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Blade-Resolved Improved Delayed Detached Eddy Simulation of a Pitching Floating Hybrid Wind-Tidal Power System: Energy-Harvesting Characteristics and Decoupling Prediction

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Abstract

Aiming at the typical pitching response of floating hybrid wind-tidal power systems, this study conducts unsteady numerical simulations using a high-fidelity blade-resolved Computational fluid dynamics (CFD) framework with the Improved delayed detached eddy simulation (IDDES) turbulence model. The disturbance effect of pitch motion on the instantaneous power output of the upper wind turbine and lower tidal turbine is investigated, the spatiotemporal evolution mechanism of the three-dimensional wake vortex field under dynamic angle-of-attack distortion is elucidated, and a decoupling prediction model is established. Results show that pitching-induced periodic tilt and angle-of-attack distortion markedly amplify nonlinear power oscillations but barely affect time-averaged energy harvesting performance, with the lower tidal turbine being more sensitive to motion excitation than the upper wind turbine; under extreme conditions, the wind turbine exhibits a 73.32% power fluctuation amplitude, whereas the tidal turbine undergoes torque reversal due to deep stall at large angles of attack, reaching a peak fluctuation of 105.93%. Pitching also disrupts the axisymmetric evolution of wake vortices, with the wind turbine wake featuring vertical spatial misalignment, while the tidal turbine wake deflects strongly and asymmetrically toward the rotation side under coupled tilt-rotation effects, accelerating vortex instability and premature breakdown. Furthermore, the least-squares-based decoupling prediction model agrees well with the improved delayed detached eddy simulation results and enables accurate evaluation of the hybrid device's energy harvesting performance. This work provides theoretical guidance for fatigue mitigation, stall suppression, and power smoothing control of multi-energy complementary marine energy systems.

Keywords: Floating hybrid wind-tidal power systems; Improved delayed detached eddy simulation; Pitching motion; Unsteady characteristics; Decoupling prediction model.

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1. Introduction

In recent years, marine renewable energy exploitation has expanded rapidly into deep offshore waters,^[1] making floating platforms a key development trend.^[2,3] However, harsh deep-sea conditions drive high initial investment in floating platforms, and single-energy devices face technical and economic bottlenecks such as high levelized cost of energy

and large power output fluctuations.^[4,5] Against this backdrop, a floating hybrid wind-tidal power system (FHWTS) integrating an upper wind turbine and a lower tidal turbine has garnered extensive attention. These systems share a floating substructure and mooring system to reduce construction and operation and maintenance costs. Meanwhile, they smooth power output *via* the spatiotemporal complementarity of wind and tidal resources, significantly improving marine space utilization and mesh integration stability.^[6]

Despite broad application prospects, such hybrid systems are inherently complex multi-physics coupled systems. In practical offshore wind and wave energy development,^[7-9] floating platforms are continuously subjected to coupled wind-wave-current loads,^[10-12] inducing large-amplitude multi-degree-of-freedom motions including pitch motion,^[13-15] surge motion^[16-18] and heave motion.^[19,20] Such periodic

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reciprocating motions drastically alter the inflow conditions of the dual rotors, causing dynamic distortion or even dynamic stall of the blade effective angle of attack. These complex unsteady aerodynamic and hydrodynamic phenomena induced by platform forced motion severely disrupt the transient load characteristics and wake evolution of the turbines. Accordingly, clarifying the fluid-structure coupling mechanism under complex platform motions is critical for improving the reliability of such floating devices and advancing their engineering deployment.

Research on standalone floating offshore wind turbines (FOWTs) and tidal turbines has been well established. For FOWTs specifically, scholars have conducted extensive investigations covering offshore wind resource assessment,^[21] unit reliability analysis,^[22,23] floating platform motion responses,^[24] unit manufacturing optimization,^[25] and composite blade materials.^[26] Among these topics, aerodynamic performance changes of FOWTs caused by platform motions have been widely explored. Tran *et al.*^[27] quantified aerodynamic characteristics under pitch excitation using overset meshes, revealing that conventional unsteady blade element momentum (BEM) theory yields notable prediction errors under large-amplitude oscillations. They further noted that simplified tools based on BEM theory and vortex methods tend to overpredict load and power fluctuations under periodic motions. Regarding fully coupled system responses, Han *et al.*^[28] elucidated the effects of second-order wave forces, wind-wave misalignment and water depth on turbine fatigue and motion responses. Bae *et al.*^[29] established a data transfer interface between FAST and CHARM3D to achieve fully coupled time-domain analysis of the wind turbine-platform-mooring system. Antonutti *et al.*^[30] demonstrated the significance of geometric nonlinearity and viscous effects using the CALHYPSON code, and indicated that wind-wave synergy dominates the dynamic response of turbines under rated conditions.

For standalone tidal turbines, existing studies have systematically investigated the interference of carrier motions with different degrees of freedom on hydrodynamic and wake characteristics. Kong *et al.*^[31] systematically explored the hydrodynamic and wake evolution of side-by-side dual horizontal-axis tidal turbines under pitch motion with wave-current coupling using the IDDES method. They established a hydrodynamic load coefficient decoupling model *via* the least squares method, and quantified the effects of wave parameters and pitch amplitude on added mass and damping forces. Wang *et al.*^[32] found that yaw motion has a minor impact on turbine energy efficiency, but amplifies instantaneous fluctuations of thrust and lateral force with increasing tip speed ratio (TSR) and yaw frequency. Zhang *et al.*^[18] noted that pitch amplitude under uniform inflow significantly impairs structural safety. Huang *et al.*^[19] investigated coaxial counter-rotating tidal

turbines using sliding and overset meshes, and found that heave-induced oscillatory flow reduces the time-averaged power and thrust by 9% and 5%, respectively, with the upstream rotor more susceptible to heave disturbance.

For the hybrid wind-tidal power system, early engineering practice dates to the SKWID developed and tested at sea in Japan in 2013.^[33] As the world's first offshore wind-tidal hybrid platform, it validated the feasibility of the hybrid wind-tidal energy concept. In theoretical and numerical studies, Yang *et al.*^[34] integrated dual tidal turbines with an FOWT, and found that the hydrodynamic damping from tidal turbines improves platform transient motion response, reduces mooring line tension fluctuations, and increases total power output by 3.84%–6.46%. Their subsequent evaluation of a 10 MW wind turbine integrated with multiple tidal turbines further confirmed the enhancement of multi-turbine configurations on total power generation.^[35] Ma *et al.*^[36] proposed and validated a semi-submersible wind-tidal hybrid device equipped with counter-rotating vertical-axis tidal turbines, confirming that the damping coupling between wind and tidal components suppresses platform surge and pitch motions while boosting total power output. Li *et al.*^[37] designed a multi-energy hybrid device comprising a Spar-type wind turbine, wave energy converters and tidal turbines, and also demonstrated that tidal turbine-induced motion suppression stabilizes wind turbine power output.

For coupled dynamic solution methods, early studies by Muliawan and Bachynski simplified wind turbine aerodynamic loads *via* wind pressure theory, and solved floating body hydrodynamics using the Morison equation or radiation-diffraction potential flow theory,^[38,39] achieving favorable prediction accuracy. Subsequently, Peiffer and Tian introduced BEM theory to calculate aerodynamic loads and incorporated control system dynamics,^[40,41] further improving the prediction accuracy of coupled dynamic responses for hybrid systems.

Overall, existing research on multi-energy hybrid systems is extensive, but conventional simplified engineering methods cannot accurately capture viscous effects and strong nonlinearity under large-amplitude oscillations. Studies on FHWTS under pitch conditions remain inadequate to support engineering applications. To address this gap, this study establishes a three-dimensional numerical model for FHWTSs under pitching conditions based on the IDDES method. The effects of pitch amplitude and frequency on the transient power fluctuations of both the upper wind turbine and lower tidal turbine are investigated, the wake vortex evolution and dissipation mechanism during pitching oscillation are characterized in detail, and a fast load prediction model is constructed to analyze the underlying mechanism of energy fluctuations.

