

Built-in wave energy converter inspired adaptive vibration control for offshore floating platform

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ABSTRACT

In addition to wave energy harvesting, there is growing interest in integrating the wave energy converter (WEC) into the offshore floating platform (OFP) for potential vibration control. However, due to the inherent dynamic coupling among the OFP, WEC, and ocean waves, it is still challenging for existing WEC-OFP combined structures to maintain effective vibration control through structural parameter optimization under changing wave conditions. To address this issue, this paper extends the energy conversion mechanism of built-in WEC to vibration control to eliminate dynamic coupling induced by ocean waves on WECs. Based on this, a multiple electromagnetic damping adaptive synergy optimization method is proposed to meet the needs of vibration control under different wave conditions. To illustrate this proposal, a detailed design is presented using a semi-submersible floating platform synergizing multiple built-in WECs. A multi-degree-of-freedom mathematical model including multiple physical domains is developed and validated. Subsequently, vibration control-oriented multiple electromagnetic damping optimization is conducted, followed by performance analysis. Hardware-in-the-loop experiments are also conducted to test the vibration control effectiveness of the proposed approach under varying wave conditions. Results show that the proposed approach is particularly effective for varying ocean wave conditions.

Nomenclature

Abbreviations			
OFP	Offshore floating platform	RAO	Response amplitude operator
WEC	Wave energy converter	MSL	Mean sea level
SSP	Semi-submersible floating platform	COG	Center of gravity
BIWEC	Built-in WEC	PMSG	Permanent magnet synchronous generator
SSP-BIWEC	SSP synergy multiple BIWECs	PTO	Power take-off
RTLAB	Real Time Laboratory	HIL	Hardware in the loop
Symbols			
m_p	Physical mass of SSP	Ω	Excitation frequency ratio

(continued on next column)

(continued)

ω	Wave frequency	γ_i ($i = 1, 2, 3$)	Natural frequency ratio between the i th BIWEC and SSP
H	Wave height	ξ_i ($i = 1, 2, 3$)	Damping ratio of the i th BIWEC
m_w	Physical mass of BIWEC	G	Heave vibration amplitude of SSP
k_{wi} ($i = 1, 2, 3$)	Stiffness coefficients of the i th BIWEC	l_b	Lead of the ball screw
c_{wi} ($i = 1, 2, 3$)	Electromagnetic damping of the i th BIWEC	ω_{mi}	PMSG's mechanical angular velocity
D_i	Duty cycle of the i th BIWEC's DC/DC converter	L_i ($i = 1, 2, 3$)	Safe motion excursion for each BIWEC
μ	Mass ratio between each BIWEC and SSP	P_{all}	Overall output power of SSP-BIWEC

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4) For the added mass:

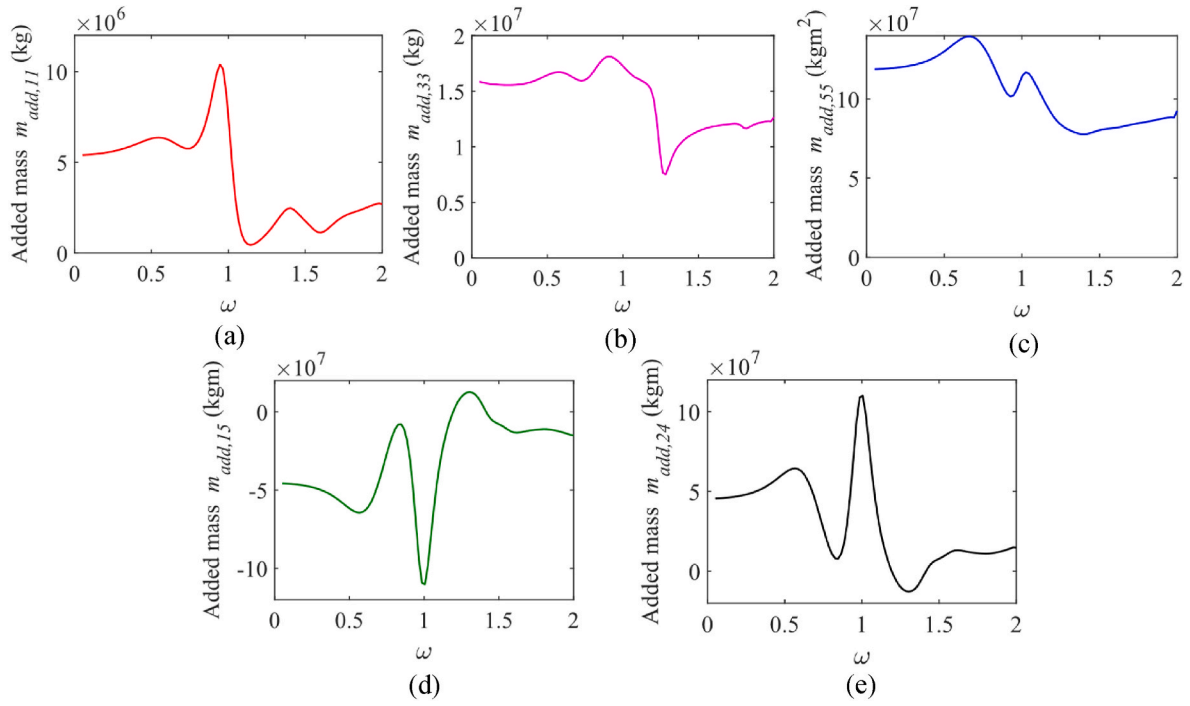


Fig. A2. The added mass of the SSP: (a) $m_{add,11}$, (b) $m_{add,33}$, (c) $m_{add,55}$, (d) $m_{add,15}$ and (e) $m_{add,24}$.

5) For the retardation function:

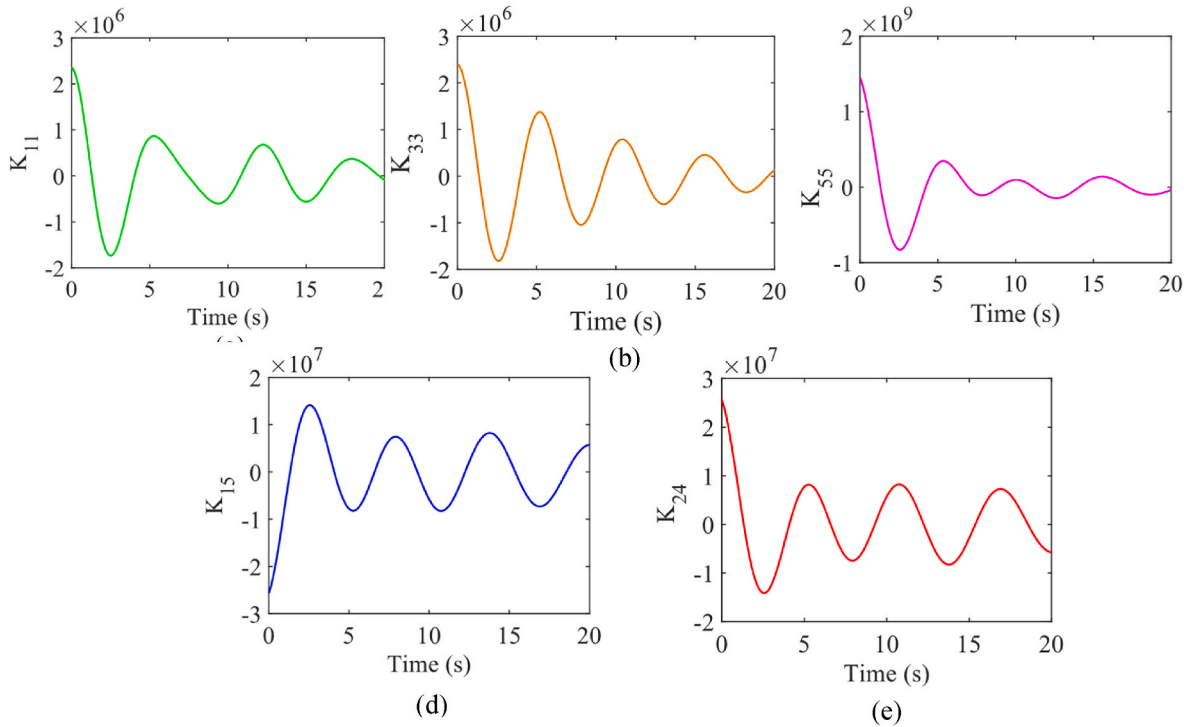


Fig. A3. The comparison of retardation function of the SSP: (a) K_{11} , (b) K_{33} , (c) K_{55} , (d) K_{15} and (e) K_{24} .

- 6) For the hydrostatic stiffness: $K_{p3} = 3.635 \times 10^3$ kN/m, $K_{p4} = 6.54 \times 10^5$ kNm/deg, $K_{p5} = 6.54 \times 10^5$ kNm/deg.
 7) For the mooring line force:

$$\mathbf{F}_{\text{wave},0} = \begin{bmatrix} 0 & 0 & -1.607 \times 10^6 \text{ N} & 0 & 0 \end{bmatrix}^T \quad (\text{Q1})$$

$$\mathbf{K}_{\text{line}} = \begin{bmatrix} 4.118 \times 10^4 \text{ N/m} & 0 & 0 & 0 & -2.821 \times 10^6 \text{ N/rad} \\ 0 & 4.118 \times 10^4 \text{ N/m} & 0 & 2.821 \times 10^6 \text{ N/rad} & 0 \\ 0 & 0 & 1.194 \times 10^4 \text{ N/m} & 0 & 0 \\ -2.816 \times 10^6 \text{ Nm/m} & 0 & 0 & 3.111 \times 10^8 \text{ Nm/rad} & 0 \\ 0 & 0 & 0 & 3.111 \times 10^8 \text{ Nm/rad} & 0 \end{bmatrix} \quad (\text{Q2})$$

1. Introduction

The ocean has abundant energy sources, including fossil fuels like oil and gas, as well as renewable options such as wind, wave, and tidal energy [1]. Over the past few decades, there has been significant extraction of energy resources from near-shore areas. However, as shallow water resources diminish and technology advances, offshore energy extraction is progressively shifting towards deeper waters [2]. Offshore floating platforms (OFPs) serve as crucial infrastructure for exploring and exploiting ocean resources in deeper waters. These platforms can host various installations, such as wind turbines, oil and gas drilling facilities, substations, wave energy converters (WECs), and so on. They fulfill the escalating energy demands accompanying rapid development and consequently attract considerable attention from numerous coastal countries [3].

According to structural configuration mode, OFP can be classified into three main types: semi-submersible floating platforms (SSP), spar floating platforms, and tension leg floating platforms [4]. Among these platforms, SSP is the most widely used in engineering due to its large deck areas, strong payload capacity, and better adaptability to different water depths [5]. However, in the deep sea, SSP still suffers from excessive heave motion caused by extreme waves [6]. This excessive heave motion not only leads to fatigue issues in major structural components but also may result in platform overturning or sinking [7]. Therefore, it is necessary to reduce the vibration response of the SSP in the heave direction to protect the equipment and ensure the safety of the operation activities. It is a critical first step towards improving the floating platform's overall operability [8].

To mitigate the excessive heave vibrations of the SSP, many structural control methods in civil or marine engineering have been applied and investigated. The structural vibration methods can be classified in three main categories: passive, active, and semi-active controls [9]. Among these strategies, semi-active and active control methods offer feedback control forces, thus excellent vibration control performance can be achieved. But these methods require additional energy inputs and mechatronic equipment with high maintenance costs. In contrast, passive vibration control methods are more stable and cost-effective, which can address the shortcomings associated with semi-active and active control approaches. The main passive vibration control methods are addressed as follows, including vibration energy consumption (such as dampers), vibration energy isolation (e.g., isolators/isolation platforms), and vibration energy transfer (e.g., dynamic vibration absorbers) [10]. However, these reported vibration control devices are all energy dissipation systems, implying that the wave energy absorbed by the SSP under ocean wave excitation is wasted. In fact, the offshore SSP

usually necessitates significant energy inputs for operating equipment, supporting living facilities, and powering scientific research equipment. Therefore, it is desired to convert the wave energy absorbed by the SSP into useful energy, such as electricity, such that vibration control and energy harvesting can be simultaneously realized.

To simultaneously achieve vibration control and energy harvesting, the SSP combined with wave energy converter (SSP-CWEC) has been

developed [11]. The design concept behind the SSP-CWEC involves deploying multiple WECs outside the platform to reduce wave loads on floating structures or introducing extra restoring moments and positive damping to the floating platform [12], thereby suppressing vibrations. Initial experimental tests demonstrate that deploying the WEC arrays around the SSP contributes to reducing platform heave motion [13]. However, these studies only conducted time-domain analysis under fixed wave frequencies, missing the dynamic interaction between the SSP and WEC under varying wave frequencies. Through subsequent analysis in Ref. [17], it can be found that the heave motion reduction of the platform only occurs within a specific wave frequency band. As wave frequencies change, the platform's vibration amplitude may even increase due to the coupling of the WEC and SSP through the ocean waves. Ocean wave incurs added mass, hydrostatic stiffness, and radiation-damping effects on both the WEC and SSP, leading to variations in their natural frequencies with changing wave conditions. When the natural frequencies of the two structures align closely, the SSP's vibration amplitude increases sharply. While certain improvements have been made to the SSP-CWEC structure, such as adjustments in WEC array configuration [14], WEC shape [15], power take-off (PTO) parameters [16] and the number of WECs [17], these methods cannot thoroughly eliminate the inherent dynamic coupling between the WEC, SSP, and ocean waves. Therefore, under varying wave conditions, the vibration control effect might not be reliable.

Based on the analysis of the above research status, it is evident that in the existing combined structural configuration, the energy absorbed by the WEC results from the combined effects of the floating platform and ocean waves. This implies that, when the WEC is used for vibration reduction, it cannot suppress the vibrations by directly absorbing the floating platform's vibration energy. As a result, even with structural parameter optimization, the WEC struggles to meet vibration control requirements under varying wave conditions.

To this end, our recently published work [18] extends the operational mechanism of the built-in WEC to the floating platform vibration control scenario, aiming to eliminate the dynamic coupling induced by ocean waves on WECs due to the structural configuration. Building on this, we proposed a passive optimization strategy where the PTO's optimal stiffness and damping coefficients in the three BIWECs are fixed. In this method, the optimization of PTO stiffness coefficients aims to adjust the BIWEC's natural frequencies, allowing more of the SSP's heave vibration energy to transfer into the BIWEC system. The optimization of PTO damping aims to efficiently convert the SSP's vibration energy into electrical power, thereby preventing the energy from being transferred back to the SSP. However, we neglected the poor energy harvesting performance of fixed damping coefficients under varying wave conditions. This limitation prevents the BIWECs from fully utilizing their synergy potential to harvest the platform's vibration energy and effectively suppress vibrations. Moreover, the analysis in previous

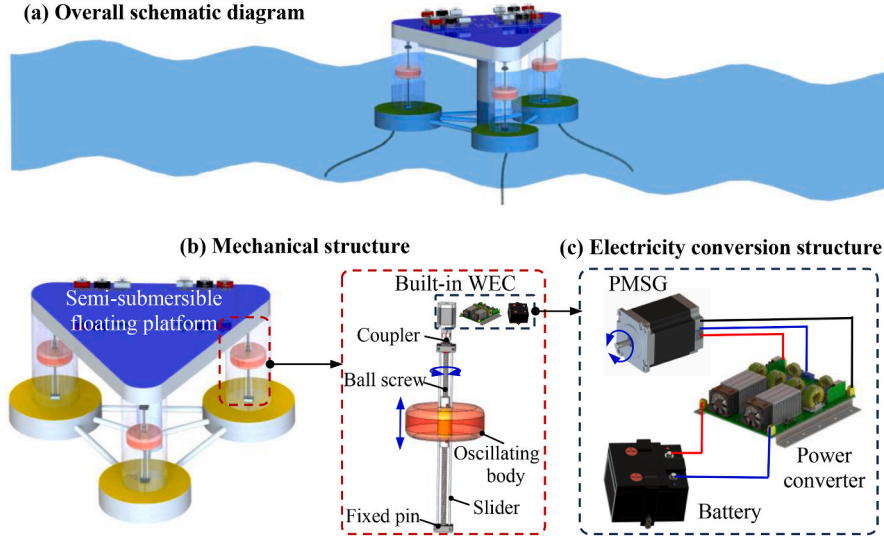


Fig. 1. Mechanical and electric conversion structure design scheme of SSP-BIWECS.

