

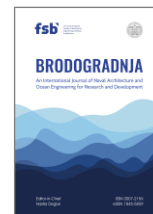


University of Zagreb  
Faculty of Mechanical  
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## Brodogradnja

An International Journal of Naval Architecture and  
Ocean Engineering for Research and Development



# Horizontal-plane pendulum-based inertial wave energy harvester with sprag-clutch rectification for unmanned surface vehicles



*Fatih Alver*

*Fatsa Vocational School, Ordu University, Ordu, Turkey*

## ARTICLE INFO

### Keywords:

Wave energy converter  
Unmanned surface vehicle  
Inertial pendulum  
Mechanical rotation rectifier  
Multi-axial excitation

## ABSTRACT

Unmanned surface vehicles (USVs) increasingly support long-duration ocean observation, yet their mission endurance remains constrained by limited onboard energy storage. Wave energy offers a continuous, weather-independent supply, although most pendulum-based harvesters respond to only a single motion axis. This study presents a horizontal-plane pendulum-based inertial wave energy harvester integrated within the USV hull, with the pendulum configured parallel to the XY-plane to simultaneously exploit pitch and roll excitation. A sprag-clutch mechanical rotation rectifier (MRR), a three-stage planetary gearbox, a hypoid stage, and a Halbach-array permanent magnet generator are used to convert the irregular, bidirectional motion of the pendulum into stable, unidirectional electrical output. A Lagrangian-based dynamic model, partitioned into torque-transmitting and freewheeling states, was developed and validated against motion-platform experiments under sinusoidal, swept-frequency, and JONSWAP irregular-wave excitation. The system delivers 13.0 W under matched-load conditions ( $f = 0.51$  Hz,  $\Phi_0 = 15^\circ$ ), exhibits a monotonic increase in output across the principal ocean-wave band (0.10-0.70 Hz), reaches 91 W at  $\Phi_0 = 30^\circ$ , and yields a fourfold power gain over single-axis operation under in-phase pitch-roll excitation. Under JONSWAP excitation ( $H_s = 1.5$  m,  $T_p = 7$  s), the mean experimental output is 7.21 W, corresponding to a motion-to-electric conversion efficiency of approximately 43 % under matched-load conditions.

## 1. Introduction

As climate change, pollution, and biodiversity loss intensify globally, the demand for continuous, long-term environmental monitoring and systematic data acquisition has grown more pressing than ever [1]. In offshore and oceanic environments, where human presence is limited and operational risks and costs are prohibitive, these activities are largely carried out by unmanned surface vehicles (USVs) [2-7]. However, USVs currently rely on batteries, fuel cells, or solar panels to meet their energy demands, imposing significant constraints on mission endurance, range, and autonomy. With existing energy storage solutions, continuous operation is typically limited to a matter of days or weeks [8]. Solar energy, while a viable supplementary source, cannot alone guarantee continuous and reliable power supply, particularly at high latitudes where irradiance is low and environmental variability is high [1]. Ocean waves, by contrast, cover the majority of

\* Corresponding author.

E-mail address: [fatihalver@odu.edu.tr](mailto:fatihalver@odu.edu.tr)

the Earth's surface and are independent of the day-night cycle, offering a high-energy-density, sustainable power source well suited to extended maritime missions [9-11]. In this context, energy harvesting technologies that capture ambient wave motion and convert it into electrical energy have emerged as a compelling pathway toward self-sufficient power infrastructure for USVs [3].

The electrical power demand of USVs varies with mission type and platform size, driven by onboard sensors, navigation and control systems, data acquisition units, and communication hardware; however, continuous power consumption typically falls within the range of a few to several tens of watts [12-14]. This requirement necessitates not merely short-duration peak supply, but sustained and stable energy delivery over extended periods. USVs operating in offshore conditions are simultaneously subjected to low-frequency, irregular dynamic excitation arising from wave-induced motion. A substantial portion of the open-ocean wave spectrum is widely reported to be concentrated within approximately 0.05-0.7 Hz, a band commonly referred to in the literature as the sea-swell regime [15]. This demands that energy harvesting systems be evaluated not solely on nominal power capacity, but on their ability to operate effectively under low-frequency, irregular excitation. Accordingly, one of the critical design criteria for wave energy harvesting systems intended for USV applications is that the dynamic and mechanical characteristics of the system enable meaningful and sustained power generation under realistic ocean wave spectra. In parallel with these supply-side considerations, recent system-level studies have made significant contributions to the modelling of USV propulsion and energy behaviour. For instance, CFD-based and experimentally validated digital twin models of underwater thrusters enable more realistic estimation of energy consumption and more accurate battery capacity sizing [16]. This approach highlights the importance of consistently and comprehensively modelling both energy supply and energy demand in autonomous maritime platforms.

The most prevalent approach in the literature for extending USV operational endurance involves hybrid power architectures that supplement battery-based storage with renewable sources such as solar and wind. The feasibility of solar-powered USVs for long-duration missions has been demonstrated in scenarios including continuous observation periods of up to 26 days and data collection under typhoon conditions; however, such solutions remain highly dependent on the day-night cycle and meteorological variability [17]. Wind-solar hybrid USV designs have similarly been reported; however, fluctuations in wind direction and speed, combined with their operational effects on the platform, make a stable standalone power supply difficult to achieve [18]. As a result, wave energy, which is abundant and continuously available across the ocean environment, has attracted growing interest as a complementary power source for USVs. Several wave-based approaches exploit wave motion directly for propulsion rather than electricity generation. The Wave Glider architecture, for instance, couples a surface float with a submerged glider via a tether, converting wave motion into thrust [19]. Yet systems of this type incorporate submerged components and additional hydrodynamic elements, limiting their direct applicability to conventional USV platforms in certain scenarios. Wave-based electricity-generation solutions have proposed external harvesting modules, such as fins or turbines mounted on the hull; however, such arrangements on mobile platforms introduce practical engineering challenges related to drag, structural integrity, and sealing [20]. In parallel, hydrokinetic turbine concepts have been investigated for extracting energy from currents and wave-induced flows, with horizontal-axis configurations reported to deliver higher hydrodynamic efficiency than their vertical-axis counterparts under comparable operating conditions [21]. Nevertheless, the externally exposed rotor and the requirement for an unobstructed inflow restrict their integration on small, mobile USV platforms, reinforcing the case for hull-internal solutions. Consequently, inertial energy harvesting systems that can be integrated within the hull have gained increasing traction in the literature [22, 23].

More broadly, wave-powered autonomous marine vehicles are currently being developed through two complementary paradigms. In the first, the conversion system is embedded directly within the vehicle, so that power is generated in situ from the platform's own wave-induced motion; recent examples include wave-powered AUVs for underwater sensor networks [24], WEC gliders [25, 26], and integrated recharging modules for AUVs [27]. In the second, a larger standalone WEC is moored or floated separately, and the vehicle periodically rendezvouses, docks, and recharges at it [28-31]. The two approaches embody a clear trade-off: standalone converters can be sized for favourable hydrodynamic scaling and therefore achieve

higher absolute wave-energy capture, but they require autonomous docking and charging infrastructure, station-keeping, and mission interruption for each recharge, whereas vehicle-integrated harvesters deliver continuous power without docking downtime or rendezvous complexity, at the cost of the limited hydrodynamic capture resulting from the small size of a marine vehicle relative to ocean wavelengths. The present work adopts the vehicle-integrated route and mitigates this limitation by harvesting inertially from the vehicle's multi-axial wave-induced motion rather than through wave-scale hydrodynamic absorption. The hull-internal horizontal-plane pendulum requires no external infrastructure, adds no docking overhead, and keeps the platform continuously self-powered throughout its mission.

Within this class, pendulum-based electromagnetic inertial energy harvesting systems are widely regarded as among the most mature and promising solutions, owing to their capacity to extract energy from low-frequency, wave-induced vibrations. Mitcheson et al. [32] comprehensively examined the advantages of pendulum-type inertial systems for low-frequency ambient vibrations, discussing the decisive influence of mechanical transmission, damping, and resonance tuning on harvesting performance. For USV applications specifically, Graves et al. [22] proposed an inertial energy harvester based on a pendulum-driven mechanical rotation rectifier (MRR)-DC generator architecture and experimentally demonstrated that electricity generation from low-frequency platform vibrations is achievable. A subsequent study by the same group reported that incorporating a counterweight pendulum configuration can lower the natural frequency and improve spectral alignment with ocean wave frequencies [23]. Nevertheless, in both studies, the pendulum configuration was primarily treated as responsive to a single motion component either pitch or roll depending on the vessel's heading. Various alternative configurations of pendulum-based inertial harvesting systems have also been investigated in the literature. Lou et al. [33] developed a two-axis pendulum structure, testing the system on a six-degree-of-freedom (6-DoF) motion platform and analysing its dynamic behaviour under multi-axial wave excitation. Similarly, the dual-wing flywheel-based inertial energy harvesting system proposed by Dai et al. [34] was evaluated primarily under heave excitation; however, the reported power levels were shown to remain limited relative to the endurance requirements of long-duration USV missions. Regarding mechanical rotation rectifier architectures, Liang et al. [35] proposed gear and one-way clutch-based solutions for converting reciprocating mechanical motion into unidirectional rotation, yet structural constraints including axial loads on bevel gears and the limited torque capacity of roller clutches were highlighted. As an alternative, the gyroscope-based inertial energy harvesting system developed by Townsend [14] aimed to simultaneously exploit pitch and roll motion; however, the system's dependence on an external energy input for continuous operation restricted net positive power generation. Broadly, the literature indicates that pendulum-based inertial energy harvesting systems represent a promising approach for USVs; however, a notable proportion of existing studies have been conducted on scaled prototypes under controlled laboratory excitation and, in most cases, with single-axis motion assumptions.

Against this backdrop, three principal research gaps emerge from the existing literature. First, many pendulum-based inertial harvesting systems employ vertically oriented configurations that are sensitive to only a single motion component either pitch or roll, depending on the vessel's heading, thereby limiting their capacity to exploit the multi-axial wave excitation that characterises real sea conditions [22, 23, 33]. Second, mechanical rotation rectifier designs have predominantly relied on roller clutch or bevel gear-based solutions, where engineering constraints including impact loading, limited torque capacity, and mechanical durability pose substantial challenges for USV integration [14, 35]. Third, most experimental validations have been conducted on scaled prototypes under controlled excitation, and direct comparisons of reported power outputs with the continuous power demands of a full-scale USV platform under multi-degree-of-freedom wave excitation have frequently rested on additional assumptions [23, 33, 34].

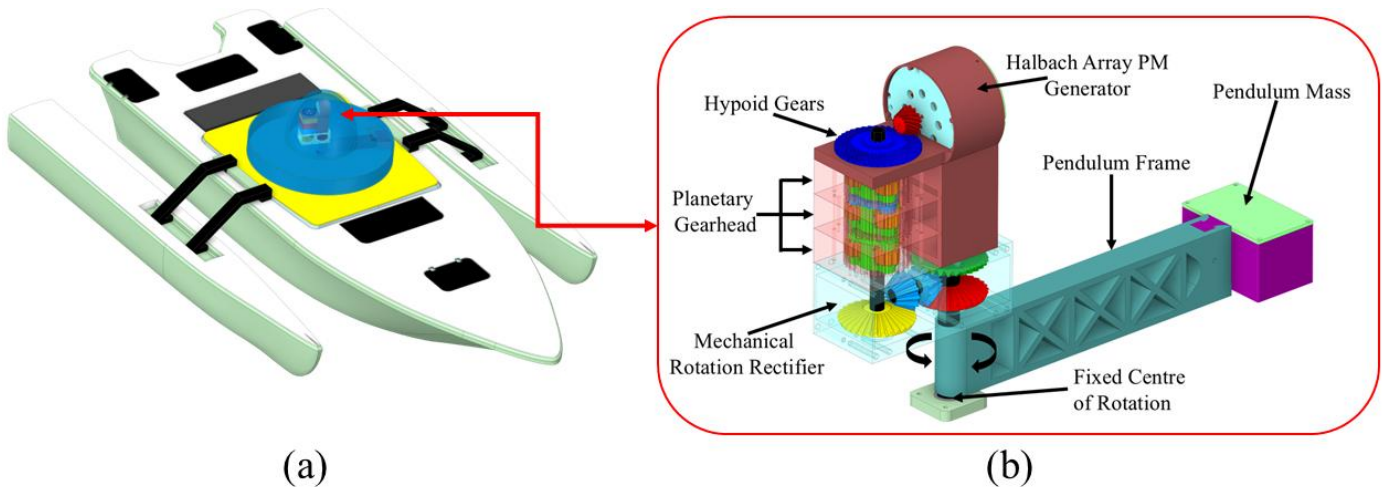
This study presents a novel horizontal-plane pendulum-based inertial wave energy harvesting system designed to address the identified gaps. In the proposed architecture, the pendulum is mounted within the USV hull parallel to the XY-plane, enabling simultaneous torque generation from both pitch- and roll-induced motions. To convert the resulting reciprocating angular displacement into stable, unidirectional output, a mechanical rotation rectifier incorporating sprag clutch elements was designed, offering high torque capacity and rapid engagement characteristics. Furthermore, unlike many studies in the literature, the system was

prototyped at a scale and mass compatible with integration into an actual USV platform, and its performance was experimentally evaluated on a 3-DoF motion platform under multiaxial excitation representative of realistic sea conditions. In these respects, the present work offers an original and complementary contribution to the pendulum-based inertial energy harvesting literature, both in terms of its kinematic scope, which captures pitch and roll simultaneously, and its realistic scale and mechanical architecture.

## 2. Mechanical design of the energy harvester

The general architecture of the pendulum-based inertial wave energy harvesting system proposed in this study, along with its integration onto the USV, is presented in Figures 1a and 1b. The system is designed for integration into the hull of an unmanned surface vehicle and comprises a horizontal-plane pendulum mechanism, a mechanical rotation rectifier (MRR), a multistage gear transmission, and a permanent magnet generator. As shown in Figure 1a, the energy harvesting unit is housed within a watertight enclosure positioned at the centre of the USV hull and incorporates no external moving components in contact with water. This arrangement allows the system to operate without increasing hydrodynamic drag or affecting the fundamental seakeeping characteristics of the platform.

Figure 1b provides a schematic overview of the principal components and power transmission chain of the energy harvesting mechanism. The pendulum arm and mass, oriented within the horizontal plane (parallel to the XY-plane), are free to rotate about a fixed pivot defined relative to the USV hull. The wave-induced pitch and roll motions are simultaneously converted into rotational torque at the pendulum by virtue of this horizontal configuration. The resulting irregular, bidirectional angular displacement is rectified into continuous unidirectional rotation via the MRR and subsequently transmitted through a planetary gearbox and a hypoid gear set to a Halbach-array permanent magnet generator, where it is converted into electrical energy. This integrated architecture aims to deliver a compact and mechanically robust energy harvesting solution under low-frequency, multiaxial wave excitation.



**Fig. 1** Overall design of the pendulum-based energy harvester: (a) Integrated into the USV, and (b) View of the main components of the energy harvesting mechanism

Figure 2 illustrates in detail the mechanical rotation rectifier (MRR) architecture of the energy harvesting system, together with the associated power transmission and electricity generation components. Figure 2a presents the input structure through which rotational torque from the pendulum mechanism is transferred to the MRR, along with its arrangement within the USV hull. This figure summarises the principal components and the operating principle of the mechanical chain responsible for converting the bidirectional, irregular angular displacement of the pendulum into stable, unidirectional rotation before transmission to the generator. The structural layout of the gear and generator configurations selected to achieve a compact design with high torque capacity is also presented within this integrated architecture.