2. Numerical simulations

2.1 Numerical method

This study adopts the IDDES method to solve the unsteady Navier–Stokes equations. The model resolves boundary layers *via* RANS and captures wake vortices *via* LES, with a hybrid

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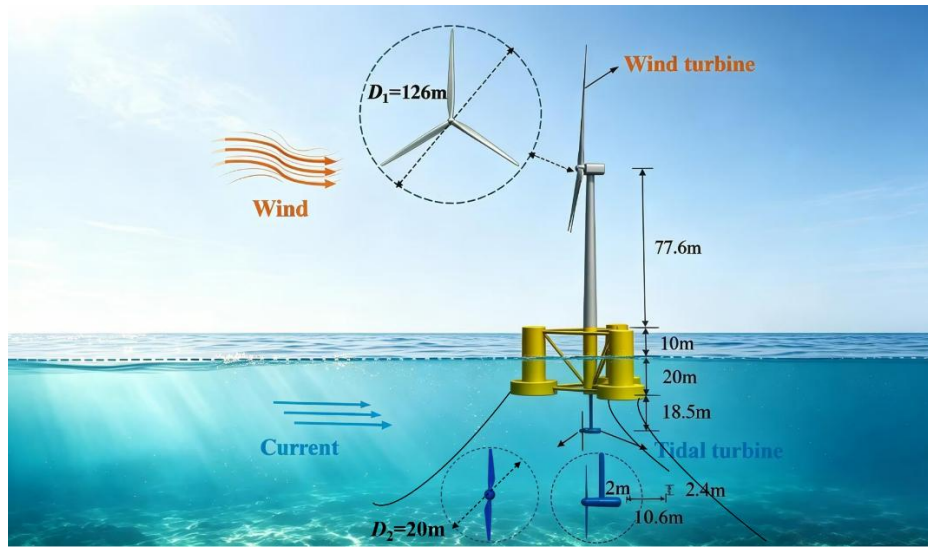


Fig. 1: Schematic diagram of FHWTS.

Table 1: Key operating parameters of the combined system.

Operating parameter	Wind turbine	Tidal turbine
Rated power	5 MW	550 kW
Rated inflow velocity	11.4 m/s	2.0 m/s
Rated rotor speed	12.1 rpm	11.5 rpm
Number of blades	3	2

length scale in the turbulent kinetic energy dissipation term. Its transport equation is presented below Eq. (1):^[42]

$$\frac{\partial(\rho k)}{\partial t} + \frac{\partial(\rho u_j k)}{\partial x_j} = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + \tau_{ij} S_{ij} - \frac{\rho k^2}{l_{IDDES}} \quad (1)$$

where k , u_j , t , μ , μ_t , σ_k , τ_{ij} and S_{ij} are turbulent kinetic energy, fluid velocity component, time, molecular viscosity coefficient, eddy viscosity coefficient, turbulent Prandtl number for k , Reynolds stress tensor, and mean strain rate tensor, respectively. l_{IDDES} is the length scale in the IDDES model, given by Eq. (2):

$$l_{IDDES} = \tilde{f}_d (l + f_e) l_{RANS} + (1 - \tilde{f}_d) l_{LES} \quad (2)$$

where l_{RANS} and l_{LES} are the length scales of the RANS and the LES, respectively. $\tilde{f}_d$ is the boundary layer identification function and f_e is the elevation function defined as Eqs. (3-7):

$$l_{RANS} = \frac{\sqrt{k}}{\beta^* \omega} \quad (3)$$

$$l_{LES} = C_{DES} \Delta \quad (4)$$

$$\Delta = \min[\max(C_W \Delta_{max}, C_W d, \Delta_{min}), \Delta_{max}] \quad (5)$$

$$\tilde{f}_d = \max((1 - f_d), f_b) \quad (6)$$

$$f_e = \max((1 - f_d), f_b) \cdot \psi \quad (7)$$

where β^* is a constant parameter, which is taken as 0.09 in the

SST- $k-\omega$ model. ω is the specific dissipation rate. C_{DES} is the model coefficient. Δ is the local mesh characteristic scale. C_W is an empirical constant; d is the nearest distance of the node from the wall. Δ_{min} is the minimum distance between the center of a mesh cell and the center of an adjacent mesh cell. Δ_{max} is the maximum mesh spacing. ψ is an empirical correction coefficient. f_b is the free shear correction factor.^[42]

2.2 Model construction

2.2.1 Model parameters

The FHWTS comprises an upper NREL 5 MW wind turbine and a lower SNL RM1 horizontal-axis tidal turbine (HATT). Fig. 1 and Table 1 present its layout, parameters and rated conditions.

2.2.2 Energy characteristic parameters

Dimensionless parameters are defined to evaluate the energy harvesting performance of the FHWTS. The tip speed ratio characterizes the power performance of both turbines. The instantaneous and time-averaged power coefficients are denoted as C_p and $\bar{C}_p$, respectively Eqs. (8-10):^[43]

$$TSR = \frac{R\omega}{V} \quad (8)$$

$$C_p = \frac{M \cdot \omega}{0.5 \rho A V^3} \quad (9)$$

$$\bar{C}_p = \frac{1}{2\pi} \int_0^{2\pi} C_p(\theta) d\theta \quad (10)$$

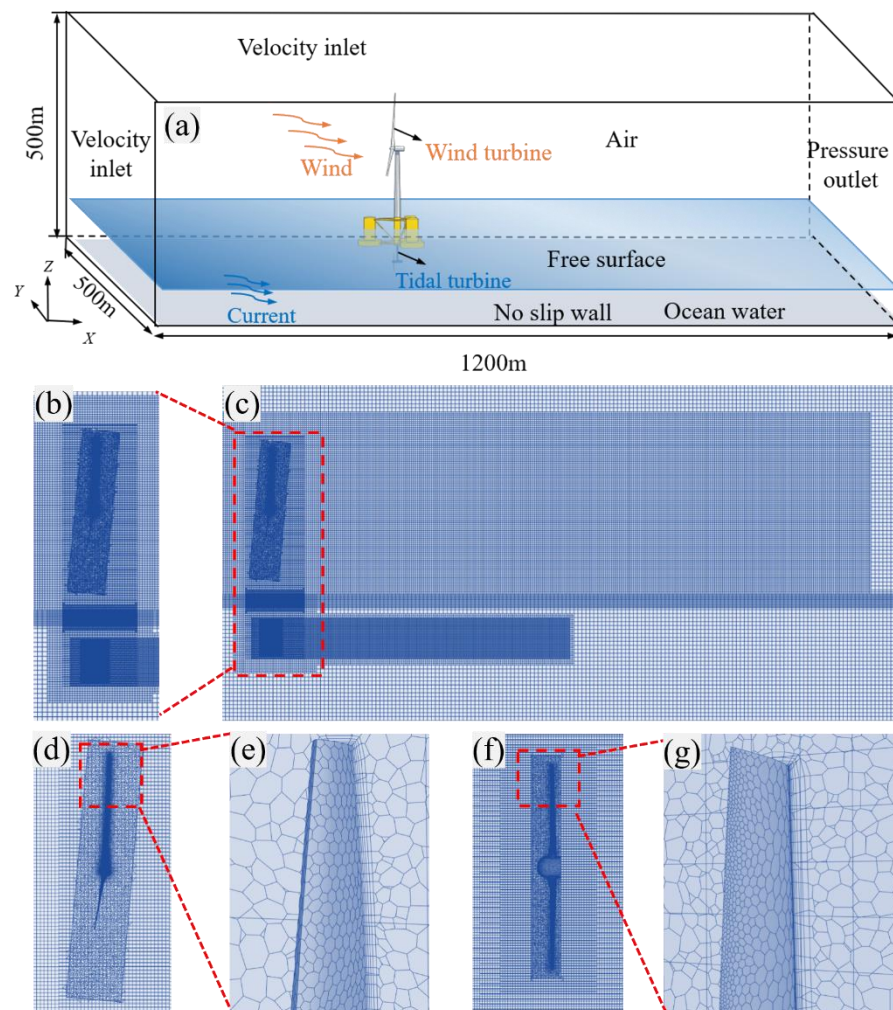


Fig. 2: Mesh division diagram. (a) boundary conditions. (b) overset domain mesh. (c) computational domain mesh. (d) wind turbine rotating domain mesh. (e) wind turbine blade boundary layer mesh. (f) tidal turbine rotating domain mesh and (g) tidal turbine blade boundary layer mesh.

where M is the rotor torque, ω is the rotor angular velocity, ρ is the fluid density, A is the rotor swept area, V is the inflow velocity, R is the rotor radius, and θ is the rotor azimuth angle.

2.3 Numerical model validation

2.3.1 Boundary conditions and mesh generation

Fig. 2 illustrates the three-dimensional CFD computational domain and mesh configuration of the FHWTS. The domain dimensions are 1200 m (length) $\times$ 500 m (width) $\times$ 500 m (height). The hybrid unit is positioned 300 m downstream of the inlet, leaving a 900 m outlet zone to facilitate wake dissipation.^[38,39] This layout complies with standard CFD practices for floating marine energy devices, guaranteeing fully developed inflow and accurate free-surface resolution by the VOF method. To investigate the effects of forced pitching motion on the aerodynamic and hydrodynamic performance of the dual rotors, the full geometries of wind and tidal turbines are retained, whereas the floating support structure is excluded to reduce computational cost. The top boundary is placed 300 m above the free surface. A velocity inlet is assigned to the upstream face for uniform wind and current inflow, the

downstream boundary serves as a pressure outlet, the bottom wall adopts a no-slip condition, and open-flow boundaries are applied to the top and lateral sides.