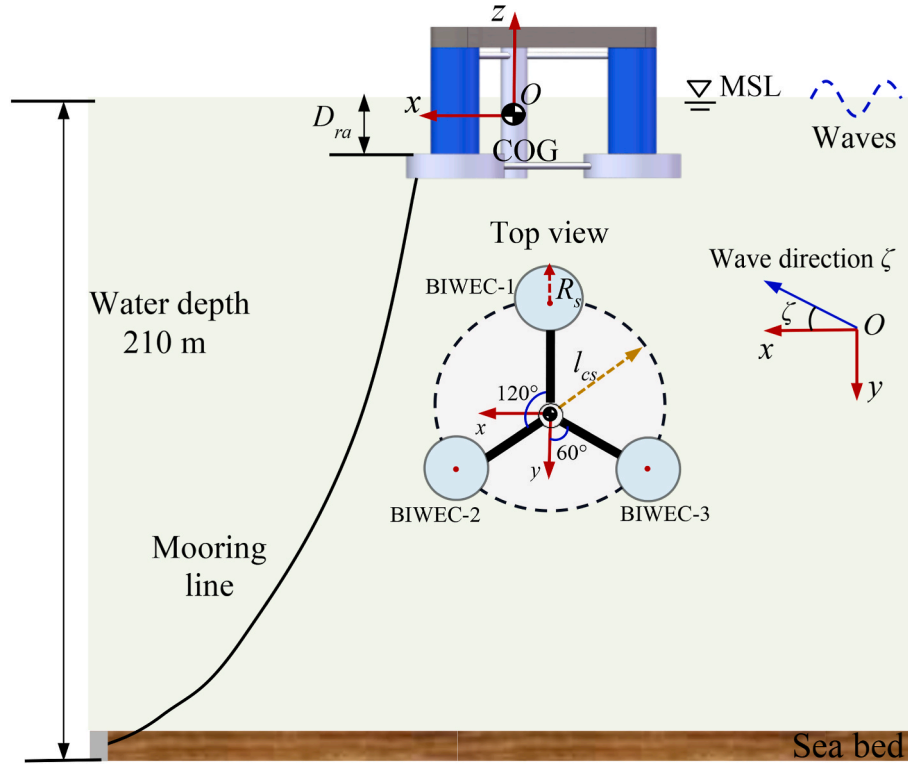


Fig. 2. Geometric structural relationship of the proposed SSP-BIWECS.

work is based solely on the single heave motion of the SSP and a fixed damping coefficient, neglecting a detailed analysis of the system's other motions and the electric energy conversion structure.

To address the above issue, this paper proposed a multiple electromagnetic damping adaptive synergy optimization method to meet the needs of vibration control under different wave conditions. To illustrate the proposal, a design example is presented using a semi-submersible floating platform synergizing multiple built-in WECs (SSP-BIWECS). An advanced theoretical model is developed that incorporates multiple degrees of freedom (DOFs) for the SSP. By including additional motion modes (surge, sway, pitch, and roll), the theoretical model can more comprehensively validate the effectiveness of the proposed approach

and enable the evaluation of its effects on other motions of the SSP. Moreover, we incorporate the electric energy conversion structure into the mathematical model, making it more representative of the actual physical system. This provides a foundation for implementing the multiple electromagnetic damping adaptive synergy optimization method and subsequent performance comparison analysis. Hardware-in-the-loop (HIL) experiments are conducted to test the operational logic of the proposed control strategy on the actual controller, while also providing essential preliminary analysis for constructing the physical model. The contributions of this study are summarized as follows:

Table 1
Main structure parameters of the of the SSP-BIWEC.

Configuration	Item	Symbol	Value
SSP	Distance from the central column to the three side columns	l_{cs}	28.87 m
	Radius of the central column	R_c	6.44 m
	Radius of the three side columns	R_s	11.88 m
	Operating draft	D_{ra}	31.5 m
	Center of gravity	COG	(0,0, 0 m)
	Mass	m_p	1500 t
	Mass moment of inertia about X axis	I_{xx}	7.53×10^8 $\text{kg} \cdot \text{m}^2$
	Mass moment of inertia about Y axis	I_{yy}	7.53×10^8 $\text{kg} \cdot \text{m}^2$
	Mass moment of inertia about Z axis	I_{zz}	1.22×10^9 $\text{kg} \cdot \text{m}^2$
	Water depth	D_p	210 m
BIWEC	Oscillating body mass	m_w	105 t
	Radius of the oscillating body	R_w	2.5 m
	Mass moment of inertia about X axis	I_{xx}	1.22×10^6 $\text{kg} \cdot \text{m}^2$
	Mass moment of inertia about Y axis	I_{yy}	1.22×10^6 $\text{kg} \cdot \text{m}^2$
	Mass moment of inertia about Z axis	I_{zz}	2.07×10^6 $\text{kg} \cdot \text{m}^2$

- (1) An active optimization approach using multiple electromagnetic damping adaptive synergy is proposed to achieve more effective vibration control and energy harvesting under varying wave conditions.
- (2) A multi DOF mathematical model of the SSP-BIWEC is developed, integrating ocean waves, multi-body dynamics, and electricity conversion to provide a more comprehensive and realistic analysis framework.
- (3) A hardware-in-the-loop (HIL) experiment for the proposed optimization approach is conducted using the advanced Real-time Laboratory (RTLAB), testing the operational logic of the control strategy on the actual controller and laying the foundation for subsequent physical prototype testing.

The rest of this paper is structured as follows: Section 2 elaborates on the detailed mechanical and electricity conversion structure design and operational principles of SSP-BIWEC. In Section 3, the multi DOF mathematical model of SSP-BIWEC is derived, and its multi-physical domain model is validated by ANSYS-AQWA and Simulink. Section 4 conducts vibration control-oriented parameters optimization, offering further performance comparison analysis. Section 5 establishes a hardware-in-the-loop experiment to validate the vibration control effect

of SSP-BIWEC under varying excitation frequencies and wave heights. Finally, Section 6 presents the conclusions drawn from this study.

2. Mechanical and electricity conversion structure design

The design scheme of the SSP-BIWEC is shown in Fig. 1, with (a) presenting an overall system schematic, (b) illustrating the mechanical structure of SSP synergized multiple built-in WECs, and (c) indicating the electricity conversion structure of built-in WEC. As shown in Fig. 1 (a) and (b), the proposed SSP-BIWEC consists of a SSP and three built-in WECs. The three BIWECs are installed on the side columns of the SSP, which are named BIWEC-1, BIWEC-2, and BIWEC-3, respectively. The protective barrier formed by the SSP's side columns ensures a reliable seal against water ingress, safeguarding the internal components of the WEC. More importantly, this internal installation structure form can decouple the WEC and ocean waves, making WEC directly absorb the vibration energy from SSP.

To be specific, the platform vibrates when waves pass through the SSP-BIWEC. Due to the inherent inertia of the object, the platform's vibration energy is transmitted to the built-in WEC. Fig. 1(b) shows that the BIWEC is composed of a permanent magnet synchronous generator (PMSG), a coupler, a ball screw, an oscillating body, and a fixed pin. The oscillating body is connected to the ball screw via a screw pair. As the oscillating body moves vertically due to the platform's vibrations, this motion is translated into the reciprocating rotation movement of the screw nut. This reciprocating rotation movement drives the PMSG to produce electricity through the coupler. The generated electricity is then stored in a battery by a power converter (See Fig. 1(c)).

To give readers a clearer overall understanding of the proposed structure, Fig. 2 illustrates the geometric structural relationships of the SSP-BIWEC concept. The platform is located in waters with a depth of 210 m and a draft of 31.5 m, and uses mooring lines for station-keeping. The fixed coordinate system O is located at the position of the SSP's Center of gravity (COG) when the SSP is in a resting state. The overall structure is axially symmetrical, and the horizontal distance l_{cs} between the center point of each BIWEC and the SSP's COG is equal to 28.87 m, and the radius of the three side columns equals 11.88 m. The detailed structure parameters are listed in Table 1. Moreover, the wave direction is defined as the clockwise angle ζ between the wave propagation direction and the positive x-axis of the SSP.

3. Modeling and validation

Under wave excitation, SSP-BIWECC comprises two primary elements:

- 1) The mechanical movement involving floating platforms and the

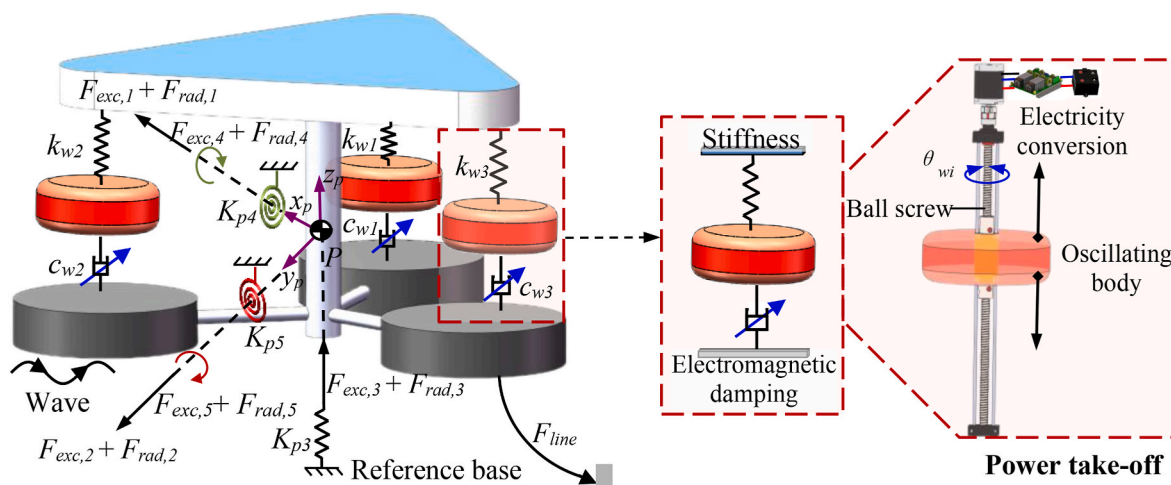


Fig. 3. Dynamic model of the SSP-BIWECC.

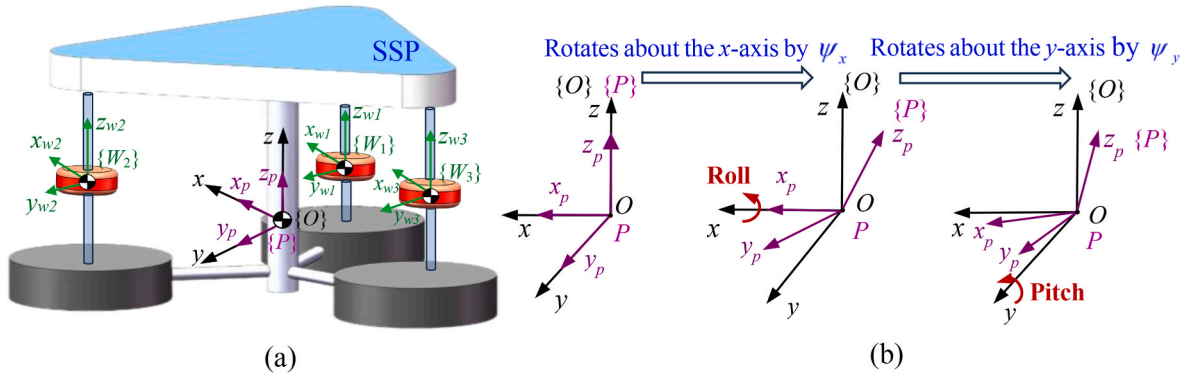


Fig. 4. Structure sketch of SSP-BIWEC: (a) establishment of the coordinate system and (b) successive rotations in the fixed coordinate system.

oscillating body of BIWEC; 2) The generation of electrical energy through the oscillating body of BIWEC driving the electricity conversion structure. To comprehensively analyze the performance of SSP-BIWEC, this section first establishes the multi-body motion equation of SSP-BIWEC based on linear potential theory [19] and the Newton-Euler method. Subsequently, according to the PMSG and power converter operation mechanism, the dynamics of the electric conversion structure are constructed. Based on this, the relation of equivalent electromagnetic damping and duty cycle is derived. Finally, the multi-domain simulation model of SSP-BIWEC is established by ANSYS-AQWA and Simulink, which is used to verify the accuracy of the theoretical modeling.

3.1. Mathematical model

The conventional SSP offers motion with six degrees of freedom, namely, surge, sway, yaw, heave, pitch, and roll. Among these motions, its heave motion is most critical since its small water plane, draft, and limited hydrodynamic damping. Note that, the SSP's yaw motion is nearly zero due to the symmetry of the SSP, and is therefore neglected in this study. Different from the traditional mechanical motion, the SSP possesses not only the physical mass of m_p , but also experiences added mass induced by the surrounding seawater. As illustrated in Fig. 3, the three oscillating bodies, each with a mass of m_w , are connected to the SSP through the power take-off (PTO) structure composed of a ball screw and PMSG. Because the operational characteristics of the PTO structure, it can generally be equivalent to a linear spring-damp-mass system with good agreement [20]. The stiffness and electromagnetic damping coefficients of three PTOs are denoted as k_{w1} , c_{w1} , k_{w2} , c_{w2} , k_{w3} and c_{w3} , respectively. Additionally, the SSP is subjected to mooring line forces F_{line} , wave excitation forces $F_{exc,j}$ and radiation damping forces $F_{rad,j}$, where j represents the motion mode of the SSP. Specifically, $j = 1, 2, 3, 4, 5$ corresponds to surge, sway, heave, roll, and pitch, respectively. Meanwhile, the SSP is connected to a reference base through hydrostatic stiffness K_{p3} , K_{p4} and K_{p5} , which correspond to the heave, pitch, and roll directions, respectively. Notably, the reference base is not an actual physical foundation, and it symbolizes the support offered by seawater.

3.1.1. Mechanical dynamic model

The proposed system consists of an SSP and three BIWECS. Due to the SSP's multiple motion DOFs under wave excitation, the relative motion between these bodies becomes more complex. To clearly establish the mechanical dynamic model of the proposed system, we constructed the coordinate frames as shown in Fig. 4(a). The fixed coordinate system $O-xyz$ is located at the position of the SSP's COG in a resting state. The moving coordinate system $P-x_p y_p z_p$ is established at COG of the SSP, while the local coordinate system $W_i - x_{wi} y_{wi} z_{wi}$ ($i = 1, 2, 3$) is established at the COG of the three BIWECS. In still water, the x_p , y_p and z_p axes are in the same direction as the x , y , and z axes. Similarly, the x_{wi} ,

y_{wi} , and z_{wi} axes are aligned with the x , y , and z axes, respectively.

As shown in Fig. 4(b), the SSP's orientation is described by successive rotations in the fixed coordinate system, with the corresponding rotation matrix given in Eq. (1)

$${}^P R_O = R_{rot(y)}(\psi_y) R_{rot(x)}(\psi_x) = \begin{bmatrix} \cos \psi_y & \sin \psi_y \sin \psi_x & \cos \psi_x \sin \psi_y \\ 0 & \cos \psi_x & -\sin \psi_x \\ -\sin \psi_y & \cos \psi_y \sin \psi_x & \cos \psi_x \cos \psi_y \end{bmatrix} \quad (1)$$

where ψ_x , and ψ_y are the Euler angles of the SSP rotating around x and y , respectively.

Note that when the SSP experiences pitch and roll motions, the three BIWECS also rotate with the SSP due to their built-in installation. Therefore, the rotation matrix ${}^{Wi} R_O$, which describes the orientation of the three BIWECS in the fixed coordinate system $O-xyz$, is equal to ${}^P R_O$. Based on this rotational relationship, the motion of the three BIWECS in local coordinate systems $W_1 - x_{w1} y_{w1} z_{w1}$, $W_2 - x_{w2} y_{w2} z_{w2}$, and $W_3 - x_{w3} y_{w3} z_{w3}$ can be obtained as follows.

$${}^{Wi} \mathbf{X}_{wi} = \mathbf{X}_p - \mathbf{X}_{wi} - {}^{Wi} R_O \mathbf{X}_{owi} \quad (2)$$

where $\mathbf{X}_p$ and $\mathbf{X}_{wi}$ represent the SSP's translation displacement vector and the BIWEC's translation displacement vector in the fixed coordinate system, respectively. $\mathbf{X}_{owi}$ represents the i -th BIWEC's initial position in the fixed coordinate system. Specifically, the $\mathbf{X}_{ow1} = [0 \quad -l_{cs} \quad 0]^T$ for BIWEC-1, $\mathbf{X}_{ow2} = [l_{cs}\sqrt{3}/2 \quad l_{cs}/2 \quad 0]^T$ for BIWEC-2, and $\mathbf{X}_{ow3} = [-l_{cs}\sqrt{3}/2 \quad l_{cs}/2 \quad 0]^T$ for BIWEC-3.