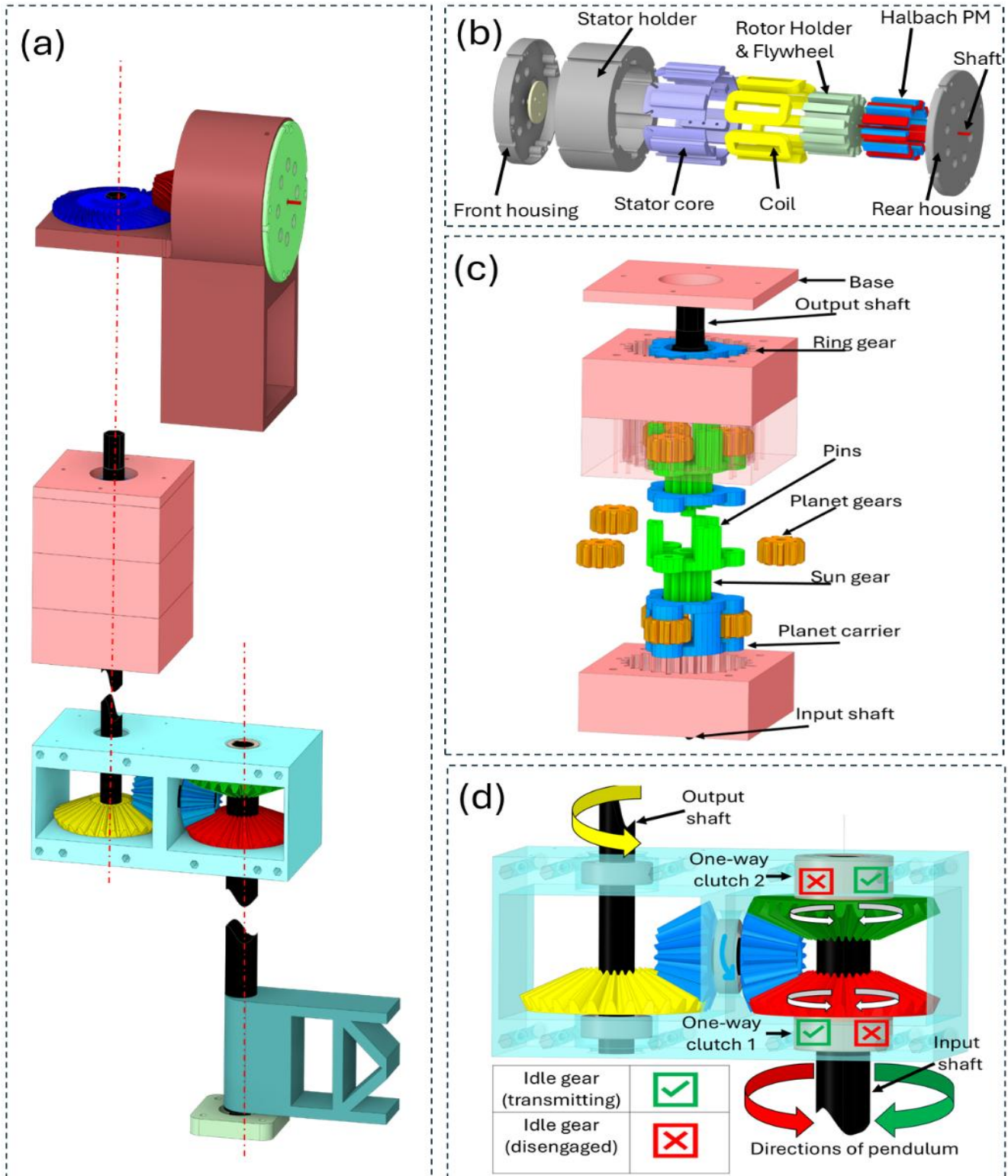
Figure 2b presents an exploded view of the Halbach-array permanent magnet generator employed in the energy harvesting system. The Halbach arrangement was selected to concentrate magnetic flux density on the

air gap side, thereby improving electromechanical energy conversion under low-speed operating conditions. The generator features a three-phase, six-slot, eight-pole topology, a configuration that supports higher electrical frequency and stable electromotive force (EMF) generation at low angular velocities. The Halbach array comprises three magnet blocks per pole, utilising a total of 24 permanent magnets, each measuring  $5 \times 10 \times 50$  mm. The stator windings are wound using 0.8-mm-diameter copper wire with 100 turns per phase. The generator has a rotor diameter of 200 mm and a stator to rotor air gap of 1 mm. This parametric configuration was designed to preserve the scalability of the generator architecture for higher power requirements.

Figure 2c presents an exploded view of the planetary gearbox integrated at the output of the mechanical rotation rectifier. In this study, a planetary gear set was selected in preference to the offset gear arrangements commonly employed in the literature. The planetary architecture distributes load across multiple simultaneously engaged gear teeth, reducing contact stresses, increasing torque capacity, and yielding a more compact power transmission path [36]. The configuration consists of a central sun gear, three orbiting planet gears, and a ring gear that carries them. The system incorporates three planetary stages connected in series, each providing a threefold speed increase, resulting in a total output shaft speed ratio of 27:1. At the output of the planetary gearbox, a hypoid gear set is employed to redirect the rotational axis by  $90^\circ$  and connect to the generator shaft, providing an additional speed increase of 3:1. The hypoid gear set was selected over spur or bevel alternatives for its ability to transmit high torque silently and continuously between nonintersecting shafts, its higher contact ratio, and its advantage in compact installation [37]. This approach shortens the power transmission path and enables more efficient integration of the system within the limited volumetric envelope of the USV hull.

Direct transmission of pendulum's bidirectional, irregular angular displacement to the generator reduces mechanical efficiency during abrupt directional reversals and can induce high stresses and potential deformation in the gear and shaft components owing to opposing force components. To address this, the MRR architecture was designed to ensure that the generator rotates continuously in a single direction regardless of the pendulum's bidirectional motion. The MRR system presented in Figure 2d comprises five offset bevel gears supported by two sprag clutches, which provide high torque capacity and rapid engagement characteristics. Bevel gears were selected for their ability to transmit torque and speed directly between intersecting shafts, typically oriented at  $90^\circ$ , thereby allowing the directional change requirement to be met compactly within the system's constrained envelope.

Power transmission follows distinct pathways depending on whether the pendulum arm rotates anticlockwise or clockwise, as indicated by the red and green arrows respectively in Figure 2d. During anticlockwise rotation, sprag clutch 2 engages to drive the idler gear, which in turn drives the output gear. During clockwise rotation, sprag clutch 1 drives the output gear directly. Positioning the gears offset from their conventional planes enables efficient power transfer with a minimal number of components, whilst minimising the gear count improves overall system efficiency and reduces backlash. Furthermore, the presence of the one-way clutches means that, for instance, when the angular velocity of the pendulum arm falls below the output speed on the generator side as it approaches the point of reversal, the clutches transition to the freewheeling state, allowing the generator to continue rotating. This feature exploits the rotational inertia of the generator to sustain electricity generation even when the input motion is momentarily interrupted, whilst preventing forced stall-and-restart cycles and reducing the high starting torque that would otherwise be required.



**Fig. 2** CAD model demonstrating working mechanism of the pendulum energy harvester

To enable structural replication of the proposed mechanism, the principal mechanical design parameters of the energy harvester are summarised in Table 1. The table specifies the module and tooth numbers of the MRR bevel gears, the commercial sprag clutch designation and its rated torque capacity, the tooth numbers of the custom-manufactured planetary and hypoid gear stages, the generator configuration, and the layout and dimensions of the watertight enclosure. It is noted that, although the input and output bevel gears ( $z = 32$ ) and

the idler gear ( $z = 16$ ) form 2:1 gear-meshing pairs, the symmetric arrangement of the gear train cancels the intermediate speed ratios, so that the net transmission ratio of the MRR remains 1:1. The overall transmission ratio of the system is therefore governed solely by the planetary and hypoid stages.

**Table 1** Mechanical design parameters of the energy harvester

| Component                                                       | Parameter                                | Value                                                                                                                             |
|-----------------------------------------------------------------|------------------------------------------|-----------------------------------------------------------------------------------------------------------------------------------|
| <b>MRR bevel gears</b><br>(DIN 3971)                            | Module, $m$                              | 3 mm                                                                                                                              |
|                                                                 | Input/output gears                       | $z = 32$ , bore $\varnothing 20$ mm                                                                                               |
|                                                                 | Idler gear                               | $z = 16$ , bore $\varnothing 15$ mm                                                                                               |
|                                                                 | Net MRR speed ratio                      | 1:1                                                                                                                               |
| <b>Sprag clutches</b><br>(2 $\times$ , Stieber CSK20P)          | Type                                     | Combined bearing/freewheel (6204 series), inner-race keyway to DIN 6885.3                                                         |
|                                                                 | Bore / outer diameter / width            | $\varnothing 20$ mm / $\varnothing 47$ mm / 14 mm                                                                                 |
|                                                                 | Nominal torque, $T_{KN}$                 | 50 N·m ( $T_{max} = 2 \times T_{KN} = 100$ N·m)                                                                                   |
| <b>Planetary gearbox</b><br>(three stages, custom-manufactured) | Sun / planet ( $\times 3$ ) / ring gears | $z = 18 / 9 / 36$                                                                                                                 |
|                                                                 | Stage ratio (fixed ring gear)            | 3:1 per stage (27:1 total)                                                                                                        |
| <b>Hypoid gear pair</b><br>(custom-manufactured)                | Pinion / gear                            | $z = 14 / 42$                                                                                                                     |
|                                                                 | Ratio / shaft angle                      | 3:1 / $90^\circ$                                                                                                                  |
| <b>Generator</b><br>(Halbach-array PMSG)                        | Topology                                 | Three-phase, 6-slot, 8-pole                                                                                                       |
|                                                                 | Permanent magnets                        | $24 \times (5 \times 10 \times 50)$ mm, three blocks per pole                                                                     |
|                                                                 | Rotor diameter / air gap                 | 200 mm / 1 mm                                                                                                                     |
|                                                                 | Stator windings                          | 100 turns per phase, $\varnothing 0.8$ mm copper wire                                                                             |
| <b>Watertight enclosure</b>                                     | Material                                 | Water-resistant reinforced polymer; gasket-sealed lids                                                                            |
|                                                                 | Pendulum housing (cylindrical)           | $\varnothing 2000$ mm $\times$ 200 mm                                                                                             |
|                                                                 | Drivetrain dome (semi-elliptical)        | $\varnothing 1000$ mm $\times$ 450 mm                                                                                             |
|                                                                 | Sealing architecture                     | MRR and MRR to planetary coupling enclosed in separate sealed sub-housings; no rotating shaft penetrates the outer enclosure wall |

### 3. Mathematical modelling

A lumped parameter mathematical model is developed to analyse the dynamic behaviour of the proposed horizontal-plane pendulum energy harvester and to establish a theoretical framework for validation against experimental findings. System dynamics are formulated using the Lagrangian mechanics framework, which offers a systematic and physically transparent approach for multibody electromechanical systems. Because the sprag-clutch-based MRR introduces piecewise switching behaviour, the model is partitioned into two distinct operating states: (i) the torque-transmitting state, in which the clutches are locked and the generator is mechanically driven by the pendulum; and (ii) the freewheeling state, in which the clutches disengage and the pendulum decouples from the generator-side inertia. All symbols, together with their units, sign conventions, and the system state variables, are collected in the Nomenclature (Appendix A).

The following modelling assumptions are adopted throughout this section:

1. All rotating components, including the pendulum arm, pendulum bob, three-stage planetary gearbox, hypoid gear pair, flywheel, and generator rotor, are treated as rigid bodies.
2. The pendulum oscillates exclusively within the horizontal XY-plane of the USV body frame; out-of-plane displacement is neglected.
3. The gear train is treated as kinematically rigid with no backlash; energy losses are lumped into an equivalent viscous mechanical damping coefficient.
4. The permanent magnet synchronous generator (PMSG) is represented by an equivalent circuit comprising an ideal back electromotive force (back-EMF) source connected in series with the armature resistance and a resistive load.
5. Wave-induced USV body motion is characterised by the pitch angle  $\varphi_p(t)$  about the body y-axis and the roll angle  $\varphi_r(t)$  about the body x-axis; both are prescribed as kinematic inputs.

The pitch and roll motions are prescribed as kinematic inputs, such that the hull and the harvester are treated as a one-way-coupled (decoupled) system. This reflects both the aim and the experimental basis of the study: the electromechanical behaviour of a prototype-scale harvester is characterised under realistic multiaxial motion reproduced on a 3-DoF motion platform, in the same manner as motion-platform and shaker testing commonly employed in the inertial energy harvesting literature [22, 23, 33]. The representative full-scale hull of Figure 1 defines the integration context and is not itself the subject of a hydrodynamic study. Moreover, the neglect of the two-way hull-harvester interaction is quantitatively justified. First, the harvested output, 7.2 W on average and approximately 90 W at peak, represents a very small fraction of the wave power available to the carrier. For the validated sea state ( $H_s = 1.5$  m), the incident wave power flux is approximately 7 kW per metre of crest, so the device extracts well below 1 % of the wave power intercepted by a metre-scale hull and therefore adds negligible damping to the hull motion. Second, because the horizontal-plane pendulum rotates about the vertical (yaw) axis, its reaction on the hull is predominantly a yaw moment, largely orthogonal to the pitch and roll modes from which energy is harvested, so any feedback into the driving motion is a second-order effect. The one-way formulation is thus an appropriate and well-bounded approximation for the present device-characterisation study.

The generalised coordinate  $\theta(t)$  denotes the angular displacement of the pendulum arm relative to the USV body frame, measured about the vertical z-axis in the horizontal XY-plane. The MRR housing is fixed to the USV body; therefore, the MRR output shaft (and hence the planetary gearbox input) rotates at the relative pendulum velocity,  $\dot{\theta}$ . The three-stage planetary gearbox has ratio  $N_{pg} = 27$ , and the hypoid gear pair introduces an additional ratio  $N_h = 3$ , giving a total gear ratio:

$$N = N_{pg} \cdot N_h = 81 \quad (1)$$

The generator shaft angular velocity is accordingly:

$$\omega_g = N\dot{\theta} \quad (2)$$

### 3.1. Torque-transmitting state

In the torque-transmitting state, the sprag clutches are engaged, and the pendulum shaft is mechanically coupled to the gear train and generator. The system comprises six rotating subassemblies. Because the pivot is fixed to the USV body and the pendulum swings in the horizontal plane, the pendulum arm and bob rotate in inertial space with the absolute angular velocity  $(\dot{\theta} + \dot{\varphi})$ , where  $\dot{\varphi}$  denotes the combined effective z-axis component of the USV angular velocity. The MRR output shaft, planetary gearbox input, and all downstream gear stages are driven by the relative pendulum velocity  $\dot{\theta}$ .

The total kinetic energy is the sum of the kinetic energies of each subassembly:

$$T = T_{arm} + T_{bob} + T_{planetary} + T_{hypoid} + T_{flywheel} + T_{rotor} \quad (3)$$

where each term is:

$$\begin{aligned}
T_{arm} &= \frac{1}{2}J_a(\dot{\theta} + \dot{\phi})^2 \\
T_{bob} &= \frac{1}{2}m_bL_b^2(\dot{\theta} + \dot{\phi})^2 \\
T_{planetary} &= \frac{1}{2}J_{pg}(N_{pg}\dot{\theta})^2 \\
T_{hypoid} &= \frac{1}{2}J_h(N\dot{\theta})^2 \\
T_{flywheel} &= \frac{1}{2}J_f(N\dot{\theta})^2 \\
T_{rotor} &= \frac{1}{2}J_r(N\dot{\theta})^2
\end{aligned} \tag{4}$$

Introducing the pendulum moment of inertia,  $J_p$ , the generator-side effective inertia,  $J_G$ , (all generator-side inertias referred to the pendulum shaft), and the total effective inertia,  $J_{eq}$ :

$$\begin{aligned}
J_p &= J_a + m_bL_b^2 \\
J_G &= N_{pg}^2J_{pg} + N^2(J_h + J_f + J_r) \\
J_{eq} &= J_p + J_G
\end{aligned} \tag{5}$$

With these definitions, the total kinetic energy (Equation (3)) simplifies to:

$$T = \frac{1}{2}J_p(\dot{\theta} + \dot{\phi})^2 + \frac{1}{2}J_G\dot{\theta}^2 \tag{6}$$

The pendulum bob moves in the horizontal plane of the USV body. When the USV undergoes pitch,  $\varphi_p(t)$ , (rotation about the body y-axis) and roll,  $\varphi_r(t)$ , (rotation about the body x-axis), the pendulum plane tilts, and gravity acquires an in-plane component that creates a position-dependent gravitational torque. Expressing the position vector of the pendulum bob in the body frame as  $(L_b\cos\theta, L_b\sin\theta, 0)^T$ , the gravitational potential energy is:

$$V = m_bgL_b[\sin\varphi_p(t)\cos\theta - \sin\varphi_r(t)\sin\theta] \tag{7}$$

For the practical range of wave-induced pitch and roll amplitudes encountered during USV operations ( $|\varphi_p|, |\varphi_r| \ll 1$  rad), the small-angle approximation  $\sin\varphi \approx \varphi$ , may be applied:

$$V \approx m_bgL_b[\varphi_p(t)\cos\theta - \varphi_r(t)\sin\theta] \tag{8}$$

The nonconservative generalised forces acting on the coordinate  $\theta$  encompass mechanical damping (bearing friction and viscous gear-train losses) and the electromagnetic reaction torque of the generator. The wave-induced excitation enters the model through the kinematic coupling in the kinetic energy (Equation (6)) and through the gravitational coupling in the potential energy (Equation (7)); it does not appear as an independent generalised force in the Lagrangian formulation.