The flow field is solved by the IDDES turbulence model with implicit time stepping. The domain consists of a background zone, an overset zone and two independent rotating zones. Cell-based linear interpolation is adopted for data exchange between the background and overset zones, with the minimum overlap ratio set to 0.25 to ensure stable interpolation. Local mesh refinement is implemented at overset interfaces and wake regions to improve calculation accuracy. Sliding meshes simulate rotor rotation, and rotating subdomains are embedded within the overset zone to realize single-degree-of-freedom forced pitching motion. Supplementary verification eliminates outlet backflow interference: the outlet cross-section maintains positive normal velocity throughout the simulation, and extended-outlet test cases produce parameter deviations below 2%.

The IDDES model is equipped with a boundary layer identification function to distinguish attached boundary layers and free shear vortices, avoiding misjudged RANS–LES

Table 2: Three sets of meshes of different basic sizes.

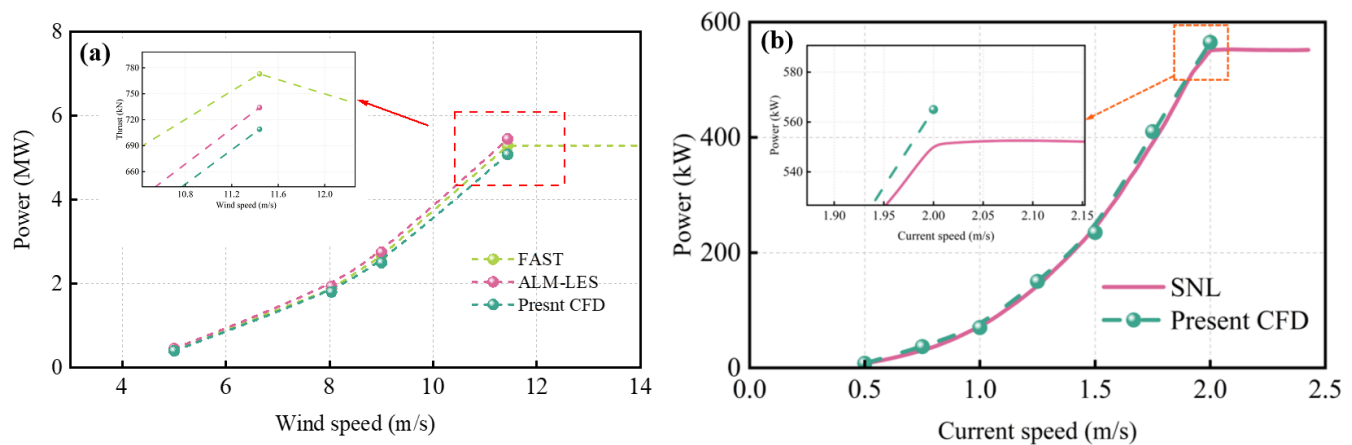
Mesh number	Base mesh size of wind turbine rotating domain (m)	Mesh count of wind turbine rotating domain ($\times 10^4$)	Base mesh size of tidal turbine rotating domain (m)	Mesh count of tidal turbine rotating domain ($\times 10^4$)	Mesh count of overset domain ($\times 10^4$)
Mesh 1	1.1	342.6	0.17	248.7	167.2
Mesh 2	0.8	504.5	0.12	347.5	227.4
Mesh 3	0.6	696.7	0.08	481.2	311.6

Table 3: Results of mesh independence verification.

Mesh number	Total number of cells ($\times 10^4$)	Wind turbine power (MW)	Relative change of wind turbine power (%)	Tidal turbine power (kW)	Relative change of tidal turbine power (%)
Mesh 1	1338.5	5.16	\	542.7	\
Mesh 2	1797.2	5.27	2.13	559.5	3.10
Mesh 3	2531.6	5.33	1.14	568.8	1.66

Table 4: Validation conditions for wind and tidal turbines.

Wind speed (m/s)	Wind turbine speed (rpm)	Current speed (m/s)	Tidal turbine speed (rpm)
5	7.39	0.5	3.34
8	9.16	1.0	6.68
9	10.3	1.5	10.03
11.4	12.1	2.0	11.5

**Fig. 3:** Validity verification. (a) Wind turbine power comparison and (b) Tidal turbine power comparison.

switching caused by rotor rotation. An all- y^+ wall treatment is adopted to guarantee robust near-wall solutions under rotational and stall flow conditions, and this model has been widely validated in existing rotating turbine simulations to capture complex separated vortex evolution accurately.

2.3.2 Mesh independence study

To eliminate spatial discretization errors while balancing computational efficiency, a mesh independence study was performed in accordance with the ASME standard with a mesh refinement ratio above 1.3.^[44] Three mesh schemes were generated by progressively reducing the base mesh size of the rotating and background domains, as presented in Table 2.

Simulations are run under rated conditions (wind speed 11.4 m/s, current speed 2.0 m/s). The total cell counts and

dual-rotor power results are listed in Table 3. The data show that the device power output exhibits a favorable convergence trend with increasing mesh density. When refining from Mesh 2 to Mesh 3, the relative variations in wind turbine and tidal turbine power are only 1.14% and 1.66%, respectively. Considering both flow field resolution accuracy and computational cost, Mesh 2 is adopted as the standard mesh for subsequent unsteady analyses.

2.3.3 Validation of wind and tidal turbines

To verify the prediction accuracy of the established numerical framework, independent aerodynamic calculations for the upper wind turbine are carried out according to Table 4. Separate validations are conducted for the wind and tidal turbines. Full-scale experimental data of the NREL 5 MW

turbine under oscillatory pitching are not publicly available, and stationary benchmarks from FAST and ALM-LES under multiple incoming wind speeds are adopted.^[45-47] Fig. 3a shows good agreement of power curves, with a relative error of 4.13% at the rated wind speed of 11.4 m/s. Static benchmark results effectively verify core modules including turbulence modelling, sliding mesh rotation and near-wall treatment. For the tidal turbine, steady validation is performed against published SNL RM1 experimental data.^[48] As shown in Fig. 3b, the predicted power curve matches experimental measurements well, with deviations below 4% at the rated current speed of 2.0 m/s, despite slight overprediction under high flow velocity. The overset mesh used to realize rigid-body pitching has been widely validated in relevant floating turbine studies.^[49,50]

The above validation results for individual rotors confirm that the proposed CFD model can accurately capture the energy conversion characteristics of the core components in the hybrid system, providing a reliable baseline model for subsequent unsteady aerodynamic-hydrodynamic coupling analysis under complex attitude deflections.

3. Aerodynamic-hydrodynamic response under pitching conditions

3.1 Case setup

As shown in Fig. 4, pitch motion refers to periodic rotational oscillation of the floating platform around its transverse Y axis under wave loads. The upper wind turbine and lower tidal turbine rotate synchronously about the center of mass of the whole hybrid system. This setup follows rigid-body dynamics and is widely used for floating energy device simulations. Reasonable vertical pivot offset merely causes quantitative differences without altering main qualitative trends. Different from uniform inflow, pitching periodically changes rotor relative inflow and generates spanwise non-uniform velocity gradients.

Pitch motion is simplified as sinusoidal harmonic motion. The angular displacement θ_p as a function of time t is given by Eq. (11):^[15]

$$\theta_p = A_p \sin(\omega_p t) \quad (11)$$

where A_p denotes the pitch amplitude and ω_p is the pitch frequency. Differentiating the displacement function with respect to time yields the pitch velocity u_p Eq. (12):

$$u_p = A_p \omega_p \cos(\omega_p t) \quad (12)$$

In the present numerical simulations, pitch amplitudes A_p of 1° , 2° and 4° and pitch frequencies ω_p of 0.79 rad/s, 0.52 rad/s and 0.39 rad/s are selected with reference to relevant studies,^[51] as detailed in Table 5. Simulations of the FHWTS under pitch motion are performed using these parameters to investigate the effects of pitch motion on its aerodynamic and hydrodynamic performance and wake characteristics.

The pitch amplitude and frequency selected in this study correspond to typical motion parameters of floating marine energy platforms. Identical sinusoidal forced pitching boundaries are widely adopted in numerical simulations of standalone wind or tidal turbines in existing literature. Nevertheless, previous studies only focus on single rotors. This study establishes a unified working condition system tailored for integrated hybrid wind-tidal devices, enabling direct comparison of aerodynamic and hydrodynamic response discrepancies between the two rotors under identical platform pitching excitation. This simulation uses prescribed forced pitching and omits mooring coupling to isolate pure pitching-induced flow interference. Actual mooring restraints adjust pitching amplitude and phase, lowering alternating load extremes and weakening asymmetric wake deflection of the tidal turbine.