Based on the motion of the three BIWECS in their local coordinate systems, the PTO translation force acting on the SSP can be calculated using the stiffness and electromagnetic damping coefficients.

$$\mathbf{F}_{pto} = \sum_{i=1}^3 (\mathbf{K}_{wi} {}^{Wi} \mathbf{X}_{wi} + \mathbf{C}_{wi} \dot{{}^{Wi} \mathbf{X}_{wi}}) \quad (3)$$

where the $\mathbf{K}_{wi}$ and $\mathbf{C}_{wi}$ are the stiffness and electromagnetic damping coefficient matrices of the BIWEC, respectively, expressed as $\mathbf{K}_{wi} = \text{diag}(k_{wi}, k_{wi}, k_{wi})$ and $\mathbf{C}_{wi} = \text{diag}(c_{wi}, c_{wi}, c_{wi})$.

The PTO moment in pitch and roll directions acting on the SSP are generated by the PTO force in the heave direction. Combined with Eq. (3) and the moment definition, we can obtain as:

$$\begin{cases} M_{ptok} = \sum_{i=1}^3 [F_{ptoi}(3,1) {}^{Wi} \mathbf{X}_{owi}(k,1)] & k = 1, 2 \\ \mathbf{M}_{pto} = [M_{pto1} \quad M_{pto2}]^T \end{cases} \quad (4)$$

The multi-body dynamic model of the SSP-BIWEC can be formulated by Newton-Euler method, which can be written as:

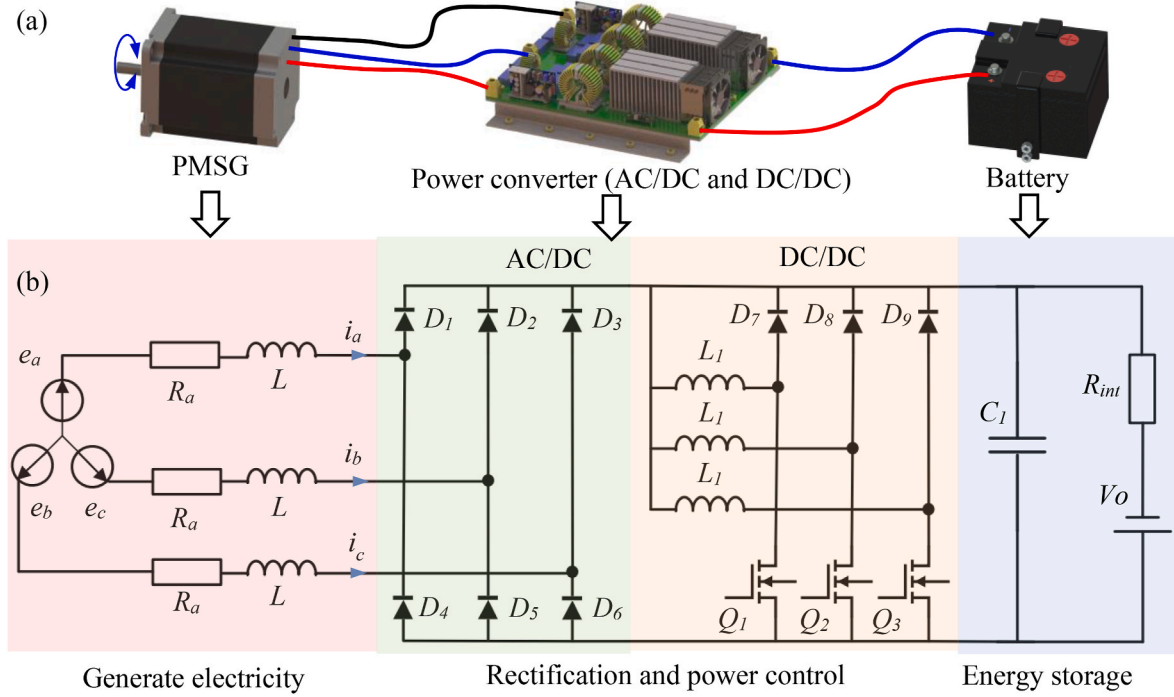


Fig. 5. Electricity conversion structure schematic: (a) physical circuit connection, (b) equivalent circuit.

$$\begin{cases} \mathbf{M}_p \begin{bmatrix} \ddot{\mathbf{x}}_p \\ \ddot{\theta}_p \end{bmatrix} = \mathbf{F}_{exc} + \mathbf{F}_{rad} + \mathbf{F}_{res} + \mathbf{F}_{line} + \begin{bmatrix} \mathbf{F}_{pto} \\ \mathbf{M}_{pto} \end{bmatrix} \\ \mathbf{M}_{wi} \ddot{\mathbf{x}}_{wi} = -\mathbf{F}_{ptoi} \end{cases} \quad (5)$$

$$\mathbf{F}_{res} = \mathbf{K}_p \begin{bmatrix} \mathbf{x}_p \\ \theta_p \end{bmatrix} \quad (10)$$

$$\mathbf{F}_{line} = \mathbf{F}_{wave,0} - \mathbf{K}_{line} \begin{bmatrix} \mathbf{x}_p \\ \theta_p \end{bmatrix} \quad (11)$$

where $\mathbf{M}_p$ is the inertia item of the SSP, and $\mathbf{M}_{wi}$ represents the inertia item of the i -th BIWEC. $\ddot{\mathbf{x}}_p$ and $\ddot{\theta}_p$ are the translation acceleration vector and rotation acceleration vector of SSP, respectively. $\ddot{\mathbf{x}}_{wi}$ and $\ddot{\theta}_{wi}$ are the translation acceleration vector and rotation acceleration vector of i -th BIWEC, respectively. $\mathbf{F}_{exc}$, $\mathbf{F}_{rad}$, $\mathbf{F}_{res}$, $\mathbf{F}_{line}$ are the wave excitation force vector, radiation force vector, restoring force vector, and the mooring force vector acting on the SSP, respectively.

Due to the rotation of SSP and BIWEC in wave excitation, its inertia items $\mathbf{M}_p$ and $\mathbf{M}_{wi}$ and can be expressed as:

$$\mathbf{M}_p = \begin{bmatrix} m_p \mathbf{E}_{3 \times 3} & \mathbf{0}_{3 \times 3} \\ \mathbf{0}_{3 \times 3} & {}^o \mathbf{R}_p {}^p \mathbf{I}_p ({}^o \mathbf{R}_p)^T \end{bmatrix}_{5 \times 5} \quad (6)$$

$$\mathbf{M}_{wi} = \begin{bmatrix} m_w \mathbf{E}_{3 \times 3} & \mathbf{0}_{3 \times 3} \\ \mathbf{0}_{3 \times 3} & {}^o \mathbf{R}_w {}^{wi} \mathbf{I}_w ({}^o \mathbf{R}_w)^T \end{bmatrix}_{5 \times 5} \quad (7)$$

where the ${}^p \mathbf{I}_p$ is the moment of inertia matrix of the SSP in the moving coordinate system $P - x_p y_p z_p$. The ${}^{wi} \mathbf{I}_w$ is the moment of inertia matrix of the BIWEC in the local coordinate system $W_i - x_{wi} y_{wi} z_{wi}$.

Based on widely derived expressions in the literature [20], the $\mathbf{F}_{exc}$, $\mathbf{F}_{rad}$, $\mathbf{F}_{res}$, and $\mathbf{F}_{line}$ can be expressed as follows:

$$\begin{cases} F_{exc,j} = \frac{H}{2} \text{Re}\{F_{RAO,j}(\omega_n)\} \cos(\omega_n t + \zeta_n) \\ \mathbf{F}_{exc} = \text{diag}(F_{exc,1}, F_{exc,2}, F_{exc,3}, F_{exc,4}, F_{exc,5}) \end{cases} \quad (8)$$

$$\mathbf{F}_{rad} = -\mathbf{M}_{add,\infty} \begin{bmatrix} \ddot{\mathbf{x}}_p \\ \ddot{\theta}_p \end{bmatrix} - \int_0^t \mathbf{K}(t-\tau) \begin{bmatrix} \dot{\mathbf{x}}_p(\tau) \\ \dot{\theta}_p(\tau) \end{bmatrix} d\tau \quad (9)$$

where H represents the regular wave height, $\text{Re}\{F_{RAO,j}(\omega_n)\}$ is wave excitation force or moment modulus per wave amplitude in the j -mode for wave frequency ω_n , with j representing the motion mode of the SSP, and ζ_n represents the corresponding phase angle. $\mathbf{M}_{add,\infty}$ is the infinite-frequency added mass, $\mathbf{K}(t)$ represents the retardation function that characterizes the radiation damping force. $\mathbf{K}_p = \text{diag}(0, 0, K_{p3}, K_{p4}, K_{p5})$, with K_{p3} , K_{p4} , K_{p5} represent the hydrostatic stiffness in heave, roll and pitch, respectively. $\mathbf{F}_{wave,0}$ is the total force exerted on the SSP by the entire mooring lines when the SSP is at rest, and $\mathbf{K}_{line}$ is the stiffness matrix of the mooring lines. The above hydrodynamic parameters can effectively analyze using the commercial software ANSYS AQWA. For brevity, the detailed solution process is not provided here, and they are shown in Appendix A.

To obtain the vibration response of the SSP-BIWEC, by substituting Eqs. (3) and (4) and (8)–(11) into Eq (5), we can obtain

$$\begin{cases} (\mathbf{M}_p + \mathbf{M}_{add,\infty}) \begin{bmatrix} \ddot{\mathbf{x}}_p \\ \ddot{\theta}_p \end{bmatrix} = \mathbf{F}_{exc} + (\mathbf{K}_p - \mathbf{K}_{line}) \begin{bmatrix} \mathbf{x}_p \\ \theta_p \end{bmatrix} \\ + \begin{bmatrix} \sum_{i=1}^3 (\mathbf{K}_{wi} {}^{wi} \mathbf{x}_{wi} + \mathbf{C}_{wi} \dot{\mathbf{x}}_{wi}) \\ \sum_{i=1}^3 [F_{ptoi}(3,1) {}^{wi} \mathbf{x}_{owi}(1,1)] \\ \sum_{i=1}^3 [F_{ptoi}(3,1) {}^{wi} \mathbf{x}_{owi}(2,1)] \end{bmatrix} + \mathbf{F}_{wave,0} - \int_0^t \mathbf{K}(t-\tau) \begin{bmatrix} \dot{\mathbf{x}}_p(\tau) \\ \dot{\theta}_p(\tau) \end{bmatrix} d\tau \\ \mathbf{M}_{wi} \ddot{\mathbf{x}}_{wi} = -\mathbf{K}_{wi} {}^{wi} \mathbf{x}_{wi} - \mathbf{C}_{wi} \dot{\mathbf{x}}_{wi} \quad i = 1, 2, 3 \end{cases} \quad (12)$$

Table 2
Main structure parameters of the electricity conversion structure.

Symbol	Parameter	Value	Unit
R_a	Stator phase resistance	2.62	Ohm
n_p	Number of pole pairs	12	–
k_e	Voltage constant	9.819	Vpeak L-L/(rad/s)
L	Inductance	1.87	mH
R_{int}	Internal resistance of battery	5.5	Ohm
l_b	Lead of ball screw	0.725	m

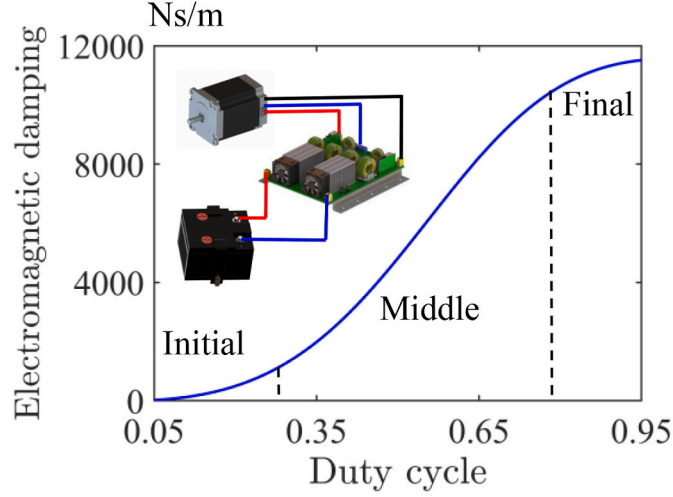


Fig. 6. The electromagnetic damping vs. duty cycle for SSP-BIWEC.

Using the typical fourth-order Runge-Kutta method [21], the second-order differential equation set Eq. (12) is solved, yielding the motion displacement of both the SSP and three BIWECs.

3.1.2. Electricity model

For the designed SSP-BIWEC, the PTO converts the oscillating body's mechanical energy into electrical energy. In this process, the PTO's output power can be obtained by analyzing PMSG's mechanical angular velocity and electromagnetic torque.

Following the above idea, we first determine the PMSG's mechanical angular velocity. As illustrated in the schematic diagram of the PTO in Fig. 3, the PMSG is connected to the oscillating body via ball screw. Thus, each PMSG's mechanical angular velocity ω_{mi} is related to the corresponding oscillating body's excitation velocity ${}^{wi}\dot{X}_{wi}$ by the following equation:

$$\omega_{mi} = 2\pi \frac{{}^{wi}\dot{X}_{wi}}{l_b} \quad (13)$$

where l_b represents the lead of the ball screw.

Moreover, since the PMSG generates electricity as alternating current (AC), when storing electric energy, the generated AC must be converted into direct current (DC) through an AC/DC rectifier. Then, it is stabilized by a DC/DC converter to charge the battery (See Fig. 5(a)). To obtain the electromagnetic torque of the PTO undergoes the above electric conversion, we present the equivalent circuit of the PTO in Fig. 5(b) based on circuit theory. Specifically, for the PMSG, continuous induced currents with a 120-degree phase difference exist between any two-phase currents. Rectifier (AC/DC) and power controller (DC/DC) are achieved using three-phase full-bridge passive circuits and three-phase interleaved parallel circuits, respectively. On the battery side, an open-circuit voltage source V_o in series with an internal resistance R_{int} models the battery's operating mechanism. When integrating circuit dynamics, the resistances of inductors, capacitors, and switching

elements are generally small and are ignored in this modeling process.

According to the circuit impedance matching principle [22], the power converter and the battery can be equivalent to the load resistance of the PMSG. Thus, based on the PMSG equivalent model [23] in Fig. 5(b), the electromagnetic force can be expressed as follows.

$$T_{elei} = \frac{3k_e^2 \omega_{mi}}{6R_a + 4R_{Li}} \quad (14)$$

where T_{elei} and R_{Li} denote the electromagnetic force and equivalent load resistance of the i th PMSG, respectively. R_a , k_e and ω_{mi} represent stator phase resistance, voltage constant and mechanical angular velocity of PMSG, respectively. It is worth noting that the PMSG, being a widely utilized motor, has been extensively studied in many literatures. For brevity, we will not repeat its detailed modeling here.

Furthermore, the rectifier (AC/DC) consists of passive circuits and has no impact on the equivalent load resistance R_L of the PMSG. While the DC/DC converter includes switching element (MOSFET), which can adjust the equivalent load resistance R_L by changing the duty cycle of MOSFET [24]. According to the circuit impedance matching principle, the relationship between the equivalent load resistance and duty cycle can be expressed as follows:

$$R_{Li} = \left(\frac{1 - D_i}{D_i} \right)^2 R_{int} \quad (15)$$

where D_i denotes the duty cycle of the i th BIWEC's DC/DC converter. Given the practical limitations of MOSFET regulation, the duty cycle D_i range is generally set between [0.05, 0.95]. The main structure parameters of the electricity conversion structure are given in Table 2.

For the above structural parameters, the following clarifications are made. Given that the proposed structure is still in the conceptual design stage, determining appropriate PTO parameters for real-world operation remains a challenge. At this stage, it is reasonable to leverage parameters from existing heave WEC prototypes and production AC motor datasheets to preliminarily evaluate system performance. To this end, we refer to the well-documented parameters of the WaveBob prototype [25], adopting its ball screw lead of 0.725 m and rated power of 1000 kW. Based on this power rating, we obtain PTO motor data from the AC motor datasheet provided by Huaneng Shantou Nan'ao Wind Power Co., Ltd. [26], a Chinese power generation equipment manufacturer.

