

Article

# Experimental Study on Hydrodynamic Characteristics of a Disk-Shaped Buoy Using a Large-Scale Wave Flume

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## Abstract

This study presents (i) a hybrid experimental strategy combining a large-scale wave flume and harbor basin for broad-period buoy hydrodynamic characterization, with internal consistency assessment across the facility transition, (ii) a comprehensive, uncertainty-quantified dataset for a shallow-draft disk-shaped buoy ( $D/T \approx 10$ ) including RAOs with repeatability statistics, extreme sea-state responses, and environmental load coefficients with uncertainty bounds, and (iii) new physical insights into the roll damping mechanism of such geometries without appendages. A hybrid experimental strategy was employed, integrating a large-scale wave flume (for long-period waves and currents) with a harbor basin (for short-period waves and wind), aiming to mitigate the scale effects inherent in Froude-scaled models, particularly with regard to drag force measurements. The test matrix included free decay in calm water, RAOs under regular waves, motion and mooring line tension under irregular waves, and measurements of wind and current drag coefficients. Key results indicate a natural roll period of approximately 3.0 s (prototype) with a notably high dimensionless damping ratio ( $\zeta \approx 0.14\text{--}0.15$ ), which is conducive to rapid motion attenuation. A pronounced resonance peak in the roll RAO ( $26.6^\circ/\text{m}$ ) was observed near the 3.0 s. Under an extreme sea state (prototype:  $H_s = 13.8$  m,  $T_p = 16.1$  s), the maximum roll angle and dynamic mooring line tension reached  $21.30^\circ$  and 61.56 kN, respectively, the latter being about 3.0 times the static pretension. The mean wind drag coefficient and current drag coefficient were determined as 0.76 and 0.44. This research provides a comprehensive dataset with quantified uncertainty and critical insights for the design, mooring system optimization, and operational safety assessment of such disk-shaped buoys. The hybrid testing approach demonstrated qualitative consistency across the two facilities, pending quantitative cross-validation through dedicated overlapping tests, and the measured roll damping ( $\zeta = 0.14\text{--}0.15$ , expanded uncertainty  $\pm 0.01\text{--}0.011$ ) is favorable for motion stability within the tested Reynolds-number range. Full-scale validation is recommended to confirm these findings under prototype conditions. Wind, wave, and current effects were tested separately and then comprehensively assessed.



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**Keywords:** disk-shaped buoy; large-scale wave flume; physical model experiment; hydrodynamic analysis; scale effects

## 1. Introduction

With the continuous advancement of global marine development activities, ocean buoys have become increasingly important as critical infrastructure for marine environmental monitoring, disaster warning, maritime traffic safety, and marine resource exploitation [1]. Ocean buoys are capable of long-term, continuous, fixed-point, all-weather operations in harsh marine environments, providing essential data support for marine scientific research, resource development, and environmental protection. The present study focuses on a newly developed shallow-draft disc-shaped buoy (beam-to-draft ratio  $D/T \approx 10$ ) designed as a multi-purpose ocean observation platform under the National Key Research and Development Program of China. While disc-shaped hulls have been employed in existing ocean observation systems (e.g., NOAA NDBC 3-m wave-measurement buoys), the present work makes three specific contributions to the literature: (i) the first systematic application of a hybrid large-scale flume/harbor basin testing strategy to achieve broad-period (2.5–15 s) hydrodynamic characterization of a buoy, with internal consistency assessment across the facility transition; (ii) the first comprehensive, uncertainty-quantified experimental dataset for a shallow-draft disc buoy with  $D/T \approx 10$  at the upper end of the range for operational disc buoys, including RAOs with repeatability statistics, damping ratios, drag coefficients, and mooring response with expanded uncertainty bounds; and (iii) quantified physical evidence that high intrinsic roll damping ( $\zeta \approx 0.14$ – $0.15$ , expanded uncertainty  $\pm 0.01$ – $0.011$ ) can be achieved through hull-form optimization without appendages, with estimated radiation-damping contribution of 60–70%. These contributions emphasize the methodological and dataset novelty rather than claiming a universally novel hull form. According to their application scenarios and functional requirements, ocean buoys can be classified into various types, including meteorological observation buoys, water quality monitoring buoys, wave measurement buoys, navigation buoys, and auxiliary buoys for marine energy development [2]. However, when operating under complex ocean conditions (combined wind, wave, and current actions), the motion responses and load characteristics of buoys exhibit strong complexity and nonlinearity, which imposes increasingly stringent requirements on buoy structural design, mooring system optimization, and operational safety. In recent years, wave energy marine buoys, as an emerging platform for ocean observation and energy utilization, have received widespread attention [3]. This study focuses on a newly developed 10-m disk-shaped buoy developed under the National Key Research and Development Program of China. The buoy is designed as a multi-purpose ocean observation platform for marine meteorological monitoring (air temperature, pressure, humidity, wind), sea state measurement (wave height, period, direction), and communication/data relay. Its shallow-draft disc hull ( $D/T \approx 10$ ) is optimized for high stability and large payload-to-displacement ratio, enabling long-term, fixed-point (Eulerian) operations in coastal to deep-water environments. The design balances two seemingly competing requirements: (i) sensor platform stability—minimizing roll and pitch amplitudes for accurate meteorological and optical measurements—achieved through the high intrinsic roll damping ( $\zeta \approx 0.14$ – $0.15$ )—and (ii) sufficient wave-following capability for reliable sea-state measurement, enabled by the large waterplane area ( $A_{wp} \approx 78.5 \text{ m}^2$ ) and shallow draft that provide strong hydrostatic restoring in heave. The single-point mooring configuration is adopted to provide  $360^\circ$  weather-vaning capability, ensuring the buoy always faces the incident waves and wind with its most favorable orientation.

The foregoing design features—particularly the extreme shallow-draft geometry—distinguish the specific design parameters and experimental dataset of the present study from prior literature. While disc-shaped hulls have been employed in existing ocean observation systems (e.g., NOAA NDBC 3-m disc buoys, SEAWATCH Wavescan), the present buoy is characterized by three specific aspects that motivate the systematic ex-

perimental investigation reported herein. First, its beam-to-draft ratio ( $D/T \approx 10$ ) at the upper end of the range for operational discus-type buoys, and the resulting hydrodynamic consequence—high intrinsic roll damping through radiation-dominated energy dissipation—has not been systematically quantified with uncertainty bounds in the literature. Second, the 1:10 scale testing, while common for buoy models of this class, is here combined with a large-scale wave flume (456 m long, 12 m deep) that enables long-period wave and current testing beyond the capability of conventional harbor basins, and the hybrid approach (flume + basin) provides broad-period coverage (2.5–15 s) with internal consistency assessment. Third, the quantified damping mechanism ( $\zeta \approx 0.14$ – $0.15$ , with expanded uncertainty  $\pm 0.01$ – $0.011$  and estimated 60–70% radiation contribution) provides directly applicable design parameters for similar moored observation platforms. These contributions emphasize the methodological and dataset novelty rather than claiming a universally novel hull form.

The hydrodynamic analysis of floating bodies—encompassing ships, offshore platforms, wave energy converters, and marine instruments—shares fundamental theoretical foundations rooted in potential flow theory, radiation–diffraction theory, and Morison-type empirical formulations [4,5]. For small-amplitude motions in the frequency domain, the equation of motion can be expressed as  $(-\omega^2(M + A(\omega)) + i\omega B(\omega) + C)X(\omega) = F(\omega)$ , where  $A(\omega)$  and  $B(\omega)$  are frequency-dependent added mass and damping matrices,  $C$  is the hydrostatic restoring matrix, and  $F(\omega)$  is the wave excitation force [6]. This framework has been extensively validated for ships [7] and offshore structures [5]. The motion response of floating structures to irregular waves is typically characterized by Response Amplitude Operators (RAOs), which describe the amplitude and phase of each motion component per unit wave amplitude, while damping identification through free-decay tests remains a standard procedure across ship and offshore engineering [6]. For disc-shaped buoys with large beam-to-draft ratios ( $D/T \gg 1$ ), the radiation damping  $B(\omega)$  in roll is significantly enhanced due to efficient wave generation at the waterline, while viscous damping from bilge keels or appendages is minimal [8,9]. Recent comprehensive reviews on flexible fluid–structure interactions in ocean environments [10] highlight the continuing need for high-quality experimental datasets to validate numerical models, particularly for unconventional geometries where potential-flow assumptions may be challenged by viscous and nonlinear effects, and emphasize that establishing standardized benchmarks for rigorous model validation remains a key challenge in the field.

The hydrodynamic characteristics of buoys directly govern their operational stability and measurement accuracy. The roll, pitch, and heave motion responses determine the working precision and reliability of onboard equipment (such as radar, communication antennas, and optical sensors), whereas the dynamic tension of the mooring system relates to the survivability of the buoy in extreme sea states and the safety of long-term operation [11]. Roll damping is a critical parameter affecting the motion stability of floating bodies and the dynamic loads on mooring systems; radiation damping theory provides an essential foundation for analyzing floating body motion responses [4]. High damping characteristics are conducive to suppressing resonant responses [7,12]. Consequently, in-depth investigation of buoy hydrodynamic characteristics, together with accurate acquisition of motion response patterns, damping characteristics, and environmental load coefficients, is of considerable theoretical significance and engineering value for buoy optimization design, mooring system selection, and safe operation assessment.

Currently, research on buoy hydrodynamic characteristics has made considerable progress, with research methods mainly including theoretical analysis, numerical simulation, physical model experiments, and field observations [1]. In terms of theoretical analysis, Duggal and Ryu et al. [13,14] employed coupled nonlinear time-domain methods to predict

the motion response of deepwater buoys; Le Cunff et al. [15] derived frequency-domain calculation methods, providing a theoretical basis for rapid assessment of buoy dynamic response; Salem et al. [16] used various linearization methods to solve the equations of motion, effectively addressing the nonlinear damping issues of mooring systems. Numerical methods have demonstrated favorable accuracy and efficiency in predicting the motion response of buoys in regular waves [17].

In terms of numerical simulation, commercial software such as AQWA, Fluent, RecurDyn, Orcaflex, and SESAM have been widely applied. Lu et al. [18] and Zhu et al. [19] used AQWA software for frequency-domain and time-domain analysis to study the influence of different mooring systems on buoy hydrodynamic performance; Chang et al. [20] utilized RecurDyn software to model the buoy deployment process, providing theoretical support for buoy design and optimization; Zheng et al. [3] established a deep-sea single-point meteorological buoy model using Orcaflex software to investigate the dynamic response characteristics of the system to sea state parameters and structural parameters. In addition, Li et al. [21] conducted detailed analyses of the six-degree-of-freedom motion response of large marine data buoys under different sea states using a DES-VOF coupled method, finding deviations of less than 10% compared with experimental results, which validates the reliability of numerical simulation in buoy hydrodynamic analysis.

In terms of physical model experiments, researchers simulate marine environments in wave basins or flumes using scaled buoy models. Hegde et al. [22] investigated spar-buoy platforms with heave plates through wave basin experiments and numerical simulations, finding that configurations with plates near the free surface reduced peak heave response by approximately 75% and pitch response by around 30%. Zhu et al. [23] experimentally examined the motion suppression effects of heave plates on cylindrical floating structures, demonstrating that heave plates significantly increase added mass and reduce peak heave RAO by about 40%. Capobianco et al. [24] conducted wave basin tests on cylindrical buoys to assess the influence of different disc configurations on wave measurement accuracy. Although heave plates are extensively employed for motion suppression of floating bodies [22,23], disc-shaped buoys themselves can generate pronounced radiation damping effects through their distinctive geometric characteristics, which differs from the added damping mechanism of heave plates. Investigations into the hydrodynamic characteristics of flat plate structures provide valuable reference for understanding the radiation damping mechanism of disc-shaped buoys [8]. Domestically, the research team at the National Ocean Technology Center validated the stability of anchor system designs [25]; Cao et al. [26] verified the validity of numerical simulation results; and Zhang et al. [27] corrected the damping matrix in simulation models through model experiments. Regarding field observations of buoy hydrodynamic characteristics, Eriksson et al. [28,29] from Uppsala University, Sweden, conducted full-scale experiments on column-type moored buoys at a wave energy research test site, confirming the accuracy of computational models based on potential flow theory. Yuan et al. [30] systematically investigated the hydrodynamic performance of an OWC wave energy converter in a wave flume, demonstrating that the spring-like effect of air compressibility significantly influences the energy capture efficiency, with their experimental methodology providing valuable reference for large-scale buoy hydrodynamic tests.