3.2 Effect of pitch amplitude

Fig. 5(a, b) present the time histories and statistical bar results of the wind turbine power coefficient under various pitch conditions. Pitch motion does not significantly alter the mean aerodynamic loads; the time-averaged power coefficient remains stable at approximately 0.528, with all mean deviations within $\pm 2.1\%$. Pitch motion affects wind turbine aerodynamic loads primarily by modifying the blade inflow

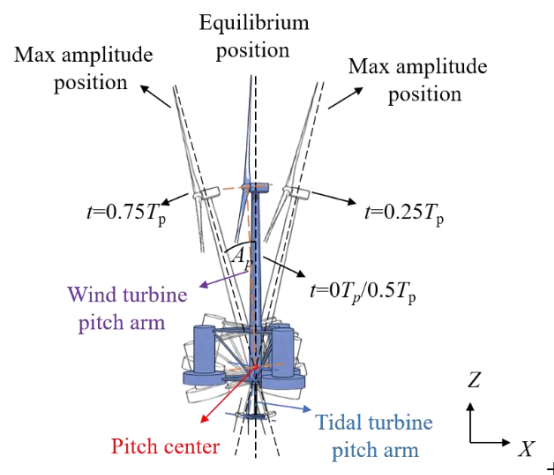
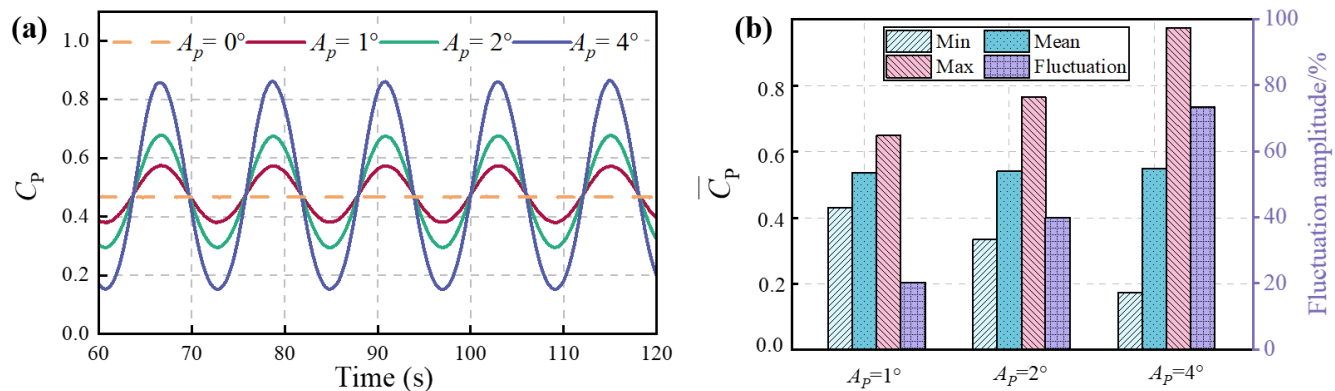
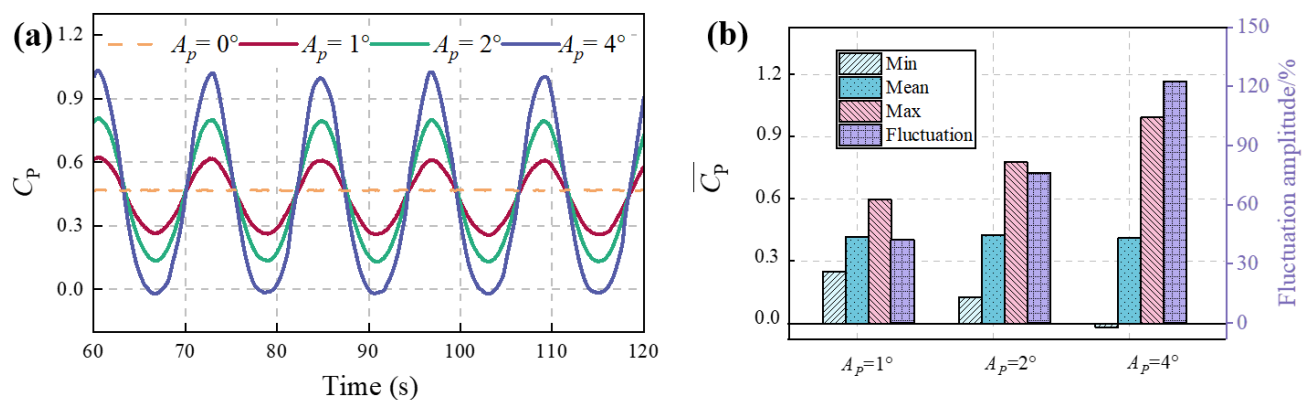


Fig. 4: Schematic diagram of pitch motion.

Table 5: Pitch motion operating condition.

Case number	Pitch amplitude ($^{\circ}$)	Pitch frequency (rad/s)
1	0	0
2	1	0.52
3	2	0.52
4	4	0.52
5	2	0.79
6	2	0.39

**Fig. 5:** Wind turbine power coefficients and their fluctuation amplitudes under different A_p . (a) Instantaneous power coefficient and (b) Average power coefficient.**Fig. 6:** Tidal turbine power coefficients and their fluctuation amplitudes under different A_p . (a) Instantaneous power coefficient and (b) Average power coefficient.

angle of attack. The periodic rotor tilt induced by pitch motion exposes blades to dynamic stall during rotation, causing severe power fluctuations. For the large-amplitude case with $A_p = 4^{\circ}$ in particular, the power fluctuation amplitude reaches 73.32%. As illustrated in Fig. 5b, increasing the pitch amplitude from 1° to 4° significantly raises the maximum power coefficient and reduces the minimum, continuously expanding the extreme value range. The power fluctuation amplitude at $A_p = 4^{\circ}$ is 3.61 times that at $A_p = 1^{\circ}$.

Overall, wind turbine power loads under pitch excitation exhibit more prominent dynamic and nonlinear characteristics.

The tidal turbine follows a similar power variation pattern to the wind turbine under pitch motion, with its fluctuation frequency governed by the platform pitch frequency. However,

the tidal turbine is more sensitive to pitch motion and exhibits larger fluctuation amplitudes. At $A_p = 4^{\circ}$, the power time history shows intense nonlinear oscillations deviating from symmetric sinusoidal profiles, with negative power occurring at certain instants due to large-amplitude pitch driving the angle of attack beyond stall, causing a sharp lift drop, drag surge and torque reversal, as shown in Fig. 6(a, b). As pitch angle increases, both maximum and minimum power values expand in tandem, widening the fluctuation range while the mean remains largely unchanged. This enhanced sensitivity arises from the higher-density water medium, where dynamic angle-of-attack variations exert a more prominent effect on torque, highlighting the need for power stability optimization in FHWTS design.

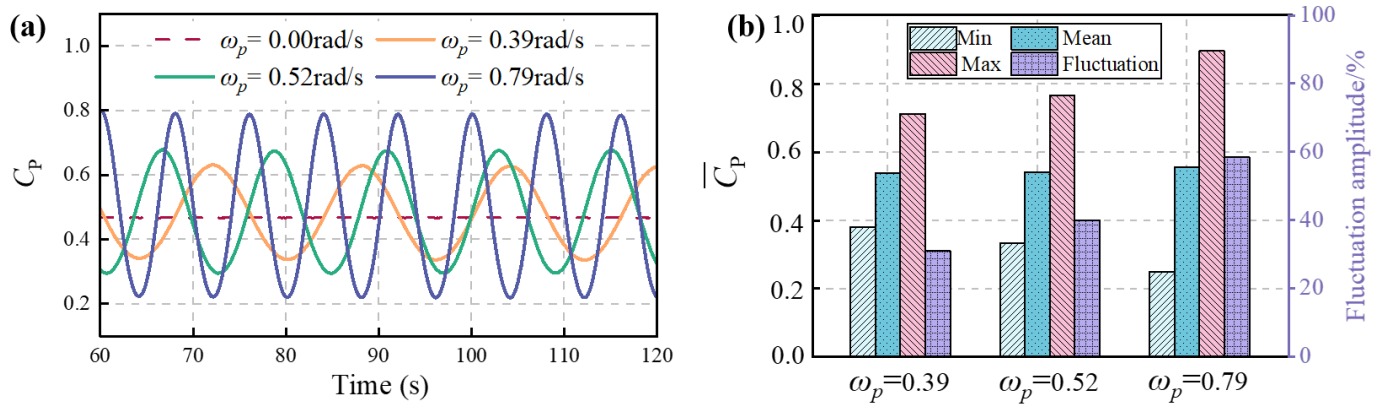


Fig. 7: Wind turbine power coefficients and their fluctuation amplitudes under different ω_p . (a) Instantaneous power coefficient and (b) Average power coefficient.

