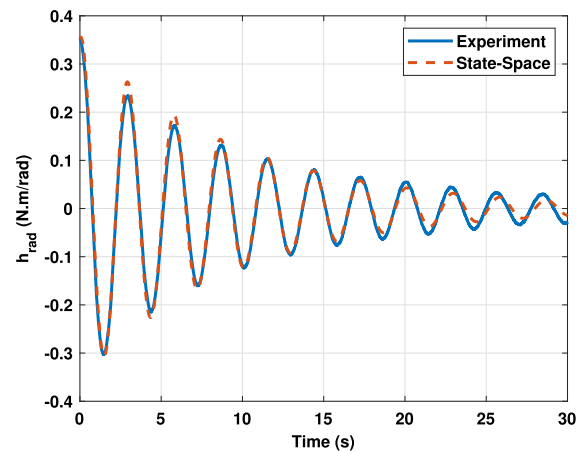


Fig. 9 Comparison between the impulse response from experiment and state-space model

The only issue with minimizing the error is the initial conditions as the accuracy of the results is affected by their values. As such, random initial conditions were assumed for different trials to get a close estimate for the parameters. Using the impulse response function, the approximation matrices were identified as:

$$A_{\text{rad}} = \begin{bmatrix} 0 & -4.83 \\ 1 & -0.21 \end{bmatrix} \quad B_{\text{rad}} = \begin{bmatrix} 0.17 \\ 0.35 \end{bmatrix} \quad C_{\text{rad}} = [0 \ 1]$$

A comparison of the impulse response function as obtained from the measurements and as estimated from the state-space model is presented in Fig. 9. The plots show that the identified state-space model can accurately predict the impulse response of the OSWEC. Under harmonic excitation, the added mass moment of inertia and nonlinear damping coefficients are functions of frequency. We perform their identification from regular forcing excitation over a broad range of excitation frequencies. Under harmonic torque excitation, Eq. 10 is written as [28]:

$$(I + I_a(\omega))\ddot{\theta} + C_r(\omega)\dot{\theta} + C_D(\omega)\dot{\theta}|\dot{\theta}| + k\theta = T_{\text{exc}} = T_o \sin(\omega t + \phi) \quad (19)$$

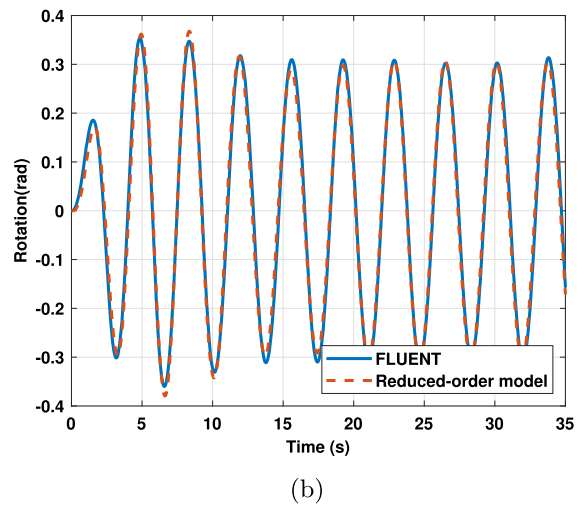
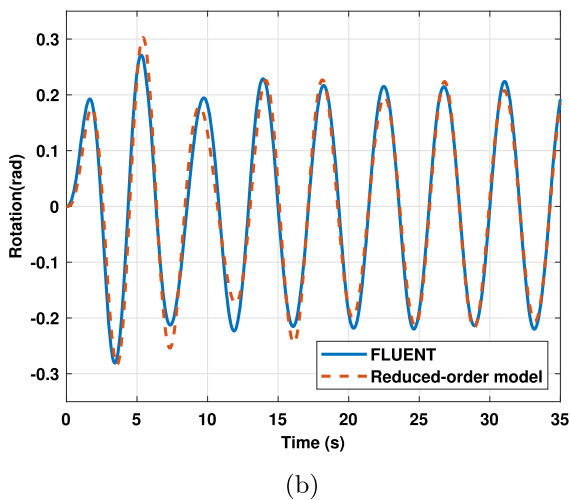
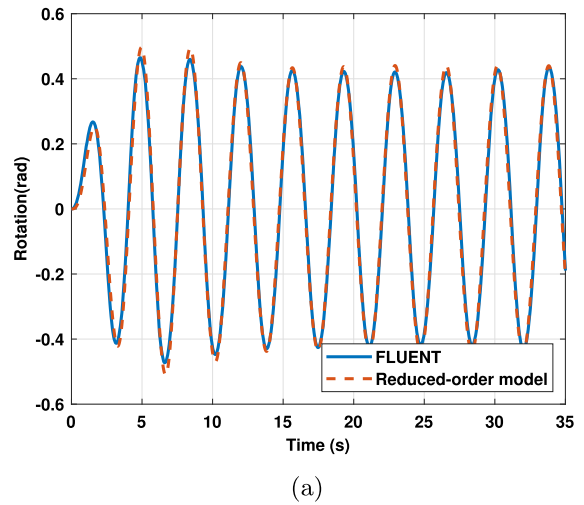
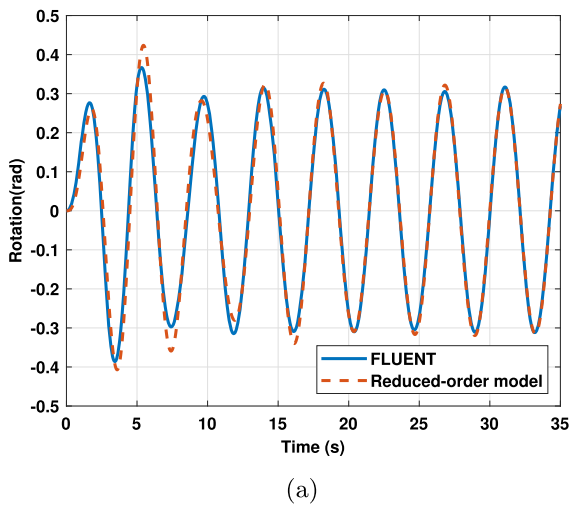
where C_r is the linear damping coefficient. Expressing the response as $\theta = \theta_o \cos(\omega t)$ and using the first term in the Fourier series expansion of the nonlinear damping term, the coefficients in Eq. (19) are related by:

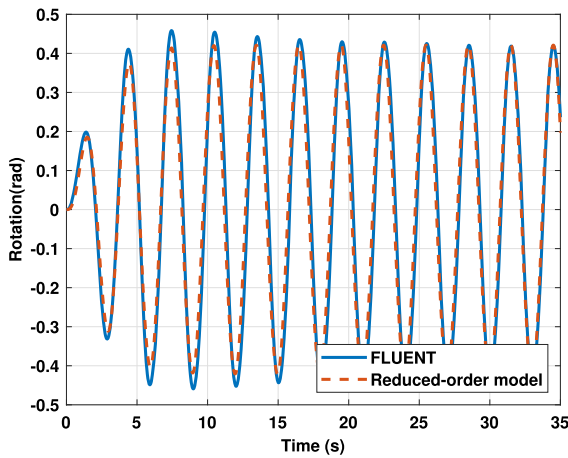
$$I_a = \frac{k}{\omega^2} - \frac{T_o \sin(\phi)}{\theta_o \omega^2} - I \quad (20)$$

$$\frac{8}{3\pi} C_D \omega \theta_o + C_r = \frac{T_o \cos(\phi)}{\theta_o \omega} \quad (21)$$

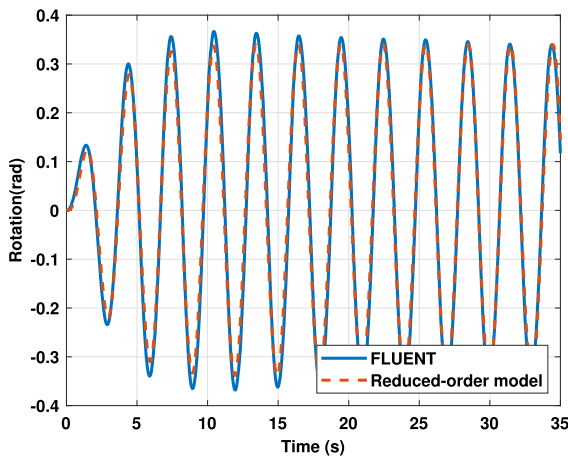
Table 3 Identified hydrodynamic coefficients at different frequencies

Parameter	ω (rad/s), ω/ω_n			
	1.73, 0.78	2.09, 0.95	2.33, 1.06	2.65, 1.2
Added mass moment of inertia coefficient (I_a)(kg m ²)	45	51	55	57
Total mass moment of inertia ($I + I_a$)(kg m ²)	55	61	65	67
Radiation damping coefficient (C_r)(N m s)	1.67	5.29	6.19	23.21
Damping ratio (ζ)	0.0066	0.020	0.022	0.083
Nonlinear damping coefficient (C_D)(N m s ²)	33.49	41.27	47.91	50.00

**Fig. 10** Comparison of the OSWEC response from the numerical simulations with the reduced-order model under regular forcing with 1.47 rad/s excitation frequency and **a** 45 N m, **b** 30 N m excitation torque amplitudes**Fig. 11** Comparison of the OSWEC response from the numerical simulations with the reduced-order model under regular forcing with 1.73 rad/s excitation frequency and **a** 45 N m, **b** 30 N m excitation torque amplitudes



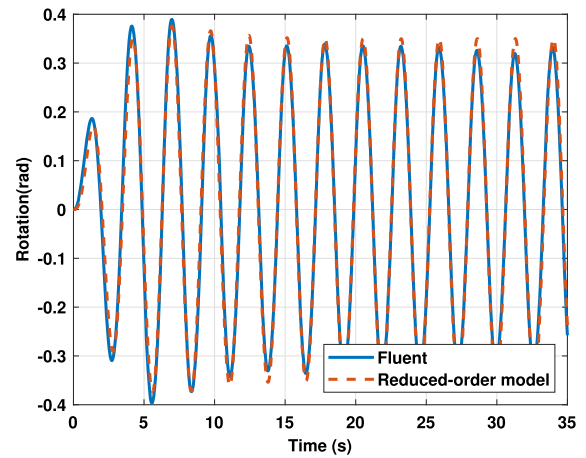
(a)



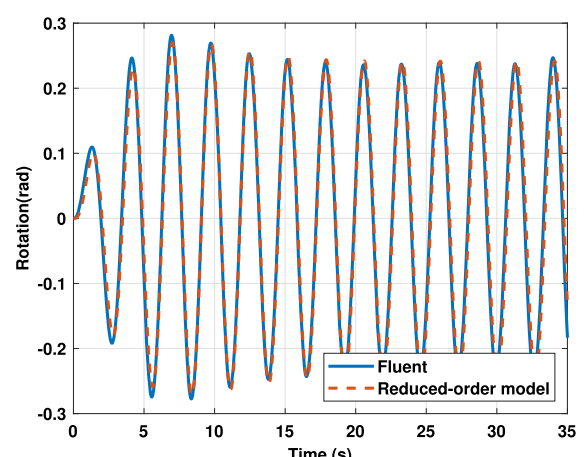
(b)

Fig. 12 Comparison of the OSWEC response from the numerical simulations with the reduced-order model under regular forcing with 2.09 rad/s excitation frequency and **a** 35 N m, **b** 23 N m excitation torque amplitudes