The PMSG produces a back-EMF proportional to its shaft angular velocity:

$$\varepsilon(t) = k_e\omega_g = k_eN\dot{\theta} \tag{9}$$

Modelling the generator as an ideal voltage source,  $\varepsilon$ , connected in series with the equivalent armature resistance,  $R_a$ , and an external load,  $R_L$ , the induced current is:

$$i(t) = \frac{\varepsilon(t)}{R_a + R_L} = \frac{k_e N \dot{\theta}}{R_a + R_L} \quad (10)$$

The electromagnetic braking torque exerted at the generator shaft arising from the back-EMF interaction is:

$$\tau_{em,g} = k_t i(t) = \frac{k_t k_e N}{R_a + R_L} \dot{\theta} \quad (11)$$

To refer the electromagnetic braking torque to the generalised coordinate  $\theta$  (the pendulum shaft), the principle of power equivalence is invoked ( $\tau_{em,g} \omega_g = \tau_{em} \dot{\theta}$ ):

$$\tau_{em} = N \tau_{em,g} = \frac{N^2 k_t k_e}{R_a + R_L} \dot{\theta} \quad (12)$$

The total generalised nonconservative moment, referred to the pendulum shaft, is:

$$Q_{nc} = -c_m \dot{\theta} - \tau_{em} = -(c_m + c_e) \dot{\theta} \quad (13)$$

where  $c_m \dot{\theta}$  is the mechanical damping torque. From Equations (12) and (13), the equivalent electrical damping coefficient referred to the pendulum shaft is:

$$c_e = \frac{N^2 k_t k_e}{R_a + R_L} \quad (14)$$

This expression shows that the electromagnetic back-reaction acting on the pendulum behaves as a linear viscous damper with coefficient  $c_e$ . Critically,  $c_e$  is load-tunable: reducing the external load resistance,  $R_L$ , increases  $c_e$  and strengthens the electromagnetic braking, whereas increasing  $R_L$  reduces  $c_e$  and permits larger pendulum oscillations. Optimal power transfer is achieved at the matched resistance condition  $R_L = R_a$ . Combining the kinetic energy (Equation (6)) and potential energy (Equation (7)), the Lagrangian of the system in the torque-transmitting state is:

$$\mathcal{L} = T - V = \frac{1}{2} J_p (\dot{\theta} + \dot{\varphi})^2 + \frac{1}{2} J_G \dot{\theta}^2 - m_b g L_b [\sin \varphi_p(t) \cos \theta - \sin \varphi_r(t) \sin \theta] \quad (15)$$

Applying the Euler-Lagrange equation with the nonconservative generalised force  $Q_{nc}$  yields:

$$\frac{d}{dt} \left( \frac{\partial \mathcal{L}}{\partial \dot{\theta}} \right) - \frac{\partial \mathcal{L}}{\partial \theta} = Q_{nc} \quad (16)$$

The required partial derivatives are evaluated as follows:

$$\frac{\partial \mathcal{L}}{\partial \dot{\theta}} = J_p (\dot{\theta} + \dot{\varphi}) + J_G \dot{\theta} = J_{eq} \dot{\theta} + J_p \dot{\varphi} \quad (17)$$

$$\frac{d}{dt} \left( \frac{\partial \mathcal{L}}{\partial \dot{\theta}} \right) = J_{eq} \ddot{\theta} + J_p \ddot{\varphi} \quad (18)$$

$$\frac{\partial \mathcal{L}}{\partial \theta} = m_b g L_b [\sin \varphi_p(t) \sin \theta + \sin \varphi_r(t) \cos \theta] \quad (19)$$

Substituting Equations (17)-(19) into Equation (16) and rearranging yields the equation of motion in the torque-transmitting state:

$$J_{eq} \ddot{\theta} + (c_m + c_e) \dot{\theta} - m_b g L_b [\sin \varphi_p(t) \sin \theta + \sin \varphi_r(t) \cos \theta] = -J_p \ddot{\varphi}(t) \quad (20)$$

The right-hand-side term,  $-J_p \ddot{\phi}(t)$ , is the inertial excitation torque produced by the wave-induced angular acceleration of the USV body; it constitutes the primary driving mechanism of the energy harvester. The left-hand side contains the effective inertial resistance,  $J_{eq} \ddot{\theta}$ , the combined viscous damping,  $(c_m + c_e) \dot{\theta}$ , and the gravitational coupling torque,  $m_b g L_b [\sin \varphi_p \sin \theta + \sin \varphi_r \cos \theta]$ , which is both position- and time-dependent and reflects the simultaneous sensitivity of the horizontal-plane pendulum architecture to pitch and roll motions.

It is emphasised that Equation (20) is inherently nonlinear owing to its trigonometric dependence on  $\theta$  and on the time-varying platform tilt angles,  $\varphi_p(t)$  and  $\varphi_r(t)$ . This nonlinearity is retained in the numerical simulation to faithfully capture large-amplitude pendulum responses that may arise under realistic sea conditions.

### 3.2. Freewheeling state

When the pendulum angular velocity,  $|\dot{\theta}|$ , falls below the angular velocity of the generator-side MRR output shaft, which occurs as the pendulum decelerates near its turning points before reversing direction, the sprag clutches disengage, and the system transitions into the freewheeling state. The mechanical connection between the pendulum and the generator-side assembly is severed, thereby separating the dynamics into two independent subsystems.

With the sprag clutches in the freewheeling state, the generator-side effective inertia,  $J_G$ , and the electromagnetic damping,  $c_e$ , are absent from the pendulum equation. The pendulum evolves solely under wave-induced inertial and gravitational excitation and is resisted only by the mechanical damping of the pendulum assembly:

$$J_p \ddot{\theta} + c_m \dot{\theta} - m_b g L_b [\sin \varphi_p(t) \sin \theta + \sin \varphi_r(t) \cos \theta] = -J_p \ddot{\phi}(t) \quad (21)$$

Comparing Equation (21) with the torque-transmitting Equation (20), two key differences are evident: (i) the effective inertia decreases from  $J_{eq}$  to  $J_p$ , allowing the pendulum to accelerate more rapidly in response to wave excitation; and (ii) the electrical damping term,  $c_e \dot{\theta}$ , vanishes, enabling the pendulum to accumulate angular momentum before the sprag clutches reengage.

After clutch disengagement, the generator-side assembly, comprising the flywheel, generator rotor, hypoid gear pair, and planetary gearbox, constitutes a mechanically isolated rotational system. Receiving no input torque, it decelerates under the combined action of the electromagnetic damping,  $c_e$ , (back-EMF braking) and the generator-side mechanical damping,  $c_{m,G}$ , (bearing friction and windage). Defining the angular velocity of the generator-side MRR output shaft as  $\Omega(t)$ , which becomes an independent variable after disengagement and is no longer equal to  $\dot{\theta}$ , the equation of motion of the generator subsystem is:

$$J_G \dot{\Omega} + (c_e + c_{m,G}) \Omega = 0 \quad (22)$$

The solution of Equation (22) is an exponential decay:

$$\Omega(t) = \Omega_0 \exp\left(-\frac{c_e + c_{m,G}}{J_G} (t - t_0)\right) \quad (23)$$

where  $\Omega_0 = \Omega(t_0)$  is the generator-side angular velocity at the instant of clutch disengagement  $t_0$ , and  $\tau_G = J_G / (c_e + c_{m,G})$  is the generator-side time constant.

Crucially, the generator continues to rotate and therefore continues to produce electrical power throughout this deceleration phase. The generator shaft angular velocity during the freewheeling state is  $\omega_g(t) = N \Omega(t)$ , and the instantaneous electrical power delivered to the load is:

$$P_L(t) = i^2(t) R_L = \frac{(k_e N \Omega)^2 R_L}{(R_a + R_L)^2} \quad (24)$$

The time constant,  $\tau_G$ , governs the duration of sustained electricity generation following clutch disengagement and can be increased by increasing the flywheel inertia,  $J_f$ , or selecting a higher gear ratio,  $N$ . This self-sustained electricity generation during the freewheeling state is a key advantage of the MRR architecture and is particularly beneficial under low-frequency, large-amplitude wave excitation, where the pendulum spends a significant proportion of its motion near its turning points.

The system alternates between the torque-transmitting state (Equation (20)) and the freewheeling state (Equations (21) and (22)), governed by the following transition criteria:

- Clutch engagement (freewheeling  $\rightarrow$  torque-transmitting):  $|\dot{\theta}| > \Omega$
- Clutch disengagement (torque-transmitting  $\rightarrow$  freewheeling):  $|\dot{\theta}| \leq \Omega$

The idealised switching representation is justified by the characteristics of the selected sprag clutches (Stieber CSK20P; Table 1). Being of the sprag type, the elements remain in continuous spring-loaded contact with both races, so engagement is backlash-free and occurs within a few arc-minutes of relative rotation, producing no impact loading. The catalogue drag torque in the freewheeling direction is 0.013 N·m, which is below 0.1 % of the peak transmitted torque of approximately 40 N·m; therefore, freewheeling hysteresis losses are negligible. Since the peak transmitted torque remains below the rated capacity of the clutch ( $T_{KN} = 50$  N·m,  $T_{max} = 100$  N·m), the sprag-race contact stresses stay within the manufacturer's design envelope, and a component-level contact analysis is not required.

### 3.3. Influence of heave excitation

The 3-DoF motion platform also permits vertical (heave) excitation, which is not treated as an independent driving axis in the present formulation. This omission is not an assumption but follows directly from the kinematics of the horizontal-plane configuration, as demonstrated below.

Consider the vessel in an upright attitude subjected to a pure heave acceleration  $a_z(t)$ . In the body frame, the position vector of the pendulum bob relative to the pivot is  $r_b = (L_b \cos \theta, L_b \sin \theta, 0)^T$ , and the corresponding inertial force acting on the bob is  $F = -m_b a_z \hat{z}$ . The moment of this force about the pivot is:

$$M = r_b \times F = (-m_b a_z L_b \sin \theta, m_b a_z L_b \cos \theta, 0)^T \quad (25)$$

The component of Equation (25) about the vertical  $z$ -axis, which corresponds to the axis of the generalised coordinate  $\theta$ , is identically zero. A purely vertical inertial force therefore produces no generalised force on the pendulum, irrespective of its magnitude; the two nonzero components act about the body  $x$ - and  $y$ -axes and are reacted by the hull rather than by the pendulum shaft.

Heave consequently enters the dynamics only at second order through the tilted-plane gravitational coupling of Equation (20), in which the gravitational acceleration is replaced by the instantaneous effective value,  $(g + a_z)$ :

$$Q_{\theta, grav} = m_b (g + a_z(t)) L_b [\sin \varphi_p(t) \sin \theta + \sin \varphi_r(t) \cos \theta] \quad (26)$$

The relative magnitude of this modulation is determined by the ratio  $a_z/g$ . Under the wave-follower assumption adopted throughout, a linear wave of amplitude  $a = H_s/2$  and peak frequency  $\omega_p$  yields  $a_z \approx a \omega_p^2$ . For the validated sea state ( $H_s = 1.5$  m,  $T_p = 7$  s), this gives  $a_z \approx 0.60$  m·s<sup>-2</sup>, corresponding to  $a_z/g \approx 0.06$ .

Furthermore, the two contributions are in quadrature. For a linear wave, the surface elevation varies as  $\eta \propto \cos(kx - \omega t)$ , so that the heave acceleration,  $\ddot{\eta} \propto -\cos(kx - \omega t)$ , whereas the wave slope, and hence the pitch and roll angles that appear in Equation (26), vary as  $\partial \eta / \partial x \propto -\sin(kx - \omega t)$ . The cycle-averaged product of the two therefore vanishes to first order, and the residual influence of heave on the mean harvested power scales as  $O[(a_z/g)^2] \approx 0.4$  %.

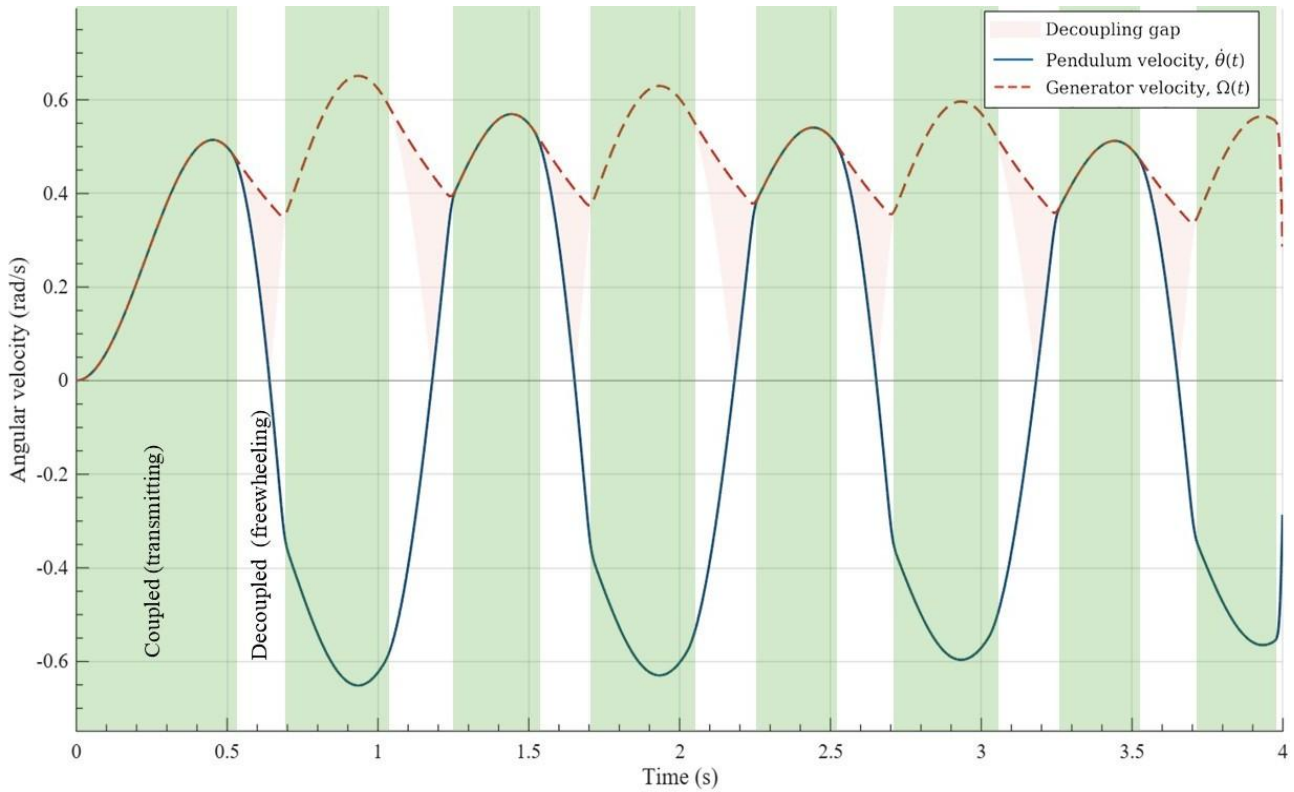
This analytical estimate is consistent with the numerical results reported in Section 5, where superimposing heave on the combined pitch-roll excitation was found to alter the mean harvested power by less than 2 %. The horizontal-plane architecture is thus inherently insensitive to heave, and the pitch-roll

excitation set employed throughout this study captures the motion axes that govern the performance of the device.

### 3.4. Numerical simulation of the mathematical model

The system of equations derived in the preceding sections, comprising the torque-transmitting equation of motion (Equation (20)), the freewheeling pendulum equation (Equation (21)), the freewheeling generator equation (Equation (22)), and the state-transition criteria, was implemented in MATLAB to numerically simulate the dynamic behaviour of the energy harvester. The governing equations were integrated in time using the variable-step solver ode45 with event detection to accurately capture the nonsmooth transitions between the torque-transmitting and freewheeling states. At each integration step, the clutch engagement condition ( $|\dot{\theta}| > \Omega$ ) and the clutch disengagement condition ( $|\dot{\theta}| \leq \Omega$ ) were monitored. Upon detection of a zero-crossing event, the solver terminated the current integration interval, updated the active equation set, transferred the state variables as the initial conditions for the new regime, and resumed the integration. The solver was configured with a relative tolerance of  $10^{-6}$ , an absolute tolerance of  $10^{-8}$ , and a maximum step size of 0.01 s. Two terminal event functions were defined: clutch engagement was detected as the upward zero-crossing of  $(|\dot{\theta}| - \Omega)$ , whereas clutch disengagement was detected as the downward zero-crossing of the transmitted clutch torque. This configuration resolves each switching instant to within the solver tolerance rather than to the nearest fixed time step.