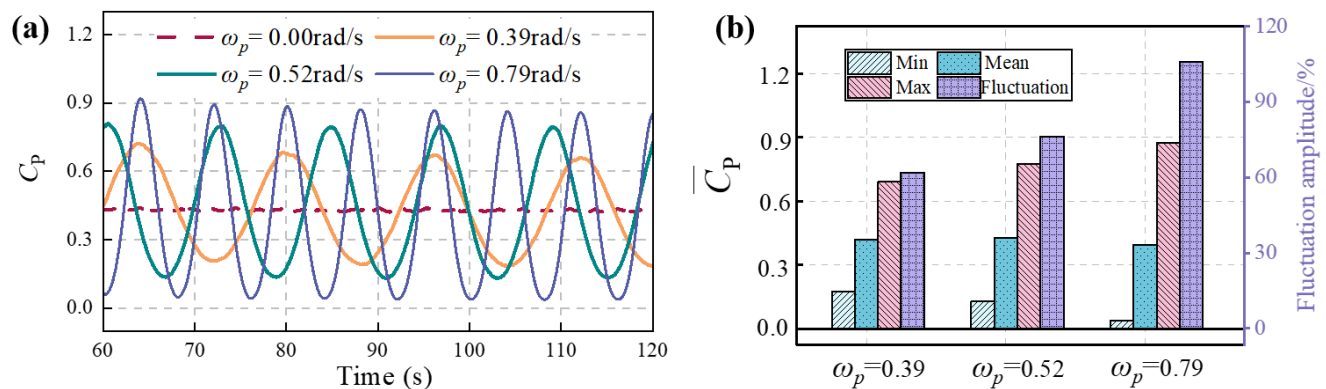


Fig. 8: Tidal turbine power coefficients and their fluctuation amplitudes under different ω_p . (a) Instantaneous power coefficient and (b) Average power coefficient.

3.3 Effect of pitch frequency

Fig 7(a, b) presents the time-history response of wind turbine power under varying pitch frequencies at $A_p = 2^\circ$. Wind turbine power shows periodic fluctuations synchronized with the pitch frequency for all tested conditions. Increasing pitch frequency induces more rapid changes in the blade relative inflow angle per unit time, exacerbating power fluctuations. Dynamic stall may occur at higher frequencies, triggering sudden power drops. The extreme values, mean values and fluctuation amplitudes of wind turbine power at different pitch frequencies are summarized in Fig. 7(a, b). Analysis of power extremes reveals that the spread between maximum and minimum power widens gradually with rising pitch frequency, reflecting a more intense aerodynamic response at higher frequencies.

The tidal turbine exhibits an identical periodic power variation pattern to the wind turbine under different pitch frequencies, with its fluctuation frequency matching the platform pitch frequency. As depicted in the time-history curves in Fig. 8(a, b), the tidal turbine power output follows a similar trend to the wind turbine, with fluctuation amplitudes rising markedly with increasing frequency. Nevertheless, at the same ω_p , the tidal turbine power displays significantly stronger fluctuations and more pronounced oscillations than the wind turbine.

Statistical results in Fig. 8 further demonstrate that pitch frequency exerts a substantial influence on the power extremes and fluctuation amplitude of the tidal turbine. As ω_p increases from 0.39 rad/s to 0.79 rad/s, the power extreme range expands continuously, and the fluctuation amplitude surges from 61.91% to 105.93%. This indicates that high-frequency pitch markedly aggravates the unsteady nature of power output, causing more dramatic variations within a single cycle. However, the mean power remains largely unchanged across all cases, indicating that pitch frequency primarily affects the transient fluctuation characteristics of power and has negligible impact on the long-term average energy harvesting performance of the tidal turbine.

3.4 Wake field analysis

Fig. 9 presents the wake velocity distribution of the FHWTS over one full pitch cycle at $\omega_p = 0.52$ rad/s and $A_p = 2^\circ$. On the X - Z plane, both wind turbine and tidal turbine wakes exhibit pronounced periodic unsteady evolution under pitch motion, with remarkable differences arising from their distinct operating media, inflow conditions and rotor characteristics. Wake contours on the X - Y plane further reveal notable disparities in the transverse evolution of dual-rotor wakes under pitch excitation. The wind turbine wake undergoes periodic lateral oscillation and asymmetric boundary

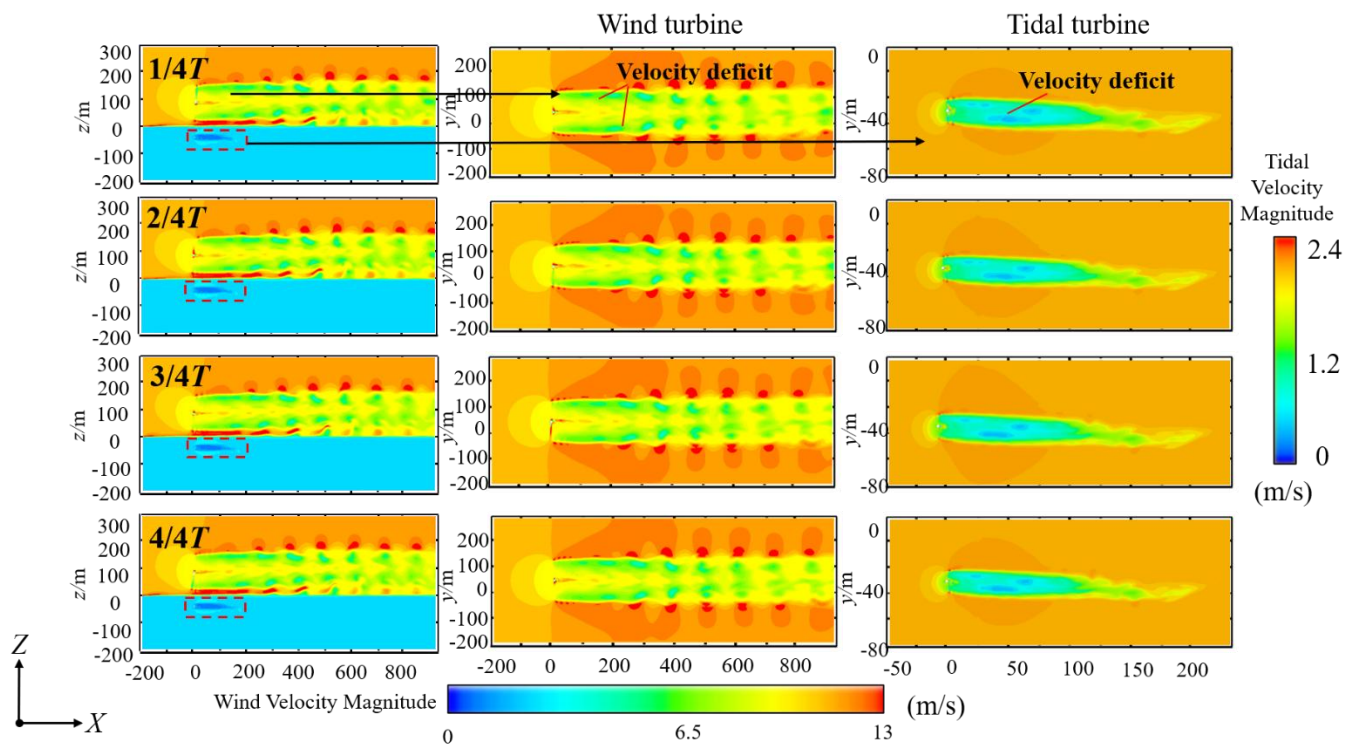


Fig. 9: Wake velocity distribution of wind and tidal turbines at four instants within one full pitch period.