To identify the coefficients, we perform numerical simulations by applying a regular torque of known amplitude and frequency on the flap without the presence of waves. The simulations were performed at four frequencies near the estimated flap's natural frequency, $\omega_n = 2.2$ rad/s. For each frequency, we performed five simulations at different excitation amplitudes. Using the amplitude of the excitation torque T_o , response amplitude θ_o , phase difference ϕ , and Eqs. 20 and 21, we performed curved fitting and identified the coefficients I_a , C_r and C_D . The stiffness coefficient was calculated as $k = 290$ N m/rad and validated from quasi-static experiments in the wave tank experiments. The



(a)



(b)

Fig. 13 Comparison of the OSWEC response from the numerical simulations with the reduced-order model under regular forcing with an 2.33 rad/s excitation frequency and **a** 35 N m, **b** 20 N m excitation torque amplitudes

identified coefficients as a function of the excitation frequency are presented in Table 3. The numbers show that the added mass moment of inertia increases as the excitation frequency is increased and assumes values that are between five and six times the dry mass moment of inertia. The damping coefficient also increases as the frequency is increased. When represented as a damping ratio, it varies from 0.7% in the low end of the range of considered excitation frequencies, to 2% in the middle of that range and higher values of 8.3% at the highest considered frequency. The nonlinear damping coefficient increases by about 50% as the excitation

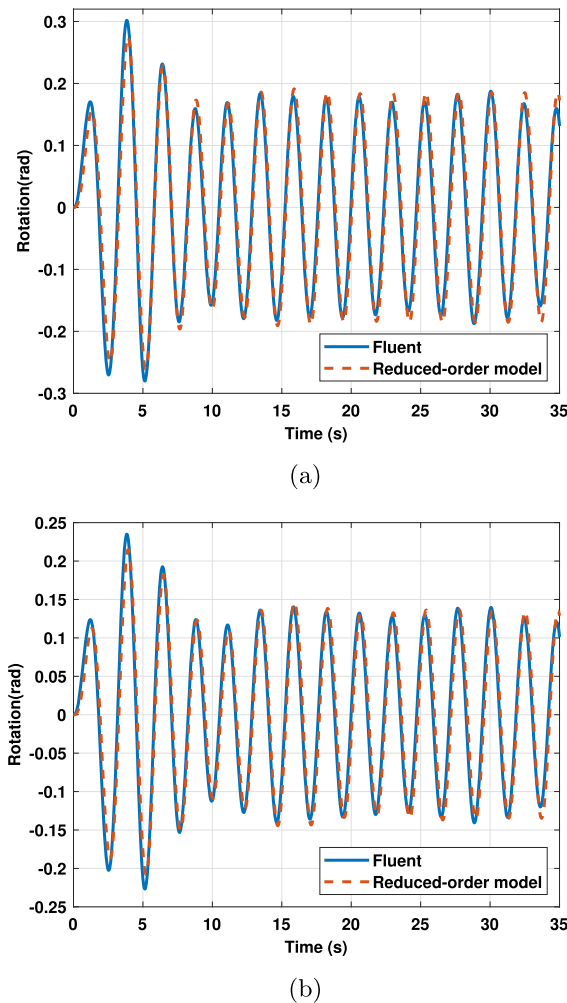


Fig. 14 Comparison of the OSWEC response from the numerical simulations with the reduced-order model under regular forcing with 2.65 rad/s excitation frequency and **a** 35 N m, **b** 25 N m excitation torque amplitudes

frequency is increased over the considered range of excitation frequencies.

The validity of the state-space model, presented in Eqs. 17 and 11, and its coefficients as identified above, is determined by comparing its predictions against those of the FLUENT simulations. Different cases of excitation amplitudes (between 20 and 45 N m) and frequencies (between 1.47 and 2.65 rad/s) are considered. The subset of compared time series presented in Figs. 10, 11, 12, 13 and 14 shows good agreement. The extent of this agreement is demonstrated in Table 4, which compares the root-mean-square (RMS) values from the reduced-order model and FLUENT simula-

tions. The numbers show a maximum error of 5%, which is acceptable for a reduced-order model that reduces the computational cost from few days to few minutes.

Still, for efficient implementation of the state-space model to predict the response under irregular wave excitation, there is a need to determine one set of values for the coefficients that are otherwise frequency-dependent. As noted from Table 3, C_D varies over a range between 30 and 50 N m s² and I_a varies between 50 and 57 kg m². Using these ranges, we conducted numerical simulations over a broad range of excitation frequencies between 1.47 and 2.65 rad/s and applied torque excitation amplitudes over a range between 5 and 45 N m. Response amplitudes from the reduced-order model assuming three C_D values, namely 30, 40 and 50 N m s² and three I_a values, namely 51, 54 and 57 kg m², were evaluated against predictions from the numerical simulations. The percentage errors between these predictions are presented in Table 5. The numbers show that the error depends on the torque amplitude and frequency. When I_a is set equal to 51 kg m² or 54 kg m², the errors are significant, especially for $\omega = 2.65$ rad/s. Setting I_a to 57 kg m², the errors are in general reduced, especially when C_D is set equal to 50 N m s². For these values, all errors except at the lowest torque values of 5 N m are less than 10%. Given that the amplitudes are small at these torque values, it was decided to use $C_D = 50$ N m s² and I_a to 57 kg m² to validate the performance of the reduced-order model under irregular wave excitation. Based on the current results, these values are valid over an excitation frequency range of 1.47 to 2.56 rad/s and excitation amplitudes larger than 5 N m.