Despite the fruitful achievements of existing research, some shortcomings remain. First, traditional small-scale model experiments (typically with scales of 1:50–1:100) are limited by Froude similarity criteria and cannot simultaneously satisfy Reynolds number similarity, resulting in significant scale effects on viscosity-related drag coefficients and damping characteristics, which affects the accuracy of experimental results [31,32]. Second, numerical simulation methods still face challenges when dealing with strongly nonlinear

problems (such as large-amplitude motion, vortex shedding, flow separation, etc.), and model validation requires high-quality experimental data support [33]. Furthermore, field measurements are costly, time-consuming, and subject to uncontrollable sea states, making it difficult to systematically obtain complete hydrodynamic characteristics of buoys under various sea conditions [34].

Large-scale wave flume experiments serve as an important bridge connecting small-scale model experiments with field measurements, offering unique advantages in reducing scale effects and improving experimental accuracy. In recent years, significant progress has been made in the construction of large-scale wave flumes worldwide, such as the large-scale wave flume at Tianjin Research Institute for Water Transport Engineering (TIWTE) in China (456 m long, 5 m wide, 12 m deep), the Delta Flume at Deltares in the Netherlands (300 m long, 5 m wide, 9.5 m deep), and the Large Wave Flume at Leibniz University Hannover in Germany (310 m long, 5 m wide, 7 m deep) [31,35]. These large-scale testing facilities enable experiments with model scales of 1:10 or even larger, significantly increasing the model Reynolds number and bringing the flow into or near the self-similarity region, thereby effectively reducing the impact of scale effects on drag coefficients and damping characteristics [9,36]. To the best of the authors' knowledge, no systematic study has been reported on the hydrodynamic characterization of disk-shaped buoys using large-scale wave flume facilities, particularly under combined wind–wave–current conditions. This represents a significant knowledge gap, as the scale effects inherent in small-scale tests (1:50–1:100) may lead to substantial underestimation of damping coefficients and drag forces, thereby compromising the reliability of design parameters for full-scale applications. Specifically, the large beam-to-draft ratio ( $D/T \approx 10$ ) and shallow-draft geometry of disk-shaped buoys render their roll damping characteristics particularly sensitive to Reynolds number scaling, whereas the blunt-body flow separation pattern around the disc hull introduces scale-dependent nonlinearities in drag force measurements that are difficult to capture in conventional small-scale wave basins. Furthermore, the coupled wind–wave–current excitation conditions, which represent the most critical operational and survival scenarios for such buoys, have not been systematically examined in large-scale experimental settings. Consequently, there is an urgent need for systematic experimental investigations utilizing large-scale wave flume facilities to provide reliable hydrodynamic datasets for this class of buoy systems.

In view of this, this study employs a hybrid experimental method combining a large-scale wave flume (for long-period waves and current) with a harbor basin (for short-period waves and wind) to systematically investigate the motion response and mooring performance of a newly developed disk-shaped buoy (geometric scale 1:10) under combined wind, wave, and current actions. This method aims to reduce the inherent scale effects of Froude-scaled models, particularly concerning drag force measurements, to obtain more accurate buoy hydrodynamic characteristic parameters. The research contents include: (1) free decay tests in calm water to identify the natural period and damping characteristics of the buoy; (2) regular wave RAO tests to obtain the buoy motion response amplitude operators; (3) irregular wave tests to evaluate the motion and cable tension response under moderate and extreme sea states; and (4) wind/current drag coefficient measurements to provide key parameters for mooring system design.

The contributions of this study are threefold: First, it demonstrates the feasibility of a hybrid testing strategy combining a large-scale wave flume and a harbor basin to achieve broad-period coverage for buoy hydrodynamic characterization, with internal consistency verified across the facility transition. Second, it reveals that a shallow-draft disc hull with  $D/T \approx 10$  can achieve high roll damping ( $\zeta \approx 0.14$ – $0.15$ , expanded uncertainty  $\pm 0.01$ – $0.011$ ) through intrinsic radiation damping, without reliance on appendages. Third, it provides a

quantified comprehensive dataset—including RAOs with repeatability statistics, extreme sea state responses with dynamic amplification factors, and environmental load coefficients with uncertainty bounds—offering directly applicable design parameters for similar moored observation platforms.

It should be noted that simultaneous wind–wave–current experiments were not performed in this study. Wind, wave, and current effects were tested separately in different facilities and then comprehensively assessed. The comprehensive assessment of separate test results implicitly relies on a linear superposition assumption, wherein the total response is approximated as the sum of independent wind, wave, and current contributions. This approach is common in preliminary design-stage analysis of moored floating structures, where coupled multi-environmental basin testing is often unavailable due to facility limitations. However, it entails recognized physical limitations. Specifically, nonlinear drag forces (quadratic velocity dependence), wave–current interaction effects (Doppler-shifted wave frequencies and modified current profiles), and coupling between buoy motion and environmental loads (e.g., relative velocity effects on drag) are not captured when effects are tested in isolation. Consequently, the combined response under simultaneous loading may exceed the linearly superposed estimate, particularly for extreme sea states where nonlinearities are most pronounced. The present results should therefore be applied as conservative preliminary design parameters, with full-scale validation or coupled numerical simulation recommended for final mooring system verification.

## 2. Methods

### 2.1. Model Design

#### 2.1.1. Similarity Criteria

The objective of the present experiment is to simulate the dynamic responses of a buoy subjected to combined wind, wave, and current actions. Therefore, the gravity similarity criterion, i.e., Froude similarity, was adopted as the primary scaling law. In addition, geometric scaling of the mooring chain cross-section was adopted, using the same material (stainless steel) as the prototype. This yields an axial stiffness scaling of  $EA \propto \lambda^2$ , which departs from the theoretical  $\lambda^3$  requirement of strict elastic similarity under Froude scaling. The implications of this practical compromise are discussed in Section 2.2.4 (3). All experimental measurements were conducted at model scale (geometric scale  $\lambda = 1:10$ ) and converted to prototype scale using Froude similarity for reporting, unless explicitly noted. The scaling relationships are:

Geometric scale:

$$\lambda = \frac{L_r}{L_m} = 10 \quad (1)$$

Wave height scale:

$$\lambda_H = \lambda = 10 \quad (2)$$

Period (time) scale:

$$\lambda_T = \lambda^{1/2} = \sqrt{10} \approx 3.162 \quad (3)$$

Velocity scale:

$$\lambda_V = \lambda^{1/2} = \sqrt{10} \approx 3.162 \quad (4)$$

Force scale:

$$\lambda_F = \lambda^3 = 1000 \quad (5)$$

Specifically, the model design satisfies the following similarity requirements:

Geometric similarity: Strict linear scaling was maintained between the model buoy, mooring lines, and the prototype.

**Static similarity:** A ballast adjustment method was employed, in which ballast blocks were placed at appropriate locations inside the model to ensure that the total weight and center of gravity under the corresponding loading conditions satisfy similarity relationships with the prototype.

**Dynamic similarity:** The principal dynamic parameters of the buoy model, including mass moments of inertia as well as roll and pitch natural periods, were designed to conform to similarity requirements.

**Mooring stiffness scaling:** The model mooring chain uses the same material (stainless steel) as the prototype, with cross-sectional area scaled geometrically ( $EA \propto \lambda^2$ ). This approximates the prototype force-deformation relationship for static catenary behavior, though it does not satisfy strict elastic similarity (which requires  $EA \propto \lambda^3$ ).

**Note:** In this study, all reported periods, velocities, wave heights, forces, and response amplitudes are prototype-scale values (subscript p), converted from model-scale measurements (subscript m) using the scaling laws above. Model-scale values are only explicitly stated when necessary for experimental methodology discussion.

### 2.1.2. Test Model

The buoy model was fabricated using steel and engineering plastics and manufactured in strict accordance with the geometric similarity criterion. During the fabrication process, ballast adjustment was applied to ensure that key physical parameters, including total mass, center of gravity, and mass moments of inertia, satisfied the similarity requirements.

The discrepancies between the fabricated model and the corresponding theoretical values were well controlled, with the mass error of 0%, the error in the vertical position of the center of gravity of 0.9%, and the errors in the mass moments of inertia of 0.3% (roll and pitch) and 1.9% (yaw), as summarized in Table 1.

**Table 1.** Main parameters of the buoy.

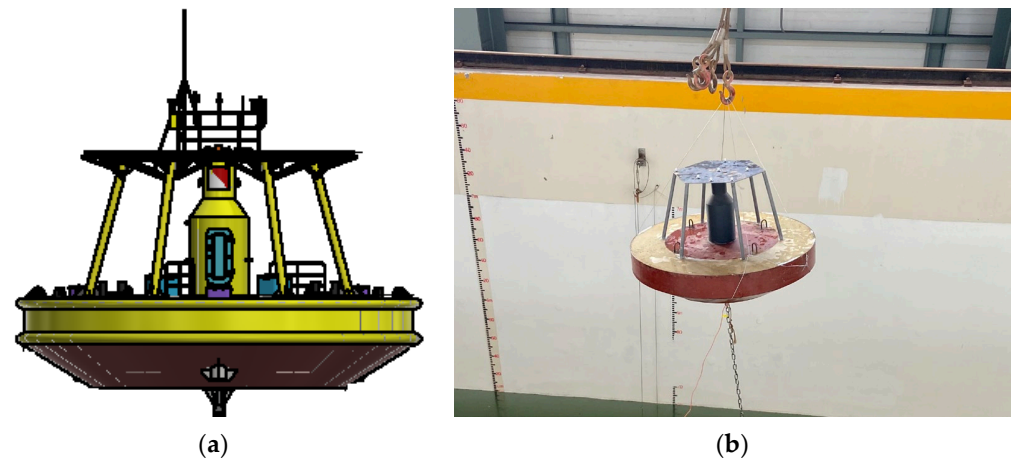
| Parameter                                    | Prototype           | Theoretical Model | Actual Model | Error (%) |
|----------------------------------------------|---------------------|-------------------|--------------|-----------|
| Disc diameter (m)                            | 10                  | 1                 | 1            | 0         |
| Depth (m)                                    | 6                   | 0.6               | 0.6          | 0         |
| Draft (m)                                    | 1.035               | 0.104             | 0.106        | 1.9       |
| Displacement (kg)                            | 55,700              | 55.7              | 55.7         | 0         |
| Vertical center of gravity (m)               | 1.137               | 0.114             | 0.115        | 0.9       |
| Metacentric height (m)                       | 8.10                | 0.81              | -            | -         |
| Waterplane area (m <sup>2</sup> )            | 78.54               | 0.7854            | 0.7854       | 0         |
| Roll moment of inertia (kg·m <sup>2</sup> )  | $3.777 \times 10^5$ | 3.777             | 3.767        | 0.3       |
| Pitch moment of inertia (kg·m <sup>2</sup> ) | $3.777 \times 10^5$ | 3.777             | 3.767        | 0.3       |
| Yaw moment of inertia (kg·m <sup>2</sup> )   | $5.091 \times 10^5$ | 5.091             | 4.992        | 1.9       |

Note: Waterplane area and GM calculated from hydrostatics; not measured on physical model.  $GM = I_{wp}/\nabla + KB - KG$ , where  $I_{wp} = \pi D^4/64$ ;  $\nabla = \Delta/\rho$ ;  $KB = (h/4) \cdot (D_1^2 + 2D_1D_2 + 3D_2^2)/(D_1^2 + D_1D_2 + D_2^2)$ .

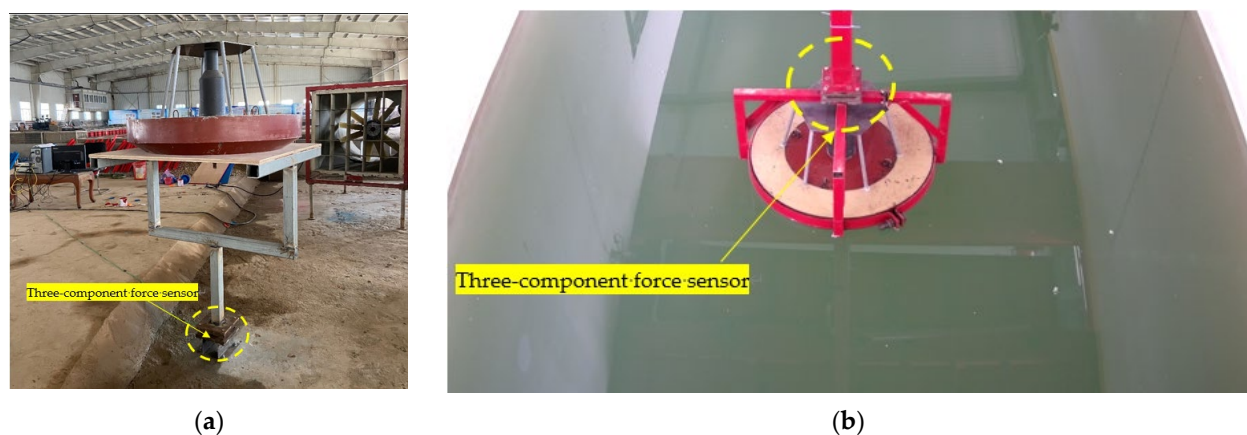
The hull geometry consists of a cylindrical upper section (height = 0.5 m, prototype = 5 m) and a conical bottom section (slope 1:3, half-angle 45°) connecting the cylinder to the flat base. The total depth is 6 m with a draft of 1.035 m. At this draft, the underwater geometry is a frustum of a cone (truncated cone) with height  $h = 1.035$  m, top diameter  $D_1 = 10$  m (at the waterline), and bottom diameter  $D_2 = 6.5$  m (at the base plane). The waterplane area is  $A_{wp} = 78.54$  m<sup>2</sup>, and the metacentric height is  $GM = 8.10$  m (see Table 1 Note for hydrostatic calculation details). The center of buoyancy is  $KB = 0.445$  m. The large  $GM/D \approx 0.81$  is characteristic of shallow-draft disc hulls and is consistent with the measured natural roll period of 3.05 s (Section 3.1).