To illustrate the piecewise switching behaviour of the system, a harmonic pitch excitation of the form  $\varphi_p(t) = \Phi_0 \sin(\omega_f t)$  was prescribed as the kinematic input, with the roll angle set to zero ( $\varphi_r = 0$ ) for clarity. The physical parameters employed in the simulation are listed in Table 2; these values represent the as-built prototype described in Section 4. Figure 3 presents the simulated time histories of the pendulum angular velocity,  $\dot{\theta}$ , (referred to the MRR input shaft) and the generator-side angular velocity,  $\Omega$ , (referred to the generator-side MRR output shaft). As the pendulum accelerates under wave excitation and its angular velocity exceeds the generator-side angular velocity, the sprag clutches engage, and the system enters the torque-transmitting state, during which both shafts are mechanically coupled. When the pendulum decelerates near its turning points and its angular velocity falls below  $\Omega$ , the sprag clutches disengage, and the system transitions to the freewheeling state. The pendulum reverses freely, while the generator-side assembly continues to rotate under its own inertia, with its angular velocity decaying exponentially in accordance with Equation (23). This alternating engagement-disengagement pattern confirms the intended rectification function of the MRR and demonstrates that the generator maintains a nonzero rotational speed, thereby enabling continuous power generation even during the pendulum reversal intervals. The simulation results are validated against experimental measurements in Section 5.



**Fig. 3** Simulated angular velocities of the pendulum shaft  $\theta(t)$  and generator shaft  $\Omega(t)$  under harmonic excitation ( $f = 1$  Hz,  $\Phi = 5^\circ$ )

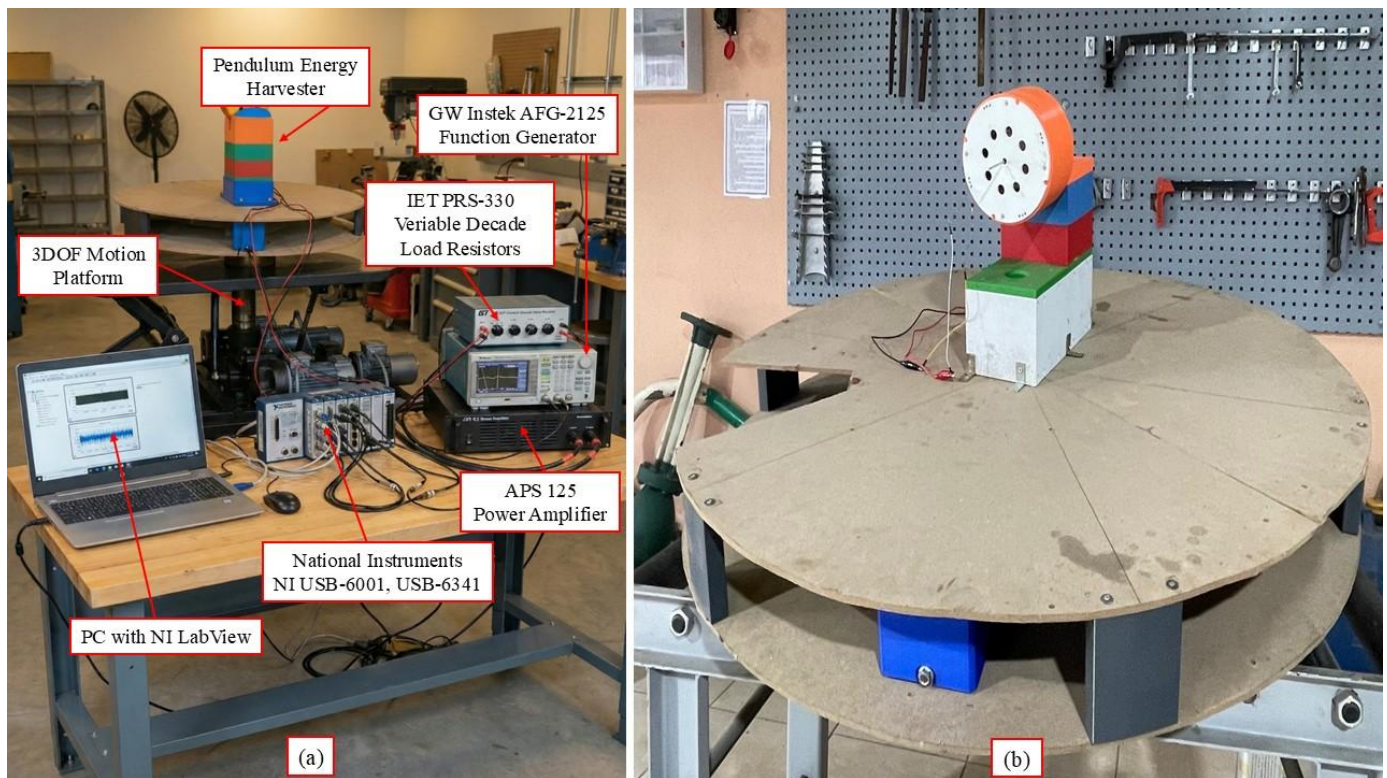
**Table 2** Physical parameters used in the numerical simulation of the energy harvester

| Symbol    | Description                                                 | Value                 | Unit                                         |
|-----------|-------------------------------------------------------------|-----------------------|----------------------------------------------|
| $J_a$     | Moment of inertia of pendulum arm about pivot               | $2.5 \times 10^{-1}$  | $\text{kg} \cdot \text{m}^2$                 |
| $m_b$     | Mass of pendulum bob                                        | 5.0                   | kg                                           |
| $L_b$     | Distance from pivot to bob centre of mass                   | 0.9                   | m                                            |
| $J_f$     | Moment of inertia of flywheel                               | $1.64 \times 10^{-2}$ | $\text{kg} \cdot \text{m}^2$                 |
| $J_r$     | Moment of inertia of generator rotor                        | $9.38 \times 10^{-3}$ | $\text{kg} \cdot \text{m}^2$                 |
| $J_{pg}$  | Equivalent moment of inertia of planetary gearbox           | $3.6 \times 10^{-3}$  | $\text{kg} \cdot \text{m}^2$                 |
| $J_h$     | Equivalent moment of inertia of hypoid gear pair            | $1.7 \times 10^{-4}$  | $\text{kg} \cdot \text{m}^2$                 |
| $N_{pg}$  | Planetary gear ratio                                        | 27                    | —                                            |
| $N_h$     | Hypoid gear ratio                                           | 3                     | —                                            |
| $N$       | Total gear ratio, $N = N_{pg} \cdot N_h$                    | 81                    | —                                            |
| $k_e$     | Generator back-EMF constant                                 | 0.95                  | $\text{V} \cdot \text{s/rad}$                |
| $k_t$     | Generator torque constant                                   | 0.95                  | $\text{N} \cdot \text{m/A}$                  |
| $R_a$     | Armature winding resistance                                 | 40                    | $\Omega$                                     |
| $R_L$     | External load resistance                                    | 0-100                 | $\Omega$                                     |
| $c_m$     | Mechanical damping coefficient                              | 0.05                  | $\text{N} \cdot \text{m} \cdot \text{s/rad}$ |
| $c_{m,G}$ | Mechanical damping coefficient                              | 0.10                  | $\text{N} \cdot \text{m} \cdot \text{s/rad}$ |
| $c_e$     | Electrical (electromagnetic) equivalent damping coefficient | 74.80                 | $\text{N} \cdot \text{m} \cdot \text{s/rad}$ |
| $g$       | Gravitational acceleration                                  | 9.81                  | $\text{m/s}^2$                               |

#### 4. Experimental setup

The experimental setup used to evaluate the performance of the pendulum-based energy harvesting mechanism is presented in Figure 4. The mechanism was mounted on a 3-DoF motion platform to enable the application of controlled, low-frequency excitation representative of wave-induced USV motions. As shown in Figure 4a, the motion platform was driven by sinusoidal signals generated by a programmable function generator (GW Instek AFG-2125), which were amplified by a power amplifier (APS 125) before being applied to the motion platform. The generator output was connected to an adjustable resistive load bank (IET PRS-330 variable decade resistor) to enable testing across a range of loading conditions. Voltage measurements were acquired using National Instruments (NI) USB-6001 and NI USB-6341 data acquisition modules, and all data were recorded using a dedicated LabVIEW programme running on a desktop computer. Figure 4b shows the fabricated prototype of the pendulum-based energy harvesting mechanism developed and tested in this study.

The generator terminal voltage, from which the back-EMF was obtained, was recorded using the differential analogue input channels of the NI USB-6341 module, while the pendulum angular position was measured using a rotary encoder mounted on the pivot shaft and acquired through the NI USB-6001 module. Both signals were synchronously acquired in LabVIEW at a sampling rate of 1 kHz, well above the highest electrical frequency of the low-speed generator. The commanded platform motion (amplitude, frequency, and pitch-roll phase), generated by the function generator and amplified by the power amplifier, provided the excitation reference against which the measured responses were compared.



**Fig. 4** Experimental test setup of the pendulum-based energy harvesting mechanism: (a) Overall experimental arrangement comprising the motion platform, measurement, and control equipment; (b) The fabricated prototype of the pendulum-based energy harvesting mechanism

The measurement uncertainty was assessed based on the instrument specifications. The generator terminal voltage was acquired at 16-bit resolution (NI USB-6341) with the channels operated within their calibrated range, and the load resistance was set with a precision decade box (IET PRS-330,  $\pm 0.1\%$ ). Since the electrical power is calculated as  $P = V^2/R$ , the propagated relative uncertainty is approximately  $\pm 5\%$ , with the voltage measurement constituting the dominant source of uncertainty. Each reported value represents a steady-state time average over multiple excitation cycles, and repeated measurements at representative operating points were consistent within this uncertainty bound. The residual differences between the

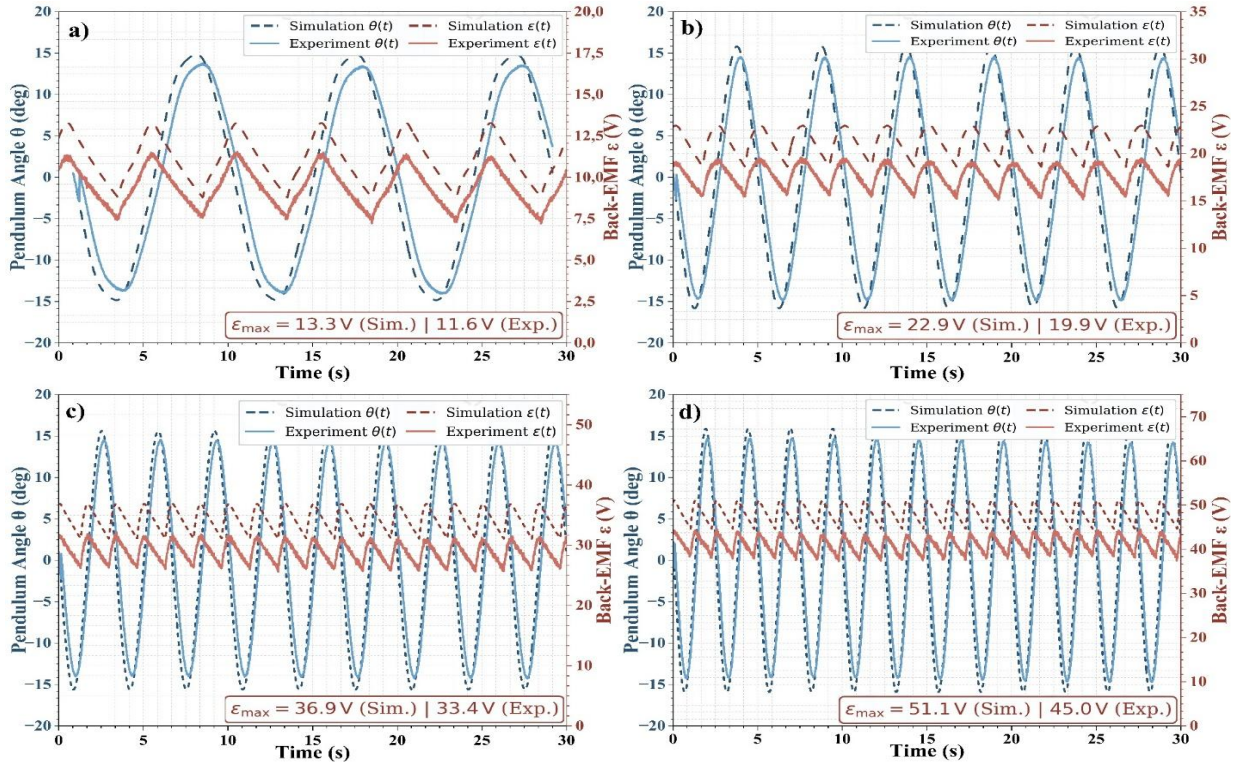
simulation and the experimental results (within  $\pm 5\%$  for output power) therefore fall within the combined measurement uncertainty.

The excitation amplitudes, frequencies, and phase relationships applied on the motion platform were selected to represent the wave-induced motion experienced by a small USV at sea. For a hull whose dimensions are small relative to the dominant wavelengths of the sea-swell band, the vessel behaves essentially as a wave follower, with its instantaneous pitch and roll tracking the local wave slope. Near the natural roll frequency, this response is amplified, and beam-sea roll amplitudes of  $15\text{--}30^\circ$  are commonly observed in small, lightly damped craft. The tested amplitude range ( $\pm 15^\circ$  to  $\pm 30^\circ$ ) therefore spans the motions characteristic of moderate to rough seas, while the excitation frequency range (0.05-1.0 Hz) covers and slightly exceeds the sea-swell band (0.05-0.7 Hz), within which ocean wave energy, and hence USV motion, is concentrated.

The phase relationship between pitch and roll in a real seaway is governed by the incident wave heading. The four tested configurations - pitch only, roll only (representative of head and beam seas), in phase ( $\psi = 0^\circ$ , corresponding to oblique or quartering seas), and in quadrature ( $\psi = 90^\circ$ ) - accordingly bracket the range of heading-dependent phase relationships encountered in directional waves. Finally, because long-endurance USV missions are characterised by low forward speeds (station keeping or slow survey,  $U \approx 0\text{--}2\text{ m}\cdot\text{s}^{-1}$ ), the Doppler shift between the wave and encounter frequencies is small. Consequently, the encounter frequency remains close to the wave frequency and within the tested band; therefore, vessel speed need not be treated as a separate excitation parameter.

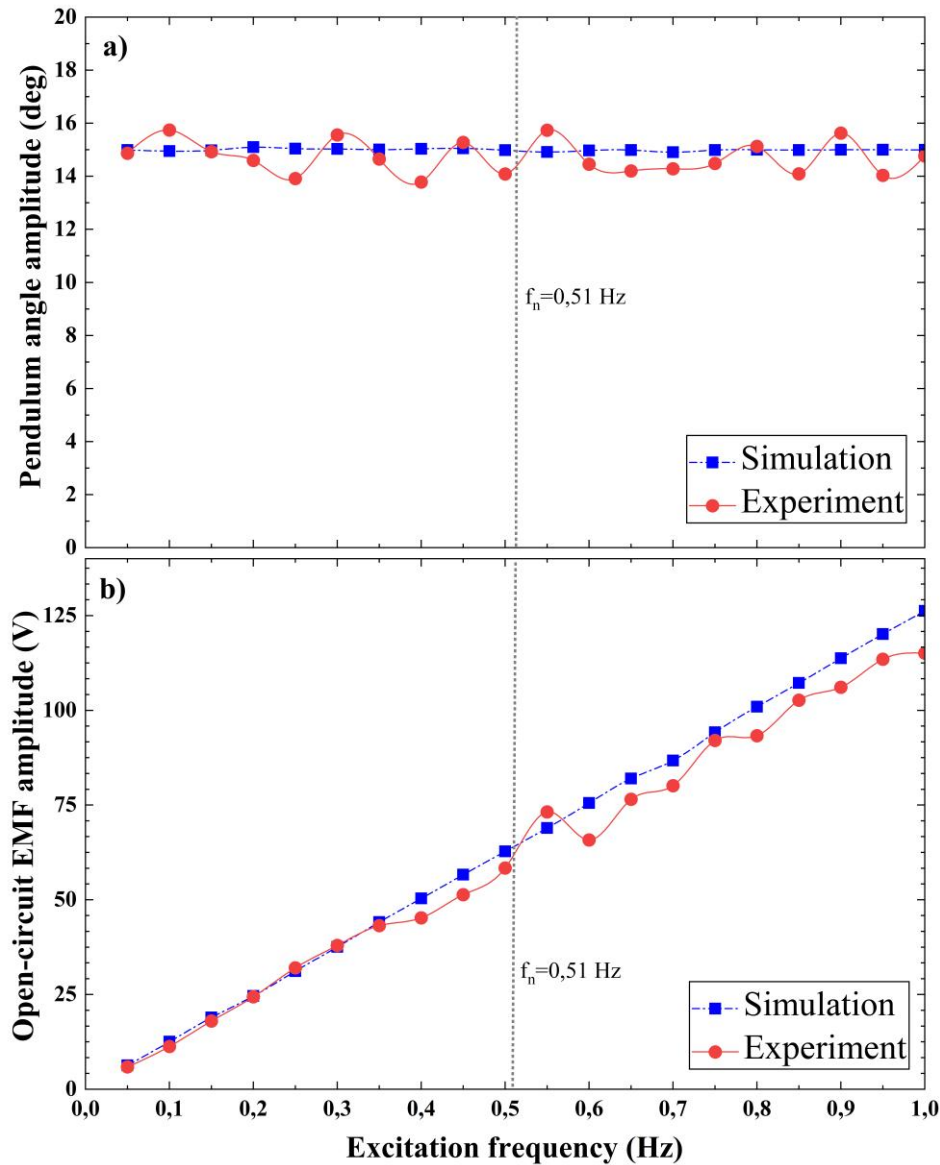
## 5. Results and discussion

This section presents the experimental and numerical characterisation of the MRR-based pendulum WEC, progressing from sinusoidal model validation to the assessment of irregular-wave performance. Figure 5 presents a time-domain comparison of the simulated and experimental pendulum angle  $\theta(t)$  and back-EMF  $\varepsilon(t)$  under sinusoidal roll excitation of  $\pm 15^\circ$  across four discrete excitation frequencies. Across all tested conditions (0.1-0.4 Hz), the simulation and experimental results show close agreement in both phase and amplitude. The peak back-EMF increases monotonically with excitation frequency, rising from 13.3 V / 11.6 V at 0.1 Hz to 51.1 V / 45.0 V at 0.4 Hz, consistent with the frequency-dependent amplification of the angular velocity by the MRR mechanism. The residual discrepancies of 12-15 % are attributed primarily to unmodelled gear-train and bearing-friction losses.