After tuning the hydrodynamic coefficients, the reduced-order model is evaluated under spectrum of excitation frequencies defined by a significant frequency. The results are compared with the numerical results generated from numerical simulations under the same conditions. The same range of frequency used in testing the regular forcing was also used to test irregular forcing. In all the numerical simulations, the values for C_D and I_a were fixed to 50 N m s² and 57 kg m², respectively. The input for the reduced-order model and the numerical simulations is an irregular torque signal for different peak frequencies. To generate these signals, we used a spectral representation of ocean waves. We replaced the significant wave height (H_s) by a significant torque amplitude. The Pierson–Moskowitz spec-

Table 4 Comparison between the rotation RMS from FLUENT and reduced-order model under regular forcing

ω (rad/s)	T_0 (Nm)	FLUENT (rad)	Reduced-order model (rad)	Error %
1.47	45	0.2199	0.2262	2.82
	30	0.1550	0.1526	0.70
1.73	45	0.3087	0.3109	0.72
	30	0.2196	0.2143	2.42
2.09	35	0.3068	0.2959	3.53
	23	0.2584	0.2385	5.01
2.33	35	0.2404	0.2473	2.89
	20	0.1722	0.1712	0.60
2.65	35	0.1250	0.1305	4.38
	25	0.0938	0.0954	2.48

Table 5 Percentage errors of predictions of reduced-order model with fixed coefficients values

$I_a = 51$																			
$C_D=30$	15.9	13.0	8.6	1.0	5.1	5.6	-16.8	-20.7	-27.6	-29.1	-39.0	-38.3	-36.9	-33.0	-34.7				
$C_D=40$	16.0	13.2	8.9	12.1	7.4	9.1	-7.2	-6.1	-11.5	-24.1	-24.7	-22.4	-36.8	-30.7	-30.6				
$C_D=50$	16.1	13.3	9.2	13.4	9.8	12.6	0.6	4.3	-0.4	-19.1	-14.0	-10.9	-36.6	-28.0	-26.6				
$T_0 = 15 T_0 = 30 T_0 = 45 T_0 = 30 T_0 = 45 T_0 = 62 T_0 = 5 T_0 = 23 T_0 = 35 T_0 = 5 T_0 = 20 T_0 = 35 T_0 = 5 T_0 = 25 T_0 = 35$																			
$\omega = 1.47$				$\omega = 1.73$				$\omega = 2.09$				$\omega = 2.33$				$\omega = 2.65$			
$I_a = 54$																			
$C_D=30$	12.5	9.4	4.8	3.6	-2.1	-0.8	-41.5	-26.0	-31.2	3.2	-22.0	-27.5	-19.0	-16.7	-19.1				
$C_D=40$	12.4	9.5	5.1	5.2	1.0	3.8	-24.1	-9.5	-13.8	-4.4	-12.6	-15.0	-18.8	-15.4	-16.8				
$C_D=50$	12.5	9.7	5.6	7.0	4.2	8.3	-11.8	1.9	-1.9	6.0	-4.8	-5.5	-18.7	-13.9	-14.3				
$T_0 = 15 T_0 = 30 T_0 = 45 T_0 = 30 T_0 = 45 T_0 = 62 T_0 = 5 T_0 = 23 T_0 = 35 T_0 = 5 T_0 = 20 T_0 = 35 T_0 = 5 T_0 = 25 T_0 = 35$																			
$\omega = 1.47$				$\omega = 1.73$				$\omega = 2.09$				$\omega = 2.33$				$\omega = 2.65$			
$I_a = 57$																			
$C_D = 30$	8.2	5.4	0.6	-5.0	-10.1	-7.6	-48.1	-27.2	-31.9	22.9	-4.7	-15.7	-4.6	-3.6	-6.3				
$C_D=40$	8.4	5.6	1.1	-2.6	-5.8	-1.6	-28.4	-10.2	-14.3	23.7	0.6	-6.4	-4.9	-2.9	-5.0				
$C_D=50$	8.6	5.8	1.6	0.0	-1.5	3.9	-14.9	1.4	-2.2	24.4	5.6	1.1	-4.9	-2.1	-3.4				
$T_0 = 15 T_0 = 30 T_0 = 45 T_0 = 30 T_0 = 45 T_0 = 62 T_0 = 5 T_0 = 23 T_0 = 35 T_0 = 5 T_0 = 20 T_0 = 35 T_0 = 5 T_0 = 25 T_0 = 35$																			
$\omega = 1.47$				$\omega = 1.73$				$\omega = 2.09$				$\omega = 2.33$				$\omega = 2.65$			

Error is based on a comparison with predictions from high-fidelity simulations

Table 6 Comparison between the rotation RMS from FLUENT and reduced-order model under irregular forcing

ω_p (rad/s)	T_{rms} (Nm)	FLUENT (rad)	Reduced-order model (rad)	Error %
1.47	30	0.2298	0.2250	2.09
1.73	30	0.2409	0.2350	2.44
2.09	30	0.2275	0.2217	2.54
2.33	30	0.2088	0.2028	2.91
2.65	30	0.1676	0.1652	1.45

trum was used to generate the irregular torque excitation. It is defined by an amplitude T_{oirr} and a peak frequency ω_p and is written as:

$$T_{\text{excirr}}(\omega) = \frac{5}{16} \frac{T_{\text{oirr}}^2 \omega_p^4}{\omega^5} e^{\left(-\frac{5\omega_p^4}{4\omega^4}\right)} \quad (22)$$

Five irregular torque signals were generated with the same significant torque amplitude but different five peak frequencies (Fig. 15). The signals were then used as the excitation input to the reduced-order model and high-fidelity numerical simulations. It is important to note here that while the high-fidelity simulations required 17 days to conduct one run under irregular conditions, the reduced-order model took only 13 min.

Table 6 shows the RMS of the applied irregular torque (T_{rms}) and compares the RMS values of the rotation of the OSWEC from the reduced-order model and FLUENT simulations under that torque. Figure 16 shows a comparison between results from the reduced-order model and numerical simulations. The error is less than 3% which indicates a high level of agreement.

4 Characterization and scaling of torque components

The above results emphasize the need to include the nonlinear damping term in the reduced-order model. In order to quantify the relative contribution of each term in the equation of motion (Eq. 19) toward the response, a scaling is applied to all components. The derived non-dimensional coefficients give better insight about the relative importance of each component under different excitation conditions. The magnitudes of each component from FLUENT simulations and reduced-order model are presented in Tables 7, 8 and 9. The numbers show that the largest contributions to the response are associated with the inertia and restoring terms. At low forcing amplitudes, the nonlinear term is about the same order in magnitude as the linear term. However, at larger amplitudes, the nonlinear term is about one order higher.