Combining Eqs. (13)–(15) with the mechanical transmission structure of the PTO, the equivalent electromagnetic damping of each PTO can be expressed as follows:

$$c_{wi} = \frac{3\pi k_e^2}{3l_b R_a + 2l_b \left(\frac{1 - D_i}{D_i} \right)^2 R_{int}} \quad (16)$$

Based on Eq. (16) and the above structural parameters, the relationship between electromagnetic damping and duty cycle can be determined. Fig. 6 shows the electromagnetic damping in three stages as the duty cycle increases: the damping gradually increases in the initial stage, approaches linear growth in the middle stage, and slows down in the final stage.

As a result, the overall output power P_{all} of SSP-BIWEC can be calculated as follows:

$$P_{all} = \sum_{i=1}^3 \frac{2\pi {}^{wi}\dot{X}_{wi} T_{elei}}{l_b} \quad (17)$$

3.2. Model validation

The proposed SSP-BIWEC is simulated under the regular wave states (wave height $H = 2$ m, wave frequency $\omega = 0.628$ rad/s, wave direction $\zeta = 150^\circ$) and compared with the numerical analysis results to verify the correctness of the theoretical modeling. System structural parameters

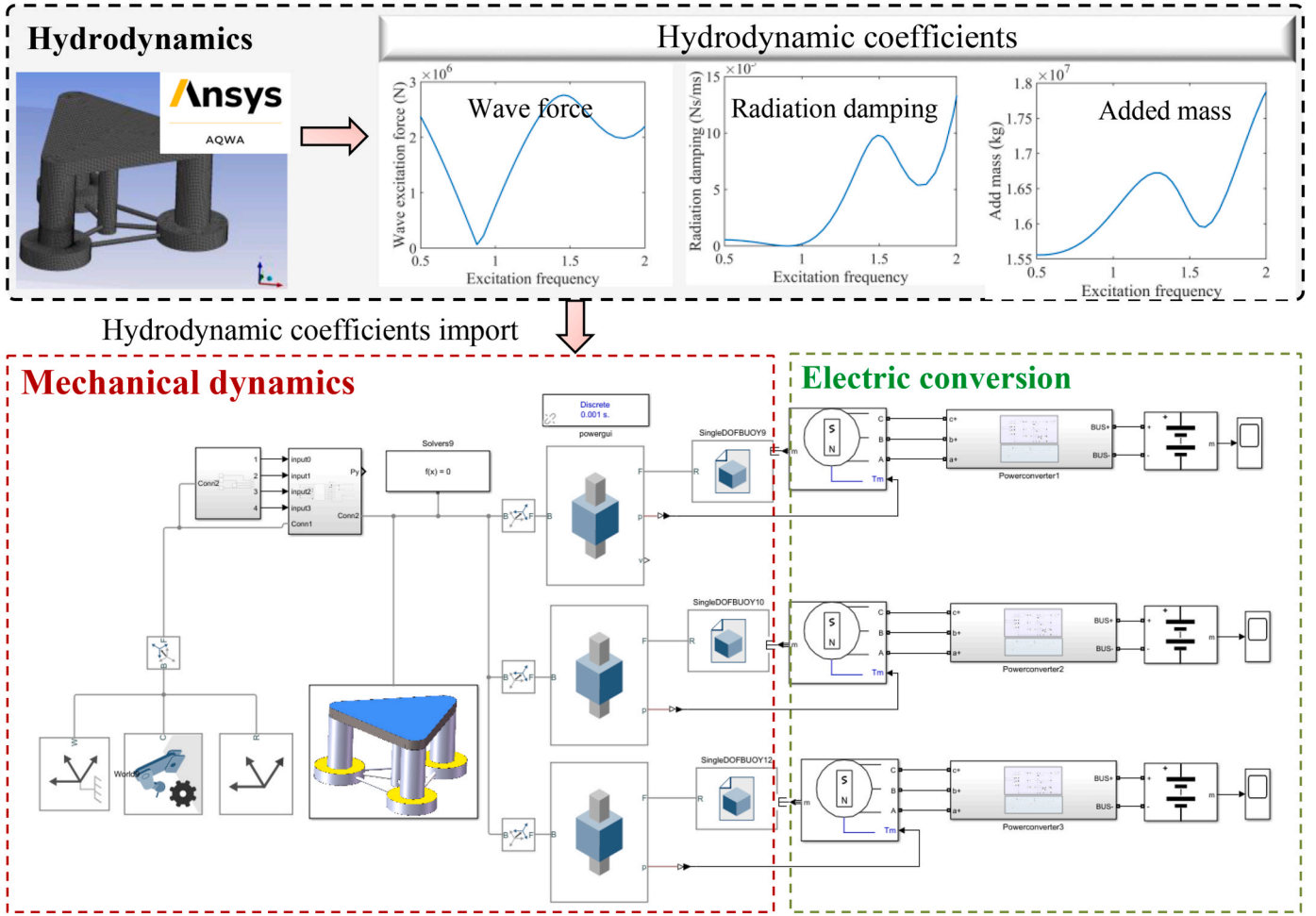


Fig. 7. Framework of the simulation model for SSP-BIWEC.

are adopted from Table 1, and the other parameters for the numerical analysis include $K_{p3} = 3.635 \times 10^3$ kN/m, $K_{p4} = 6.54 \times 10^5$ kNm/deg, $K_{p5} = 6.54 \times 10^5$ kNm/deg, $k_{w1} = 26.3$ kN/m, $c_{w1} = 4.64$ kN/m, $k_{w2} = 20.7$ kN/m, $c_{w2} = 4.08$ kN/m, $k_{w3} = 23.2$ kN/m, and $c_{w3} = 4.34$ kN/m.

For the different damping coefficients and spring stiffnesses of the three BIWECs used in the simulations, the following explanation is provided: For the simulation validation, the main objective is to assess the accuracy of the theoretical model. If the same PTO coefficients are used for all three BIWECs, there is a risk that the agreement between the theoretical analysis and the simulation results might be coincidental. In other words, the agreement would not result from the correctness of the theoretical model but rather from the specific choice of coefficients. To reduce this potential risk, we intentionally selected different PTO coefficients during the simulation. This approach improves the reliability of the validation process and strengthens confidence in the theoretical model's validity.

1) Simulation model construction

The SSP-BIWEC simulation involves three mutually coupled nonlinear systems: hydrodynamic, mechanical dynamics and energy conversion. To achieve this goal, the ANSYS-AQWA and Simulink-SimMechanics/electric are chosen for integration in simulation. This modelling technique allows to easily integrate the multi-body model with other blocks defining hydrostatic/hydrodynamic forces and the behavior of the electric conversion [27]. Fig. 7 shows the detailed simulation block diagram in Simulink, which illustrates the concrete implementation of the simulation system model of SSP-BIWEC.

- Regarding hydrodynamics, hydrodynamic coefficients such as added mass, radiation damping, and wave excitation force are obtained through commercial software ANSYS-AQWA. For brevity, these parameters are shown in Appendix A.
- Regarding mechanical dynamics, the multi-body dynamics module in SimMechanics is utilized to import mechanical structure parameters and establish constraints and motion relationships between the floating platform and the three BIWECs.
- Regarding electricity conversion, the power converter and battery model is constructed by power electronics simulation module in Simulink. The motion of SSP and output power of each BIWEC are simulated in combination with the model of hydrodynamics and mechanical.

2) Model verification

Based on the above structural parameters, the SSP's motion, the heave motions of the three BIWECs relative to the SSP, and their output power are obtained through simulation analysis in Simulink. From Figs. 8 and 9, the displacement and output power obtained by the theoretical analysis method (refer to Section 3.1) are in good agreement with the simulation results, demonstrating the correctness of the present theoretical analysis and calculation program.

4. Vibration control-oriented electromechanical parameters optimization

The dynamic model of SSP-BIWEC is derived in Section 3. Based on

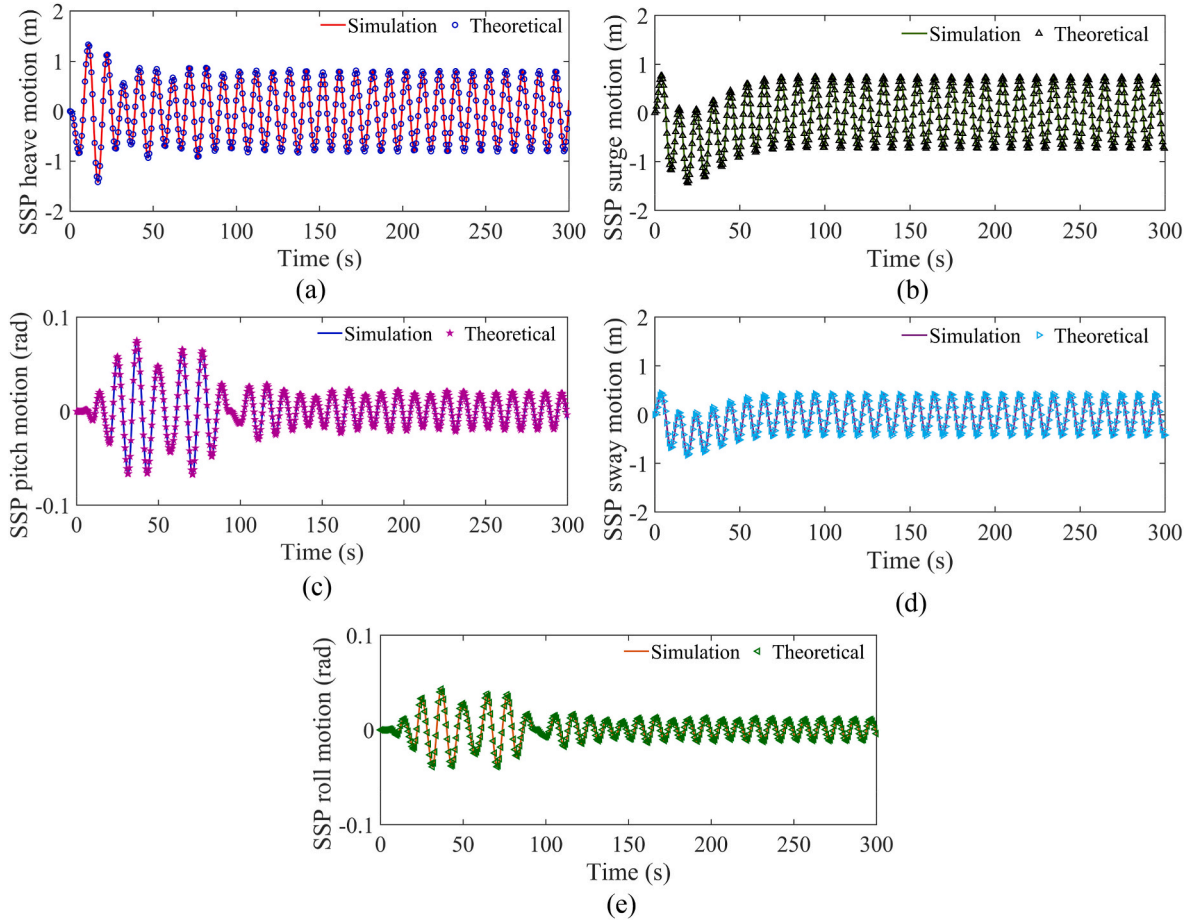


Fig. 8. Theoretical model verification of SSP motion: (a) heave motion, (b) surge motion, (c) pitch motion, (d) sway motion and (e) roll motion.

this, this section first formulates the two-stage optimization model oriented to vibration control. Subsequently, the multi variable hierarchical optimization procedure of the SSP-BIWEC is established, and optimal electromechanical parameters are obtained. Finally, the performance comparison analysis is conducted based on the above optimization results.

4.1. Two-stage objective function and constraint

The fundamental philosophy of vibration control is uncomplicatedly and efficiently regulating and reallocating vibration energy of target protected system [28]. For the proposed SSP-BIWEC, this vibration control process is achieved by adjusting the stiffness and electromagnetic damping of the PTO of each BIWEC. However, only the electromagnetic damping of the PTO can be adjusted via the duty cycle of the power converter (Refer to Eq. (16) in Section 3), whereas the stiffness of the PTO is generally fixed and difficult to alter once initially set. As a result, as shown in Fig. 10, the first step involves determining the stiffness of each BIWEC by minimizing the maximum vibration amplitude of the SSP in the most dangerous frequency range (i.e., around the resonance frequency). This step aims to transfer the vibration energy of the SSP to the BIWEC. Subsequently, the second step entails optimizing the electromagnetic damping of the each BIWEC to minimize the SSP's vibration amplitude at each wave frequency for different wave heights. This optimization process aims to convert the BIWEC's vibration energy

into electrical energy in response to changing external wave conditions.

Moreover, to make the optimization analysis results more universal, the non-dimensional parameters used here are given below:

$$\begin{cases} \mu = m_w / m_p, \Omega = \omega / \sqrt{K_{p3} / (m_p + m_{add,33})} \\ \gamma_i = \sqrt{k_{wi} / m_w} / \sqrt{K_{p3} / (m_p + m_{add,33})}, \xi_i = c_{wi} / 2\sqrt{k_{wi} m_w} \quad i = 1, 2, 3 \end{cases} \quad (18)$$

where μ is the mass ratio between each BIWEC and SSP; Ω is the excitation frequency ratio between the ocean wave and SSP; γ_i is the natural frequency ratio between each BIWEC and SSP; ξ_i is the damping ratio of each BIWEC. $m_{add,33}$ and K_{p3} denote the added mass and hydrostatic stiffness of SSP in heave direction, respectively.

In optimization solution tasks, it is imperative to consider the actual physical constraints of the system to ensure the practicality of the optimization results. As depicted in Fig. 11, for the SSP-BIWEC, there are two physical constraints: one related to mechanical motion and the other to the adjustable range of electromagnetic damping. For mechanical motion, it is necessary to ensure that the BIWEC operates within its maximum motion range to avoid equipment damage. For PTO's electromagnetic damping, the adjustment capacity of the actual power converter must be taken into account.

As a result, in this paper, the optimization task for SSP-BIWEC is established as follows:

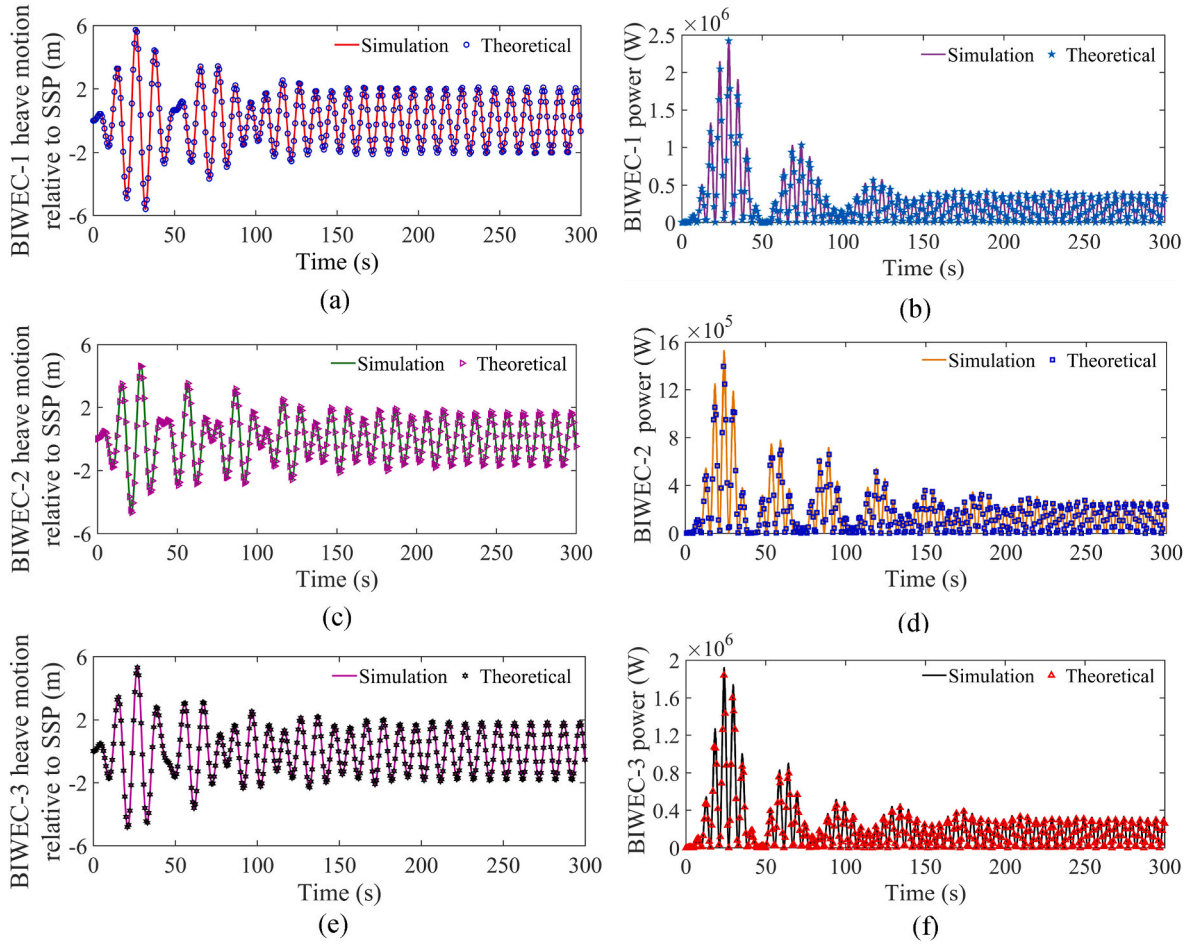


Fig. 9. Theoretical model verification of BIWEC motion and output power: (a) BIWEC-1 heave motion relative to SSP (b) BIWEC-1 output power, (c) BIWEC-2 heave motion relative to SSP, (d) BIWEC-2 output power, (e) BIWEC-3 heave motion relative to SSP and (f) BIWEC-3 output power.