**Fig. 5** Time-domain model validation of pendulum angle  $\theta(t)$  and back-EMF  $\varepsilon(t)$  under sinusoidal roll excitation ( $\pm 15^\circ$ ) at excitation frequencies of (a) 0.1 Hz, (b) 0.2 Hz, (c) 0.3 Hz, and (d) 0.4 Hz

The frequency-domain characteristics of the MRR-based pendulum system were investigated under sinusoidal roll excitation with the excitation frequency swept from 0.05 to 1.0 Hz at a constant amplitude of  $\pm 15^\circ$ , and the results are presented in Figure 6. As shown in Figure 6a, the pendulum angle amplitude remains essentially constant at  $\theta \approx 15^\circ$  across the entire frequency range, confirming that the MRR effectively decouples the generator inertia from the pendulum dynamics and suppresses resonance-driven amplitude amplification. Figure 6b shows that the open-circuit EMF amplitude increases monotonically with excitation frequency, rising from 6.3 V at 0.05 Hz to 126.2 V at 1.0 Hz, consistent with the proportional relationship between the induced EMF and the angular velocity at a constant angular amplitude. The close agreement between the simulation and the experimental results across the full frequency range further validates the proposed dynamic model.

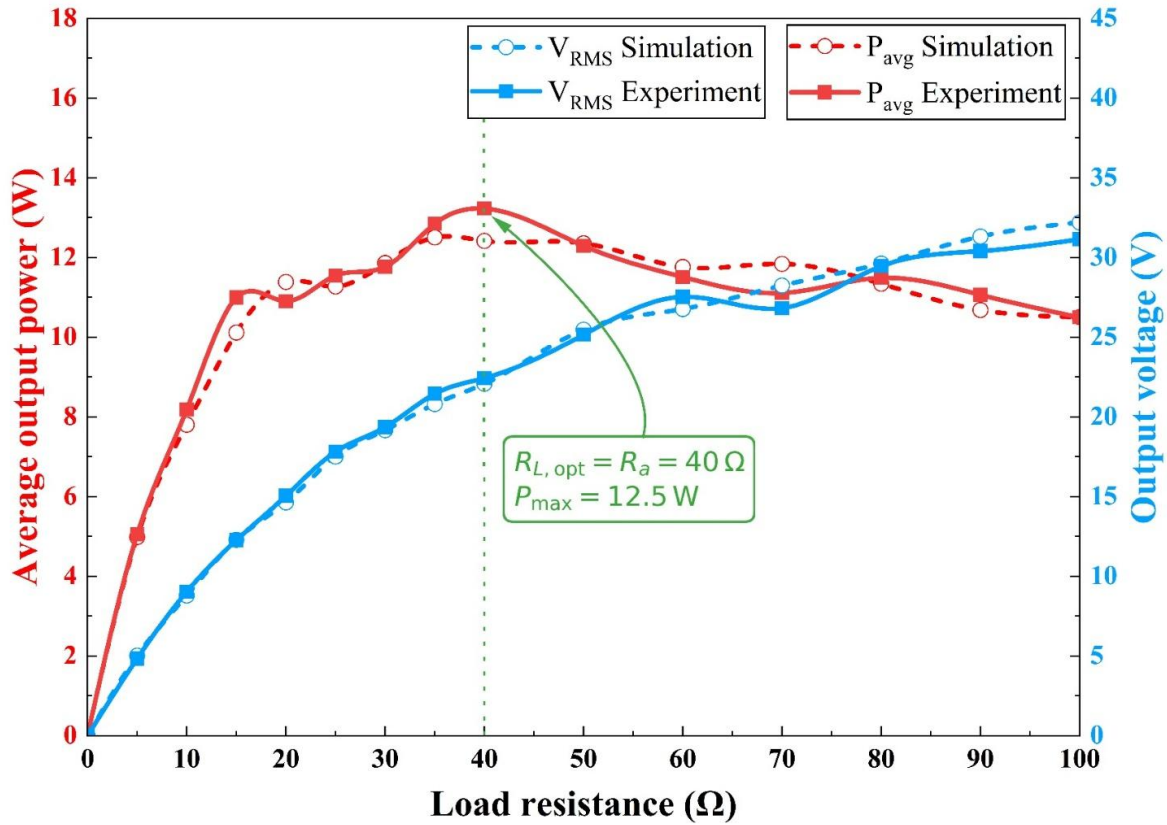


**Fig. 6** Frequency-domain response of the system under sinusoidal roll excitation ( $\pm 15^\circ$ ) from 0.05 to 1.0 Hz: (a) Pendulum angle amplitude; (b) Open-circuit EMF amplitude

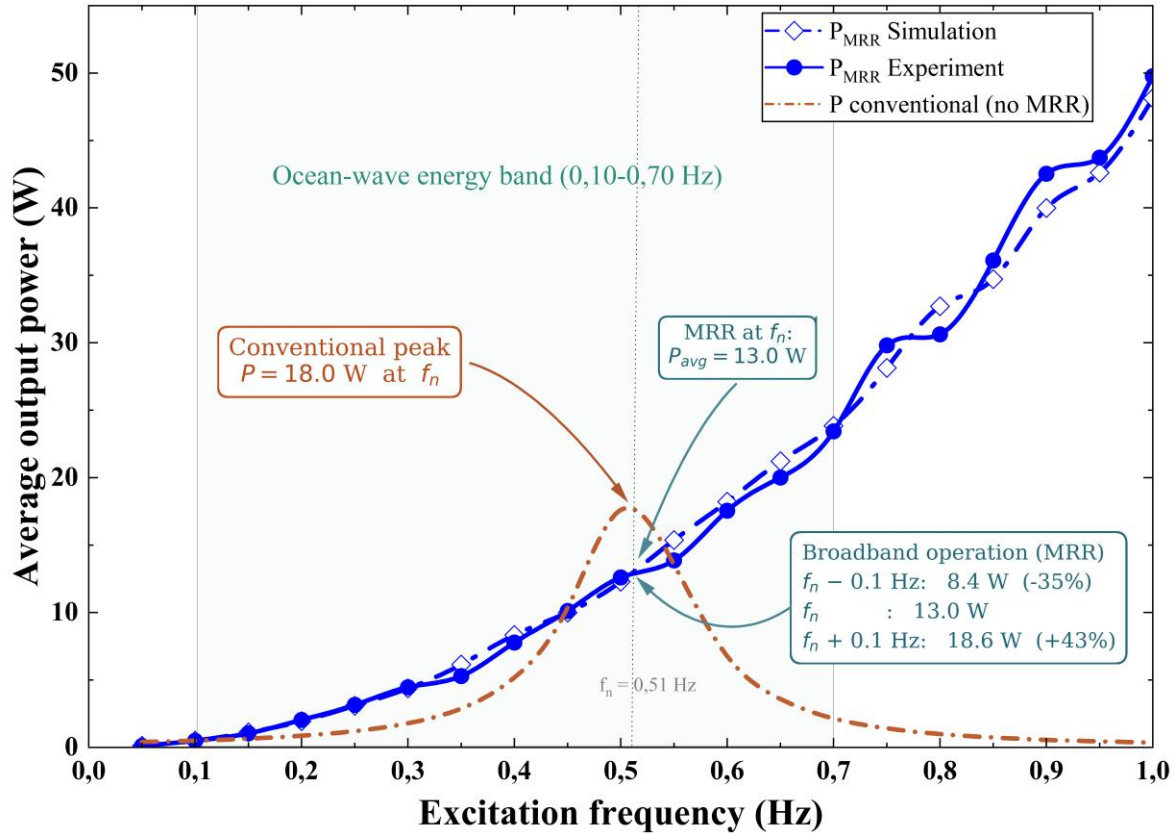
The dependence of the average output power,  $P_{avg}$ , and the RMS output voltage,  $V_{RMS}$ , on the load resistance,  $R_L$ , was examined under resonant excitation at  $f = 0.51$  Hz and  $\Phi_0 = 15^\circ$ , and the results are presented in Figure 7. Consistent with the maximum power transfer theorem,  $P_{avg}$  exhibits a well-defined peak at  $R_{L,opt} = 40 \Omega$ , equal to the armature resistance,  $R_a$ , with a maximum output power of  $P_{max} = 12.5$  W. Beyond this optimal point, power declines monotonically as the impedance mismatch increases, while  $V_{RMS}$  continues to rise with  $R_L$  throughout the tested range. The simulation and experimental results are in close agreement within  $\pm 5\%$ , thereby validating the electromechanical model and establishing  $R_L = 40 \Omega$  as the optimal operating point for all subsequent characterisation tests.

A key advantage of the MRR-based design is its broadband power-generation capability, which is demonstrated in Figure 8 through a direct comparison with a conventional pendulum harvester (without an MRR) over the 0.05-1.0 Hz frequency range ( $\Phi_0 = 15^\circ$ ,  $R_L = 40 \Omega$ ). For a fair comparison, the two configurations are identical in every respect, including pendulum mass and inertia, mechanical damping, generator, load resistance ( $R_L = 40 \Omega$ ), gear ratio, and mechanical losses, and differ only in the presence of the MRR. In the conventional configuration, the pendulum is directly coupled to the generator inertia. The broadband advantage is therefore attributable to the mechanical rotation rectifier alone rather than to any difference in tuning or damping. Whereas the conventional system produces a sharply peaked output confined

to the vicinity of the resonant frequency,  $f_n = 0.51$  Hz, reaching 18.0 W at resonance and falling below 2 W at a detuning of  $\pm 0.1$  Hz, the MRR-based harvester delivers power that increases monotonically across the entire ocean-wave energy band (0.10-0.70 Hz). At resonance, the MRR system yields  $P_{avg} = 13.0$  W, and the output increases to 18.6 W (+43 %) at  $f_n + 0.1$  Hz, whereas the output of the conventional system falls below 2 W. This broadband characteristic eliminates the need for active frequency retuning under variable sea states, representing a significant operational advantage for USV deployment.

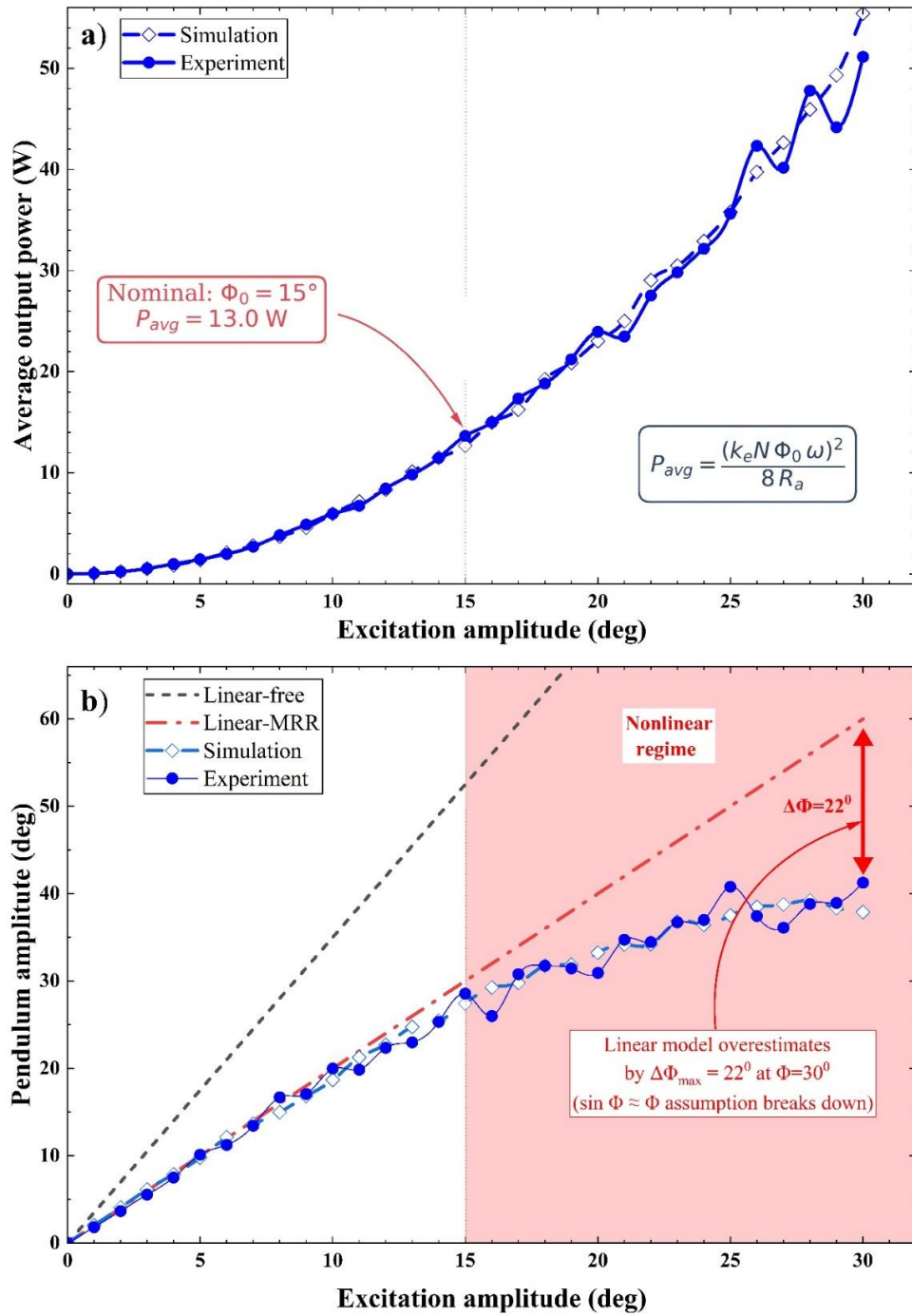


**Fig. 7** Average output power  $P_{avg}$  and RMS output voltage  $V_{RMS}$  as a function of load resistance  $R_L$  under resonant excitation ( $f = 0.51$  Hz,  $\Phi_0 = 15^\circ$ )



**Fig. 8** Average output power  $P_{avg}$  of the MRR-based pendulum WEC and a conventional pendulum harvester as a function of excitation frequency ( $\Phi_0 = 15^\circ$ ,  $R_L = 40 \Omega$ )

The effect of excitation amplitude on system performance is characterised in Figure 9 under resonant conditions ( $f = 0.51 \text{ Hz}$ ,  $R_L = 40 \Omega$ ). Figure 9a shows that the average output power scales quadratically with the excitation amplitude,  $\Phi_0$ , increasing from the nominal value of  $13.0 \text{ W}$  at  $\Phi_0 = 15^\circ$  to approximately  $52.1 \text{ W}$  at  $\Phi_0 = 30^\circ$ , in close agreement with both the simulation and the analytical predictions. This quadratic dependence implies that moderate increases in wave amplitude yield disproportionately large gains in harvested power, a characteristic particularly advantageous under energetic sea states. Figure 9b shows that, under free oscillation, the pendulum amplitude reaches 3.5 times the excitation amplitude, whereas the electromagnetic damping introduced by the MRR-coupled generator reduces this amplification factor to 2.0, thereby providing an inherent mechanical safety benefit. Beyond  $\Phi_0 \approx 15.8^\circ$ , the system enters a nonlinear regime in which the linear model progressively overestimates the pendulum amplitude, reaching a maximum deviation of  $\Delta\Phi = 22^\circ$  at  $\Phi_0 = 30^\circ$ , whereas the simulation and experimental results consistently remain below the linear-model prediction.