For a simple linear mass damper system, the critical damping condition and natural frequency of a system are defined as [29]:

$$C_{\text{cr}} = 2\sqrt{Ik} \quad (23)$$

$$\omega_n = \sqrt{k/I} \quad (24)$$

The linear damping ratio (ζ) is defined as the ratio between the linear damping of the system and its critical

damping (Eq. 25).

$$\zeta = \frac{C_r}{C_{\text{cr}}} = \frac{C_r}{2\sqrt{Ik}} \quad (25)$$

Because the OSWEC is excited under different frequencies, we define the frequency ratio (ω_r) as the ratio between the excitation frequency (ω_e) and that of free oscillations (ω_n) and write

$$\omega_r = \frac{\omega_e}{\omega_n} = \frac{\omega_e}{\sqrt{k/I}} = \frac{\omega_e}{2.08} \quad (26)$$

The nonlinear damping term should be expected to be a function of the frequency and amplitude of the oscillations. As the amplitude increases, the strength of the shed vortices due to flow separation at the edges of the flap should increase, which means an increase in the drag and energy loss that must be accounted for in the contribution of the nonlinear damping term to the response. An assessment of this qualitative description can be obtained by deriving a relationship between the nonlinear damping coefficient (C_D) and the torque amplitude (T_o). To derive this relation, we introduce the two non-dimensional variables for time and rotation as:

$$\bar{t} = \frac{t}{T_o} \quad (27)$$

$$\bar{\theta} = \frac{\theta}{\theta_0} \quad (28)$$

Substituting the two new variables in Eq. 19, we obtain

$$\begin{aligned} (I + I_a) \frac{\theta_0}{T_o^2} \frac{d^2 \bar{\theta}}{d\bar{t}^2} + C_r \frac{\theta_0}{T_o} \frac{d\bar{\theta}}{d\bar{t}} + C_D \frac{\theta_0^2}{T_o^2} \frac{d\bar{\theta}}{d\bar{t}} \left| \frac{d\bar{\theta}}{d\bar{t}} \right| \\ + k\theta_0 \bar{\theta} = T_o \sin(\omega_r \bar{t} + \phi) \end{aligned} \quad (29)$$

From Eq. 24, T_o can be defined as

$$T_o^2 = \frac{I}{k} \quad (30)$$

and θ_0 can be defined as

$$\theta_0 = \frac{T_o}{k} \quad (31)$$

Substituting Eqs. 30 and 31 into Eq. 29, the non-dimensionalized equation of motion is written as:

$$\ddot{\bar{\theta}} + 2\zeta \dot{\bar{\theta}} + 2\beta \dot{\bar{\theta}} |\dot{\bar{\theta}}| + \bar{\theta} = \sin(\omega_r \bar{t} + \phi) \quad (32)$$

where β is a non-dimensional coefficient representing the nonlinear damping and is given by

$$\beta = \frac{T_o C_D}{2Ik} = 0.0013 T_o \quad (33)$$

The above system identification shows that the linear damping ratio ζ is about 0.02 (2%) regardless