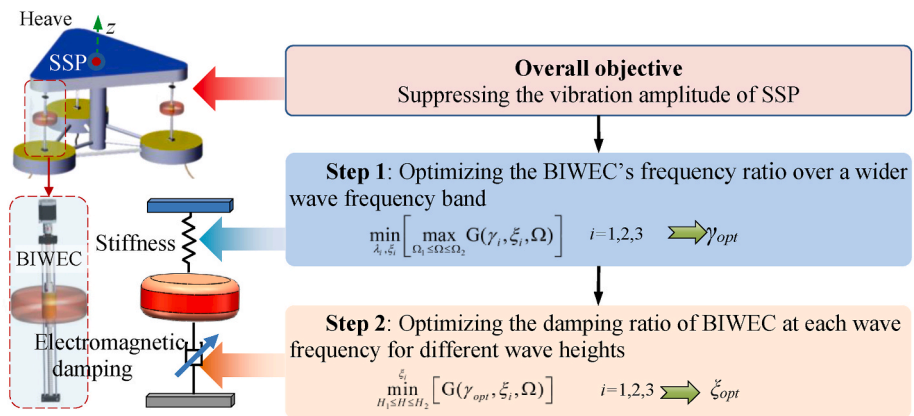


Fig. 10. The two-stage optimization objective.

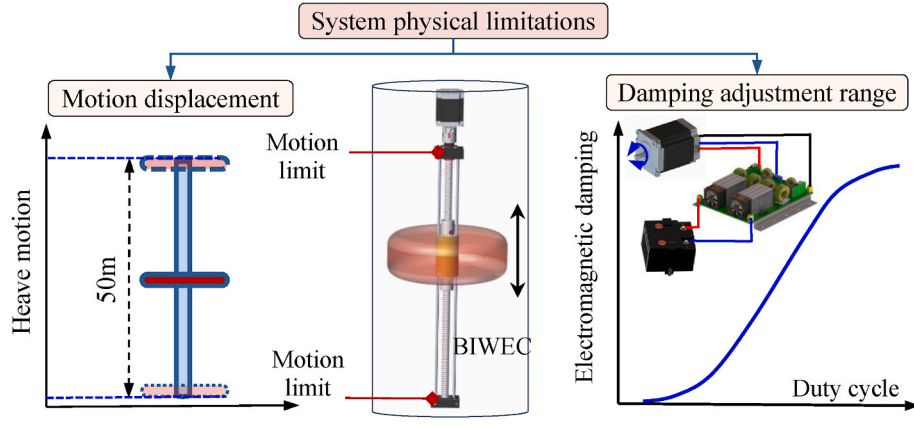


Fig. 11. The constraints for system parameter optimization.

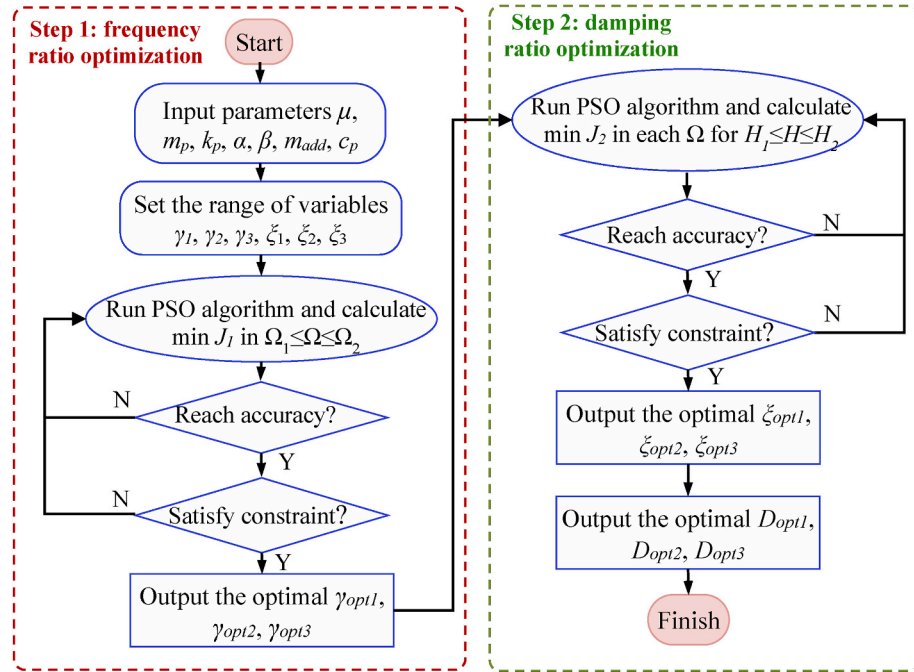


Fig. 12. The multivariable hierarchical optimization procedure.

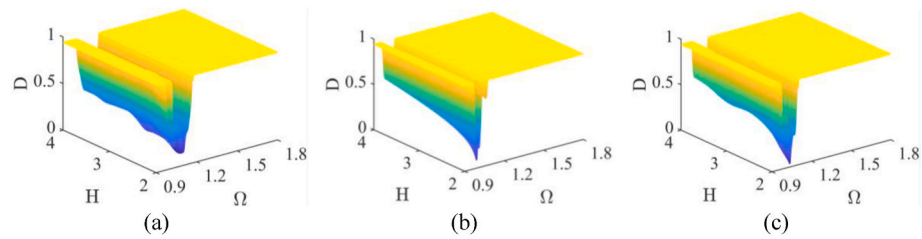


Fig. 13. The optimal duty cycles under different wave conditions: (a) power converter of BIWEC-1, (b) power converter of BIWEC-2, (c) power converter of BIWEC-3.

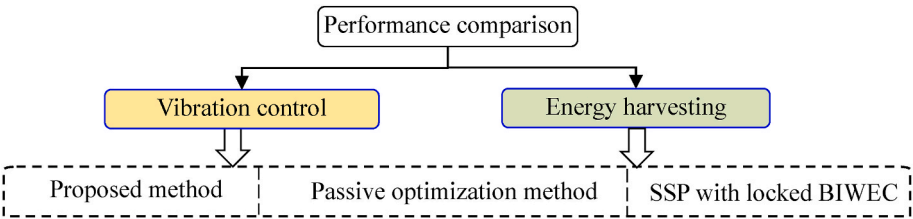


Fig. 14. Framework of performance comparison analysis of proposed method.

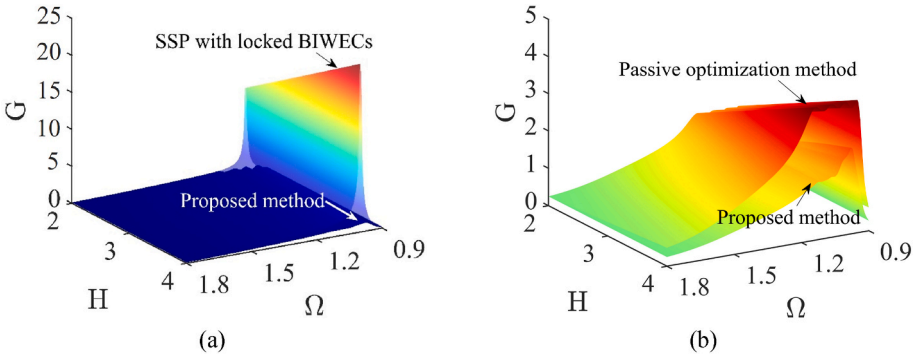


Fig. 15. Vibration amplitude G comparison under different wave conditions: (a) SSP with locked BIWEC and proposed method, (b) proposed method and passive optimization method.

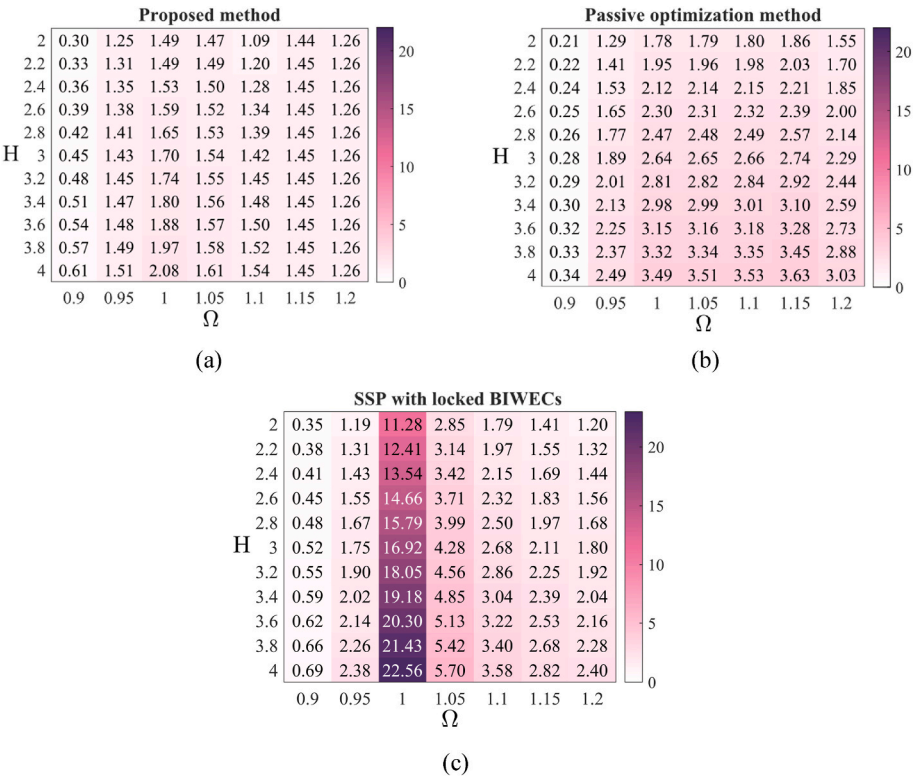


Fig. 16. Scatter diagram of platform vibration amplitude G under different wave conditions: (a) the proposed method, (b) the passive optimization method, (c) the SSP with locked BIWECs.

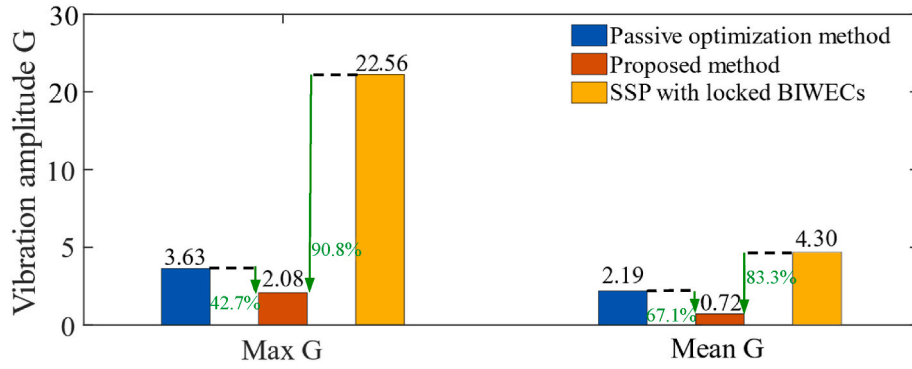


Fig. 17. The maximum and mean of the vibration amplitude G for the proposed method, the passive optimization method, and the SSP with locked BIWECS under different wave conditions.

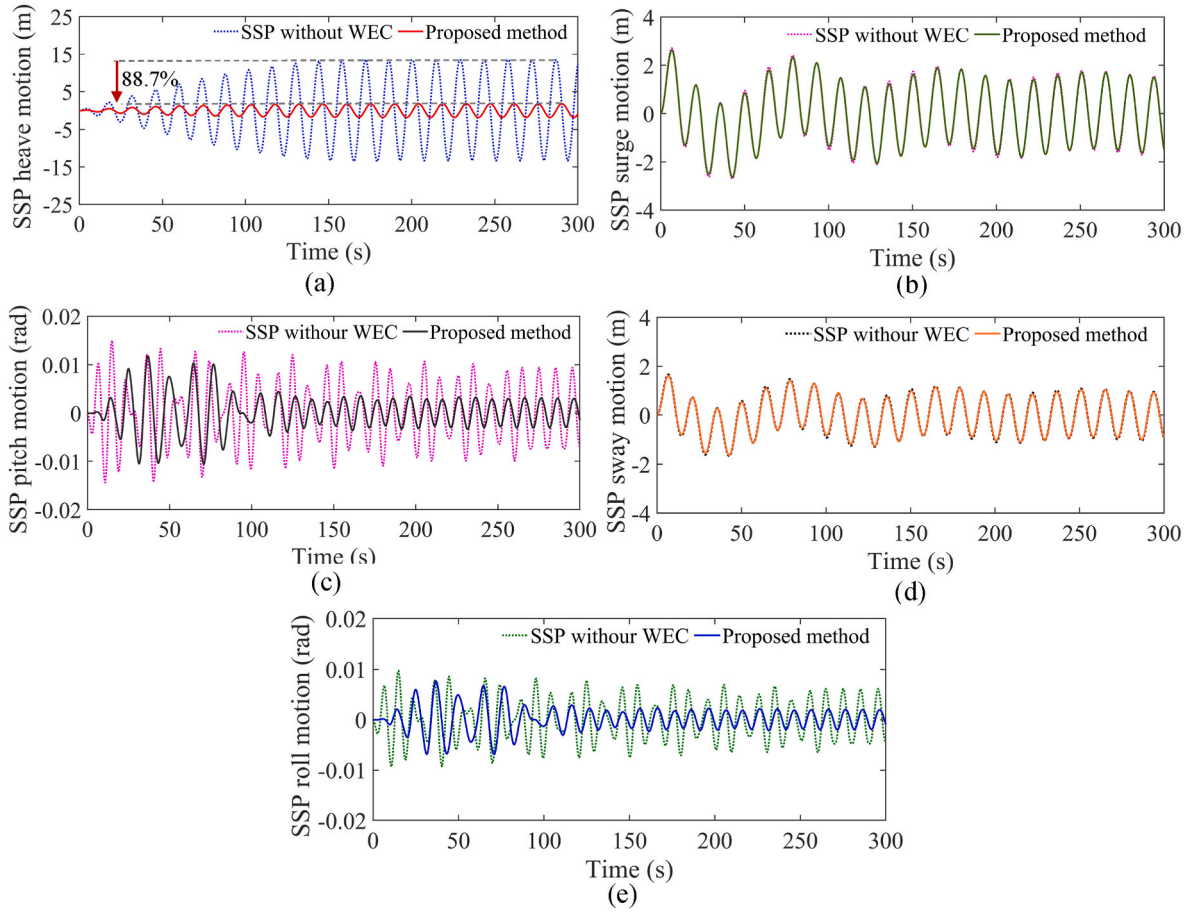


Fig. 18. The comparison of SSP motion for the SSP without WEC and the proposed method: (a) heave motion, (b) surge motion, (c) pitch motion, (d) sway motion and (e) roll motion.

$$\text{Step1} : J_1 = \min_{\gamma_i} \left[\max_{\Omega_1 \leq \Omega \leq \Omega_2} G(\gamma_i, \xi_i, \Omega) \right]$$

$$\text{Step2} : J_2 = \min_{H_1 \leq H \leq H_2} \left[G(\gamma_{opt}, \xi_i, \Omega) \right]$$

$$s.t \begin{cases} \max_{\Omega_1 \leq \Omega \leq \Omega_2} L_i \leq L_{\max} & i = 1, 2, 3 \\ 0 < \gamma_i < \alpha \\ 0 < \xi_i < \beta \end{cases}$$

(19)

where the L_i is the motion displacement for each BIWEC. $L_{\max}$ represents the maximum motion displacement for each BIWEC. The α, β are the maximum natural frequency ratio and maximum damping ratio in the optimization process, respectively. In this optimization task, the values of α, β are set to 0.8 and 0.2, respectively.