**Fig. 9** Average output power  $P_{avg}$  (a) and pendulum amplitude  $\theta_{max}$  (b) as a function of excitation amplitude  $\Phi_0$  ( $f=0.51$  Hz,  $R_L=40 \Omega$ )

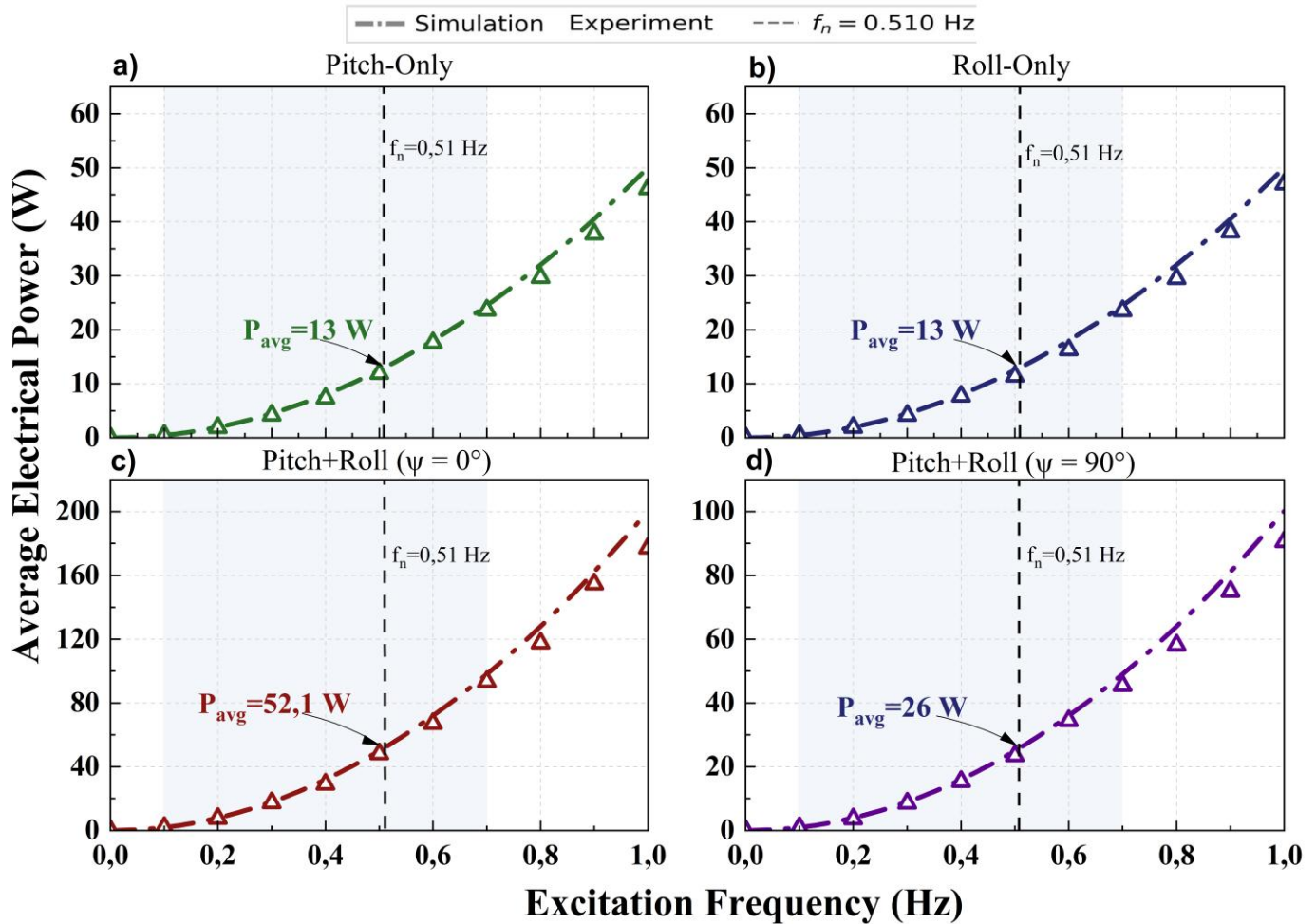
The influence of the pitch-roll phase angle,  $\psi$ , on the average output power was investigated across the 0-1.0 Hz frequency range ( $\Phi_0 = 15^\circ$ ,  $R_L = 40 \Omega$ ), and the results for four distinct motion configurations are presented in Figure 10. Under single-axis excitation, both the pitch-only and the roll-only cases yield identical outputs of  $P_{avg} = 13.0$  W at resonance, as shown in Figures 10a and 10b, confirming the symmetric response of the 2-DoF pendulum to each excitation axis independently. When both axes are excited in phase ( $\psi = 0^\circ$ ), Figure 10c shows that the output rises to  $P_{avg} = 52.1$  W at resonance, representing a fourfold increase over the single-axis case due to the constructive superposition of the angular velocities at the MRR input shaft. For the in-quadrature case ( $\psi = 90^\circ$ ), Figure 10d shows  $P_{avg} = 26.0$  W at resonance, representing a twofold increase over the single-axis output. Across all configurations, power increases monotonically with excitation frequency, confirming that the broadband harvesting characteristic of the MRR mechanism is preserved regardless of the phase relationship between motion axes. These results demonstrate that the 2-DoF

architecture provides between a twofold and a fourfold increase in output power relative to single-axis operation, depending on the pitch-roll phase relationship.

This behaviour follows directly from the gravitational coupling term of Equation (20). Because pitch and roll enter the generalised force through the orthogonal projections  $\sin \theta$  and  $\cos \theta$ , each excitation axis drives an independent response component at the pendulum shaft. In the linearised regime, these components superpose, such that  $\dot{\theta} = \dot{\theta}_p + \dot{\theta}_r$ , with a relative phase,  $\psi$ . Since the harvested power scales with the square of the shaft speed (Equation (24)), the theoretical magnification ratio relative to single-axis excitation of the same amplitude is:

$$\Gamma(\psi) = \frac{\langle \dot{\theta}^2 \rangle_{\text{coupled}}}{\langle \dot{\theta}^2 \rangle_{\text{single}}} = 2(1 + \cos \psi) = 4\cos^2(\psi/2) \quad (27)$$

Equation (27) predicts a fourfold gain for in-phase excitation ( $\psi = 0^\circ$ ), a twofold gain for quadrature excitation ( $\psi = 90^\circ$ ), and complete cancellation for antiphase excitation ( $\psi = 180^\circ$ ). The measured outputs of 52.1 W and 26.0 W, relative to the single-axis value of 13.0 W, correspond to magnification ratios of  $\Gamma = 4.01$  and  $\Gamma = 2.00$ , respectively, in close agreement with the theoretical prediction.

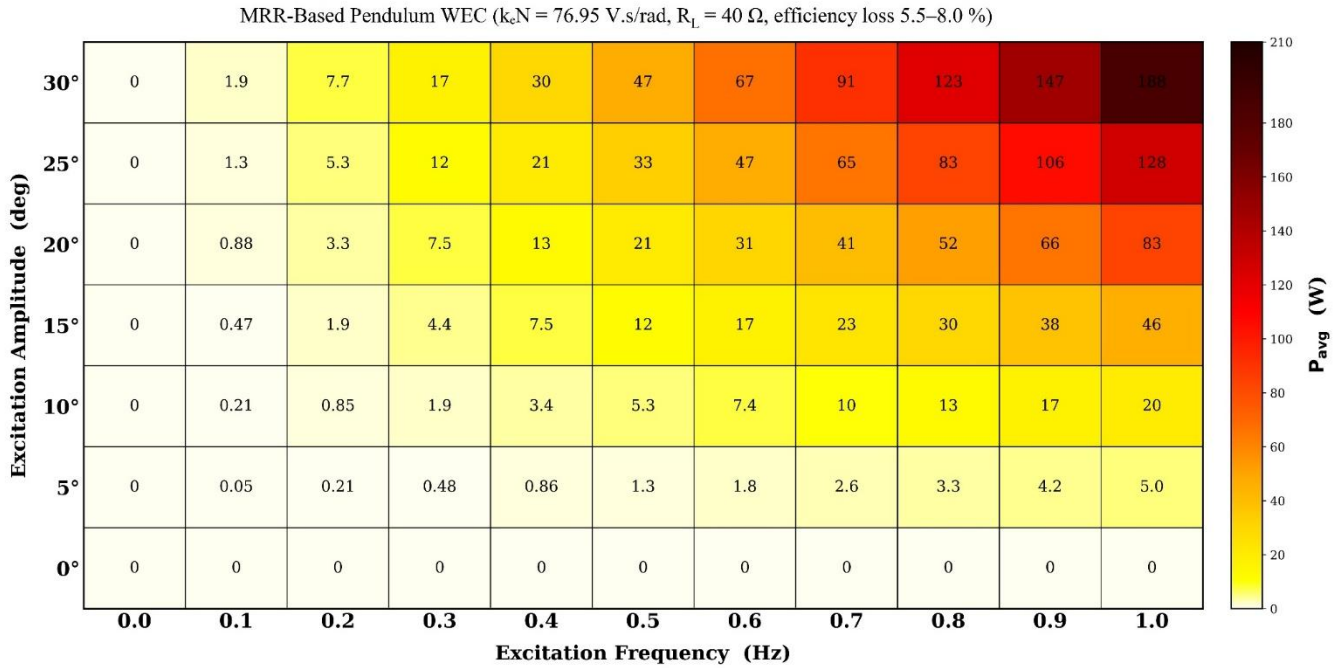


**Fig. 10** Average output power  $P_{\text{avg}}$  as a function of excitation frequency for four motion configurations ( $\Phi_0 = 15^\circ$ ,  $R_L = 40 \Omega$ ): (a) Pitch-only, (b) Roll-only, (c) Combined pitch+roll in phase ( $\psi = 0^\circ$ ), and (d) Combined pitch+roll in quadrature ( $\psi = 90^\circ$ )

The 3-DoF platform also permits heave excitation, which was not applied as a separate driving axis in Figure 10. As derived in Section 3.3, a purely vertical inertial force produces no moment about the pivot of the horizontal-plane pendulum and therefore cannot drive it directly; heave enters the dynamics only as a second-order effect through the modulation of the effective gravitational acceleration,  $(g + a_z)$ , in the pitch-roll gravitational coupling term. Consistent with that analysis, superimposing representative heave

accelerations ( $a_z$  of the order of 0.05-0.1  $g$ ) on the combined pitch-roll excitation changes the mean harvested power by less than 2 % in the validated model, confirming that the pitch-roll configurations shown in Figure 10 capture the excitation axes governing the performance of the device.

To provide a comprehensive performance map of the MRR-based pendulum WEC, the average output power,  $P_{avg}$ , was evaluated across the full excitation-frequency (0-1.0 Hz) and excitation-amplitude ( $0^\circ$ - $30^\circ$ ) parameter space under matched-load conditions ( $R_L = 40 \Omega$ ,  $k_e N = 76.95 \text{ V.s/rad}$ ), and the results are presented as a two-dimensional power map in Figure 11. The map confirms a monotonic increase in harvested power with both frequency and amplitude, consistent with the quadratic dependence established in Figure 9. Within the ocean-wave energy band, the system delivers up to 91 W at  $\Phi_0 = 30^\circ$  and  $f = 0.70 \text{ Hz}$ , demonstrating its practical viability under realistic sea-state conditions.



**Fig. 11** Two-dimensional average output power map of the MRR-based pendulum WEC as a function of excitation frequency and amplitude

The irregular-wave cases were driven by a JONSWAP spectrum with a significant wave height of  $H_s = 1.5 \text{ m}$ , a peak period of  $T_p = 7 \text{ s}$ , and the standard peak-enhancement factor  $\gamma = 3.3$ :

$$S(\omega) = \frac{\alpha g^2}{\omega^5} \exp \left[ -1.25 \left( \frac{\omega_p}{\omega} \right)^4 \right] \gamma^b \quad (28)$$

where  $\omega_p = 2\pi/T_p$ ,  $b = \exp [-(\omega - \omega_p)^2 / (2\sigma^2 \omega_p^2)]$ , with  $\sigma = 0.07$  for  $\omega \leq \omega_p$  and  $\sigma = 0.09$  otherwise, and where  $\alpha$  is scaled such that  $4\sqrt{m_0} = H_s$ ;  $m_0$  denotes the zeroth spectral moment. The spectrum was discretised into  $N$  components with a frequency spacing of  $\Delta\omega$ , spanning the energetic wave band, and the surface elevation was synthesised using the random-phase spectral method:

$$\eta(t) = \sum_{i=1}^N a_i \cos(\omega_i t + \delta_i), \quad a_i = \sqrt{2S(\omega_i)\Delta\omega} \quad (29)$$

where the phases,  $\delta_i$ , are drawn uniformly from the interval  $[0, 2\pi]$ . Under the wave-follower assumption, the platform pitch and roll motions track the local wave slope. For deep water, the wavenumber of each spectral component is  $k_i = \omega_i^2/g$ , such that:

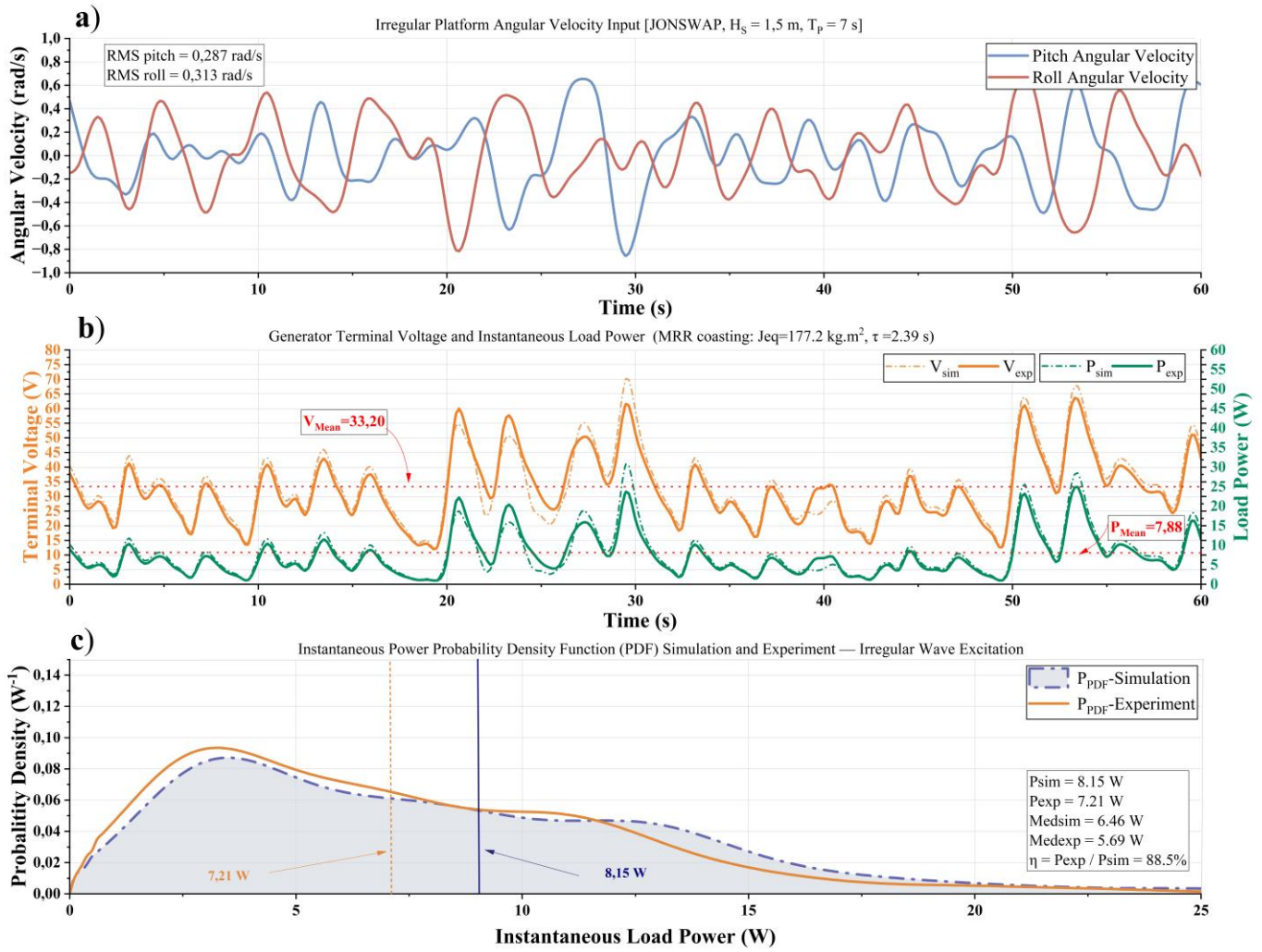
$$\varphi_p(t) = \lambda \sum_{i=1}^N k_i a_i \cos(\omega_i t + \delta_i^p), \quad \varphi_r(t) = \lambda \sum_{i=1}^N k_i a_i \cos(\omega_i t + \delta_i^r) \quad (30)$$

Independent phase sets,  $\delta_i^p$  and  $\delta_i^r$ , were used for the two excitation axes to represent their directional decorrelation in a real seaway. The factor,  $\lambda$ , accounts for the amplification of the hull response relative to the pure wave slope near the natural roll frequency and was calibrated such that the synthesised records reproduce the root-mean-square angular velocities of  $0.287 \text{ rad}\cdot\text{s}^{-1}$  (pitch) and  $0.313 \text{ rad}\cdot\text{s}^{-1}$  (roll) shown in Figure 12a. The corresponding angular velocities, obtained by differentiating Equation (30) with respect to time, were prescribed as the kinematic inputs to the model, which was then integrated using the ode45 event-detection scheme described in Section 3.4.