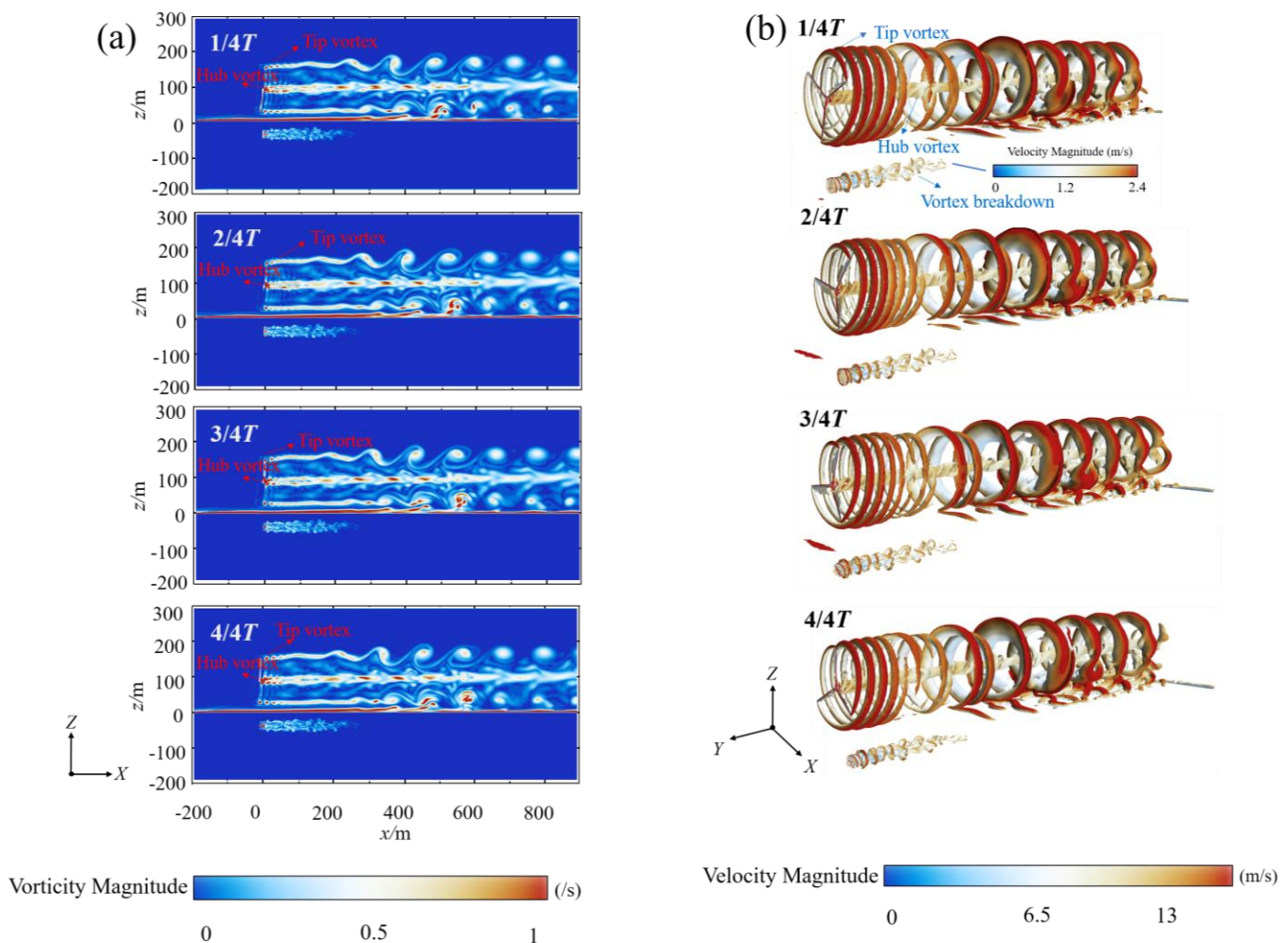


Fig. 10: Wake vortex evolution of turbines during a single pitch period: (a) Vorticity distribution and (b) 3D wake vortex topology.

deformation, but its downstream velocity deficit recovers rapidly owing to high inflow wind speed and strong turbulent diffusion. By contrast, the deep coupling between periodic rotor tilt and counterclockwise blade rotation in the tidal turbine induces local asymmetric loads under dynamic angle of attack disturbances. The relative velocity difference between advancing and retreating blades, combined with the inherent helical wake structure formed by tip and hub vortices, ultimately causes a distinct lateral deflection of the tidal turbine's velocity deficit region toward the rotation side (negative Y -direction).

Fig. 10(a, b) illustrate the vorticity distribution and three-dimensional wake vortex structure of the FHWTS at different instants within one pitch cycle, respectively. Driven by pitch motion, the rotor inflow angle of attack varies periodically, so the intensity and spatial position of high-vorticity regions evolve continuously throughout the cycle. Vorticity intensity is significantly amplified at phases with maximum angular velocity, such as $1/4T$ and $3/4T$. The tidal turbine vorticity distribution shows stronger unsteadiness, with vorticity concentrated toward the lower downstream region. Regarding wake vortex topology, the helical vortex system composed of tip and hub vortices in the wind turbine wake experiences periodic tilting and compression–stretching variations during the pitch cycle. Its helical axis oscillates with the rotor tilt angle, and the vortex scale and downstream persistence alternately strengthen and weaken across phases. The tidal turbine wake vortices exhibit more localized features: pitch motion induces uneven vertical development, and the vortices undergo breakdown within a short downstream distance. Overall, pitch motion governs the unsteady evolution of vorticity intensity and vortex structure in both wind and tidal turbine wakes primarily by modifying rotor angle of attack and spatial attitude.

4. Decoupling and prediction analysis of power coefficient

4.1 Decoupling model

Under pitch excitation, the power output of wind and tidal turbines couples pitch and rotor rotational frequencies, exhibiting multi-frequency periodic fluctuations. Based on Fourier series expansion theory, the turbine power coefficient is expressed as a trigonometric series Eq. (13):^[52]

$$C_p = C_p^0 + \sum_{i=1}^{M_1} C_p^i \sin(iN\omega_T t + \psi_p^i) + \sum_{k=1}^{N_1} \left[C_{p1}^{0k} + \sum_{i=1}^{M_2} C_{p1}^{ik} \sin(iN\omega_T t + \psi_{p1}^i) \right] \sin(k\omega_p t) + \sum_{k=1}^{N_2} \left[C_{p2}^{0k} + \sum_{i=1}^{M_3} C_{p2}^{ik} \sin(iN\omega_T t + \psi_{p2}^i) \right] \cos(k\omega_p t) \quad (13)$$

Least squares fitting yields simplified Eq. (14) by retaining dominant low-frequency components.

$$C_p = C_p^0 + C_p^1 \sin(N\omega_T t + \psi_p^1) + [C_{p1}^{01} + C_{p1}^{11} \sin(N\omega_T t + \psi_{p1}^1)] \sin(\omega_p t) + [C_{p2}^{01} + C_{p2}^{11} \sin(N\omega_T t + \psi_{p2}^1)] \cos(\omega_p t) \quad (14)$$

To further reveal the physical composition of loads, dimensionless pitch velocity and acceleration are introduced, defined as Eqs. (15, 16):

$$\bar{u}_p = \frac{u_p}{V} = \frac{A_p \omega_p}{V} \cos(\omega_p t) \quad (15)$$

$$\bar{a}_p = \frac{a_p D}{2V} = -\frac{A_p \omega_p^2 D}{2V^2} \sin(\omega_p t) \quad (16)$$

Rewriting Eqs. (14, 17) decomposes hydrodynamic loads into steady, viscous damping, and added mass components.

Notably, this decomposition operates on dimensionless C_p instead of dimensional power. It adopts the mathematical form of hydrodynamic splitting merely for waveform separation, rather than direct mechanical analogy to force/moment decomposition. All terms are dimensionless to satisfy dimensional consistency. The steady term denotes static aerodynamic/hydrodynamic contribution; the damping term represents velocity-synchronous energy dissipation from angle-of-attack oscillation; the added-mass term describes acceleration-correlated fluctuations originating from unsteady inertial flow.

$$C_p = C_p^0 + C_p^1 \sin(N\omega_T t + \psi_p^1) + [n_p^0 + n_p^1 \sin(N\omega_T t + \psi_{p1}^1)] \bar{u}_p + [m_p^0 + m_p^1 \sin(N\omega_T t + \psi_{p2}^1)] \bar{a}_p \quad (17)$$

Fitting curve 1 (FIT1) and Fitting curve 2 (FIT2) denote the fitted power coefficient time histories from Eqs. (14, 17), respectively.^[52] The coefficient of determination R^2 is adopted to quantify their agreement with CFD results. Table 6 summarizes the fitting performance. Both models maintain high accuracy across all pitch amplitude and frequency cases, with all R^2 values above 0.98 and reliable reproduction of CFD-derived power fluctuation patterns. FIT2 is selected for subsequent decoupling analysis. As shown by the time-domain comparison in Fig. 11(a-h), both fitted curves accurately capture the periodic profile, amplitude, phase and overall trend of the power coefficient, and are highly consistent with numerical simulation results.

4.2 Effect of pitch amplitude

The equivalent viscous damping coefficient is a core parameter for evaluating the energy dissipation and load vibration characteristics of the FHWTS under pitch motion, and quantitative analysis of the power viscous damping coefficient n_p^0 at varying pitch amplitudes can clarify the dominant influence of amplitude on system damping performance. Table 7 lists the energy decoupling expansion coefficients under three pitch amplitude cases (1° , 2° and 4°),

Table 6: Coefficient of determination R^2 of fitting models against IDDES results under different A_p and ω_p .