4.2. Multivariable hierarchical optimization procedure

The SSP-BIWEC contains damping effect due to wave excitation, making it difficult to obtain theoretical optimal solutions for the above optimization tasks. If using the traditional loop iteration method [29],

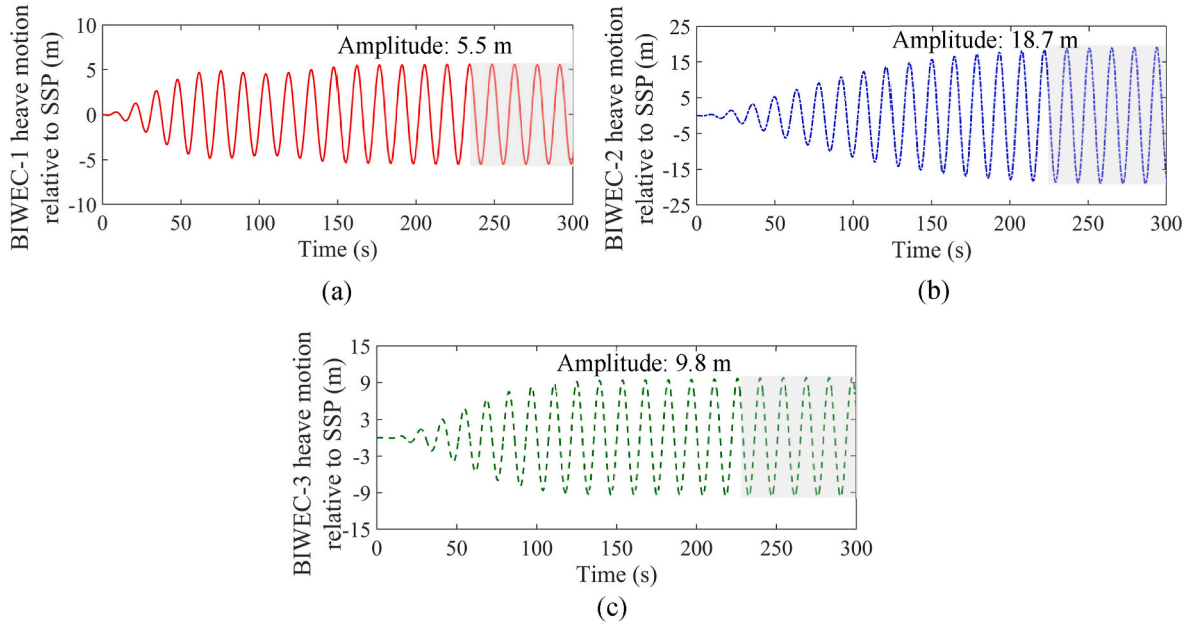


Fig. 19. The BIWEC's motion in the proposed method: (a) BIWEC-1 heave motion relative to SSP, (b) BIWEC-2 heave motion relative to SSP and (c) BIWEC-3 heave motion relative to SSP.

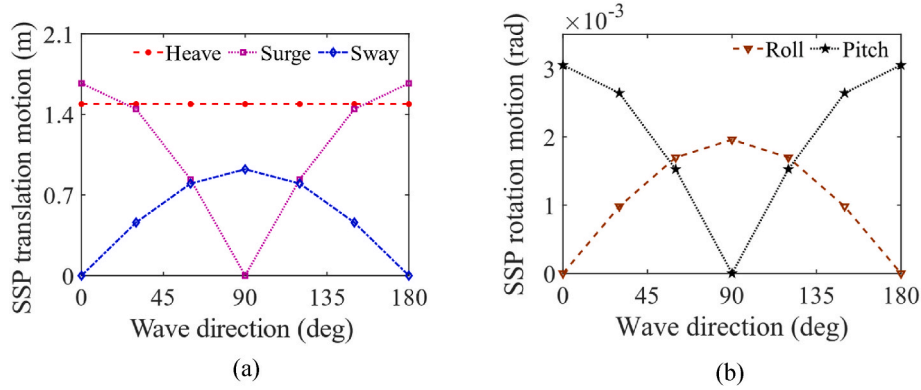


Fig. 20. The SSP's motion of the SSP-BIWEC under different wave direction: (a) translation motion, and (b) rotation motion.

this multivariable optimization task will take a long time to solve and hardly achieve relatively high accuracy. Conversely, as a heuristic algorithm for multivariable optimization, particle swarm optimization (PSO) has short search times and high-precision global search capabilities. Given this, it is relatively appropriate to choose PSO for this two-stage multivariable optimization task which lacks theoretical optimal solutions. Fig. 12 presents the multivariable hierarchical optimization method using PSO. The specific steps are as follows:

As illustrated in Fig. 12, the optimization procedure comprises two distinct phases. Initially, the optimal frequency ratios (γ_{opt1} , γ_{opt2} , γ_{opt3}) that minimize the objective function J_1 within the range of $\Omega_1 \leq \Omega \leq \Omega_2$ are identified using the PSO algorithm. Subsequently, based on the determined optimal frequency ratios, the PSO algorithm is employed again to obtain the optimal damping ratios (ξ_{opt1} , ξ_{opt2} , ξ_{opt3}) at each excitation frequency ratio for varying wave heights. The corresponding optimal duty cycles (D_{opt1} , D_{opt2} , D_{opt3}) are then calculated according to Eq. (16), facilitating the modulation of the power converter.

Before executing the optimization procedure, the system structural parameter ranges in the optimization programming are set as $\gamma_i \in (0, 0.8)$, $\xi_i \in (0, 0.2)$ and $\Omega \in [0.9, 1.8]$. Other system structural parameters are fixed as $\mu = 0.07$. By employing the proposed optimization procedure in MATLAB 2022b, we obtain the optimal frequency ratios as

0.321, 0.328 and 0.387. The corresponding optimal duty cycles under different excitation frequency ratios and wave heights are illustrated in Fig. 13.

4.3. Performance optimization analysis

Given the randomness of ocean wave motion, as depicted in Fig. 14, the optimization performance of the proposed method is studied under different excitation frequency ratios and wave heights. Its performance is compared with the previously passive optimization method [18] and the SSP with locked WEC. Specifically, the analysis includes two dimensions: (i) performance analysis for vibration control and (ii) performance analysis for energy harvesting.

4.3.1. Vibration control performance of SSP in heave

Effectively suppressing the vibration of SSP in different wave conditions is a prerequisite to ensure its safe operation. In this subsection, we analyze the impacts of varying wave heights (ranging from 2 to 4 m), excitation frequency ratios (ranging from 0.9 to 1.8) and wave direction (150°) on the vibration amplitude G of the SSP, as illustrated in Figs. 15–17. The real Oman Sea [30] as the operational sea area to investigate proposed system performance, where the dominant wave

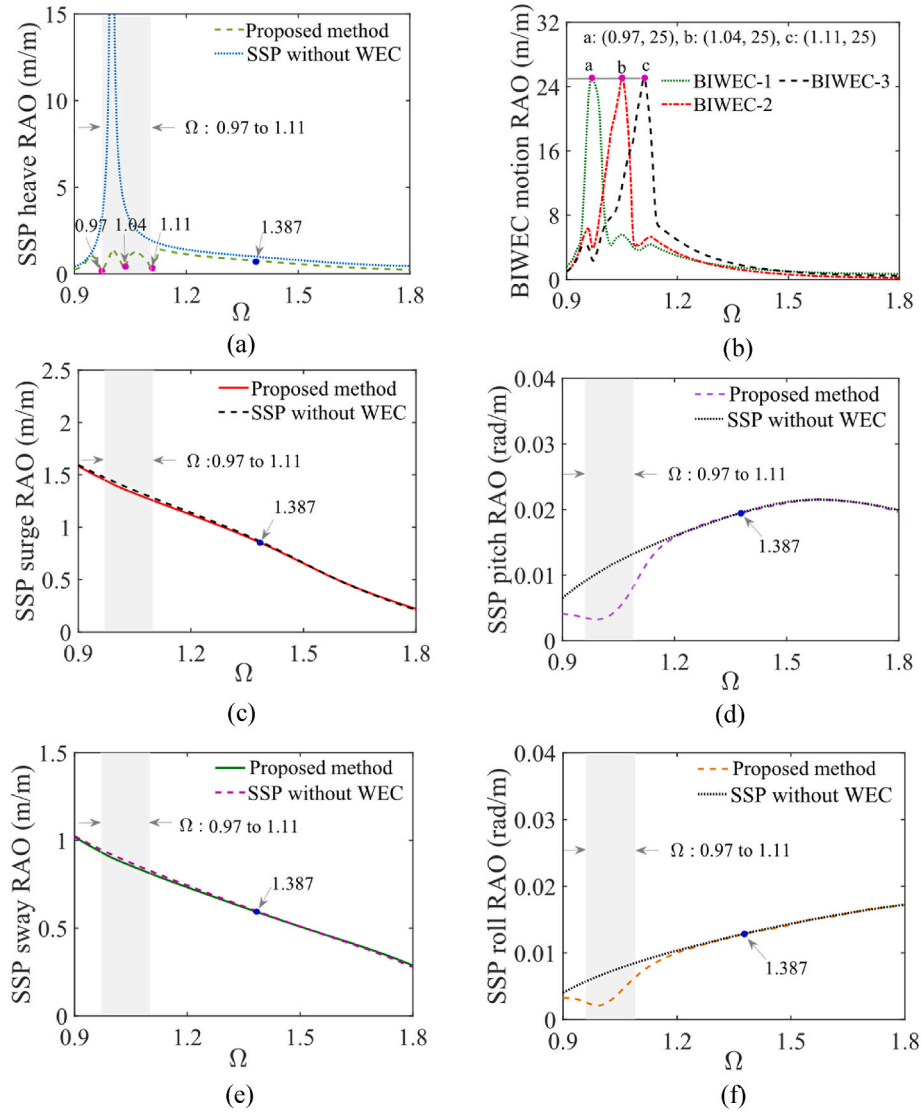


Fig. 21. Frequency-domain response amplitude operator (RAO): (a) SSP heave RAO, (b) BIWEC motion RAO, (c) SSP surge RAO, (d) SSP pitch RAO, (e) SSP sway RAO and (f) SSP roll RAO.

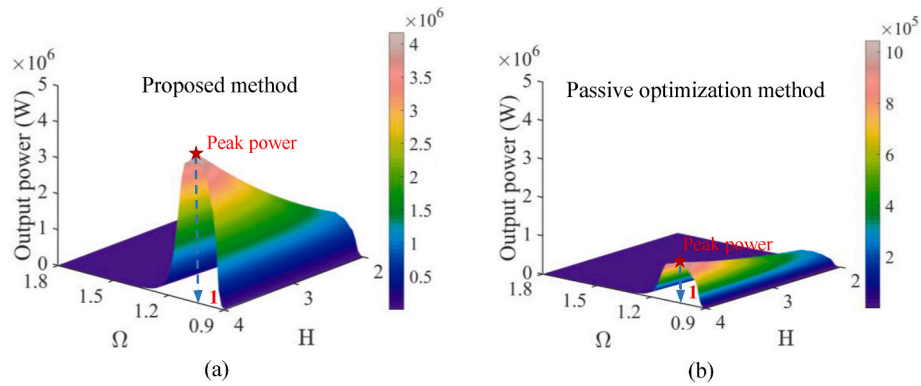


Fig. 22. Output power comparison under different wave conditions: (a) the proposed method, (b) the passive optimization method.

frequency is 0.628 rad/s (i.e., $\Omega = 1.387$). Using the commercial software ANSYS AQWA, we obtain the SSP's natural frequency in the heave direction to be 0.385 rad/s. The excitation frequency ratios used for optimization range from 0.9 to 1.8, corresponding to wave frequencies of 0.413–0.803 rad/s. This range is set to ensure it includes both the

dangerous frequency bands that the SSP may encounter and the dominant wave frequency (0.628 rad/s) of the operational sea area.

As shown in Fig. 15, compared to the SSP with locked BIWECs and the passive optimization method, the SSP's vibration amplitude (G) using the proposed method is consistently lower under various wave

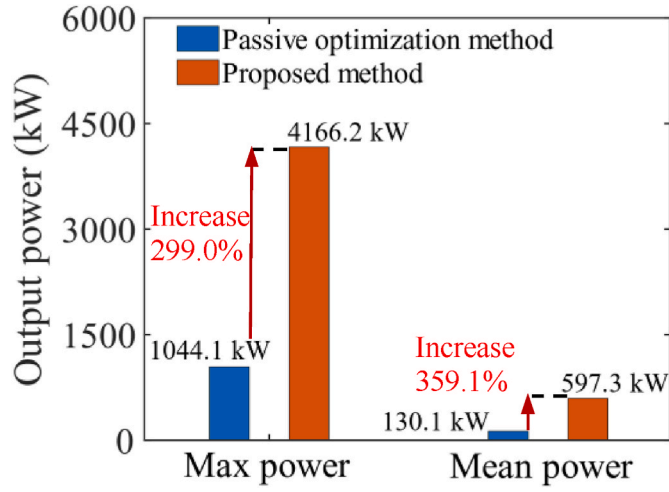


Fig. 23. The peak power and mean power for the proposed method and the passive optimization method under different wave conditions.

heights (H) and excitation frequency ratios (Ω). Furthermore, for a comprehensive comparison of the stability between these vibration control methods, the system's sensitivity to variations in wave conditions is also investigated. Fig. 16 shows that the proposed method exhibits significantly smaller vibration amplitude variations compared to both the passive optimization method and the SSP with locked BIWECS. Even in the dangerous resonant state (i.e., $\Omega = 1$), the SSP's vibration amplitude in the proposed method increases by only 0.59 m as the wave height rises from 2 m to 4 m. This increment is significantly smaller than the increase observed in the passive optimization method (1.71 m) and in the SSP with locked BIWECS (11.28 m). Further analysis of the data in Fig. 17 reveals that compared to the passive optimization method and SSP with locked BIWECS, the proposed method achieves a reduction of 42.7 % and 90.8 % in maximum vibration amplitude, and a reduction of 67.1 % and 83.3 % in mean vibration amplitude. These results further demonstrate the vibration control effectiveness of the proposed method and its robust adaptability to different wave conditions, which are highly desirable for practical applications.

4.3.2. Effects of wave direction on SSP's multi DOF motion

The wave direction changes in the real ocean environment and could impact the SSP's motion. To help readers better understand the effect of wave direction, this section presents a motion analysis of the SSP-BIWECS under both specific and varying wave directions, following a specific-to-general approach. The wave height is set to 2.4 m as an example. Specifically:

(1) At specify wave direction

The dominant wave direction (150°) observed in the real Oman Sea [30] is selected. Additionally, a wave frequency of 0.438 rad/s (i.e., near the SSP's heave resonant frequency) is chosen to better demonstrate the vibration suppression effect of the proposed method. As shown in Fig. 18, compared with the SSP without the WEC, the proposed method enables the SSP to experience an 88.7 % reduction in vibration amplitude in the heave direction. Interestingly, although the BIWECS have only a single heave degree of freedom (DOF), the vibration amplitudes of the SSP in roll and pitch also show a certain reduction. This can be attributed to two factors: i) the differences in vibration amplitude among the three BIWECS during operation (see Fig. 19), which generate moments acting on the SSP. This provides a control foundation for suppressing its roll and pitch vibrations, and ii) through PTO stiffness optimization, the natural frequencies of the three BIWECS are adjusted to match the SSP's heave resonant frequency (i.e., approaching the current set wave frequency). This adjustment mode enables vibration energy in roll and pitch to be effectively transferred into the BIWECS system, thereby reducing the SSP's vibration amplitude in these directions. On the other hand, the vibration amplitudes in the surge and sway directions remain nearly unchanged despite the presence of the BIWECS. This is because the roll and pitch angles are relatively small, resulting in negligible reactive forces in the surge and sway directions generated by the BIWECS. Compared to the SSP's reduced vibrations in heave, roll, and pitch, the three BIWECS exhibit much larger vibration amplitudes. Since changes in vibration amplitude reflect changes in energy, these results further confirm that the SSP's vibration energy is

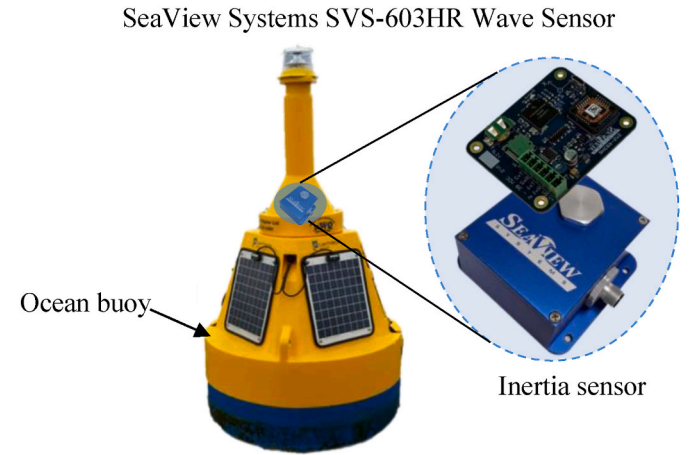


Fig. 25. The external view of the SVS-603HR wave sensor from Sea-View Systems.