The resulting system performance is presented in Figure 12. Figure 12a shows the stochastic platform angular-velocity inputs in the pitch and roll directions used as the kinematic excitation of the model. Figure 12b presents the corresponding generator terminal voltage and instantaneous load power over a 60 s record, for which the simulation and experimental results are in close agreement throughout. The MRR flywheel ( $J_{eq} = 177.2 \text{ kg}\cdot\text{m}^2$ ,  $\tau = 2.39 \text{ s}$ ) provides kinetic energy storage that sustains the terminal voltage during low-energy wave intervals, resulting in a mean terminal voltage of 33.2 V. Figure 12c presents the probability density function of the instantaneous load power for both the simulation and the experiment, revealing a right-skewed distribution characteristic of broadband irregular-wave excitation. The simulated and experimental mean powers are 8.15 and 7.21 W, with corresponding median values of 6.46 and 5.69 W, respectively. The ratio of the experimental to the simulated mean power is 88.5%. This value quantifies the agreement between the analytical model and the experimental measurements - that is, the predictive accuracy of the model - and should not be interpreted as a physical energy-conversion efficiency.

To avoid ambiguity, the distinct efficiency metrics of the system are defined separately from the model-agreement metric. The mechanical transmission efficiency of the drivetrain - comprising the three-stage planetary gearbox, the hypoid stage, and the supporting bearings - is estimated to be  $\eta_{mech} \approx 85 \%$ , consistent with typical per-stage gear efficiencies and with the 12-15 % discrepancy attributed to unmodelled friction discussed above. The generator electrical efficiency at the matched-load operating point ( $R_L = R_a = 40 \Omega$ ) is  $\eta_{elec} = R_L / (R_a + R_L) = 50 \%$ . This is an intrinsic consequence of the maximum-power-transfer condition, under which half of the generated electrical power is dissipated in the armature resistance. The matched load was selected deliberately to maximise the harvested power rather than the efficiency: operating at a higher load resistance would raise  $\eta_{elec}$  toward about 80 % (at  $R_L \approx 4R_a$ ) at the expense of reduced output power.

Combining these stages, the overall motion-to-electric conversion efficiency from the mechanical power delivered at the pendulum shaft to the electrical power delivered to the load is given by  $\eta_{m \rightarrow e} = \eta_{mech} \times \eta_{elec} \approx 0.85 \times 0.50 \approx 43 \%$  at the matched-load operating point. It is emphasised that this represents the motion-to-electric conversion efficiency. The wave-to-motion capture efficiency of the hull lies outside the scope of the present study; therefore, no wave-to-electric efficiency is claimed. In summary, the 88.5 % value represents the agreement between the analytical model and the experimental measurements, whereas the physical conversion performance is characterised by  $\eta_{mech} \approx 85 \%$ ,  $\eta_{elec} = 50 \%$ , and  $\eta_{m \rightarrow e} \approx 43 \%$  at the maximum-power operating point.



**Fig. 12** System performance under irregular JONSWAP wave excitation ( $H_s = 1.5$  m,  $T_p = 7$  s): (a) Platform angular velocity inputs in pitch and roll; (b) Generator terminal voltage and instantaneous load power; (c) Probability density function of instantaneous load power for simulation and experiment

The time-domain validation presented in Figure 12 was carried out for a single JONSWAP realisation selected to represent a moderate sea state. Operational deployments, however, expose a USV to a broad spectrum of wave climates, and a hull-integrated harvester intended for such platforms must deliver useful power throughout this range rather than at a single design point. To characterise this behaviour without resorting to an exhaustive experimental campaign, the mean output was projected across the range of sea states using the response laws validated in the preceding sections. The projection is underpinned by two experimentally confirmed relationships. As demonstrated in Figure 9, the average power scales quadratically with excitation amplitude, whereas Figures 8 and 11 show that the induced electromotive force, and hence the delivered power, increases monotonically with excitation frequency across the ocean wave band.

For a JONSWAP sea, the root-mean-square platform angular velocity is proportional to  $H_s/T_p$ . Since the generator back-electromotive force is proportional to the shaft angular velocity, and the load power to its square, the mean harvested power obeys  $P_{avg} \propto (H_s/T_p)^2$ . Anchoring this scaling relationship to the experimentally validated operating point ( $H_s = 1.5$  m,  $T_p = 7$  s,  $P_{avg} = 7.21$  W) determines the proportionality constant and yields the projections summarised in Tables 3 and 4.

**Table 3** Projected mean harvested power  $P_{\text{avg}}$  (W) over the  $H_s$ - $T_p$  envelope; the shaded cell marks the validated case

| $H_s \setminus T_p$ | 5 s  | 6 s  | 7 s  | 8 s  | 9 s  | 10 s | 11 s |
|---------------------|------|------|------|------|------|------|------|
| 0.5 m               | 1.6  | 1.1  | 0.8  | 0.6  | 0.5  | 0.4  | 0.3  |
| 1.0 m               | 6.3  | 4.4  | 3.2  | 2.5  | 1.9  | 1.6  | 1.3  |
| 1.5 m               | 14.1 | 9.8  | 7.2  | 5.5  | 4.4  | 3.5  | 2.9  |
| 2.0 m               | 25.1 | 17.4 | 12.8 | 9.8  | 7.8  | 6.3  | 5.2  |
| 2.5 m               | 39.3 | 27.3 | 20.0 | 15.3 | 12.1 | 9.8  | 8.1  |
| 3.0 m               | 56.5 | 39.3 | 28.8 | 22.1 | 17.4 | 14.1 | 11.7 |

Table 3 shows that the harvested power increases markedly with sea severity, rising by approximately a factor of four for each doubling of the significant wave height, in agreement with the quadratic dependence on excitation amplitude demonstrated in Figure 9. At a fixed significant wave height, the output decreases as the peak period increases, reflecting the lower platform angular velocities associated with longer-period, gentler swells. Importantly, the output varies smoothly and monotonically throughout the entire operating envelope, confirming that the system performance is governed by the broadband characteristics of the MRR rather than by a single resonant condition.

**Table 4** Mean harvested power and RMS platform angular velocity for representative sea states

| Sea state    | $H_s$ range (m) | Nominal $H_s$ (m) | $T_p$ (s) | RMS platform angular velocity (rad s <sup>-1</sup> ) | Mean power $P_{\text{avg}}$ (W) |
|--------------|-----------------|-------------------|-----------|------------------------------------------------------|---------------------------------|
| SS2-Smooth   | 0.10-0.50       | 0.4               | 5.5       | 0.144                                                | 0.8                             |
| SS3-Slight   | 0.50-1.25       | 0.9               | 6.5       | 0.274                                                | 3.0                             |
| SS4-Moderate | 1.25-2.50       | 1.5               | 7.0       | 0.425                                                | 7.2                             |
| SS4-Moderate | 1.25-2.50       | 2.0               | 8.0       | 0.495                                                | 9.8                             |
| SS5-Rough    | 2.50-4.00       | 3.0               | 9.5       | 0.626                                                | 15.7                            |

Note: SS1 ( $H_s < 0.1$  m) is excluded owing to negligible wave excitation ( $< 0.05$  W).

When referenced against the continuous power demand of small- to medium-sized USV platforms, typically ranging from a few to several tens of watts [12-14], the harvester meets or exceeds this requirement throughout the slight-to-rough sea-state range (SS3-SS5), while still providing a useful trickle charge under the calmest conditions (SS2). The system therefore sustains a positive energy balance across the sea states most frequently encountered in service, and does so without active frequency retuning, an operational advantage for autonomous, long-endurance missions.

Because the 60 s experimental record presented in Figure 12 spans only a limited number of wave cycles, the statistical robustness of the stochastic response was examined using the validated model. Thirty independent JONSWAP realisations ( $H_s = 1.5$  m,  $T_p = 7$  s), each with a duration of 30 min (1800 s) and generated using a different random-phase seed, were simulated. The instantaneous load power has a coefficient of variation of approximately 0.54 and an integral time scale of approximately 5.3 s. Accordingly, the standard error of the mean estimated from a single record scale as  $CV \cdot (2\tau_{\text{int}}/T)^{0.5}$ . This yields a relative standard error of approximately 23 % for a single 60 s record, confirming that such a short record is statistically limited. The corresponding value decreases to approximately 4 % for a single 30 min record and, for an ensemble of thirty 30 min realisations, to a 95 % confidence interval of  $\pm 1.5$  % for the mean power.

The resulting confidence intervals for the principal output quantities are summarised in Table 5. The mean load power, RMS terminal voltage, and peak load power are all determined within a few percent, and the mean values obtained from the 60 s experiment (7.21 W and 33.2 V) lie within these confidence intervals. The probability density functions obtained from the independent realisations are mutually consistent, retaining the right-skewed distribution observed in Figure 12c. By smoothing the shaft speed, the MRR flywheel limits

the output variance and thereby narrows these confidence intervals. These results confirm that the reported performance is statistically representative rather than an artefact of the record length.

**Table 5** Mean values and 95 % confidence intervals of the principal output quantities, obtained from thirty independent 30 min JONSWAP realisations ( $H_s = 1.5$  m,  $T_p = 7$  s)

| Quantity             | Mean   | 95 % confidence interval    |
|----------------------|--------|-----------------------------|
| Mean load power      | 7.21 W | $\pm 0.11$ W ( $\pm 1.5$ %) |
| RMS terminal voltage | 33.2 V | $\pm 0.25$ V ( $\pm 0.7$ %) |
| Peak load power      | 27.4 W | $\pm 1.1$ W ( $\pm 4$ %)    |

Beyond the electrical performance, the mechanical loading imposed on the drivetrain under severe excitation also warrants consideration. The largest torque occurs at the low-speed pendulum (MRR) shaft, where it is bounded by the peak driving torque acting on the pendulum. Under the most severe tested condition ( $\Phi_0 = 30^\circ$ , combined in-phase pitch and roll), the plane-tilt gravitational torque reaches approximately 31 N·m, while the inertial excitation associated with the peak platform angular acceleration ( $5.4 \text{ rad}\cdot\text{s}^{-2}$ ) contributes a comparable 23 N·m. Consequently, the peak torque transmitted through the sprag clutches remains on the order of 40 N·m. Because the 81:1 speed increasing transmission reduces torque in inverse proportion to its ratio, the corresponding torque at the generator shaft and at the downstream planetary and hypoid stages is only about 0.5 N·m, leaving these components with a substantial capacity margin. The critical mechanical loading is therefore concentrated at the sprag-clutch-and-bevel-gear MRR stage, which was selected specifically for its high torque capacity and rapid engagement. Quantitatively, the selected sprag clutches (Stieber CSK20P) provide a rated torque of 50 N·m within a 47 mm outer-diameter envelope, whereas roller-type one-way clutches of comparable size are typically limited to a fraction of this capacity. Moreover, their engagement requires the roller to traverse the ramp, introducing several degrees of angular lost motion during each reversal [35]. Because the sprag elements remain in continuous spring-loaded contact with both races, engagement is backlash-free, and the lost motion is reduced to a few arc-minutes. This characteristic is particularly advantageous at the low excitation frequencies considered in the present study, where the pendulum reverses direction twice during each wave cycle.

The freewheeling architecture further mitigates impact loading. By decoupling the generator inertia during each pendulum reversal, the architecture eliminates stall-and-restart cycles and the abrupt counter-torque reversals that burden roller clutches and fixed bevel-gear rectifiers, allowing torque to be transmitted progressively rather than through shock loading. With respect to fatigue, the low-frequency wave excitation (approximately 0.1-0.5 Hz) imposes on the order of  $10^7$  load cycles per year of continuous operation. Combined with the modest torque levels established above, the components therefore operate well within the high-cycle fatigue regime. This assessment can be made quantitative. At the most severe tested condition the pivot shaft carries a von Mises equivalent stress of approximately 80 MPa, of which the alternating bending component is about 66 MPa. For a medium-carbon steel shaft, the corrected endurance limit, after accounting for surface finish, size, and stress concentration, is of the order of 120-150 MPa. The operating stress therefore remains below the endurance limit, placing the shaft in the infinite-life regime rather than merely within the high-cycle fatigue regime. A further advantage is realised in the gear stages. Because the MRR rectifies the motion, each gear path is loaded unidirectionally, such that the tooth-root stress follows a repeated (zero-to-maximum) loading cycle rather than the fully reversed cycle imposed by fixed bevel-gear rectifiers, for which the permissible stress is substantially lower. The downstream planetary and hypoid stages, carrying only about 0.5 N·m, retain a large margin in any case. The minimal gear count of the MRR limits backlash. The assembly is grease-lubricated within a sealed, watertight enclosure that isolates it from seawater, while the low shaft speeds and modest torque levels render frictional heat generation negligible. The sealing arrangement is likewise favourable with regard to potential failure modes. Since no rotating shaft penetrates the outer enclosure wall, the harvester requires no dynamic shaft seal, thereby eliminating by design the wear-driven rotary-seal failure that dominates the sealing reliability of marine machinery; only static gasket seals at the enclosure lids remain, and these are amenable to inspection at scheduled maintenance intervals. The reinforced-polymer housing is not susceptible to galvanic corrosion in seawater, and the drivetrain operates in

a grease-lubricated condition within the sealed enclosure, with the freewheel units supplied prelubricated and rated for continuous operation between +5 °C and +60 °C. A detailed contact-stress and stress-life fatigue assessment, together with accelerated durability testing addressing clutch wear, lubrication, and performance degradation under sustained cyclic marine operation, is identified as an important direction for future work.

The structural integrity of the load-bearing members at the 30° excitation-amplitude limit was verified through a quasi-static assessment of the 20 mm steel pivot shaft, which carries the highest combined loading. Under this condition, the shaft transmits a peak drive torque of approximately 40 N·m (torsional shear stress  $\approx$  25 MPa), while the dynamic and gravitational transverse loads generated by the pendulum bob impose a bending moment of approximately 52 N·m at the pendulum-arm root (bending stress  $\approx$  66 MPa). The resulting von Mises equivalent stress of approximately 80 MPa corresponds to a safety factor of approximately 3 relative to the yield strength of conventional structural steel ( $\approx$  250 MPa), increasing to approximately 4 for a medium-carbon shaft steel. The pivot shaft therefore operates at approximately one-third of its yield strength, confirming a comfortable structural margin at the upper excitation limit. A detailed finite-element analysis of the complete assembly is left for future work.

A practical requirement for USV integration is that the harvester output voltage should be compatible with charging the on-board lithium battery. Although the permanent magnet generator delivers a variable-frequency, variable-amplitude three-phase output, its voltage level is well suited to this application. At the matched-load resonance point, the RMS terminal voltage is  $V_{\text{RMS}} = 22 \text{ V}$  ( $R_L = 40 \text{ } \Omega$ ), increasing to approximately 46 V under the most energetic tested conditions, while the mean terminal voltage under irregular JONSWAP excitation is 33.2 V (Figure 12). These voltage levels fall within or exceed the charging-voltage ranges of the lithium-battery configurations most commonly used on small- to medium-sized USV platforms: approximately 12-16.8 V for a 4S pack, 21-29.4 V for a 24 V (7S) system, and 39-54.6 V for a 48 V (13S) system.