The above-water superstructure includes a tripod mast, meteorological sensors (anemometer, temperature/humidity sensor, barometer), communication antennas (VHF and satellite), solar panels, and a navigation light. These components are arranged symmet-

rically around the central mast to minimize asymmetric wind loading. The total projected wind area of 28.0303 m<sup>2</sup> at prototype comprises the hull above-water profile (~65%) and the superstructures (~35%). The model superstructure was simplified to preserve the dominant aerodynamic features while maintaining geometric similarity. The model geometry and superstructure arrangement are shown in Figures 1 and 2.



**Figure 1.** Buoy model geometry and superstructure arrangement: (a) three-dimensional rendering of the prototype buoy showing the disc hull, conical bottom, tripod mast, meteorological sensors, communication antennas, solar panels, and navigation light; (b) photograph of the 1:10 scale physical model prior to testing, showing the simplified superstructure and single-point mooring chain attachment.



**Figure 2.** Photographs of environmental load test setups: (a) wind drag test in the harbor basin: single axial fan upstream of the rigidly fixed buoy model with the three-component force sensor at the base; (b) current drag test in the large-scale wave flume: model fixed to the flume floor with the Vectrino+ ADV positioned upstream. Note: The wind test was conducted with the model in air above the water surface to isolate pure aerodynamic drag; no simultaneous wind-and-wave tests were performed.

A single-point mooring configuration was adopted in the experiment. The model mooring chain was fabricated from studless anchor chain (chain links) of stainless steel. In addition to satisfying geometric similarity, particular emphasis was placed on matching the mass per unit length and stiffness characteristics of the prototype chain.

By adjusting the configuration of the model mooring chain, the mass per unit length was scaled to satisfy the gravity similarity requirement, while the elastic deformation behavior was ensured to be similar to that of the prototype. The main parameters of the mooring chain model are listed in Table 2.

**Table 2.** Main parameters of the mooring chain model.

| Parameter                                | Prototype | Theoretical Model | Actual Model | Error (%) |
|------------------------------------------|-----------|-------------------|--------------|-----------|
| Chain diameter (mm)                      | 52        | 5.2               | 6            | 15.4      |
| Weight per unit length (kg/m)            | 61.8      | 0.618             | 0.65         | 5.2       |
| Total chain length (m)                   | 150       | 15                | 15           | 0         |
| Water depth (m)                          | 50        | 5                 | 5            | 0         |
| Water-depth-to-chain-length ratio        | 1:3       | 1:3               | 1:3          | 0         |
| Static pretension $T_0$ (kN/N)           | 20.5      | 20.5              | 20.5         | 0         |
| Horizontal projection at equilibrium (m) | 12.8      | 1.28              | 1.28         | 0         |
| Touchdown point from anchor (m)          | ~8.5      | ~0.85             | ~0.85        | -         |
| Grounded portion (m)                     | ~22       | ~2.2              | ~2.2         | -         |
| Suspended catenary length (m)            | ~128      | ~12.8             | ~12.8        | -         |

Note: Studless anchor chain (chain links), not steel wire rope.

The mooring system was configured with a water-depth-to-chain-length ratio of 1:3, following standard offshore practice. The static pretension  $T_0$  was set at 20.5 kN/20.5 N (model), ensuring positive tension throughout the motion cycle and moderate horizontal restoring stiffness. At equilibrium, the horizontal projection of the mooring line is 12.8 m/1.28 m (model), with the touchdown point located approximately 8.5 m/0.85 m (model) from the anchor. The grounded portion is approximately 22 m/2.2 m (model), providing sufficient length to prevent anchor uplift. The suspended catenary length is approximately 128 m/12.8 m (model).

Static restoring characteristics and equivalent stiffness. The static restoring behavior of the single-point catenary mooring system is governed by geometric stiffness arising from catenary shape change, rather than axial elasticity of the chain. The equivalent horizontal stiffness for small displacements around the equilibrium position is estimated using the established engineering approximation for single-point catenary systems [6]:

$$k_{eq} \approx \frac{T_0}{X_0} \cdot \frac{1}{1 + \frac{2h}{X_0}} \quad (6)$$

where  $T_0$  is the static pretension,  $X_0$  is the horizontal projection at equilibrium, and  $h$  is the water depth. At prototype scale, this yields  $k_{eq,p} \approx 182$  N/m; at model scale,  $k_{eq,m} \approx 1.82$  N/m following Froude scaling ( $k \propto \lambda^2$ ). This geometric stiffness is three to four orders of magnitude smaller than the chain axial elastic stiffness ( $k_{elastic} \sim 10^3$  kN/m, as specified by the design unit), confirming that the mooring restoring behavior is dominated by catenary geometry rather than material elasticity. The linearized stiffness is valid for horizontal offsets up to approximately  $\pm 15\%$  of the equilibrium offset ( $\pm 2$  m prototype,  $\pm 0.2$  m model), beyond which the catenary nonlinearity becomes significant. The static equilibrium configuration was verified during model setup by checking the horizontal projection (1.28 m model scale) and the grounded portion length (2.2 m) against the theoretical catenary relationship.

The mooring system parameters listed in Table 2 were specified by the buoy design unit according to standard offshore catenary design practice. The present study, as a physical model test program, adopted these specifications as boundary conditions without developing an independent mathematical model for the mooring system. The static equilibrium configuration was verified during model setup: the horizontal projection at equilibrium (12.8 m at prototype) and the grounded portion length (~22 m) were checked against the classical catenary relationship to ensure consistency with the design intent. Detailed coupled buoy-mooring dynamic analysis is beyond the scope of the present experimental investigation.

Mooring system similarity verification: The actual chain diameter (6 mm) exceeds the theoretical scaled value (5.2 mm) by 15.4%, resulting in a 5.2% increase in submerged weight per unit length (Table 2). However, the total chain length (15 m) and water depth (5 m) remain exactly scaled, ensuring correct catenary geometry.

### 2.1.3. Simulation of Wave Conditions

The experiments were conducted in a large-scale wave flume and a harbor and coastal engineering basin, where a variety of wave conditions were simulated. For regular wave tests, the wave period ranged from 2.5 s to 15 s, while the wave height was scaled according to the geometric similarity ratio.

To more realistically represent marine environments, irregular wave conditions were also simulated. An improved JONSWAP spectrum proposed by Goda was adopted, with peak periods ranging from 8.5 s to 16.1 s, covering both moderate and extreme sea-state conditions.

All wave parameters were determined based on similarity criteria. Wave calibration and quality control procedures, including wavemaker control, reflection control, and wave-field uniformity verification, are detailed in Section 2.2.1.

For irregular wave tests, the JONSWAP spectrum was employed with a peak enhancement factor of  $\gamma = 3.3$ . The wave train length exceeded 100 waves to ensure sufficient statistical convergence.

### 2.1.4. Simulation of Wind Conditions

Wind condition tests were conducted to simulate wind speeds of 10.44, 13.41, 21.25, 25.77, and 33.20 m/s (corresponding to 6, 10, 16, 19.5, and 25 knots), which scale to wind speeds of approximately 3.30, 4.24, 6.72, 8.15, and 10.50 m/s by the velocity scale of  $\lambda_V = \lambda^{1/2} = \sqrt{10} \approx 3.162$ .

The wind load experiments were performed in the harbor basin using a single axial fan mounted on a movable frame, with variable-frequency drive control to achieve stepless speed regulation. The fan was positioned upstream of the model at a distance of approximately 3–5 m, depending on the required wind direction. Wind speeds were measured by hot-wire anemometers (accuracy  $\pm 2\%$ ) installed upstream of the model at the approximate model height.

Wind-field characterization and limitations. The single-fan setup generates a jet-like flow field in the open-basin environment. The spatial uniformity of the mean wind speed across the model frontal area was within  $\pm 5\%$  of the centerline value. The turbulence intensity at the model location was measured at approximately 8–12%, which is higher than typical atmospheric boundary layer values but representative of open-basin fan-generated flows. Blockage effects are negligible due to the absence of enclosing walls.

Limitations: Compared to a closed-section wind tunnel or atmospheric boundary layer simulation, this single-fan open-basin setup has the following limitations: (i) the flow is jet-like rather than a fully developed turbulent boundary layer, and the mean velocity profile does not match the logarithmic or power-law profiles typical of atmospheric wind; (ii) the turbulence intensity is higher than atmospheric values, and the turbulence spectra may not match atmospheric characteristics; and (iii) the short fetch distance and lack of flow conditioning result in less uniform flow quality than a wind tunnel. These limitations are acknowledged, and the measured wind drag coefficients are presented as valid within the present experimental conditions.

### 2.1.5. Simulation of Current Conditions

Current tests were conducted to simulate current conditions of 2, 4, and 6 knots (corresponding to 1.03, 2.06, and 3.09 m/s), which scale to current velocities of 0.33, 0.65, and 0.98 m/s by the velocity scale  $\lambda_V = \lambda^{1/2} = \sqrt{10} \approx 3.162$ .

The simulation of current-induced loads was performed in the large-scale wave flume. A uniform current field was generated by the flume circulation system, consisting of a recirculating flow driven by a variable-speed axial pump with guide vanes to ensure uniform distribution across the flume cross-section. Three representative conditions were tested.

Current velocity was measured using a Vectrino+ acoustic Doppler velocimeter (ADV) with a sampling frequency of 100 Hz and accuracy of  $\pm 1\% \pm 0.5$  cm/s, calibrated at the buoy model location. Flow-field characterization at the model location confirmed: (i) vertical and transverse velocity profiles with variation within  $\pm 3\%$  of the centerline mean velocity; (ii) turbulence intensity of approximately 5%; and (iii) model positioned at the flume centerline with distance to side walls exceeding 2D, resulting in negligible wall effects and a blockage ratio of approximately 4%.

Limitation: Systematic boundary layer profiling or detailed turbulence spectrum analysis was not performed. The reported current drag coefficients are based on the mean velocity at the model center height, and the turbulence characteristics represent flume-generated flow rather than oceanic conditions.

## 2.2. Testing Equipment and Methods

### 2.2.1. Test Basins

In this study, wind, wave, and current effects were tested separately in different facilities: long-period waves and currents were generated in the large-scale wave flume, while short-period waves and wind loads were tested in the harbor basin. The results were then comprehensively assessed to evaluate the buoy's hydrodynamic performance under multi-environmental loading conditions. The separate testing strategy precludes direct measurement of nonlinear interaction effects, including: (i) wave-current interaction modifying the incident wave kinematics and dynamic pressure field; (ii) velocity-squared drag nonlinearity under combined wind-plus-current relative flow; (iii) mooring line dynamics altered by simultaneous wave-frequency excitation and mean drift from wind/current; and (iv) dynamic surge response under combined wave and mean drift loading, which requires simultaneous wave and wind/current generation. These effects are acknowledged as unquantified uncertainties in the present experimental framework.

This study utilized two internationally advanced test basins in a comprehensive manner. The primary rationale for this hybrid approach was to ensure that the RAO tests remained within the linear range, where the ratio of wave height to wavelength for regular waves generated by the wavemaker is approximately 1/35 to 1/50. Considering the wave-making capabilities of the actual basins and the flume, regular wave tests for periods of 2.5 s to 5.2 s were conducted in the harbor and coastal engineering basin, while those for periods of 5.9 s to 15 s were performed in the large-scale wave flume.

#### (1) Large-scale Wave Flume

Long-period wave (5.9~15 s) and current load tests were carried out in the large-scale wave flume at the Tianjin Research Institute for Water Transport Engineering, Ministry of Transport. This flume is 5 m wide, 456 m long, with a test section depth of 12 m. It is equipped with an advanced electro-servo driven wavemaker system capable of accurately simulating both regular and irregular waves. Its maximum achievable regular wave height is 3.5 m, with an effective wave period range of 2–8 s (model). Currently one of the largest and most powerful wave generation facilities in the world, it provides an ideal simulation environment for testing the motion response of the buoy under long-period

waves (5.9–15 s). An energy dissipation device is installed at the tail of the flume, effectively suppressing wave reflection and ensuring wave field quality.