Condition	FIT1 R^2 of wind turbine C_p	FIT2 R^2 of wind turbine C_p	FIT1 R^2 of tidal turbine C_p	FIT2 R^2 of tidal turbine C_p
$A_p=1^\circ$, $\omega_p=0.52$ rad/s	0.9996	0.9995	0.9827	0.9675
$A_p=2^\circ$, $\omega_p=0.52$ rad/s	0.9991	0.9987	0.9912	0.9845
$A_p=4^\circ$, $\omega_p=0.52$ rad/s	0.9975	0.9964	0.9925	0.9852
$\omega_p=0.39$ rad/s, $A_p=2^\circ$	0.9925	0.9987	0.9826	0.9814
$\omega_p=0.52$ rad/s, $A_p=2^\circ$	0.9956	0.9872	0.9866	0.9812
$\omega_p=0.79$ rad/s, $A_p=2^\circ$	0.9932	0.9915	0.9986	0.9832

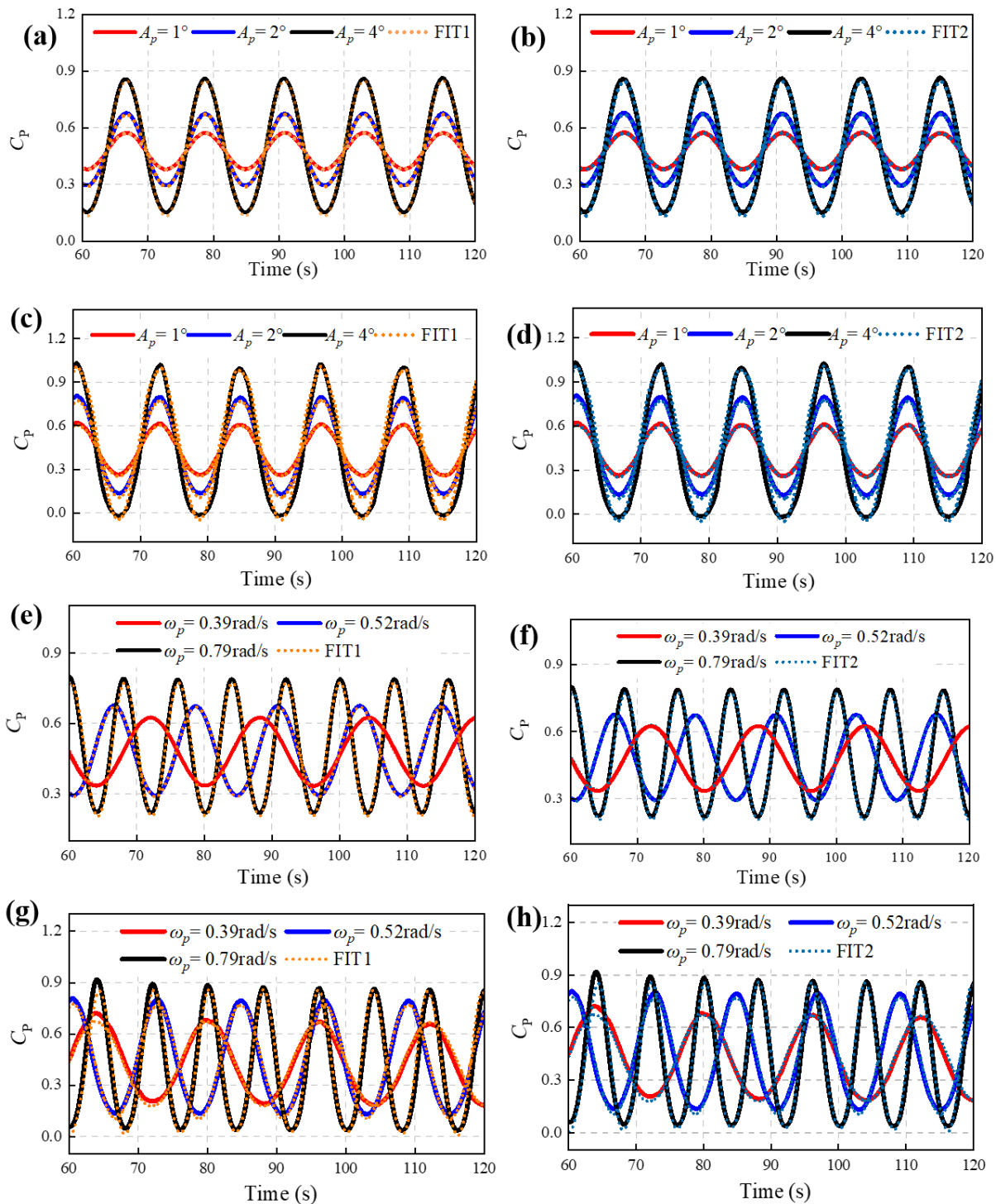
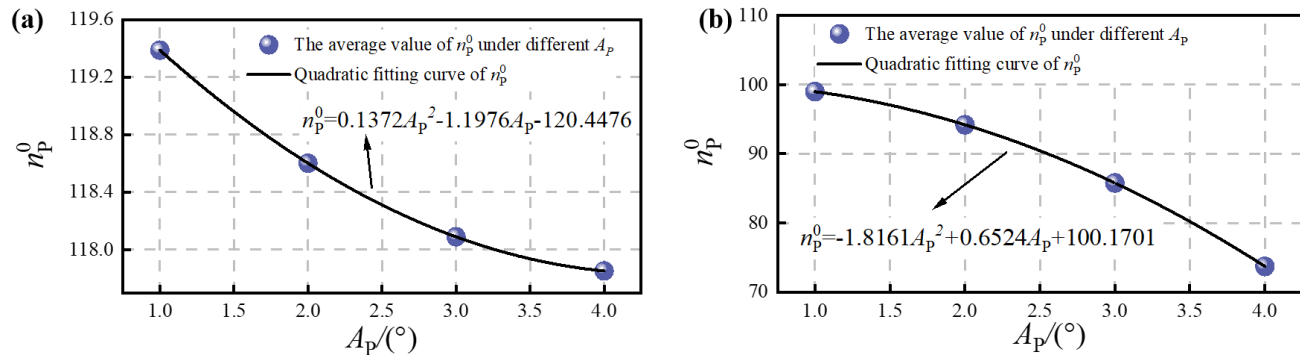


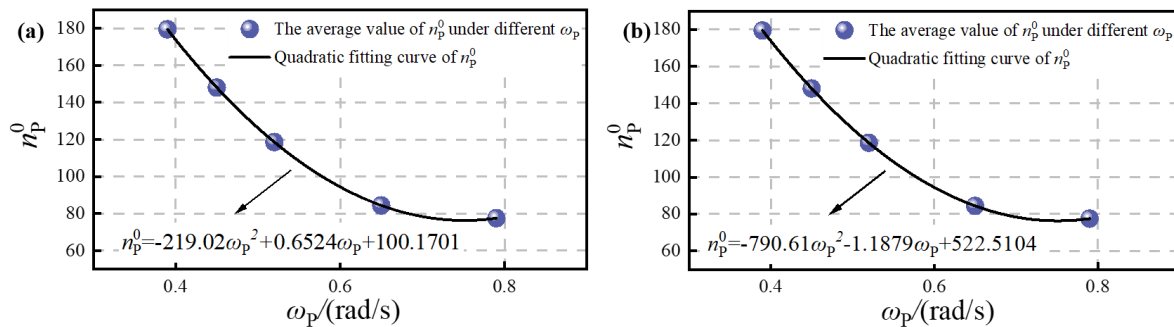
Fig. 11: Comparison between CFD results and fitted power coefficient curves: (a-d) Wind turbine FIT1, FIT2 and Tidal turbine FIT1, FIT2 under varying pitch amplitude, (e-h) Results under varying pitch frequency.

Table 7: Expansion analysis of power coefficient under different A_p .

A_p	C_p^0	C_p^1	ψ_p^1	n_p^0	n_p^1	m_p^0	m_p^1
1°-FOWT	0.4755	-0.0003	1.2351	119.3872	0.0134	6.1726	0.0217
2°-FOWT	0.4796	-0.0002	-5.2399	118.6012	0.7201	5.9256	-0.0097
4°-FOWT	0.4875	-0.0004	0.9619	117.8524	0.7982	5.2993	-0.0319
1°-HATT	0.4288	0.0032	1.2032	99.0064	0.5200	2.2833	0.0313
2°-HATT	0.4382	0.0022	0.6785	94.2104	0.6162	2.2236	-0.0143
4°-HATT	0.4719	0.0088	0.5528	73.7215	1.4674	1.4683	0.1759

**Fig. 12:** Variations of coefficient n_p^0 with different A_p . (a) Wind turbine n_p^0 and (b) Tidal turbine n_p^0 .**Table 8:** Expansion analysis of power coefficient under different ω_p .