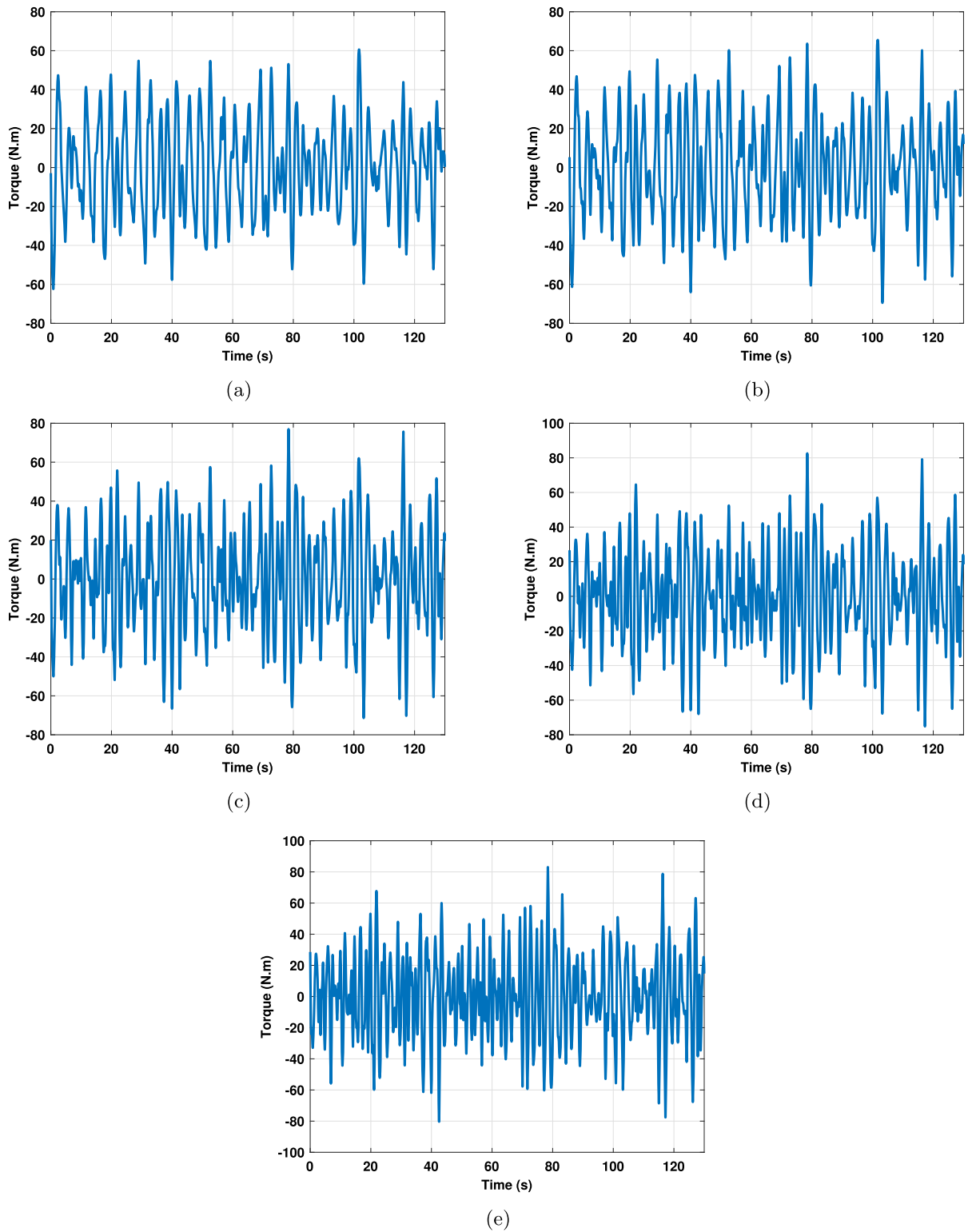


Fig. 15 Input torque for irregular forcing with a significant frequency of **a** 1.47 rad/s, **b** 1.73 rad/s, **c** 2.09 rad/s, **d** 2.33 rad/s, and **e** 2.65 rad/s

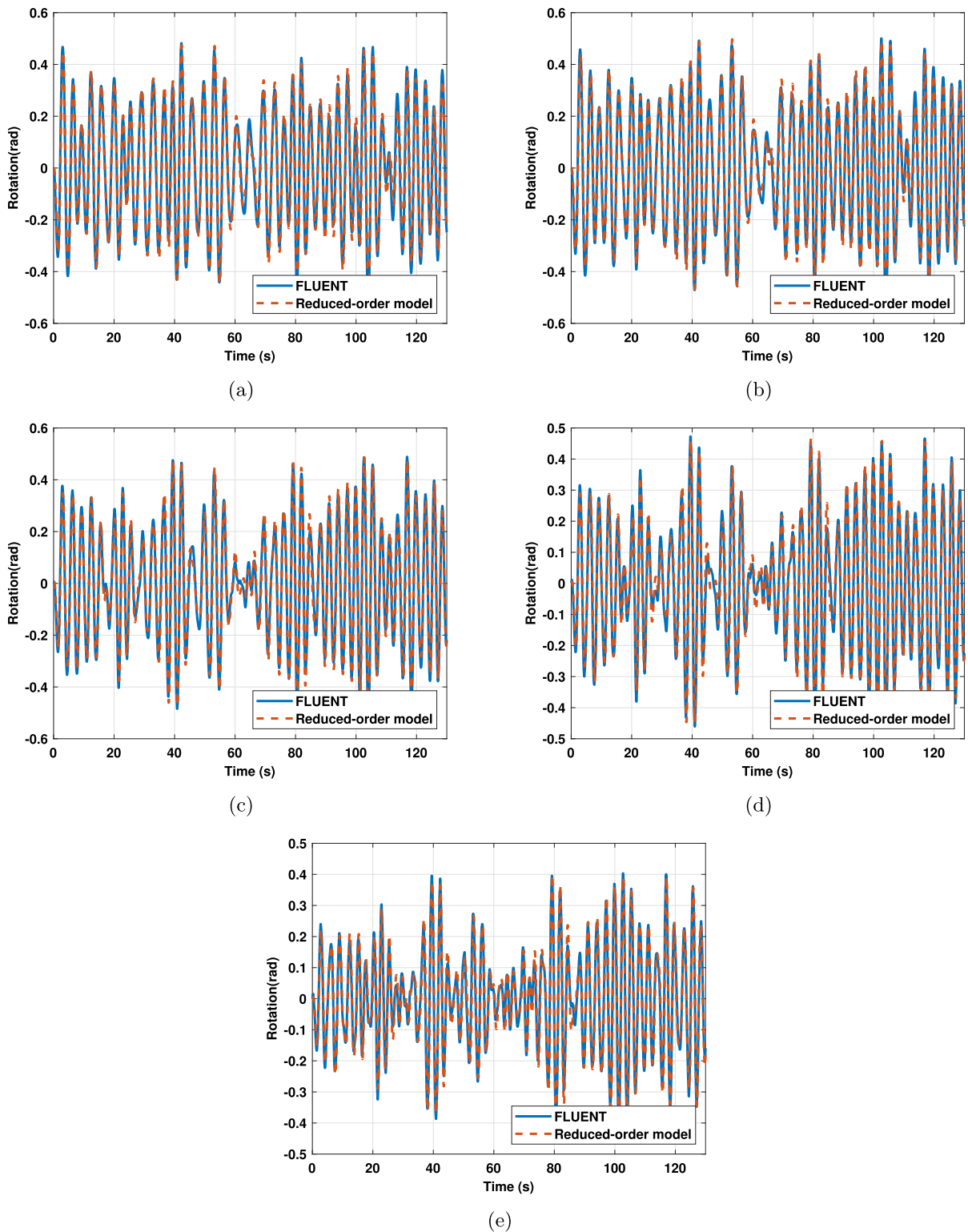


Fig. 16 Comparison of the OSWEC response from the numerical simulations with the reduced-order model under irregular forcing with a significant frequency of **a** 1.47 rad/s, **b** 1.73 rad/s, **c** 2.09 rad/s, **d** 2.33 rad/s, and **e** 2.65 rad/s

Table 7 Torque components at $\omega = 1.73$ rad/s from FLUENT (top) and reduced-order model (bottom)