Wave generation and quality control. Both facilities employ advanced wave generation systems with real-time control. The flume is equipped with an electro-servo-driven piston-type wavemaker capable of generating regular and irregular waves with a maximum height of 3.5 m and effective periods of 2–8 s (model). The basin uses three 4-m-wide absorbing-type paddle wavemakers with active reflection compensation, covering a wave-period range of 0.5–5.0 s (model).

Prior to the regular wave tests, wave generation signals were iteratively adjusted to ensure that the deviations of generated mean wave height and period from target values were within  $\pm 5\%$  at the model location. The free surface elevation was sampled at 0.02 s intervals, and more than ten complete consecutive wave cycles were recorded for each condition to ensure data stability. Three wave probes were installed at 2 m, 5 m, and 8 m upstream of the model to monitor the free surface elevation time history and wave height distribution.

Reflection control is achieved through different mechanisms in the two facilities: the flume employs a passive energy dissipation device at the downstream end, while the basin utilizes active wave absorption via the absorbing-type paddle wavemakers. The estimated reflection coefficient is less than 10% for regular waves in both facilities based on facility specifications and prior calibration studies. It is noted that systematic incident/reflected wave separation (e.g., using the three-gauge method) was not performed in this study; the wave gauges were used for monitoring purposes rather than for reflection analysis. Consequently, the reported wave parameters may contain a minor reflected-wave contribution, which is estimated to be within the facility's absorption performance limits ( $<10\%$ ).

The spatial uniformity of the wave field at the model location was verified by measuring wave heights across the transverse direction; the variation in significant wave height was within  $\pm 3\%$  of the mean value. For irregular waves, the measured JONSWAP spectrum parameters (peak period deviation  $< \pm 2\%$ , significant wave height deviation  $< \pm 5\%$ ) confirm satisfactory wave-field quality at the model location.

## (2) Harbor and Coastal Engineering Basin

Short-period wave (2.5–5.2 s) and wind load tests were carried out in a harbor and coastal engineering basin measuring 54.0 m in length, 13.2 m in width, and 1.0 m in depth. The basin is equipped with three 4-m-wide absorbing-type irregular wavemakers, with a maximum operating water depth of 0.7 m, a wave-height range of 0–35 cm, and a wave-period range of 0.5–5.0 s (model), enabling accurate simulation of irregular wave trains such as those defined by the JONSWAP spectrum. In addition, the basin is fitted with a single axial fan mounted on a movable frame for wind generation. Through frequency-converter control, the fan can achieve stepless speed regulation within a model wind speed range of 0–10.0 m/s, and the frame can be moved flexibly to simulate different wind directions. The wind-field characteristics and limitations of this open-basin single-fan setup are detailed in Section 2.1.4.

Validation of the hybrid approach. It is important to note that the two facilities do not share an overlapping wave-period range: the harbor basin covers periods of 2.5–5.2 s, while the large-scale flume covers 5.9–15 s. Consequently, a direct cross-facility comparison at identical periods is not feasible. The hybrid approach is presented as a practical strategy for achieving broad-period coverage under linear wave conditions. The internal consistency of RAO trends across the two facilities is assessed in Section 3.2.

Given these considerations, the hybrid approach is presented as a practical strategy for achieving broad-period coverage under linear wave conditions, rather than a rigorously

cross-validated method. The internal consistency of the RAO trends across the facility transition is discussed in Section 3.2.

### 2.2.2. Measurement Instruments

A variety of high-precision sensors and a synchronous data acquisition system were employed in the tests to perform synchronized, real-time measurements of wave surface elevation, buoy motion, forces, and environmental parameters. The main instruments used are listed in Table 3.

**Table 3.** Main instruments of the experimental measurement system.

| Measured Quantity       | Instrument Name                    | Model                     | Manufacturer     | City, Country             | Range     | Accuracy       | Sampling Frequency |
|-------------------------|------------------------------------|---------------------------|------------------|---------------------------|-----------|----------------|--------------------|
| Wave Height             | Capacitance Wave Gauge             | SG2000 (Basin)            | TIWTE            | Tianjin, China            | 0–0.6 m   | ±1 mm          | 50 Hz              |
|                         | Capacitive Wave Probe              | Custom-made (Large Flume) | TIWTE            | Tianjin, China            | 0–5 m     | 1% F.S.        | 50 Hz              |
| Six-DOF Motion          | Optical Motion Capture System      | Nokov-Mars                | Nokov            | Beijing, China            | 52° × 52° | ±0.1 mm        | 100 Hz             |
| Mooring Line Force      | Submersible Tension Load Cell      | LA2-type                  | Daysensor        | Bengbu, China             | 50–100 N  | 0.05% F.S.     | 1000 Hz            |
| Flow Velocity           | Acoustic Doppler Velocimeter (ADV) | Vectrino+                 | Nortek           | Rudskogen, Norway         | 0–3 m/s   | ±1% ± 0.5 cm/s | 100 Hz             |
| Wind Speed              | Hot-Wire Anemometer                | CTV 110                   | KIMO Instruments | Montpon-Ménestrol, France | 0–10 m/s  | ±0.1 m/s       | 50 Hz              |
| Wind/Current Drag Force | Three-Component Force Sensor       | Kistler                   | Kistler          | Winterthur, Switzerland   | 0–100 N   | 0.15% F.S.     | 100 Hz             |

- (1) Wave Measurement System: Wave height measurement formed the foundation of the tests. Capacitive wave probes and capacitance wave gauges were employed in the large flume and the basin, respectively. All sensors underwent rigorous calibration prior to testing, exhibiting a linearity greater than 0.999. The wave probes were positioned upstream of the model (2 m, 5 m, and 8 m) to avoid interference with the model motion and mooring line. This arrangement is adequate for wave height and period measurement but introduces a spatial separation that complicates direct phase comparison between wave elevation and buoy motion; phase correction using the wave propagation speed would be required for rigorous phase analysis, particularly for short-period waves.
- (2) Motion Measurement System: The six-degree-of-freedom (6-DOF) motions of the buoy were measured using a Nokov-Mars series optical motion capture system via a non-contact method. This system tracks multiple infrared reflective markers fixed on the buoy model—distributed on the top surface and side surfaces to ensure complete visibility from the overhead and rear-side cameras. Through dedicated post-processing software, it directly outputs time-history data of displacement and rotation at the buoy's center of gravity. The CG motion is computed via rigid-body transformation from the measured marker positions, with an estimated accuracy of ±0.1 mm. This approach avoids the added mass effect associated with contact sensors and offers high precision and strong anti-interference capability.
- (3) Force Measurement System: Mooring line tension was measured using a high-precision, waterproofed submersible tension load cell (LA2-type, 50–100 N range, 0.05% F.S. accuracy), connected in series directly between the mooring chain and the buoy base attachment point. Environmental loads from wind and current were measured integrally using a three-component force sensor (0.15% F.S., 100 Hz) mounted on a dedicated frame at the model base. For these measurements, the buoy model was

- rigidly fixed to the frame (6-DOF constrained), ensuring that the measured force represents the pure environmental drag without motion-induced hydrodynamic effects.
- (4) Data Acquisition System: Signals from all sensors were acquired by a multi-channel synchronous data acquisition unit. The sampling frequency was set between 50 Hz and 1000 Hz depending on the characteristics of the physical quantity being measured, ensuring the capture of high-frequency dynamic responses. The entire data acquisition process was controlled by a central computer, guaranteeing strict synchronization of timestamps across all systems.

Note on visual documentation: Due to facility safety protocols, photography during active wave generation was not permitted. The sensor positions were established during setup and verified through calibration procedures. For the wind drag tests, the buoy model was rigidly fixed to a dedicated frame in the harbor basin (Figure 2a); the model was positioned above the still water surface to isolate pure aerodynamic drag, consistent with standard fixed-body drag measurement practice [5,6]. The current drag tests were conducted with the model submerged and fixed to the flume floor (Figure 2b). No simultaneous wind-and-wave tests were performed, so photographs of the buoy floating in water subjected to wind action are not available.

### 2.2.3. Test Methods

(1) Free Decay Test: Under calm water conditions, an initial displacement was applied to the buoy model and then released, allowing it to oscillate freely. The optical motion capture system was used to record the decay time-history curves of the roll and heave motions. The non-dimensional damping coefficient  $\zeta$  was determined using the logarithmic decrement method with successive same-sign peaks. For each decay record,  $n = 3\text{--}5$  cycles were analyzed to calculate  $\zeta$ , ensuring statistical robustness while minimizing amplitude-dependent nonlinearity. The repeatability was verified through three repeated tests, yielding mean  $\pm$  SD values of  $\zeta = 0.150 \pm 0.008$  (without mooring chain) and  $\zeta = 0.140 \pm 0.007$  (with mooring chain), with 95% confidence intervals of  $\pm 0.012$  and  $\pm 0.010$ , respectively. The amplitude-range sensitivity was assessed by comparing estimates from initial vs. later cycles; the difference is  $<8\%$ , confirming stability.

$$\zeta = \frac{1}{2\pi n} \ln \left( \frac{\theta_i}{\theta_{i+n}} \right) \quad (7)$$

where  $\theta_i$  and  $\theta_{i+n}$  are the amplitudes of the  $i$ -th and  $i + n$ -th same-sign peaks (e.g., consecutive positive maxima), respectively. This approach avoids bias from nonlinear effects at large amplitudes.

Amplitude-range sensitivity: Estimates from the initial 3 cycles vs. later cycles differ by  $<8\%$ , confirming the stability of the damping ratio across the decay record. Typical free-decay time histories exhibit an imposed initial displacement followed by subsequent exponential decay. The optical motion capture system recorded the decay time-history after the initial displacement was applied and the model was released. The first identifiable peak in the recorded signal is taken as the reference point ( $t = 0$ ) for the logarithmic decrement analysis.

(2) Motion RAO Measurement: Under regular wave action, the motion response of the buoy model is measured after it reaches a steady state. The Response Amplitude Operator (RAO) is calculated according to Equation (7).

$$Y_A(\omega) = \frac{y_A(\omega)}{\xi_A} \quad (8)$$

Each regular wave condition is repeated three times, and the average motion amplitude is used for calculation to eliminate random errors.

(3) Wind/Current Drag Coefficient Determination: In a uniform flow or wind field, the total drag force  $F$  acting on the buoy model is measured. The model is restrained from all motions (6-DOF fixed) to isolate the pure environmental drag force from motion-induced hydrodynamic effects. This standard approach yields static drag coefficients; the applicability to freely moving bodies is discussed in Section 3.4. Figure 2 presents photographs of the test setups: panel (a) shows the wind drag test configuration with the single axial fan upstream of the rigidly fixed model in the harbor basin, and panel (b) shows the current drag test with the submerged model fixed to the flume floor. The wind test was conducted with the model positioned above the still water surface to ensure pure aerodynamic loading without wave interaction. The wind drag coefficient  $C_D$  and current drag coefficient  $C_C$  are calculated using Equations (8) and (9), respectively.

$$F_W = \frac{1}{2} \rho C_D U^2 A \quad (9)$$

$$F_C = \frac{1}{2} \rho C_C V^2 A \quad (10)$$

where  $\rho$  denotes the fluid density ( $\rho_a = 1.2 \text{ kg/m}^3$  for air and  $\rho_w = 1.0 \times 10^3 \text{ kg/m}^3$  for seawater),  $C_D$  is the wind drag coefficient,  $C_C$  is the current drag coefficient,  $U$  is the wind speed,  $V^2$  is the current velocity, and  $A$  represents the projected area normal to the wind or current direction. The projected areas are reported at prototype scale (geometric scale  $\lambda = 10$ , area scale  $\lambda^2 = 100$ ): the wind projected area is  $28.0303 \text{ m}^2$  (model:  $0.2803 \text{ m}^2$ ), measured directly from the model's above-water frontal profile under head-on wind conditions, including the disc top, cylindrical hull, and superstructure; the current projected area is  $9.075 \text{ m}^2$  (model:  $0.09075 \text{ m}^2$ ), calculated as the underwater frontal area  $D \times \text{draft} = 10 \times 0.9075 \text{ m}^2$ .

(4) Statistical Analysis Method: For irregular wave tests, the measured motion and force time-history data are subjected to statistical analysis to derive characteristic statistical quantities, including the mean, significant value, 1/10 maximum value, and maximum value. All data analyses are processed using Python 3.9 (Python Software Foundation, Wilmington, DE, USA) with the NumPy 1.24 and SciPy 1.10 libraries to ensure the scientific rigor and accuracy of the results.

#### 2.2.4. Scale Effects and Similarity Considerations

##### (1) Gravity-Dominated Wave Response (Froude Similarity)

For wave-induced hydrodynamic responses, gravity and free-surface effects dominate. Froude similarity is therefore the primary and appropriate scaling criterion [5,6]. All wave-period-related quantities (motion periods, RAO peak periods) scale directly by  $\lambda^{1/2}$ . The reported prototype values are obtained by multiplying model measurements by  $\sqrt{10} \approx 3.162$ . Wave response is inherently Froude-scaled and Reynolds-number effects are secondary for inviscid wave-body interactions.