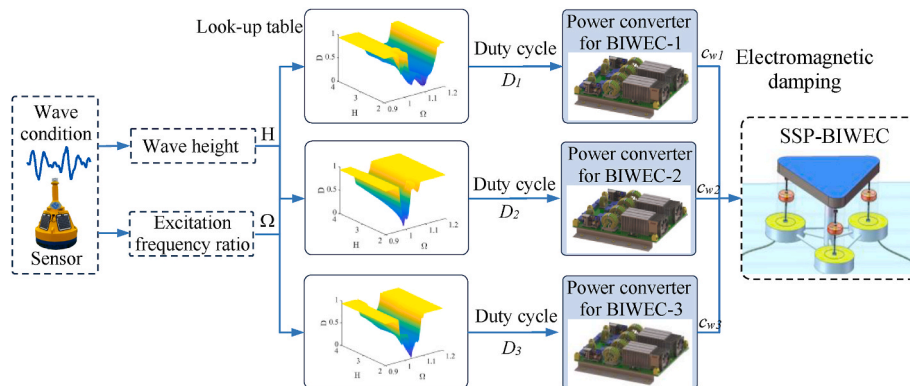


Fig. 24. The configuration of the SSP-BIWECS vibration controller based on lookup table.

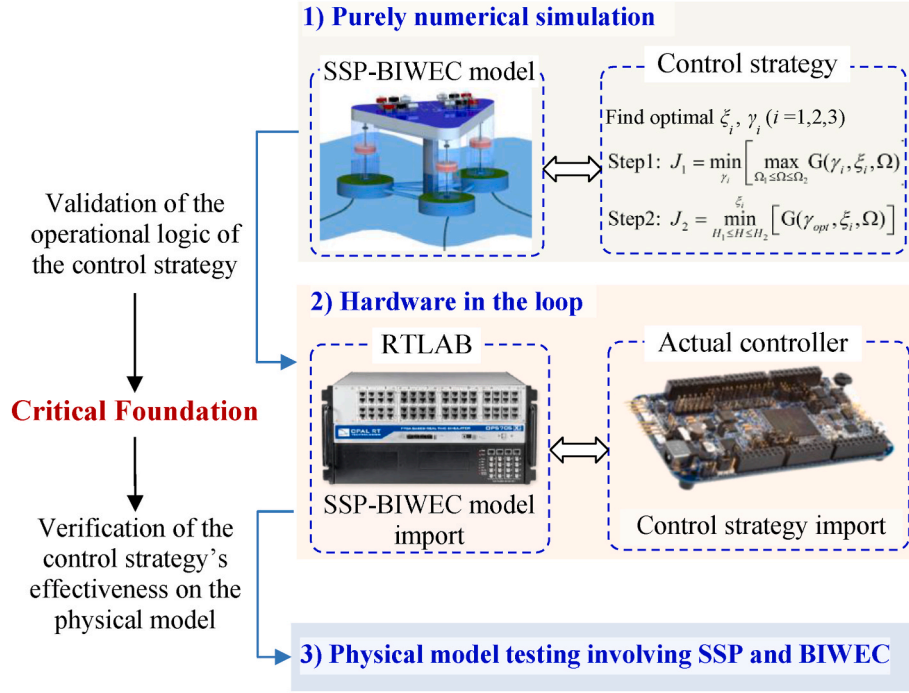


Fig. 26. Role of HIL testing to the development of the SSP-BIWEC system.

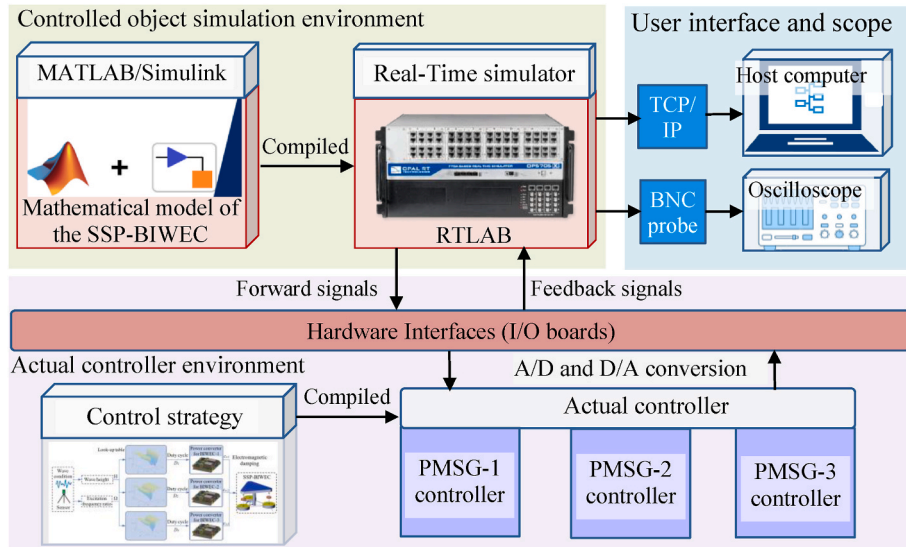


Fig. 27. The HIL test architecture for SSP-BIWEC.

effectively transferred to the BIWECs.

2) At varying wave direction

Considering the axial symmetry of the proposed SSP-BIWEC, we further analyze the multi-DOF motion of the SSP under wave directions ranging from 0° to 180° . As shown in Fig. 20, the SSP's vibration amplitude in the heave direction is nearly unaffected by changes in wave direction. This is due to the unidirectional nature of the heave response, that is, wave direction has a small impact on the heave hydrodynamics of the axially symmetric SSP-BIWEC system. In contrast, the SSP's motions in surge, sway, pitch, and roll are significantly influenced by variations in wave direction due to complex hydrodynamic coupling. The

maximum vibrations in sway and roll occur at a wave direction of 90° , while the maximum vibrations in surge and pitch are observed at 0° and 180° .

4.3.3. Response amplitude operator of SSP in multi DOF direction

This section further analysis the SSP's response amplitude operators (RAOs) in the heave, surge, pitch, sway, and roll directions (see Fig. 21). Note that the SSP's yaw motion is nearly zero due to its symmetry, and is therefore neglected in this study. As shown in Fig. 21(a), the proposed method can significantly reduce the SSP's heave RAO peak in the range near the resonant frequency (0.97–1.11). And the originally sharp single peak transforms into a smoother curve with multiple waves. Observations show that the frequencies at the troughs between adjacent waves

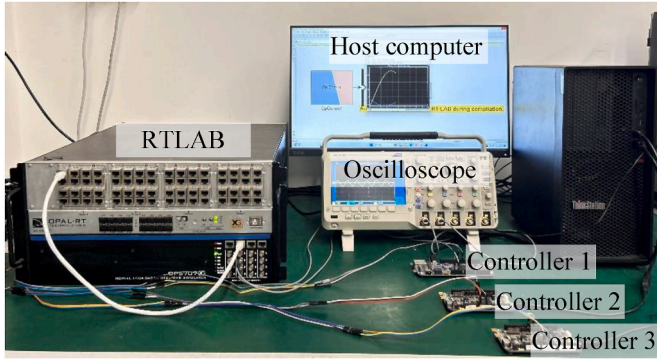


Fig. 28. HIL vibration control experiment bench of SSP-BIWE.

Table 3

Device configuration in the HIL test bench.

Hardware	Specifications
Real Time Laboratory (RTLAB)	4 cores/3.8 GHz processor, Xilinx Virtex-7 FPGA board, 16 analog channels, 64 digital channels
PMSG low-lever controller	2 × 160 MHz Power Architecture NXP 5744P
Oscilloscope	Tektronix-DPO 2024B, 200 MHz, 4 channels
Host computer	Intel i7 3.2 GHz, 16 GB memory

align exactly with the vibration peak frequencies of the three BIWECS (see Fig. 21(b)). Benefiting from this frequency-matching mode, the three BIWECS exhibit significantly larger vibration response peaks. Since changes in vibration amplitude reflect changes in energy, these results confirm that the proposed method effectively transfers and disperses the SSP's vibration energy to the three BIWECS.

Interestingly, although the BIWECS have only a single heave DOF, the proposed method reduces the vibration amplitudes of the SSP in the roll and pitch directions to a certain extent. This can be attributed to two factors: i) the differences in vibration amplitude among the three

BIWECS during operation (see Fig. 21(b)), which generate moments acting on the SSP. This provides a control foundation for suppressing its roll and pitch vibrations, and ii) through PTO stiffness optimization, the natural frequencies of the three BIWECS are adjusted to match the SSP's heave resonant frequency. As a result, the SSP's vibration energy in roll and pitch directions is effectively transferred to the BIWECS system, thereby reducing the SSP's vibration amplitudes in these directions.

In addition to the above dangerous frequency range, we also investigated the SSP's motion under the dominant wave frequency of 0.628 rad/s (i.e., $\Omega = 1.387$) based on real Oman Sea conditions. At this excitation frequency, the proposed method has almost no effect on the SSP's vibration amplitudes in the heave, pitch, and roll directions. This can be attributed to the aforementioned frequency-tuning mechanism: when the wave frequency deviates from resonance, the BIWECS absorb only a limited amount of vibration energy from the SSP, resulting in a small effect on the SSP's motion in these directions. Moreover, we also observed that the SSP's vibration amplitudes in the surge and sway directions remain nearly unchanged across the entire frequency range (0.9–1.8). This is because the roll and pitch angles are relatively small, causing the BIWECS to generate almost no reactive forces acting on the SSP in the surge and sway directions.

4.3.4. Energy harvesting performance of BIWECS

In addition to vibration control, SSP-BIWECS incorporates energy harvesting capabilities through its WEC. This section analyzes proposed method's energy harvesting performance under various wave conditions.

As shown in Figs. 22 and 23, while the peak power for both the proposed method and the passive optimization method occurs at an excitation frequency ratio of 1, the proposed method demonstrates a 299.0 % increase in peak power and a 359.1 % increase in mean output power. This phenomenon can be attributed to the optimization method for PTO stiffness and damping coefficients. Specifically, the optimal PTO stiffness coefficients in both methods are fixed to match the SSP's resonant frequency, allowing both methods to harvest significant vibration energy into the BIWECS system at the resonant frequency (i.e., $\Omega = 1$). However, the proposed method adaptively tunes the damping

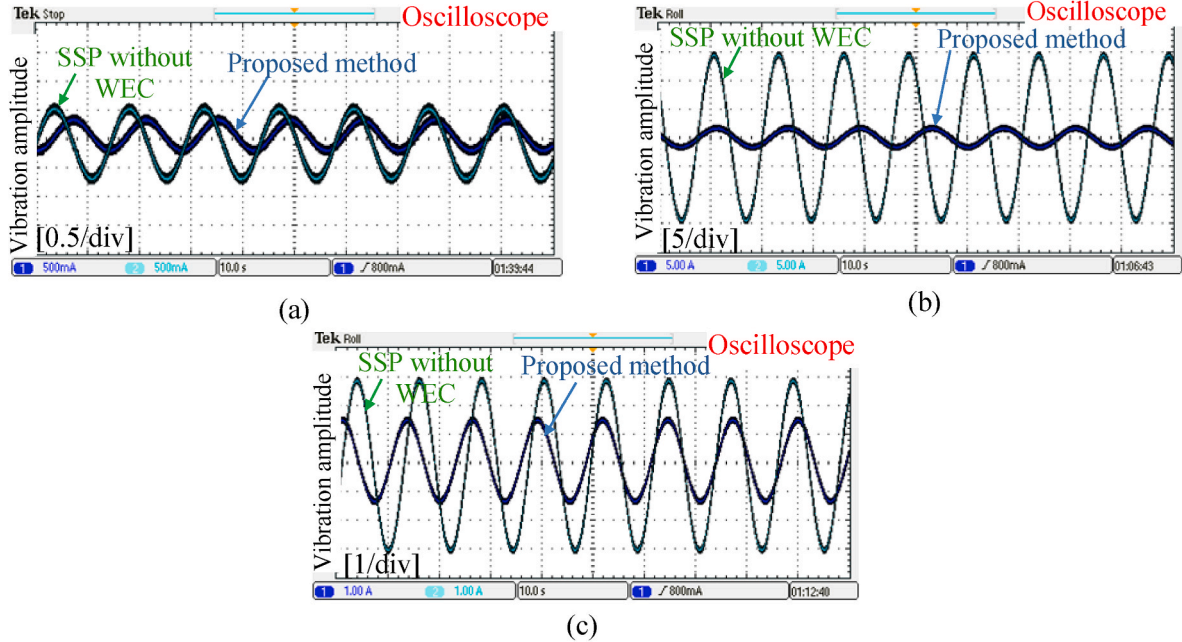


Fig. 29. Experimental waveform of vibration amplitude for the proposed method and SSP without control under varying excitation frequency ratio Ω : (a) under $\Omega = 0.9$, $H = 3$ m, (b) under $\Omega = 1$, $H = 3$ m, (c) under $\Omega = 1.1$, $H = 3$ m.

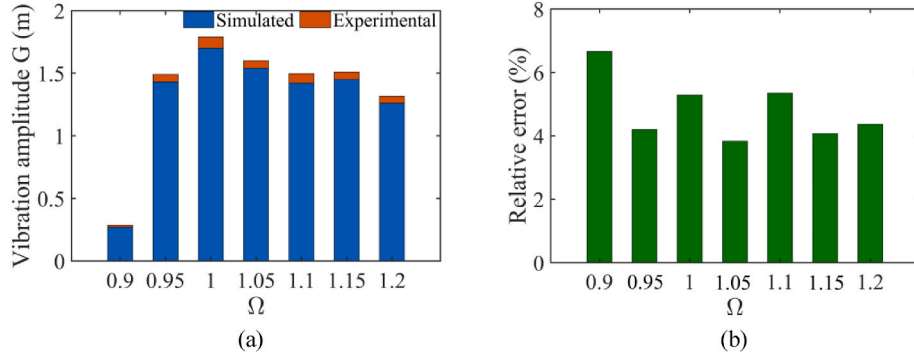


Fig. 30. Comparison of experiment and simulation for the proposed method with different excitation frequency ratio Ω : (a) vibration amplitude of platform, (b) relative error percentage.