T (Nm)	θ (rad)	$\dot{\theta}$ (rad/s)	$\ddot{\theta}$ (rad/s ²)	$I_t \ddot{\theta}$ (Nm)	$C_r \dot{\theta}$ (Nm)	$C_D \dot{\theta} \dot{\theta}$ (Nm)	$k \theta$ (Nm)	Total damping	I_t (kg m ²)	55
62	0.487	0.842	1.457	97.635	1.407	23.769	141.201	25.176	C_r (Nm s)	1.67
45	0.425	0.736	1.273	85.283	1.229	18.135	123.337	19.364	C_D (Nm s ²)	33.5
30	0.308	0.533	0.923	61.822	0.891	9.530	89.407	10.421	k (Nm/rad)	290
T (Nm)	θ (rad)	$\dot{\theta}$ (rad/s)	$\ddot{\theta}$ (rad/s ²)	$I_t \ddot{\theta}$ (Nm)	Y_{rad} (Nm)	$C_D \dot{\theta} \dot{\theta}$ (Nm)	$k \theta$ (Nm)	Total Damping		
62	0.543	0.949	1.742	116.773	0.322	45.541	159.208	45.863	I_t (kg m ²)	67
45	0.432	0.742	1.336	89.538	0.251	27.456	125.202	27.707	C_D (Nm s ²)	50
30	0.308	0.519	0.922	62.623	0.179	14.192	89.234	14.371	k (Nm/rad)	290

Table 8 Torque components at $\omega = 2.09$ rad/s from FLUENT (top) and reduced-order model (bottom)

T (Nm)	θ (rad)	$\dot{\theta}$ (rad/s)	$\ddot{\theta}$ (rad/s ²)	$I_t \ddot{\theta}$ (Nm)	$C_r \dot{\theta}$ (Nm)	$C_D \dot{\theta} \dot{\theta}$ (Nm)	$k \theta$ (Nm)	Total damping	I_t (kg m ²)	61
35	0.425	0.889	1.858	124.499	4.712	32.623	123.366	37.335	C_r (Nm s)	5.3
15	0.281	0.588	1.228	82.268	3.114	14.245	81.519	17.358	C_D (Nm s ²)	41.27
5	0.143	0.298	0.623	41.734	1.579	3.666	41.354	5.245	k (Nm/rad)	290
T (Nm)	θ (rad)	$\dot{\theta}$ (rad/s)	$\ddot{\theta}$ (rad/s ²)	$I_t \ddot{\theta}$ (Nm)	Y_{rad} (Nm)	$C_D \dot{\theta} \dot{\theta}$ (Nm)	$k \theta$ (Nm)	Total damping		
35	0.429	0.905	1.959	131.286	1.078	40.857	126.093	41.935	I_t (kg m ²)	67
15	0.284	0.586	1.258	84.274	0.698	17.597	82.253	18.295	C_D (Nm s ²)	50
5	0.1802	0.376	0.7804	52.964	0.447	5.844	52.256	6.291	k (Nm/rad)	290

Table 9 Torque components at $\omega=2.33$ rad/s from FLUENT (top) and reduced-order model (bottom)

T (Nm)	θ (rad)	$\dot{\theta}$ (rad/s)	$\ddot{\theta}$ (rad/s ²)	$I_t \ddot{\theta}$ (Nm)	$C_r \dot{\theta}$ (Nm)	$C_D \dot{\theta} \dot{\theta}$ (Nm)	$k \theta$ (Nm)	Total Damping	I_t (kg m ²)	65
35	0.333	0.777	1.810	121.233	4.807	28.894	96.657	33.701	C_r (Nm s)	6.19
20	0.236	0.551	1.283	85.951	3.408	14.523	68.527	17.931	C_D (Nm s ²)	47.91
5	0.088	0.205	0.477	31.972	1.268	2.010	25.491	3.277	k (Nm/rad)	290
T (Nm)	θ (rad)	$\dot{\theta}$ (rad/s)	$\ddot{\theta}$ (rad/s ²)	$I_t \ddot{\theta}$ (Nm)	Y_{rad} (Nm)	$C_D \dot{\theta} \dot{\theta}$ (Nm)	$k \theta$ (Nm)	Total damping		
35	0.330	0.763	1.815	121.212	0.829	29.398	93.199	30.227	I_t (kg m ²)	67
20	0.223	0.518	1.166	80.293	0.579	13.596	62.359	14.175	C_D (Nm s ²)	50
5	0.067	0.153	0.361	23.930	0.167	1.196	19.243	1.363	k (Nm/rad)	290

the amplitude of the excitation. As the amplitude of the excitation increases, the effect of the linear term becomes negligible relative to the other terms. On the other hand, because the nonlinear damping term is linearly proportional to the excitation torque, its impact becomes larger and its inclusion in the equation of motion is required to get accurate prediction for the response of the OSWEC.

5 Conclusions

Modeling the dynamic response of body-activated wave energy converters, such as the OSWEC, in the time domain can be challenging and computationally demanding due to the complex interactions between the flap motion and wave-induced forces. While CFD can give accurate results, its computational cost is almost prohibitive when it comes to using it in the

early design phase. Alternatively, reduced-order models can be more effective in determining the response while yielding an acceptable level of accuracy. Here, we implemented an approach to determine the coefficients, including the added mass moment of inertia, radiation damping, and nonlinear damping, of a reduced-order model that yields the response of an OSWEC. It was shown that coefficients determined from regular forcing can be used to determine the response under irregular forcing. The error in the predicted the response under irregular forcing was less than 3%. This was achieved at a computational cost of about 13 min, which is a significant reduction when compared to a cost of 17 days for the RANS simulation. In addition to reducing the computational cost of the response prediction, the reduced-order model provided significant insight into the governing dynamics. It was found that the nonlinear damping term has a more significant effect on the motion than the linear term. The nonlinear damping coefficient depended linearly on the torque amplitude, which emphasizes its importance, especially that high-amplitude torques are desired for increased energy generation.

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Author contributions [AA, LZ, MH] contributed to conceptualization; [AA, MH] were involved in methodology; [AA] provided software and contributed to validation; [AA, LY, JH, AS, RD] were involved in investigation; [AA] contributed to formal analysis and data curation; [RD, MH] were involved in resources; [Alaa Ahmed] contributed to writing—original draft preparation; [RD, LZ, MH] were involved in writing—review and editing; [RD, LZ, MH] contributed to supervision; and [MH] was involved in project administration and funding acquisition.