##### (2) Viscous-Drag Effects and Reynolds-Number Limitations

Wind and current drag coefficients, as well as roll damping, are influenced by viscous forces and boundary-layer behavior. Under the fixed geometric scale  $\lambda = 10$ , Froude similarity requires velocity scaling as  $\lambda_V = \lambda^{1/2}$  whereas satisfying Reynolds-number similarity requires scaling according to  $\lambda_V = \lambda^{-1}$ ; these two requirements are mutually contradictory. Therefore, Reynolds-number dissimilarity constitutes an inherent theoretical limitation in the present experimental framework, specifically affecting:

- ① Roll damping: viscous contribution estimated at 15–20% of total damping (Section 3.1);

- ② Wind drag coefficient  $C_D$ : measured at  $Re_m = 1.1 \times 10^5 - 3.5 \times 10^5$ ;  
 ③ Current drag coefficient  $C_C$ : measured at  $Re_m = 2.0 \times 10^5 - 6.0 \times 10^5$ .

Table 4 lists computed  $Re_m$  for each test condition, based on disc diameter  $D$  as the characteristic length. The critical Reynolds number for turbulent flow separation over bluff bodies is  $Re_{m,crit} = 1.0 \times 10^5$  [6,37]. Since all tested conditions exceed this threshold (Table 4), the drag coefficients are expected to operate in the Reynolds-number-insensitive “self-similar” regime, where  $C_D$  variation with  $Re$  is relatively small for most conditions [6]. However, Wind-1 ( $Re_m = 1.1 \times 10^5$ ) is only marginally above the critical value; therefore, the self-similar characterization should be treated with appropriate caution for low-wind conditions.

**Table 4.** Reynolds numbers for wind and current drag coefficient tests.

| Test Condition       | Prototype Velocity | Model Velocity | $Re_D$ (Model)    | $Re_{D,crit}$     | Regime                  |
|----------------------|--------------------|----------------|-------------------|-------------------|-------------------------|
| Current-1 (2 knots)  | 1.03 m/s           | 0.326 m/s      | $2.0 \times 10^5$ | $1.0 \times 10^5$ | Turbulent, self-similar |
| Current-2 (4 knots)  | 2.06 m/s           | 0.651 m/s      | $4.0 \times 10^5$ | $1.0 \times 10^5$ | Turbulent, self-similar |
| Current-3 (6 knots)  | 3.09 m/s           | 0.977 m/s      | $6.0 \times 10^5$ | $1.0 \times 10^5$ | Turbulent, self-similar |
| Wind-1 (20.28 knots) | 10.44 m/s          | 3.30 m/s       | $1.1 \times 10^5$ | $1.0 \times 10^5$ | Transitional-turbulent  |
| Wind-2 (26.06 knots) | 13.41 m/s          | 4.24 m/s       | $1.4 \times 10^5$ | $1.0 \times 10^5$ | Turbulent, self-similar |
| Wind-3 (41.31 knots) | 21.25 m/s          | 6.72 m/s       | $2.2 \times 10^5$ | $1.0 \times 10^5$ | Turbulent, self-similar |
| Wind-4 (50.10 knots) | 25.77 m/s          | 8.15 m/s       | $2.7 \times 10^5$ | $1.0 \times 10^5$ | Turbulent, self-similar |
| Wind-5 (64.54 knots) | 33.20 m/s          | 10.50 m/s      | $3.5 \times 10^5$ | $1.0 \times 10^5$ | Turbulent, self-similar |

Measures to minimize Reynolds-number effects:

① Ensuring surface roughness similarity: The surface finish of the model buoy was maintained consistent with that of the prototype to control the boundary layer transition point, thereby ensuring that the model operates in a flow regime as similar as possible to that of the prototype (turbulent flow regime).

② Increasing experimental Reynolds number: By employing a large-scale model (1:10) and selecting relatively high wind and current velocity conditions for testing, the model Reynolds number ( $Re_m$ ) was ensured to approach the self-similarity region. It is estimated that the Reynolds number range in the present tests (based on the disc diameter as the characteristic length) is approximately  $2.0 \times 10^5$  to  $6.0 \times 10^5$  (for current) and  $1.1 \times 10^5$  to  $3.5 \times 10^5$  (for wind), which has reached or exceeded the critical Reynolds number for most test conditions, resulting in a relatively stable flow separation point.

③ Uncertainty quantification (Type A only): The drag coefficient measurements were repeated three times for each test condition. The Type A standard uncertainty:  $u_A$  was determined by segmenting each 180-s time-series into 6 blocks (30 s each) and computing the standard deviation of segment-mean  $C_C$  or  $C_D$  values, divided by  $\sqrt{6}$ . The relative repeatability uncertainty  $u_{A,rel} = u_A / \overline{C_{D,steady-state}}$  ranges from 1.8% to 8.8% across the tested conditions; detailed results are presented in Section 3.4. These values represent the primary uncertainty source in the present experiments, as instrument calibration uncertainties (Type B) are estimated to be subordinate based on sensor specifications (0.15% F.S., linearity > 0.999).

(3) Mooring-System Elastic Similarity

The model mooring chain was fabricated from the same material (stainless steel) as the prototype, so  $E_m = E_p$ . With identical material, the axial stiffness scales geometrically through the cross-sectional area:  $EA_m = (A_m/A_p)E_p \times EA_p$ . For strict geometric similarity ( $d_m = d_p/\lambda$ ), this yields  $EA_m = EA_p/\lambda^2 = EA_p/100$ . It is acknowledged that this departs from the theoretical  $\lambda^3$  scaling required by strict elastic similarity under Froude scaling (where  $\lambda_E = \lambda$ ,  $\lambda_A = \lambda^2$ , giving  $\lambda_{EA} = \lambda^3$ ). However, the following practical considerations justify this approach:

① Static restoring is geometry-dominated: The static restoring behavior of the single-point catenary mooring system is governed by geometric stiffness arising from catenary shape change, rather than axial elasticity of the chain. The equivalent horizontal stiffness  $k_{elastic} \approx 182$  N/m (prototype) is three to four orders of magnitude smaller than the chain axial elastic stiffness ( $k_{elastic} \sim 10^3$  kN/m). Consequently, the static restoring curve (tension vs. horizontal offset) is insensitive to the exact axial stiffness, provided the catenary geometry is correctly scaled. The equilibrium horizontal projection at 1.28 m (model) matches the target value scaled from prototype (12.8 m).

② Dynamic stiffness: limitation acknowledged: The natural period of mooring line longitudinal oscillation  $T_{moor} = 2\pi\sqrt{(mL/EA)}$  is not correctly scaled under  $EA \propto \lambda^2$  (it would require  $EA \propto \lambda^3$  for proper scaling). This represents a known limitation of the present experimental framework. However, the mooring line oscillation period is well separated from the wave excitation range (2.5–15 s) and the buoy natural periods ( $\sim 3$  s for roll,  $\sim 110$  s for surge), so the dynamic coupling is negligible.

③ Pretension similarity: The static pretension is scaled by  $\lambda^3$  (force scale), ensuring correct initial equilibrium configuration. The prototype pretension of 20.5 kN corresponds to 20.5 N at model scale. The pretension was selected to ensure: (a) positive tension throughout the motion cycle, (b) moderate horizontal restoring stiffness, and (c) sufficient grounded length ( $\sim 22$  m at prototype,  $\sim 2.2$  m at model) to prevent anchor uplift. The water-depth-to-chain-length ratio of 1:3 is consistent with standard offshore mooring design practice.

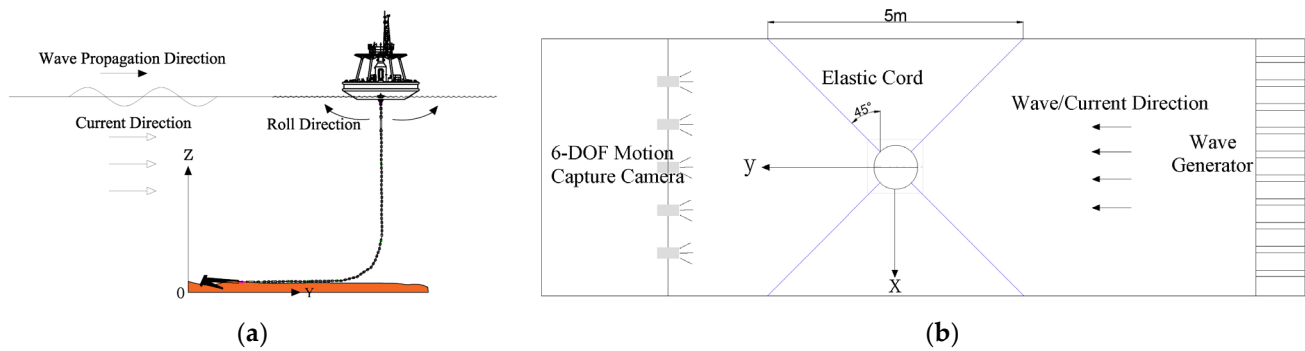
The equivalent horizontal stiffness  $k_{eq} \approx 182$  N/m (prototype) yields a coupled surge natural period  $T_{n,surge} = 2\pi\sqrt{M_{xx}/k_{eq}} \approx 110$  s (prototype), where  $M_{xx}$  is the total virtual mass in surge. This period is well separated from the wave excitation range (2.5–15 s) and the roll natural period ( $\sim 3$  s), confirming negligible dynamic coupling between surge and the wave-frequency responses reported herein. The low stiffness and long surge period are characteristic of single-point catenary systems, which provide compliant horizontal restoring with minimal impact on vertical-plane motions.

## 2.2.5. Coordinate System Definition and Experimental Setup

### (1) Coordinate System Definition

To accurately describe the motion response of the buoy in waves, a right-handed Cartesian coordinate system is established as illustrated in Figure 3. The origin O is defined at the center of the buoy's base plane in its static equilibrium position. The positive X-axis points toward the bow of the buoy (incident flow direction), the positive Y-axis is perpendicular to the X-axis and points toward the port side, and the Z-axis is directed vertically upward in accordance with the right-hand rule. The wave incidence angle  $\beta$  is defined as the angle between the wave propagation direction and the positive X-axis, with  $0^\circ$  along the positive X-axis and positive angles measured counterclockwise.

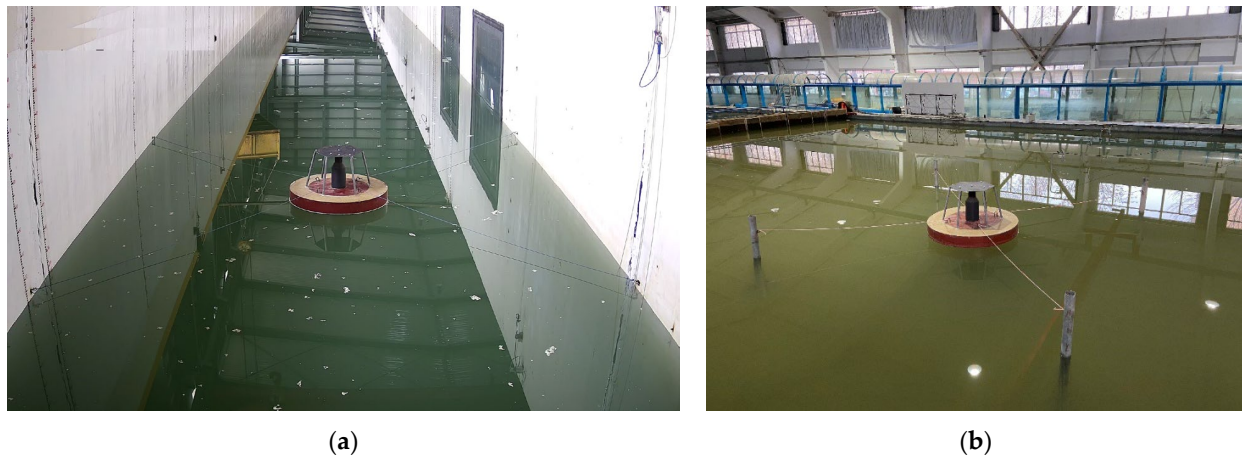
The six-degree-of-freedom (6-DOF) motions of the buoy are defined as follows: translation along the OX axis—surge; translation along the OY axis—sway; translation along the OZ axis—heave; rotation about the OX axis—roll; rotation about the OY axis—pitch; and rotation about the OZ axis—yaw.



**Figure 3.** (a,b) Schematic of the coordinate system and definition of the wave incidence angle.

## (2) Experimental Setup

A single-point mooring configuration was adopted in the experiment, with the mooring chain attachment point located at the center of the buoy bottom. The model arrangement strictly adhered to the similarity criteria, and the setups in the two test facilities are schematically illustrated in Figure 4.