ω_p	C_p^0	C_p^1	ψ_p^1	n_p^0	n_p^1	m_p^0	m_p^1
0.39-FOWT	0.4701	-0.0005	1.0952	179.4941	0.2807	10.6133	0.1205
0.52-FOWT	0.4796	-0.0002	-5.2399	118.6012	0.7201	5.9256	-0.0097
0.79-FOWT	0.4899	-0.0007	1.6097	77.5182	0.2252	18.8045	-0.2786
0.39-HATT	0.4243	-0.0017	1.4180	71.2122	0.5423	24.6549	0.2968
0.52-HATT	0.4382	0.0022	0.6785	64.2104	0.6162	2.2236	-0.0143
0.79-HATT	0.4115	0.0030	-1.0133	26.0141	2.4381	21.7303	0.1426

**Fig. 13:** Variations of coefficient n_p^0 with different ω_p . (a) Wind turbine n_p^0 and (b) Tidal turbine n_p^0 .

presenting the variations of the baseline term, first-order modulation term and damping coefficient; the viscous damping term n_p^0 exhibits the most significant response to A_p and dominates the energy dissipation and modulation process. As shown in Fig. 12, quadratic fitting is performed on the extracted average viscous damping coefficient n_p^0 : as the A_p increases from 1° to 4°, n_p^0 decreases monotonically from 99.0064 to 73.7215, and the downward-opening quadratic fitting curve indicates an accelerating decay rate with rising amplitude, presenting distinct nonlinear attenuation characteristics. Pitch amplitude dominates the damping performance, and increasing amplitude continuously weakens

the energy dissipation capacity of the system's equivalent viscous damping. Larger pitch amplitudes intensify blade angle-of-attack fluctuations and aggravate dynamic stall, expanding the flow separation region on blade surfaces and enhancing wake interference, which reduces the fluid energy dissipation level during rotor operation and weakens the damping effect. Meanwhile, large-amplitude pitch alters the relative motion between the rotor and inflow, inducing nonlinear deviation in the modulation characteristics of hydrodynamic loads, which ultimately results in the quadratically accelerating decay of the equivalent viscous damping coefficient n_p^0 with increasing A_p .

4.3 Effect of pitch frequency

The equivalent viscous damping coefficient evaluates energy dissipation and load oscillation of the FHWTs under pitching. Quantitative analysis of the power viscous damping coefficient n_p^0 under varying pitch angular frequencies reveals how angular frequency governs the system's damping characteristics. Table 8 lists the energy decoupling expansion coefficients under three pitch angular frequency cases (0.39, 0.52 and 0.79 rad/s); as shown in Fig. 13(a, b), quadratic fitting is performed on the extracted average viscous damping coefficient n_p^0 of the wind turbine and tidal turbine, respectively. For the tidal turbine, the results indicate that as the ω_p increases from 0.39 rad/s to 0.79 rad/s, n_p^0 decreases monotonically from 71.2122 to 26.0141, with the downward-opening quadratic fitting curve showing an accelerating decay rate at higher frequencies and distinct nonlinear attenuation characteristics. Pitch angular frequency dominates the damping performance, and rising frequency continuously weakens the energy dissipation capacity of the system's equivalent viscous damping: increasing pitch angular frequency accelerates blade angular velocity and intensifies instantaneous angle-of-attack fluctuations, aggravating dynamic stall, expanding the flow separation region on blade surfaces and deepening wake interference, which reduces the fluid energy dissipation level during rotor operation and weakens the damping effect; meanwhile, high-frequency pitch alters the relative motion between the rotor and inflow and induces nonlinear deviation in the modulation characteristics of hydrodynamic loads, ultimately leading to the quadratically accelerating decay of the equivalent viscous damping coefficient n_p^0 with increasing ω_p .

5. Conclusion

This study adopts the IDDES method to analyze the effects of pitch motion on the energy harvesting performance, load characteristics and wake evolution of the FHWTs for engineering reliability improvement. A fast power prediction model is developed *via* least-squares decoupling, and its fitting accuracy and application reliability are evaluated. The main conclusions are as follows:

Pitch motion induces severe unsteady fluctuations in the aerodynamic and hydrodynamic power of the system by periodically altering the rotor inflow angle of attack. It exerts a minor effect on the time-averaged power level, while large-amplitude dynamic tilt easily drives blades into the dynamic stall regime and causes torque variation. Under large-amplitude excitation, the wind turbine power fluctuation amplitude reaches 73.32%, 3.61 times that under small-amplitude conditions, exhibiting strong dynamic nonlinear response characteristics. Increasing pitch frequency amplifies the change rate of relative inflow angle experienced by blades per unit time. At a fixed $A_p = 2^\circ$, as ω_p rises from 0.39 rad/s to 0.79 rad/s, the power extreme range of the tidal turbine expands continuously, and its power fluctuation amplitude increases from 61.91% to 105.93%. Such high-frequency

forced tilt leads to more dramatic alternating load variations of the tidal turbine within a single cycle, fully indicating that pitch frequency variation mainly affects the fluctuation extremes of transient loads and has negligible impact on the long-term time-averaged energy harvesting capacity of the tidal turbine.

Restricted by the high density of seawater and low rotor inertia, the lower tidal turbine exhibits far higher sensitivity to pitch-induced angle-of-attack distortion than the wind turbine. With increasing pitch angle, the fluctuation range of tidal turbine hydrodynamic loads widens significantly. Particularly at $A_p = 4^\circ$, its power time history presents intense asymmetric oscillations; at specific phases, the instantaneous angle of attack far exceeds the stall critical point, causing a sharp drop in lift and surge in drag, which leads to torque reversal and negative power.

Pitch motion directly affects the spatial topology and vortex evolution mechanism of the dual-rotor wake field by modifying rotor attitude. Although the wind turbine wake undergoes periodic lateral oscillation and compression–stretching deformation, its helical vortex structure recovers rapidly downstream due to strong turbulent diffusion. In contrast, the coupling between tidal turbine rotor tilt and counterclockwise rotation induces local asymmetric loads. The relative velocity difference between advancing and retreating blades, combined with the inherent helical wake structure, causes a distinct lateral deflection of the wake velocity deficit region toward the rotation side. In addition, tidal turbine wake vortices develop unevenly in the vertical direction and break down within a short downstream distance, showing stronger localization and unsteady vorticity dissipation characteristics.

The hydrodynamic prediction model achieves high fitting accuracy, with a coefficient of determination $R^2 \geq 0.98$ for all cases. It can accurately reproduce the power fluctuation law of turbines under pitch motion with favorable reliability. Pitch amplitude and pitch frequency are key factors dominating the damping characteristics of the FHWTs. As A_p and ω_p increase, the equivalent viscous damping coefficient of the tidal turbine exhibits significant nonlinear accelerated attenuation. Increased pitch motion intensity intensifies blade angle-of-attack fluctuations and dynamic stall, expands flow separation and wake interference, and continuously weakens fluid energy dissipation capacity, ultimately leading to a continuous decline in system damping level and vibration suppression performance with rising motion excitation intensity.

This study clarifies pitching-induced flow mechanisms of floating hybrid wind-tidal turbines and establishes an accurate power decoupling prediction model to guide fatigue suppression and power smoothing for marine multi-energy systems. This work has clear limitations: simulations run at fixed rotor speed without active control, and only single-degree forced pitching is adopted while six-DOF platform coupling, mooring stiffness, damping and dynamic feedback are ignored. Real mooring constraints reduce load extremes

and weaken asymmetric tidal turbine wake deflection. Floating structures and actual turbulent wave inflows are simplified, and the decoupling model is only calibrated over limited parameters without a complete engineering application framework.

Future research will integrate active control schemes, full platform geometric models and realistic marine inflow conditions. Extensive parametric analyses will be conducted to broaden the applicable scope of the decoupling model, formulate standardized operating procedures and strengthen its generalization for real engineering deployment.

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Data Availability

The data that support the findings of this study are available from the corresponding author upon reasonable request.

Conflict of Interest

There is no conflict of interest.

Declaration of Generative AI and AI-assisted Technologies in the Manuscript Preparation Process

During the preparation of this work, the authors did not use generative AI tools or AI-assisted technologies. The authors take full responsibility for the content of the published article.

Supporting Information

Not Applicable.

CRedit Statement

Renwei Ji: Investigation, Methodology, Project administration, Writing – Original draft, Funding acquisition. **Miankui Wu:** Investigation, Methodology, Software, Data curation, Writing – Original draft. **Xiangquan Li:** Investigation, Software, Validation, Formal analysis, Writing – Review & editing. **Jonas W. Ringsberg:** Conceptualization, Methodology, Supervision. **Ho-Seong Yang:** Investigation, Visualization, Resource.

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