Because the generator voltage generally exceeds these charging-voltage ranges, a front-end three-phase rectifier followed by either a step-down (buck) or a buck-boost DC-DC converter - both standard, high-efficiency (>90 %) power-electronic building blocks is sufficient to regulate the fluctuating output to a stable charging voltage. This conditioning burden is further eased by the flywheel, which sustains the shaft speed, and hence the terminal voltage, during low-energy wave intervals, as reflected in the 33.2 V mean maintained under irregular excitation. The detailed design and closed-loop control of this stage remain subjects for future work. Nevertheless, the present results confirm that the harvester output is voltage-compatible with direct lithium-battery charging across the common 12-48 V range.

The one-way (decoupled) treatment of the hull and the harvester adopted in Section 3 can be examined quantitatively through two independent budgets: the reaction moment exerted on the hull, and the equivalent roll damping added to it.

**Reaction-moment budget:** The largest destabilising action that the harvester can impose on the carrier arises from the lateral displacement of the pendulum bob, whose worst-case heeling moment is  $m_b g L_b = 44 \text{ N}\cdot\text{m}$ . This is to be compared with the hydrostatic restoring moment of the host platform,  $\Delta g GM \sin\phi$ . Even for a modest platform with a displacement of 2 t and a metacentric height of 1 m, the restoring moment at a heel angle of 15° is approximately 5.1 kN·m. The harvester reaction therefore amounts to less than 1% of the restoring capacity, while the representative multihull platform shown in Figure 1, whose waterplane moment of inertia and displacement are substantially larger, is expected to be stiffer still. The 44 N·m figure is therefore an upper bound that is not approached in practice.

**Added-damping budget:** The mechanical power extracted at the pendulum shaft under the validated sea state is obtained from the mean electrical output and the motion-to-electric conversion efficiency.  $P_{\text{mech}} = P_{\text{avg}}/\eta_{m \rightarrow e} = 7.21/0.43 \approx 16.8 \text{ W}$ . When referred to the measured RMS platform angular velocity of  $0.425 \text{ rad}\cdot\text{s}^{-1}$  (Table 5), this corresponds to an equivalent added roll-damping coefficient of:

$$c_{\text{add}} = \frac{P_{\text{mech}}}{\langle \dot{\phi}^2 \rangle} \approx 93 \text{ N}\cdot\text{m}\cdot\text{s}\cdot\text{rad}^{-1} \quad (31)$$

The critical roll damping of a platform of the present class,  $2(I_{xx} + A_{44})\omega_n$ , is of the order of  $10^4$ - $10^5 \text{ N}\cdot\text{m}\cdot\text{s}\cdot\text{rad}^{-1}$ , such that the damping introduced by the harvester is two to three orders of magnitude lower. Moreover, in the wave-follower regime, the excitation frequency lies well below the natural roll frequency ( $\omega \ll \omega_n$ ), where the response is stiffness-dominated and the motion amplitude is governed primarily by the wave slope and the hydrostatic restoring moment rather than by damping. Consequently, the sensitivity of the motion amplitude to a damping perturbation is itself of the order of  $(\omega/\omega_n)^2$ . The two budgets are mutually consistent and indicate that the error introduced by the one-way formulation is bounded to within at most a few percent of the reported power.

These estimates are deliberately expressed as upper bounds rather than as predictions for a specific hull, since both the reaction and damping ratios decrease with increasing displacement and metacentric height. A resolved treatment of the two-way interaction, in which the incident waves, the hull, and the pendulum are solved simultaneously, would require a CFD-based or coupled boundary-element formulation and is identified in the Conclusion as a dedicated avenue for future work.

Taken together, these results also inform the practical integration of the harvester. The unit is housed entirely within the hull, with no external moving parts, and its reaction and mounting loads are bounded by the peak transmitted torque of approximately 40 N·m, which is reacted through the 20 mm pivot shaft with a safety factor of approximately 3. Survivability under extreme motions is enhanced by the electromagnetic damping provided by the MRR, which limits the pendulum amplitude to approximately twice the excitation amplitude. Additional protection may be provided by mechanical end-stops, while compatibility with the on-board battery and power-electronic systems has been established above. The remaining vessel-level effects including added mass, centre-of-gravity shift, occupied volume fraction, and their influence on stability depend on the mass properties of the specific host platform. Since the present study characterises a prototype-scale device on a 3-DoF motion platform emulating the wave-induced motion of a representative hull, their quantitative assessment is deferred to the platform-integration and seakeeping study identified in the Conclusion.

## 6. Conclusion

This study presented a horizontal-plane pendulum-based inertial wave energy harvester intended for integration within the hull of a USV. By orienting the pendulum parallel to the XY-plane and coupling it to a sprag-clutch-based mechanical rotation rectifier (MRR), a three-stage planetary gearbox, a hypoid gear stage, and a Halbach-array permanent magnet generator, the proposed architecture was configured to convert the multiaxial, low-frequency motion of the USV into a stable, unidirectional electrical output without external moving parts in contact with the water. A Lagrangian-based mathematical model was developed and partitioned into torque-transmitting and freewheeling states to capture the piecewise switching behaviour introduced by the sprag clutches. The model was subsequently validated against experimental measurements obtained on a motion platform under sinusoidal, swept-frequency, and JONSWAP-based irregular-wave excitation.

The system delivered a maximum average power of 12.5 W under the matched-load condition ( $R_L = 40 \Omega$ ) during resonant single-axis excitation at  $f = 0.51 \text{ Hz}$  and  $\Phi_0 = 15^\circ$ , with simulation and experimental results agreeing within  $\pm 5 \%$ . Frequency-domain testing confirmed that the mechanical rotation rectifier effectively decouples the generator inertia from the pendulum, thereby suppressing resonance-driven amplitude amplification and yielding a monotonic increase in output power across the principal ocean-wave band (0.10-0.70 Hz). At  $f_n + 0.1 \text{ Hz}$ , the proposed system delivered 18.6 W, whereas a conventional pendulum harvester without rectification produced less than 2 W. The two-degree-of-freedom (2-DoF) architecture further enhanced performance under combined pitch-and-roll excitation, producing up to a fourfold increase in output power relative to single-axis operation (52.1 W at  $\psi = 0^\circ$ ) and at least a twofold increase for any pitch-roll phase relationship. The two-dimensional power map demonstrated practical viability across realistic sea states, reaching 91 W at  $\Phi_0 = 30^\circ$  and  $f = 0.70 \text{ Hz}$ . Under JONSWAP-based irregular-wave excitation ( $H_s = 1.5 \text{ m}$ ,  $T_p = 7 \text{ s}$ ), the harvester sustained a mean experimental output of 7.21 W, with the flywheel preserving the terminal voltage during low-energy intervals. The experimental-to-simulation mean-power

ratio of 88.5 % confirmed the predictive capability of the model under stochastic excitation. The physical conversion performance was characterised separately by a mechanical transmission efficiency of approximately 85 %, a matched-load generator efficiency of 50 %, and an overall motion-to-electric conversion efficiency of approximately 43 %.

Taken together, these results indicate that a full-scale hull-integrated harvester employing horizontal-plane pendulum kinematics and sprag-clutch rectification can meet the continuous power requirements of small- to medium-sized USV platforms while remaining robust under variable sea states without requiring active frequency tuning. Future work will address open-sea validation under representative wave climates and explore power-electronic conditioning strategies for direct integration with on-board energy storage. The coupled hull-harvester hydrodynamics also warrant dedicated investigation. A CFD-based model could resolve the two-way interaction among the incident waves, the USV hull, and the pendulum. Such an analysis would validate and extend the present motion-platform results under deployed operating conditions, building on recent CFD studies of USV hydrodynamic response in waves [16, 38]. Such an investigation lies beyond the scope of the present study and therefore constitutes an important direction for future research.

### Declaration of competing interest

The author declares that the mechanical mechanism described in this manuscript is the subject of a Turkish patent application (No. 2025/007756), filed by Ordu University, with the author as the sole inventor. No other competing interests are declared.

### Appendix A. Nomenclature

| Symbol                                | Description                                                                                                                                                           | Unit                |
|---------------------------------------|-----------------------------------------------------------------------------------------------------------------------------------------------------------------------|---------------------|
| <b>State variables and kinematics</b> |                                                                                                                                                                       |                     |
| $\theta$                              | Pendulum angular displacement about the vertical z-axis in the horizontal XY-plane                                                                                    | rad                 |
| $\dot{\theta}$                        | Pendulum angular velocity (state variable; MRR input-shaft speed)                                                                                                     | rad·s <sup>-1</sup> |
| $\ddot{\theta}$                       | Pendulum angular acceleration                                                                                                                                         | rad·s <sup>-2</sup> |
| $\Omega$                              | Generator-side angular velocity referred to the pendulum shaft; $\Omega =  \dot{\theta} $ in the transmitting state, decaying independently in the freewheeling state | rad·s <sup>-1</sup> |
| $\omega_g$                            | Generator-shaft angular velocity, $\omega_g = N\Omega$                                                                                                                | rad·s <sup>-1</sup> |
| $\varphi_p$                           | Wave-induced pitch angle (about body y-axis; prescribed input)                                                                                                        | rad                 |
| $\varphi_r$                           | Wave-induced roll angle (about body x-axis; prescribed input)                                                                                                         | rad                 |
| $\dot{\varphi}$                       | Combined effective z-component of the USV angular rate                                                                                                                | rad·s <sup>-1</sup> |
| $\psi$                                | Pitch-roll phase angle                                                                                                                                                | deg                 |
| $\Phi_0$                              | Excitation (pitch/roll) amplitude                                                                                                                                     | deg                 |
| $f$                                   | Excitation frequency                                                                                                                                                  | Hz                  |
| $a_z$                                 | Heave acceleration of the vessel                                                                                                                                      | m·s <sup>-2</sup>   |
| <b>Inertias</b>                       |                                                                                                                                                                       |                     |
| $J_p$                                 | Pendulum moment of inertia about the pivot (arm + bob), $J_p = J_a + m_b L_b^2$                                                                                       | kg·m <sup>2</sup>   |
| $J_a$                                 | Moment of inertia of the pendulum arm about the pivot                                                                                                                 | kg·m <sup>2</sup>   |
| $J_G$                                 | Generator-side effective inertia referred to the pendulum shaft                                                                                                       | kg·m <sup>2</sup>   |
| $J_{eq}$                              | Total effective inertia (transmitting state), $J_{eq} = J_p + J_G$                                                                                                    | kg·m <sup>2</sup>   |
| $J_f$                                 | Flywheel moment of inertia                                                                                                                                            | kg·m <sup>2</sup>   |
| $J_r$                                 | Generator-rotor moment of inertia                                                                                                                                     | kg·m <sup>2</sup>   |
| $J_{pg}$                              | Equivalent moment of inertia of the planetary gearbox                                                                                                                 | kg·m <sup>2</sup>   |
| $J_h$                                 | Equivalent moment of inertia of the hypoid gear pair                                                                                                                  | kg·m <sup>2</sup>   |

| Symbol                                       | Description                                                                                             | Unit                                                     |
|----------------------------------------------|---------------------------------------------------------------------------------------------------------|----------------------------------------------------------|
| <b>Damping coefficients</b>                  |                                                                                                         |                                                          |
| $c_m$                                        | Mechanical damping coefficient, pendulum side                                                           | $\text{N}\cdot\text{m}\cdot\text{s}\cdot\text{rad}^{-1}$ |
| $c_{m,G}$                                    | Mechanical damping coefficient, generator side                                                          | $\text{N}\cdot\text{m}\cdot\text{s}\cdot\text{rad}^{-1}$ |
| $c_e$                                        | Electromagnetic equivalent damping referred to the pendulum shaft,<br>$c_e = k_e k_t N^2 / (R_a + R_L)$ | $\text{N}\cdot\text{m}\cdot\text{s}\cdot\text{rad}^{-1}$ |
| $c_{add}$                                    | Equivalent roll damping added to the host platform by the harvester                                     | $\text{N}\cdot\text{m}\cdot\text{s}\cdot\text{rad}^{-1}$ |
| <b>Electrical parameters</b>                 |                                                                                                         |                                                          |
| $k_e$                                        | Generator back-EMF constant                                                                             | $\text{V}\cdot\text{s}\cdot\text{rad}^{-1}$              |
| $k_t$                                        | Generator torque constant                                                                               | $\text{N}\cdot\text{m}\cdot\text{A}^{-1}$                |
| $\varepsilon$                                | Back-EMF                                                                                                | V                                                        |
| $i$                                          | Load / armature current                                                                                 | A                                                        |
| $R_a$                                        | Armature winding resistance                                                                             | $\Omega$                                                 |
| $R_L$                                        | External load resistance                                                                                | $\Omega$                                                 |
| <b>Gear ratios, geometry and mass</b>        |                                                                                                         |                                                          |
| $N_{pg}$                                     | Planetary-gearbox ratio                                                                                 | —                                                        |
| $N_h$                                        | Hypoid-gear ratio                                                                                       | —                                                        |
| $N$                                          | Total gear ratio, $N = N_{pg} N_h$                                                                      | —                                                        |
| $m_b$                                        | Pendulum-bob mass                                                                                       | kg                                                       |
| $L_b$                                        | Pivot-to-bob-centre distance                                                                            | m                                                        |
| $m$                                          | Gear module                                                                                             | mm                                                       |
| $z$                                          | Number of gear teeth                                                                                    | —                                                        |
| $T_{KN}$                                     | Rated (nominal) torque capacity of the sprag clutch                                                     | $\text{N}\cdot\text{m}$                                  |
| <b>Excitation, sea state and time scales</b> |                                                                                                         |                                                          |
| $H_s$                                        | Significant wave height                                                                                 | m                                                        |
| $T_p$                                        | Spectral peak period                                                                                    | s                                                        |
| $U$                                          | Vessel forwards speed                                                                                   | $\text{m}\cdot\text{s}^{-1}$                             |
| $S(\omega)$                                  | JONSWAP wave-energy spectral density                                                                    | $\text{m}^2\cdot\text{s}$                                |
| $\gamma$                                     | JONSWAP peak-enhancement factor                                                                         | —                                                        |
| $\alpha$                                     | JONSWAP energy-scale (Phillips) parameter                                                               | —                                                        |
| $\sigma$                                     | JONSWAP spectral width parameter                                                                        | —                                                        |
| $m_0$                                        | Zeroth spectral moment                                                                                  | $\text{m}^2$                                             |
| $\eta$                                       | Free-surface elevation                                                                                  | m                                                        |
| $a_i$                                        | Amplitude of the $i$ -th spectral component                                                             | m                                                        |
| $\delta_i$                                   | Random phase of the $i$ -th spectral component                                                          | rad                                                      |
| $k_i$                                        | Deep-water wavenumber of the $i$ -th component, $k_i = \omega_i^2/g$                                    | $\text{m}^{-1}$                                          |
| $\lambda$                                    | Scale factor relating the synthesised hull response to the wave slope                                   | —                                                        |
| $\tau_G$                                     | Generator-side (flywheel) time constant                                                                 | s                                                        |
| $\tau_{int}$                                 | Integral time scale of the load power                                                                   | s                                                        |
| <b>Derived quantities</b>                    |                                                                                                         |                                                          |
| $P_{avg}$                                    | Mean (average) output power                                                                             | W                                                        |
| $P_L$                                        | Instantaneous power delivered to the load                                                               | W                                                        |

| Symbol                   | Description                                                                                   | Unit                           |
|--------------------------|-----------------------------------------------------------------------------------------------|--------------------------------|
| $V_{RMS}$                | RMS terminal (output) voltage                                                                 | V                              |
| $\Gamma(\psi)$           | Power magnification ratio of coupled pitch-roll excitation relative to single-axis excitation | —                              |
| $\eta_{mech}$            | Mechanical transmission efficiency                                                            | —                              |
| $\eta_{elec}$            | Generator electrical efficiency                                                               | —                              |
| $\eta_{m \rightarrow e}$ | Overall motion-to-electric conversion efficiency                                              | —                              |
| $g$                      | Gravitational acceleration                                                                    | $\text{m} \cdot \text{s}^{-2}$ |

Sign conventions and state transitions: The pendulum angle,  $\theta$ , is defined as positive in the counter-clockwise direction about the +z axis. The pitch angle,  $\varphi_p$ , and the roll angle,  $\varphi_r$ , are defined as positive about the body +y-axis and +x-axis, respectively. Angular velocities are defined as positive in the direction of increasing angular displacement. The dynamic state variables are  $(\theta, \dot{\theta})$  for the pendulum and  $\Omega$  for the generator-side assembly during the freewheeling state. The sprag clutch engages during the transition from the freewheeling to the torque-transmitting state when  $|\dot{\theta}| > \Omega$  and disengages during the transition from the torque-transmitting to the freewheeling state when  $|\dot{\theta}| \leq \Omega$ . Pitch-roll coupling is introduced through the gravitational term  $m_b g L_b (\sin \varphi_p \sin \theta + \sin \varphi_r \cos \theta)$ , which couples both wave-induced motion axes to the horizontal-plane pendulum.

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