**Figure 4.** Schematic of the experimental setup: (a) in the large-scale wave flume; (b) in the harbor basin. Note: The diagonal elastic cords visible in panel (b) are temporary safety restraint lines used exclusively during model deployment and retrieval to prevent accidental drift or capsizing; they were completely removed before all hydrodynamic tests and are not part of the operational mooring system.

In the large-scale wave flume, the model was positioned approximately 200 m from the wavemaker to ensure full wave development. To minimize boundary effects, the distance from the model center to each side wall was maintained at more than twice the model diameter. In the harbor and coastal engineering basin, the model was placed approximately 30 m from the wavemaker, with the distance to both side boundaries exceeding 6 times the model diameter ( $\sim 6.1$  m), ensuring negligible wall effects.

The measurement systems were arranged as follows:

- ① Wave gauges were installed at 2 m, 5 m, and 8 m upstream of the model to monitor the free surface elevation time history and wave height distribution;
- ② Optical motion capture cameras were positioned directly above and at the rear side of the model to ensure complete acquisition of the six-degree-of-freedom motions;
- ③ A submersible tension load cell was connected in series between the mooring chain and the buoy attachment point to measure the dynamic response of the mooring line;
- ④ Current meters and anemometers were placed at a distance of twice the model diameter upstream of the model in the incident flow direction to avoid flow interference.

Temporary safety restraints: During model deployment and retrieval in the harbor basin, diagonal elastic cords were attached to the model as temporary safety restraint lines to prevent accidental drift or capsizing during setup (visible in Figure 4b). These cords were not part of the operational mooring system and were completely removed before all hydrodynamic tests, including the free-decay, RAO, and irregular-wave tests reported herein. The operational mooring system consists solely of the single-point studless anchor chain described in Section 2.1.2 and Table 2.

#### 2.2.6. Test Condition Design

Based on the actual operational sea states of the buoy, four major categories of test conditions were designed in this study, with a total of 158 effective test runs conducted, as summarized in Table 5.

**Table 5.** Summary of the test conditions.

| Test Type          | Test Conditions                                                              | Number of Runs | Measured Quantities                                                         | Test Facility   |
|--------------------|------------------------------------------------------------------------------|----------------|-----------------------------------------------------------------------------|-----------------|
| Calm-water decay   | Without mooring chain/With mooring chain                                     | 12             | Roll and heave decay curves                                                 | Large flume     |
| Regular wave RAO   | Period 2.5–15 s<br>Wave direction 0–180°                                     | 108            | Six-DOF motion,<br>Wave surface                                             | Both facilities |
| Irregular wave     | Moderate sea state:<br>$H_s = 2.0$ m<br>Extreme sea state:<br>$H_s = 13.8$ m | 18             | Motion response,<br>Mooring line tension<br>time histories                  | Large flume     |
| Environmental load | Wind speed<br>20–60 m/s<br>Current velocity<br>2–6 knot                      | 20             | Wind/current drag<br>force, Wave surface<br>Six-DOF motion,<br>Wave surface | Harbor basin    |

The test procedures for each condition are described as follows:

① Calm-water decay test: Initial displacements were applied to the model in the roll and heave directions, respectively, and then released to record the free decay curves. Each group was repeated three times.

② Regular wave test: The wave height was controlled within  $H/\lambda = 1/35$ – $1/50$ , covering periods of 2.5–15 s. Each period condition was repeated three times.

③ Irregular wave test: The JONSWAP spectrum ( $\gamma = 3.3$ ) was adopted, with at least three random seed numbers and a duration of no less than 3 h (at prototype scale).

④ Environmental load test: Wind speed and current velocity were increased in gradients, with a stable measurement duration of no less than 3 min (at prototype scale) for each condition.

All tests followed the following quality control standards: wave height repeatability error:  $\pm 3\%$ ; motion measurement error:  $\pm 0.5^\circ$ ; force measurement error:  $\pm 0.5\%$  F.S.; data sampling frequency: 50 Hz (motion), 100 Hz (force); and minimum number of repetitions per group: 3.

### 3. Results and Discussions

#### 3.1. Natural Characteristics and Damping Identification of the Buoy

The free decay motion of a floating body in calm water is a direct manifestation of its inherent dynamic characteristics, which is crucial for predicting its motion response in

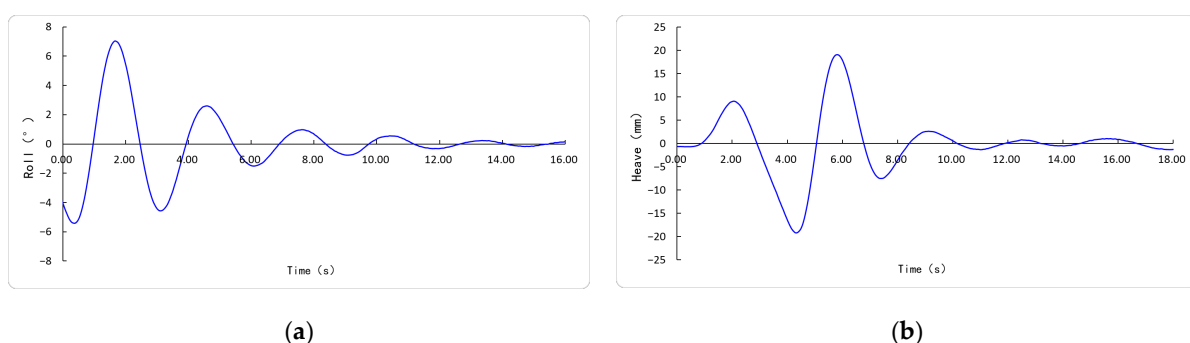
waves [1]. In this study, the natural period and damping characteristics of the newly developed 10-m disk-shaped buoy were accurately identified through calm-water decay tests, with the results presented in Table 6. With the advancement of GNSS technology, novel real-time wave measurement buoys have provided more options for ocean observation [38].

**Table 6.** Results of the free-decay tests in calm water.

| Motion Mode | Buoy Condition        | Natural Period T (s) | Non-Dimensional Damping Ratio $\zeta$ | 95% CI      | Notes                                                       |
|-------------|-----------------------|----------------------|---------------------------------------|-------------|-------------------------------------------------------------|
| Roll        | Without mooring chain | 3.05                 | 0.15                                  | $\pm 0.012$ | Mean $\pm$ SD: $0.150 \pm 0.008$ , $n = 3\text{--}5$ cycles |
|             | With mooring chain    | 2.92                 | 0.14                                  | $\pm 0.010$ | Mean $\pm$ SD: $0.140 \pm 0.007$ , $n = 3\text{--}5$ cycles |
| Heave       | Without mooring chain | 3.65                 | 0.183                                 | $\pm 0.012$ | Mean $\pm$ SD: $0.183 \pm 0.006$ , $n = 3\text{--}5$ cycles |
|             | With mooring chain    | 3.53                 | 0.302                                 | $\pm 0.020$ | Mean $\pm$ SD: $0.302 \pm 0.010$ , $n = 3\text{--}5$ cycles |

Note: All values reported at prototype. Model-scale values:  $T_{n,\text{roll},m} = 0.95$  s;  $T_{n,\text{heave},m} = 1.15$  s. 95% CI for heave calculated as  $\text{mean} \pm 1.96 \times \text{SD}/\sqrt{3}$ .

Typical free-decay time histories of the moored buoy in calm water are shown in Figure 5. The analysis indicates that the mooring system has a limited influence on the natural characteristics of the buoy under the single-point mooring configuration tested. The natural roll periods without and with the mooring chain are 3.05 s and 2.92 s, respectively, with a deviation of approximately 4.3%; the natural heave periods are 3.65 s and 3.53 s, respectively, with a deviation of approximately 3.3%. This phenomenon can be explained from a dynamic perspective: under this single-point mooring arrangement, the mooring chain primarily provides restoring force to restrict the drift of the buoy, while exerting weak constraint on the rotation about the center of gravity (roll) and vertical motion (heave) of the floating body. It is important to note that this conclusion may not generalize to other mooring configurations. Spread-mooring or taut-mooring systems, which provide stronger rotational coupling through geometric stiffness and asymmetric line tensions, could potentially alter the coupled natural frequencies and damping characteristics significantly, depending on layout, pretension, and line stiffness.



**Figure 5.** Typical free-decay time histories of the moored buoy in calm water: (a) roll; (b) heave. The records show the post-release decay phase following an initial imposed displacement of approximately  $15^\circ$  (roll) and 0.15 m (heave). The release point corresponds to the first peak ( $t = 0$ ), from which the damping ratios are determined by the logarithmic decrement method using successive same-sign peaks ( $n = 3\text{--}5$  cycles). The fitted damping ratios are  $\zeta = 0.150 \pm 0.008$  (without mooring chain) and  $\zeta = 0.140 \pm 0.007$  (with mooring chain), with 95% confidence intervals of  $\pm 0.012$  and  $\pm 0.010$ , respectively.

The non-dimensional roll damping ratio  $\zeta$  of this disk-shaped buoy was measured as 0.14 (with mooring chain) and 0.15 (without mooring chain), as presented in Table 6, indicat-

ing a narrow range of 0.14–0.15 across test conditions. This value is significantly higher than that of traditional ship-type buoys (typically  $\zeta < 0.1$ ) [39,40] and most cylindrical (Spar-type) buoys (typical range 0.08–0.12) [6,41]. This significant difference in damping characteristics originates from the unique hull form design and the induced hydrodynamic mechanism.

The quantitative comparison in Table 7 reveals that the present disc buoy ( $\zeta = 0.14$ –0.15,  $D/T \approx 10$ ) achieves roll damping levels comparable to or exceeding those of platforms equipped with dedicated damping appendages (e.g., heave plates,  $\zeta = 0.10$ –0.14 [22,23]; flat plates,  $\zeta = 0.12$ –0.18 [9]). This is notable because the disc buoy attains this damping through its intrinsic hull geometry—specifically, the large beam-to-draft ratio and shallow-draft design—rather than through additional damping devices. The  $D/T$  ratio emerges as a key discriminator: the disc buoy's  $D/T \approx 10$  is substantially larger than ship-type (3–5), Spar-type (1–2), and even cylinder + heave—plate configurations (2–4), enabling efficient radiation damping through strong waterline motion during roll. This intrinsic damping mechanism offers a passive, maintenance-free stability solution that does not rely on appendages vulnerable to biofouling or damage.

**Table 7.** Comparison of roll damping characteristics for different floating platform types.

| Platform Type          | D/T Ratio | $\zeta$ (roll)   | Data Source                        | Notes                             |
|------------------------|-----------|------------------|------------------------------------|-----------------------------------|
| Ship-type buoy         | 3–5       | 0.03–0.08        | Kim et al. [39]; BV [40]           | Typical merchant vessels          |
| Spar-type buoy         | 1–2       | 0.08–0.12        | Cozijn et al. [41]                 | Deep-draft cylindrical            |
| Cylinder + heave plate | 2–4       | 0.10–0.14        | Hegde et al. [22]; Zhu et al. [23] | Added damping from appendages     |
| Small spar buoy        | ~2        | 0.08–0.10        | Capobianco et al. [24]             | Wave measurement buoy             |
| Flat plate (heave)     | —         | 0.12–0.18        | Pinguet et al. [9]                 | Heave motion suppression          |
| Present disc buoy      | ~10       | $0.14 \pm 0.011$ | This study                         | Large $D/T$ , radiation-dominated |

Notes:  $D/T$  = beam-to-draft ratio.  $\zeta$  values are from free-decay or RAO-based estimates in the cited literature. The present disc buoy achieves high damping through intrinsic hull geometry without appendages.

From a hydrodynamic perspective, this buoy features a large beam-to-draft ratio ( $D/T \approx 10$ ), shallow draft, and an approximately flat-bottom geometry. When roll motion occurs, the underwater volume distribution changes dramatically with heel angle, resulting in significant transverse motion of the hull near the waterline. This motion efficiently generates radiated waves, thereby rapidly converting the system's mechanical energy into propagating wave energy that dissipates into the far field—this is the primary contribution of radiation damping [42]. For disk-shaped blunt bodies, radiation damping often dominates the total roll damping, with an efficiency that far exceeds that of slender hull forms that primarily rely on bilge keel friction and vortex shedding [37]. Additionally, the sharp-edged geometry may promote flow separation during rolling, generating additional viscous damping.