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Data availability Data will be made available on reasonable request.

Declarations

Conflict of interest The authors have no relevant financial or non-financial interests to disclose.

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References

1. Babarit, A.: A database of capture width ratio of wave energy converters. *Renew. Energy* **80**, 610–628 (2015)
2. López, I., Andreu, J., Ceballos, S., De Alegría, I.M., Kortabarria, I.: Review of wave energy technologies and the necessary power-equipment. *Renew. Sustain. Energy Rev.* **27**, 413–434 (2013)
3. Drew, B., Plummer, A.R., Sahinkaya, M.N.: *A Review of Wave Energy Converter Technology*. SAGE, London (2009)
4. Ahmed, A., Mi, J., Haug, J., Hajj, M.R., Datla, R., Zuo, L.: Multi-fidelity modeling and simulation of dual-flap oscillating surge wave energy converter. In: *OCEANS 2023—MTS/IEEE U.S. Gulf Coast*, pp. 1–6 (2023)
5. Dual-flap floating oscillating surge wave energy converter: modelling and experiment evaluation. In: *IFAC-PapersOnLine. 9th IFAC Symposium on Mechatronic Systems MECHATRONICS 2022* vol. 55(27), pp. 138–143 (2022)
6. Cummins, W.: *The Impulse Response Function and Ship Motions*. Technical report, David Taylor Model Basin, Washington DC (1962)
7. Ringwood, J.V., Zhan, S., Faedo, N.: Empowering wave energy with control technology: possibilities and pitfalls. *Ann. Rev. Control* **55**, 18–44 (2023)
8. Folley, M., Babarit, A., Child, B., Forehand, D., O'Boyle, L., Silverthorne, K., Spinneken, J., Stratigaki, V., Troch, P.: A review of numerical modelling of wave energy converter arrays. In: Woon, E. (ed.) *International Conference on Off-shore Mechanics and Arctic Engineering*, vol. 44946, pp. 535–545. American Society of Mechanical Engineers, New York (2012)
9. Babarit, A., Clément, A.H.: Optimal latching control of a wave energy device in regular and irregular waves. *Appl. Ocean Res.* **28**(2), 77–91 (2006)
10. Schmiechen, M.: On state space models and their application to hydrodynamic systems. *Naut report* 5002 (1973)
11. Yu, Z., Faldes, J.: State-space modelling of a vertical cylinder in heave. *Appl. Ocean Res.* **17**(5), 265–275 (1995)
12. Taghipour, R., Perez, T., Moan, T.: Hybrid frequency–time domain models for dynamic response analysis of marine structures. *Ocean Eng.* **35**(7), 685–705 (2008)
13. Kristiansen, E., Hjulstad, Å., Egeland, O.: State-space representation of radiation forces in time-domain vessel models. *Ocean Eng.* **32**(17–18), 2195–2216 (2005)
14. Sutulo, S., Soares, C.G.: An implementation of the method of auxiliary state variables for solving seakeeping problems. *Int. Shipbuild. Prog.* **52**(4), 357–384 (2005)

15. Ricci, P., Saulnier, J.-B., Falcao A, Pontes, M.T.: Time-domain models and wave energy converters performance assessment. In: International Conference on Offshore Mechanics and Arctic Engineering, vol. 48234, pp. 699–708 (2008)
16. Perez, T.: Ship Motion Control: Course Keeping and Roll Stabilisation Using Rudder and Fins. Springer, London (2006)
17. Hals, J., Taghipour, R., Moan, T.: Dynamics of a force-compensated two-body wave energy converter in heave with hydraulic power take-off subject to phase control. In: Proceedings of the 7th European Wave and Tidal Energy Conference, Porto, Portugal, pp. 11–14 (2007)
18. Holappa, K., Falzarano, J.: Application of extended state space to nonlinear ship rolling. *Ocean Eng.* **26**(3), 227–240 (1998)
19. Kristiansen, E., Egeland, O.: Frequency-dependent added mass in models for controller design for wave motion damping. *IFAC Proc.* Vol. **36**(21), 67–72 (2003)
20. Jordán, M.A., Beltrán-Aguado, R.: Optimal identification of potential-radiation hydrodynamics for moored floating structures-a new general approach in state space. *Ocean Eng.* **31**(14–15), 1859–1914 (2004)
21. McCabe, A., Bradshaw, A., Widden, M.: A time-domain model of a floating body using transforms. In: 6th European Wave and Tidal Energy Conference, pp. 281–288. Mathworks Inc (2005)
22. Craik, A.D., et al.: The origins of water wave theory. *Ann. Rev. Fluid Mech.* **36**(1), 1–28 (2004)
23. Hirt, C.W., Nichols, B.D.: Volume of fluid (VOF) method for the dynamics of free boundaries. *J. Comput. Phys.* **39**(1), 201–225 (1981)
24. Mathur, S., Murthy, J.: A pressure-based method for unstructured meshes. *Numer. Heat Transf.* **31**(2), 195–215 (1997)
25. Nichita, B.A., Zun, I., Thome, J.R.: A level set method coupled with a volume of fluid method for modeling of gas-liquid interface in bubbly flow. *J. Fluids Eng.* **132**(8), 081302 (2010)
26. Manual: Ansys fluent 2020 r2. Theory Guide (2020)
27. Snyder, D., Koutsavdis, E., Anttonen, J.: Transonic store separation using unstructured CFD with dynamic meshing. In: 33rd AIAA Fluid Dynamics Conference and Exhibit, p. 3919 (2003)
28. Todalshaug, J.H.: Hydrodynamics of WECs. Handbook of ocean wave energy. pp. 139–158 (2017)
29. Inman, D.J., Singh, R.C.: Engineering Vibration, vol. 3. Prentice Hall, Englewood Cliffs, NJ (1994)

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