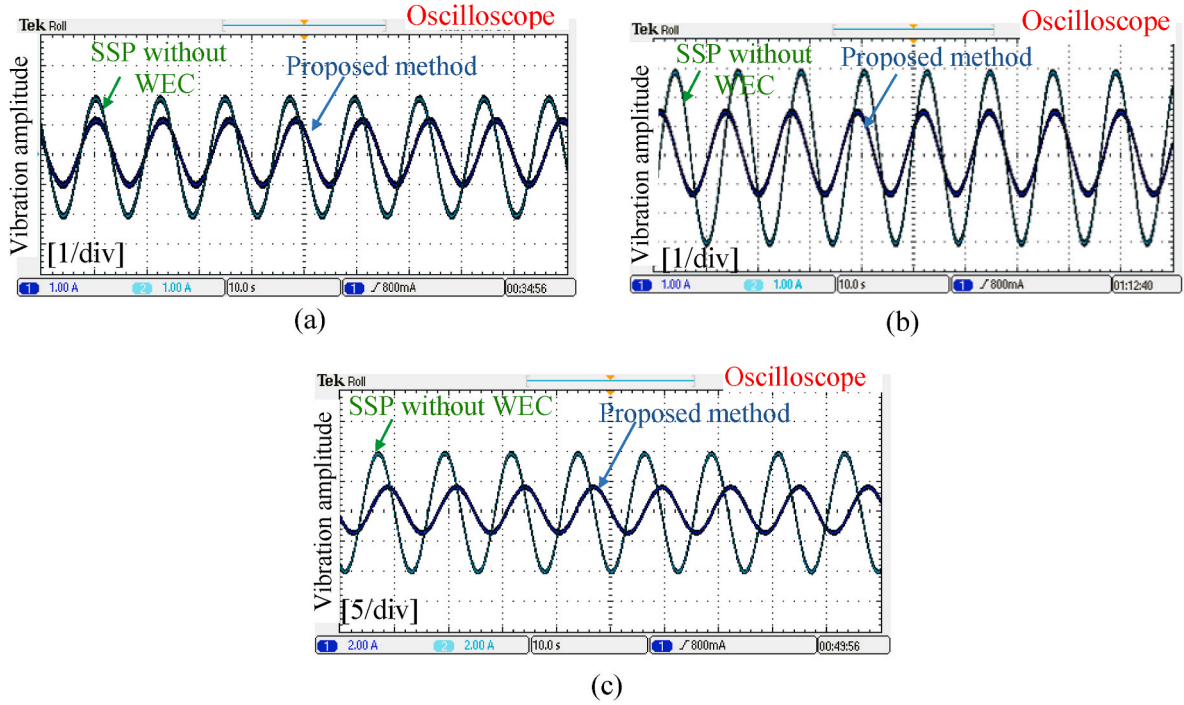


Fig. 31. Experimental waveform of vibration amplitude for the proposed method and SSP without WEC control under varying wave height H : (a) under $H = 2$ m, $\Omega = 1.1$, (b) under $H = 3$ m, $\Omega = 1.1$, (c) under $H = 4$ m, $\Omega = 1.1$.

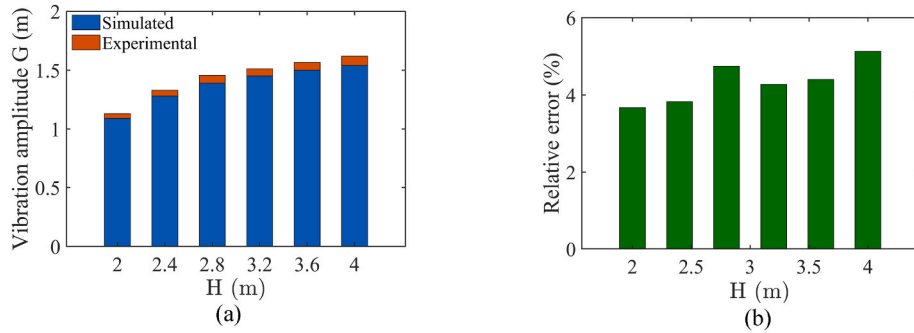


Fig. 32. Comparison of experiment and simulation for the proposed SSP-BIWECC with different wave height H : (a) vibration amplitude of platform, (b) relative error percentage.

coefficients according to changes in external wave conditions, enabling more of the absorbed vibration energy to be converted into electric power rather than being stored as mechanical energy in the BIWECC

system. To sum up, the above results demonstrate that the proposed method can effectively adapt to changing wave conditions, and its superior energy harvesting capability enhances its application potential.

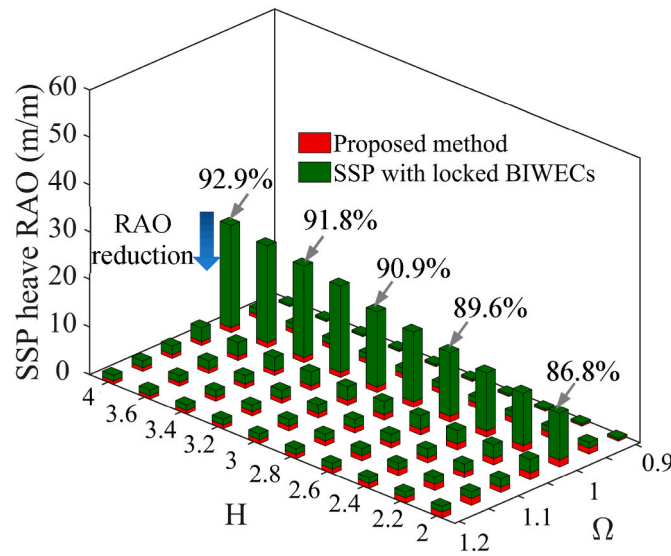


Fig. 33. Heave RAOs of the SSP for the proposed method and the SSP with locked BIWECS under different wave conditions.

5. Hardware-in-the-loop experiment

To further evaluate the vibration control effectiveness of the proposed method, this section first designs the vibration controller based on the optimization results (In section 4.2). Subsequently, the hardware-in-the-loop (HIL) experimental setup is established using advanced Real Time Laboratory (RTLAB) from OPAL-RT. Three specific cases are selected to test the control effect: validation under excitation frequency ratio change, validation under wave height change and heave RAO of SSP under varying wave conditions.

5.1. Optimization-based adaptive vibration controller design

The SSP-BIWECS is a complex coupled system involving mechanical, electrical, and nonlinear fluid dynamics. Based on the parameter optimization in Section 4, we can adjust the damping ratio of the three BIWECS based on wave conditions to suppress the SSP's vibration amplitude. On the other hand, PMSG controller is generally low-level controller that lack the capability to handle the complex mathematical computations required for intelligent control methods. Traditional PID strategies are unsuitable for managing such multi-variable, nonlinear, and strongly coupled systems [31]. In contrast, the lookup table method offers efficiency, ease of implementation, and adaptability to complex coupled systems, making it suitable for meeting the real-time operation requirements of low-level controllers.

Based on the preceding analysis, Fig. 24 illustrates the configuration of the SSP-BIWECS vibration controller utilizing the lookup table method. As depicted in Fig. 24, sensors on the platform capture the wave height and excitation frequency ratio, which are then fed into the three look-up tables developed in Section 4.2. The corresponding duty cycles (D_1 , D_2 , and D_3) are subsequently determined and employed to regulate the power converters of BIWECS-1, BIWECS-2, and BIWECS-3, respectively. This process effectively adjusts the electromagnetic damping of each BIWECS, thereby achieving vibration control of the SSP-BIWECS under different wave conditions.

In measuring wave height, the SVS-603HR wave sensor from Sea-View Systems is adopted and placed in the targeted sea area for measurement (see Fig. 25). During the measurement process, this device utilizes an internal inertia sensor to capture the buoy's vertical motion acceleration under wave excitation. By integrating the acceleration data, the buoy's vertical displacement is obtained, which is then inverted to calculate the wave height data [32]. With proper calibration, this wave

sensor can detect wave heights ranging from 0.1 m to 25 m, with a resolution of 0.001 m.

5.2. Experiment setup

As shown in Fig. 26, in the HIL test, the proposed control strategy is implemented on an actual controller and interacts with the virtual SSP-BIWECS system simulated in RTLAB. Compared with the previous purely numerical simulation research, the HIL test validates the operational logic of the implemented control strategy on the actual controller and ensures its functionality and reliability. Furthermore, the validated stable control logic serves as a critical foundational step for transitioning to physical model testing and reduces potential risks and uncertainties in later development stages.

Thus, this paper conducted a HIL test is conducted using RTLAB equipment to verify the performance of the proposed control strategy. The HIL test architecture is shown in Fig. 27. The mathematical model of the SSP-BIWECS is compiled into RTLAB as a virtual controlled object, while the control strategy is downloaded to the actual controller. During the control process, I/O boards are used to enable communication between RTLAB and the actual controller, and the signals are monitored in real time by a host computer and an oscilloscope. Nevertheless, it should be noted that, the HIL test only validates the implementation of the control strategy on a virtual SSP-BIWECS system simulated in RTLAB. The effectiveness of the proposed control strategy still requires validation on a physical model of the SSP-BIWECS system in real wave conditions. This limitation provides clear research direction for us, and our future work will focus on practical physical model testing.

Based on the above architecture, the real equipment is constructed as shown in Fig. 28, it mainly including RTLAB, three PMSG low-level controllers, an oscilloscope and a host computer. Detailed technical specifications of the HIL experimental bench are provided in Table 3.

5.3. Experiment validation of control effect

5.3.1. Validation under excitation frequency ratio change

In practical ocean waves, the wave motion frequency changes due to the influence of external factors. When the natural frequencies of the SSP and wave frequency close, the SSP's vibration amplitude increases sharply. To verify the effectiveness of the proposed vibration controller, we test the vibration amplitude of the proposed method under some dangerous frequency ratios it may encounter. The experimental

parameters are set as follows: the wave height of 3 m, excitation frequency ratios of 0.9, 1 and 1.1, and wave direction of 150° . A multi-channel oscilloscope is used to capture the SSP vibration amplitude waveforms to better demonstrate the vibration control effect. It observed from Fig. 29 that the proposed method could significantly reduce platform vibration amplitude compared to the SSP without WEC control. Specifically, there is an approximate 71.3 % reduction in vibration amplitude when the SSP is in the most dangerous resonant state (i.e., $\Omega = 1$). These experimental results further demonstrate the excellent vibration control effect of the proposed method, which can significantly reduce operational risk due to resonance.

Fig. 30 presents the comparison results between the HIL test and numerical simulation. Clearly, for the SSP's vibration amplitude under different excitation frequency ratios, the HIL test achieves nearly identical results to the simulation. Although there are slight deviations between the results, these discrepancies are within acceptable limits. The minor relative error may be caused by the inevitable transmission error of the PMSG controller during the A/D and D/A conversion process.

5.3.2. Validation under wave height change

In addition to wave frequency change, wave height varies depending on the ocean wind intensity. This section analyzes the vibration amplitude of the proposed method under different wave heights. The experiment parameters are set as follows: the dangerous excitation ratio is set to 1.1, the wave height is set to 2 m, 3 m, and 4 m, and the wave direction is set to 150° . It is observed from the oscilloscope (Fig. 31) that the proposed method has a lower vibration amplitude compared with the SSP without WEC control under different wave heights. Also, even at a maximum wave height condition ($H = 4$ m), the vibration amplitude reduction is nearly 61.3 %. This test result further validated that the proposed method has a good vibration control effect. Additionally, Fig. 32 compares the performance of the simulation and the HIL test, and shows that the HIL results track the simulation well.

5.3.3. Heave RAO of SSP under varying wave conditions

As shown in Fig. 33, the HIL test compares the SSP's heave RAOs under the proposed method with those of the SSP without WEC control across different wave conditions. The wave direction is set to 150° . The results demonstrate that the proposed method consistently reduces the vibration amplitude of the SSP under different wave conditions. Notably, the RAO reduction reaches its maximum at the resonant frequency (i.e., $\Omega = 1$). This highlights the excellent vibration control performance of the proposed method, which significantly mitigates the SSP's operational risks caused by resonance. Moreover, RAO reduction at the resonant frequency slightly increases with rising wave heights, further demonstrating the strong adaptability of the proposed method to varying wave conditions.

6. Conclusion

In this paper, the energy harvesting mechanism of the built-in wave energy converter (WEC) is extended to serve vibration control purposes, and a multiple electromagnetic damping adaptive synergy optimization approach is proposed to achieve effective vibration suppression under

varying wave conditions. To demonstrate its feasibility, a semi-submersible floating platform integrating multiple built-in WECs (SSP-BIWEC) is presented as a case study. A comprehensive multi-degree-of-freedom model that couples multiple physical domains is developed, validated, and subjected to parameter optimization and comparative performance analysis. Hardware-in-the-loop (HIL) experiments are then conducted to evaluate the effectiveness of the proposed method.

The results show that by adaptively adjusting the electromagnetic damping, the SSP-BIWEC system can consistently operate at an optimal damping level, thereby avoiding instability caused by damping detuning. The proposed method is particularly effective in suppressing heave resonance. At the same time, under the dominant wave condition of the target sea area, it does not increase the platform's motion in other directions. Additionally, it contributes to further reduction in roll and pitch vibrations within the resonance frequency range. Across a range of wave conditions, the proposed approach consistently outperforms the previously developed passive optimization method, demonstrating strong robustness and adaptability. Notably, the platform maintains stable heave responses even under abrupt changes in wave conditions.

In addition to vibration control, the proposed method also enables the BIWEC to achieve significantly improved energy harvesting performance compared to the passive approach, thereby enhancing its engineering applicability. Through HIL experiments, the good vibration control effectiveness of the proposed approach is validated under changing wave conditions. Additionally, the satisfactory agreement between HIL and numerical simulation confirms the practical applicability of the proposed approach for the actual controller. Overall, the proposed approach demonstrates excellent performance in both vibration suppression and energy harvesting, offering promising potential for real-world applications. Future work will focus on the fabrication of a physical prototype and conducting large-scale experimental testing in realistic marine environments.

CRedit authorship contribution statement

Yongxing Zhang: Writing – review & editing, Writing – original draft. **Jing Bian:** Writing – review & editing, Writing – original draft, Supervision, Conceptualization. **Zhicong Huang:** Writing – review & editing, Supervision, Project administration.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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Appendix A. Hydrodynamic parameters of SSP-BIWEC

Using the ANSYS AQWA solver, the hydrodynamic parameters of the SSP-BIWEC are calculated. Notably, since the proposed system is axisymmetric, the hydrodynamic parameters for surge and sway are identical, as are those for pitch and roll. Therefore, the hydrodynamic parameters for the sway and roll directions are not repeated in this study. Specifically,

3) For the wave excitation force or moment:

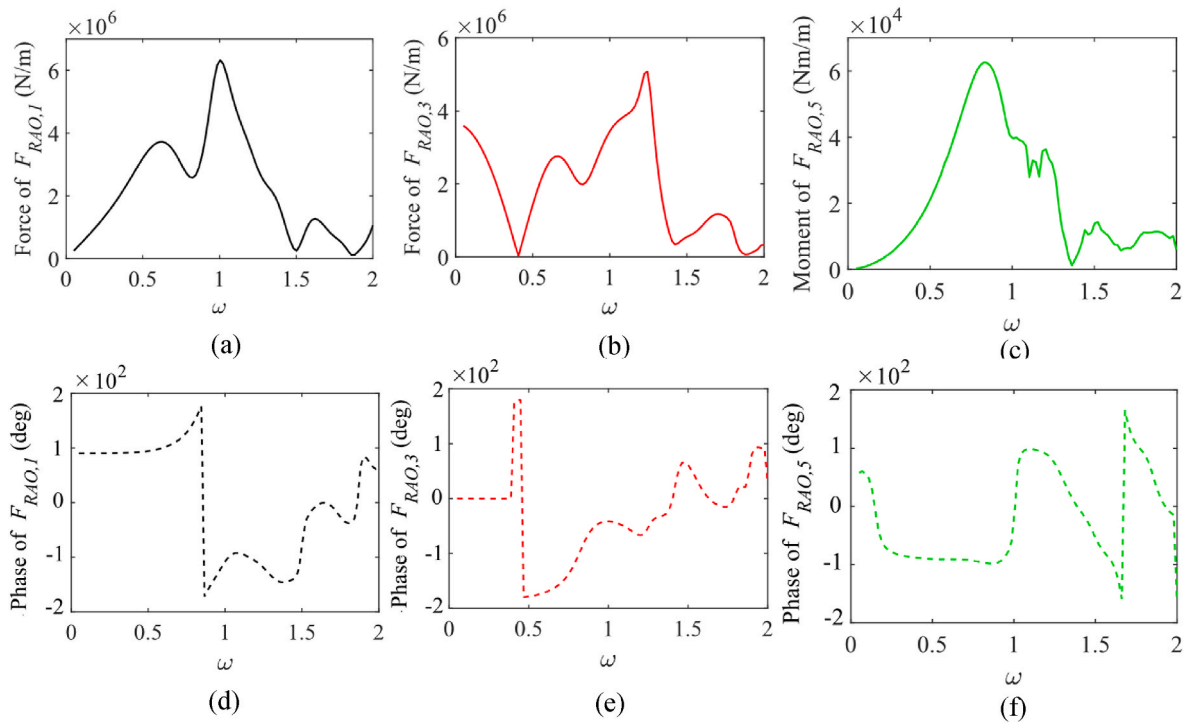


Fig. A1. The wave excitation force/moment: (a) magnitude of $F_{RAO,1}$, (b) magnitude of $F_{RAO,3}$, (c) magnitude of $F_{RAO,5}$, (d) phase of $F_{RAO,1}$, (e) phase of $F_{RAO,3}$ and (f) phase of $F_{RAO,5}$.

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