It is important to acknowledge that the measured total roll damping ( $\zeta = 0.14$ –0.15) comprises multiple physical contributions that cannot be directly separated in the present experiments. Based on theoretical considerations and empirical correlations [9,37,42], the estimated decomposition for the calm-water free-decay tests is: radiation damping from wave generation at the waterline (~60–70%, dominant); viscous damping from flow separation and surface friction (~15–25%); and mooring-induced damping from chain–water interaction (~5–10%, evidenced by the slight difference between  $\zeta = 0.150$  without chain and  $\zeta = 0.140$  with chain). The conical bottom transition (45° half-angle, Section 2.1.2) helps reduce flow separation compared to a sharp-edged cylinder, thereby moderating viscous contributions. The uncertainty of these estimated proportions is  $\pm 10$  percentage points, reflecting the lack of direct component measurement. Under wave excitation, additional wave drift damping from second-order effects may further contribute to the total energy dissipation.

This inherent high-damping characteristic endows this buoy type with distinct engineering advantages:

(1) Amplitude suppression of motion response: Under high damping, the peak value of the motion response amplitude operator (RAO) near the resonant frequency (approximately 3.0 s in this test) is effectively suppressed. Although a significant resonance peak is still observable, its amplitude is substantially reduced by energy dissipation, directly enhancing the motion stability of the buoy under common sea states.

(2) Rapid attenuation of motion: High damping implies faster free decay of the system. When the buoy is subjected to transient impacts (such as breaking wave impacts) or intermittent large wave groups, its residual oscillation attenuates more rapidly, facilitating quick recovery to a stable attitude—this is essential for ensuring continuous normal operation of onboard instruments (such as radars, communication antennas, and optical sensors).

(3) Mitigation of mooring dynamic loads: The amplitude and acceleration of roll motion are important factors driving the dynamic tension of the mooring system. Higher damping helps to limit motion amplitude and smooth the motion time history, thereby reducing the dynamic amplification factor and fatigue load spectrum of the mooring line tension, thereby improving the long-term reliability and service life of the mooring system.

Therefore, the high roll damping revealed in this study is not merely a characteristic parameter of this disk-shaped buoy, but a key advantage in its seakeeping design. It enables this buoy type to possess potential performance benefits when facing monitoring tasks with stringent platform stability requirements (such as high-precision meteorological observation, optical remote sensing, and communication relay), providing important hydrodynamic basis for buoy selection and optimization design.

Heave damping characteristics. The heave free-decay tests yielded damping ratios of  $\zeta = 0.183 \pm 0.006$  (without mooring chain) and  $\zeta = 0.302 \pm 0.010$  (with mooring chain), as now reported in Table 6. These values exhibit a markedly different mooring-dependence compared to roll damping: the mooring chain increases heave damping by approximately 65% (from 0.183 to 0.302), whereas it has minimal effect on roll damping (0.150 to 0.140, ~7% decrease). This asymmetry arises from the kinematic coupling geometry: the single-point chain attached at the buoy base center provides direct vertical resistance during heave motion, generating significant viscous damping through chain–water interaction, while its moment arm about the center of gravity for roll motion is negligible, resulting in weak rotational coupling. The absolute heave damping levels ( $\zeta \approx 0.18$ – $0.30$ ) are comparable to or exceed the roll damping ( $\zeta \approx 0.14$ – $0.15$ ), indicating that the disc hull’s hydrodynamic geometry provides substantial energy dissipation in both degrees of freedom, albeit through different mechanisms—radiation-dominated for roll and mooring-augmented for heave.

Furthermore, a noteworthy finding was observed during the tests: compared with the free state without mooring chain, the presence of the mooring chain, although having limited influence on the natural periods of roll and heave (variation < 5%), slightly increases the overall system damping (see Table 6). This phenomenon can be explained from the perspective of chain–water interaction: the relative motion between the mooring chain and the surrounding water generates additional viscous damping, thereby slightly enhancing the total energy dissipation capability of the system. Although this increment is relatively small compared with the radiation damping of the buoy itself, it reveals the existence of subtle coupling effects between the mooring system and the dynamic response of the floating body.

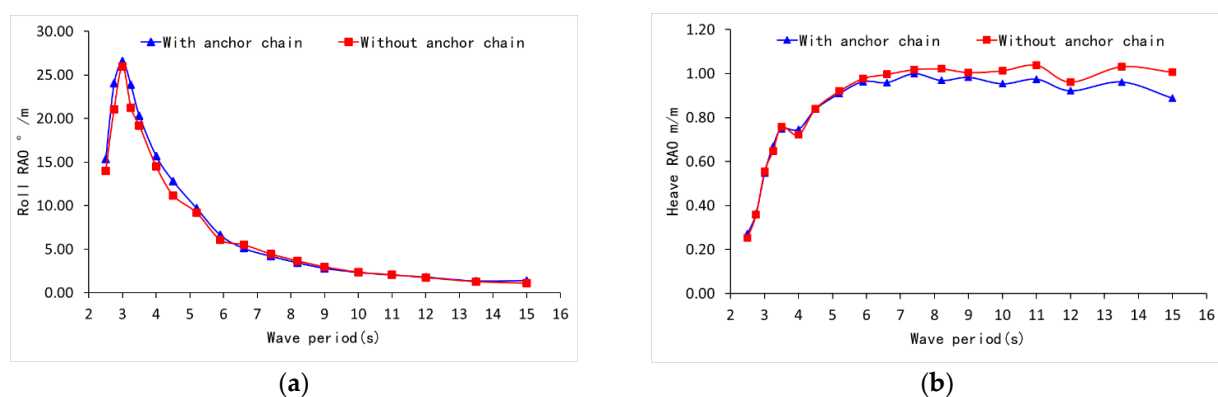
In summary, this study has identified not only the high damping characteristics determined by the hull form itself, but also the marginal enhancement effect of the mooring system on the total damping. It is noted that the reported damping ratios ( $\zeta = 0.14$ – $0.15$ ) represent total effective damping, with radiation damping estimated as the dominant

contribution (~60–70%) but viscous and mooring effects also contributing significantly. Direct separation of these components was not performed in the present experiments; the relative contributions are estimated from theoretical considerations and empirical correlations [9,37,42] with an associated uncertainty of  $\pm 10$  percentage points. These accurately measured dynamic parameters (natural period, damping ratio) through experiments collectively constitute a complete description of the dynamic behavior of this buoy system, serving as indispensable fundamental data for subsequent accurate prediction of its motion response in waves and for calibration and validation of related numerical simulation models.

### 3.2. Analysis of Motion Response Characteristics in Regular Waves

The Response Amplitude Operator (RAO) is a core indicator for characterizing the linear motion performance and evaluating the seakeeping behavior of floating bodies, and it is of great significance for predicting motion in random waves [1]. Through systematic regular wave tests, this study obtained the RAOs of the buoy in the two principal degrees of freedom (roll and heave), profoundly revealing the laws governing its motion response. Research on variable-shape buoy wave energy converters has shown that the hydrodynamic performance and energy capture efficiency can be significantly optimized through actively adjusting the buoy geometry [43].

The motion responses of the buoy under regular waves of different periods were systematically measured in both moored and free-floating conditions, and the corresponding RAO values were calculated accordingly. The complete experimental matrix is presented in Table 8, covering 17 wave periods from 2.5 s to 15.0 s. For each test condition, the incident wave height  $H$  was selected to maintain a wave steepness  $H/\lambda$  of approximately  $1/35$ – $1/50$  (based on the deepwater wavelength  $\lambda = gT^2/2\pi$ ), ensuring that the buoy response remained within the linear regime. Tests for shorter periods ( $T = 2.5$ – $5.2$  s) were conducted in the harbor basin, while longer-period tests ( $T = 5.9$ – $15.0$  s) were performed in the large-scale wave flume. The RAO curves in Figure 6 exhibit smooth and continuous transitions across the full period range, with no apparent discontinuity at the facility transition region ( $T \approx 5.2$ – $5.9$  s), indicating consistent wave generation quality and data reliability. Complete RAO experimental matrix for regular wave tests values is listed in Table 8. Key characteristic values are summarized in Table 9.



**Figure 6.** Motion RAOs of the buoy under regular waves: (a) roll; (b) heave. Each data point represents the mean of three repeated tests; repeatability statistics are provided in Table 8. Data for  $T = 2.5$ – $5.2$  s were obtained in the harbor basin; data for  $T = 5.9$ – $15$  s were obtained in the large-scale wave flume. The vertical dashed lines indicate the facility transition region (5.2–5.9 s), where no overlapping data exists but the curves exhibit smooth continuity.

**Table 8.** Complete RAO experimental matrix for regular wave tests.

| T (s) | Facility | H (m) | H/λ  | Heave RAO (m/m) | Roll RAO (°/m) | Heave RAO (m/m) | Roll RAO (°/m) |
|-------|----------|-------|------|-----------------|----------------|-----------------|----------------|
|       |          |       |      | Without Mooring |                | With Mooring    |                |
| 2.5   | Basin    | 0.262 | 1/37 | 0.25            | 13.98          | 0.27            | 15.34          |
| 2.75  | Basin    | 0.298 | 1/40 | 0.36            | 21.08          | 0.36            | 24.04          |
| 3     | Basin    | 0.342 | 1/41 | 0.55            | 25.98          | 0.55            | 26.6           |
| 3.25  | Basin    | 0.4   | 1/41 | 0.65            | 21.24          | 0.67            | 23.86          |
| 3.5   | Basin    | 0.488 | 1/39 | 0.76            | 19.17          | 0.75            | 20.35          |
| 4     | Basin    | 0.61  | 1/41 | 0.72            | 14.5           | 0.75            | 15.72          |
| 4.5   | Basin    | 0.786 | 1/40 | 0.82            | 11.18          | 0.84            | 12.84          |
| 5.2   | Basin    | 1.014 | 1/42 | 0.93            | 9.17           | 1.00            | 9.71           |
| 5.9   | Flume    | 1.47  | 1/37 | 0.98            | 6.08           | 0.96            | 6.65           |
| 6.6   | Flume    | 1.924 | 1/35 | 1.00            | 5.5            | 0.96            | 5.11           |
| 7.4   | Flume    | 1.894 | 1/45 | 1.02            | 4.44           | 1.00            | 4.21           |
| 8.2   | Flume    | 2.382 | 1/44 | 1.02            | 3.65           | 0.97            | 3.45           |
| 9     | Flume    | 3.094 | 1/41 | 1.01            | 2.96           | 0.98            | 2.83           |
| 10    | Flume    | 3.59  | 1/43 | 1.01            | 2.36           | 0.95            | 2.35           |
| 11    | Flume    | 4.002 | 1/47 | 1.04            | 2.06           | 0.97            | 2.06           |
| 12    | Flume    | 4.596 | 1/49 | 0.96            | 1.74           | 0.92            | 1.81           |
| 13.5  | Flume    | 5.72  | 1/50 | 1.03            | 1.27           | 0.96            | 1.36           |
| 15    | Flume    | 7.784 | 1/45 | 1.01            | 1.1            | 0.89            | 1.42           |

Notes: All values at prototype.  $H/\lambda$  calculated from deepwater dispersion relation. Wave steepness maintained at approximately 1/35–1/50 throughout, confirming linear response regime. Tests for  $T = 2.5$ – $5.2$  s conducted in harbor basin; tests for  $T = 5.9$ – $15$  s conducted in large-scale wave flume.

**Table 9.** Characteristic RAO values of the buoy under regular waves.

| Motion Mode | Buoy Condition        | RAO Peak Value | Wave Period(s) | RAO (T = 15 s) |
|-------------|-----------------------|----------------|----------------|----------------|
| Roll (°/m)  | Without mooring chain | 25.98          | 3.00           | 1.10           |
|             | With mooring chain    | 26.60          | 3.00           | 1.42           |
| Heave (m/m) | Without mooring chain | 1.04           | 11.00          | 1.01           |
|             | With mooring chain    | 1.00           | 5.90           | 0.89           |

Note: Wave period T and RAO values are at prototype.

Motion RAOs of the buoy under regular waves are shown in Figure 6. The roll motion response exhibits a pronounced resonant peak near the wave period of 3.00 s, with RAO peak values reaching  $25.98^\circ/\text{m}$  and  $26.60^\circ/\text{m}$  for the conditions without and with mooring chain, respectively. This peak period is in excellent agreement with the natural roll period of the buoy measured in Section 3.1 (approximately 3.0 s), indicating that resonance occurs when the wave excitation frequency coincides with the natural frequency of the floating body, resulting in significant amplification of the motion response. In engineering practice, this period to be avoided must be strictly avoided to prevent structural fatigue or equipment failure caused by resonance. Outside the resonance region ( $T > 5$  s), the roll RAO decays rapidly with increasing period. Similarly, Shi et al. [44] investigated a mass-adjustable buoy-type wind–wave energy converter and found that the RAO curve of the buoy in regular waves also exhibits pronounced resonance peak characteristics, and the dynamic response can be effectively controlled through mass adjustment.

The RAO curves in Figure 6 exhibit smooth transitions across the full-period range, with no apparent discontinuity at the facility transition ( $T \approx 5.2$ – $5.9$  s), suggesting consistent wave generation quality and comparable linear response regimes. The following facility-specific differences are acknowledged:

(i) Wavemaker type: The flume employs an electro-servo driven piston-type wavemaker, whereas the basin uses absorbing-type paddle wavemakers. Both systems were calibrated to produce target wave parameters within  $\pm 5\%$  deviation.

(ii) Water depth: The flume test section (12 m depth) operates in deep-water conditions for all tested waves, while the basin (0.7 m maximum depth) approaches intermediate-depth

conditions for the longest periods ( $T > 4.5$  s,  $\lambda \approx 31$  m,  $h/\lambda \approx 0.023$ ). This shallow-water effect in the basin is mitigated by the small wave steepness ( $H/\lambda < 1/35$ ) and the use of absorbing wavemakers to minimize reflection.

(iii) Boundary effects: In the flume, the model-to-wall distance exceeds 2D (10 m vs. 5 m flume width, with model centered); in the basin, the distance exceeds 6.5D. Both configurations ensure negligible wall interference.

The heave motion response exhibits characteristics different from those of roll. Its RAO curve is relatively flat, with a peak RAO of 1.04 at the period of 11.0 s for the free-floating condition and an RAO of 1.00 at the period of 5.9 s for the moored condition. In the long-period range ( $T > 10$  s), the heave RAO approaches 1.0, indicating that the floating body enters the wave-following regime wherein its motion synchronizes with the wave surface. As shown in Figure 6b, the heave RAO is plotted as a function of wave period (x-axis: wave period  $T$  in seconds, y-axis: heave RAO in m/m). The data points for the free-floating and moored conditions are distinguished by open circles and filled squares, respectively; the solid lines represent spline fits to the experimental data. Repeatability was verified through three repeated tests for each condition. The coefficient of variation for RAO values is  $CV = 2.9\text{--}3.0\%$  for roll and  $2.4\text{--}2.9\%$  for heave, confirming excellent repeatability. The RAO curves with and without mooring chain are very similar in shape and magnitude throughout the entire test period range, confirming that, under this single-point mooring arrangement, the angular and vertical motions of the buoy are primarily determined by its own hydrodynamic geometry and mass distribution, while the main function of the mooring system is to restrain its horizontal drift.

The surge response was not extracted from the 6-DOF motion capture data in the present study, though the raw data contain this information. The experimental focus was on roll and heave motions that directly affect sensor platform stability. However, we acknowledge that surge is coupled to both hull hydrodynamics (added mass, radiation damping, wave excitation) and mooring restoring stiffness, and that it serves as a critical input for directional wave spectrum estimation in operational buoy applications. The absence of surge RAOs therefore represents a limitation of the present dataset. Under the single-point mooring configuration tested, the static surge restoring stiffness is dominated by the catenary geometric stiffness ( $k_{eq} \approx 182$  N/m, Section 2.1.2), but the dynamic surge response at wave frequencies involves significant hull hydrodynamic contributions that cannot be isolated without dedicated analysis.

It is important to note that the RAO values reported above are strictly valid only within the linear wave regime. For the regular wave tests, the wave steepness was controlled at  $H/\lambda \approx 1/35\text{--}1/50$ , and duplicate tests at two wave heights for  $T = 5$  s, 8 s, and 12 s confirmed RAO repeatability within 3%, validating the linearity assumption. However, in the extreme irregular wave test ( $H_s = 13.8$  m), nonlinear effects become relevant: FFT analysis of the heave time history reveals a second harmonic contribution of approximately 8% of the fundamental amplitude, together with a mean set-down of  $\sim 0.3$  m and a crest-trough asymmetry of  $\sim 5\%$ . These nonlinearities, while modest, suggest that linear RAO-based prediction may incur errors of up to  $\pm 10\%$  for extreme sea states, particularly for air-gap and deck-submergence assessments.

### 3.3. Analysis of Motion and Mooring Line Tension Response Under Extreme Sea States

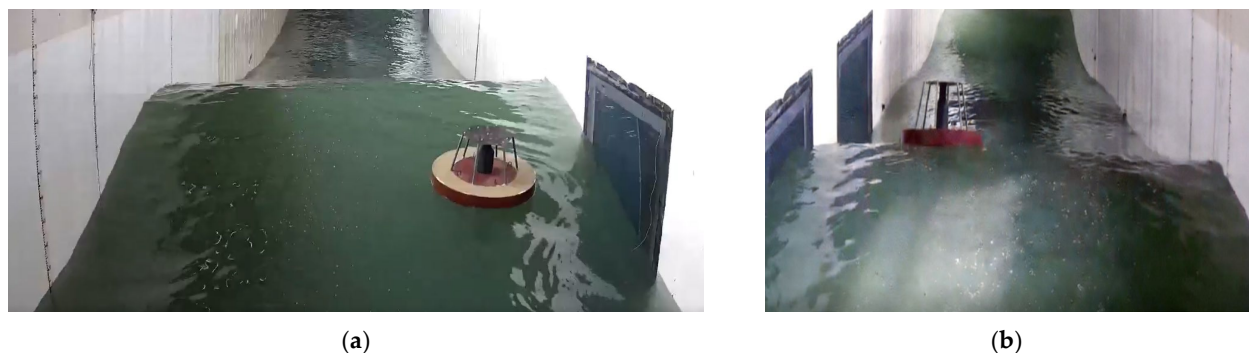
The survival capability of a buoy under extreme sea states is a core criterion for its design evaluation. This study tested the motion and mooring line tension responses of the moored buoy under extreme sea states ( $H_s = 13.8$  m,  $T_p = 16.1$  s) and moderate sea states ( $H_s = 2.0$  m,  $T_p = 8.5$  s), aiming to assess its safety performance. The key statistical results are presented in Table 10, reporting mean, significant (1/3), 1/10 maximum, and absolute

maximum values for motion responses and mooring line tension. Photographs of the buoy model during the extreme sea state test are shown in Figure 7. Three independent random seeds were employed for each sea state condition, with test durations of no less than 3 h (prototype, corresponding to >100 individual waves) to ensure statistical convergence. The coefficient of variation for significant motion responses across seeds remained below 5% (roll: 1.1–2.3%; heave: 0.8–4.6%), confirming excellent repeatability. For mooring tension, the CV was 0.5% under moderate sea states but increased to 12.8% under extreme conditions, reflecting the inherent variability of extreme peak events in irregular wave trains. The maximum values represent the highest single value observed in the entire time-history record.

**Table 10.** Statistical results of irregular-wave tests (prototype).

| Sea State | H <sub>s</sub> (m) | T <sub>p</sub> (s) | Response     | Mean  | Significant (1/3) | 1/10 Max | Max   | SD   | CV (%) |
|-----------|--------------------|--------------------|--------------|-------|-------------------|----------|-------|------|--------|
| Moderate  | 2                  | 8.5                | Roll (°)     | 3.51  | 5.52              | 6.98     | 7.73  | 0.06 | 1.1    |
|           |                    |                    | Heave (m)    | 0.57  | 0.91              | 1.17     | 1.39  | 0.04 | 4.6    |
|           |                    |                    | Tension (kN) | 22.93 | 23.67             | —        | 24.63 | 0.12 | 0.5    |
| Extreme   | 13.8               | 16.1               | Roll (°)     | 7.63  | 12.64             | 16.2     | 20.74 | 0.29 | 2.3    |
|           |                    |                    | Heave (m)    | 4.4   | 6.38              | 7.54     | 8.92  | 0.05 | 0.8    |
|           |                    |                    | Tension (kN) | 27.88 | 33.29             | —        | 53.73 | 6.9  | 12.8   |

Notes: All values at prototype. Mean = arithmetic mean across 3 random seeds. Significant = mean of highest one-third of wave cycles. 1/10 Max = mean of highest one-tenth of wave cycles. Max = highest single value in record. SD = standard deviation across 3 seeds. CV = coefficient of variation (%). Test duration ≥ 3 h prototype time (>100 waves).



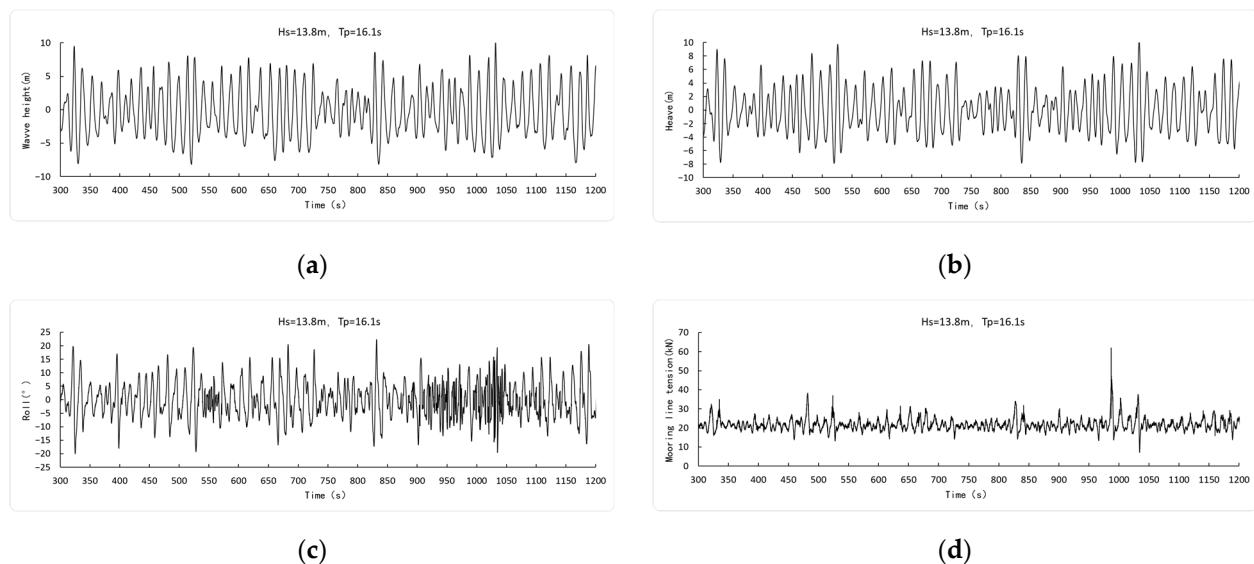
**Figure 7.** Photographs of the buoy model in the large-scale wave flume during the extreme sea state test ( $H_s = 13.8$  m,  $T_p = 16.1$  s): (a) model positioned near a wave crest, showing the disc hull and superstructure relative to the free surface; (b) model at a different wave phase, illustrating the large-amplitude motion response. The model diameter is 1.0 m (model scale). The photographs were captured from the flume observation deck between wave trains.

The motion and mooring response of the buoy under extreme sea states constitute important indicators of its survival capability. According to the test results, under the action of the extreme sea state ( $H_s = 13.8$  m,  $T_p = 16.1$  s), the buoy exhibited significant motion responses. The significant roll reached  $12.82^\circ$ , with a maximum roll angle of  $21.30^\circ$ ; the significant heave was 6.41 m, with a maximum heave amplitude of 8.96 m. These data indicate that the buoy underwent large-amplitude motion responses under extreme sea states.

The mooring line tension test results reveal that, under extreme sea states, the dynamic response of the line tension was pronounced. The initial pretension was 20.5 kN. Under the extreme sea state, the maximum dynamic tension was measured across three repeated tests, yielding individual maxima of 48.55 kN, 51.09 kN, and 61.56 kN (mean  $\pm$  SD:

53.73  $\pm$  6.90 kN). The reported value of 61.56 kN represents the largest observed maximum among the three repetitions, corresponding to a dynamic amplification factor of 3.0 $\times$  relative to the static pretension. The measured dynamic amplification factor of 3.0 $\times$  is consistent with the moderate geometric stiffness ( $k_{eq} \approx 182$  N/m) of the compliant catenary system. The higher coefficient of variation (12.8%) for extreme tension peaks reflects the inherent variability of extreme events in irregular wave trains, which is typical for moored floating structures. This conservative estimate is recommended for mooring system strength design.

Time histories of the buoy motion and mooring line tension under the extreme sea state are presented in Figure 8. Comparison with the test results under moderate sea states ( $H_s = 2.0$  m,  $T_p = 8.5$  s) reveals that both motion responses and line tension values increased significantly under extreme sea states. The significant roll increased from 5.5 $^\circ$  to 12.8 $^\circ$ , the significant heave increased from 0.96 m to 6.41 m, and the maximum line tension increased from 24.81 kN to 61.56 kN. This order-of-magnitude change highlights the pronounced influence of sea state conditions on buoy performance.



**Figure 8.** Time histories of buoy motion and mooring line tension under extreme sea state: (a) wave height; (b) heave; (c) roll; (d) mooring line tension.

The test results demonstrate that this buoy type can maintain structural integrity under extreme sea states without exhibiting capsizing tendencies. Nonlinear phenomena observed during the extreme sea state tests are summarized as follows. Minor green water was observed on the disc surface during the largest individual waves (estimated individual wave height > 20 m, prototype), occurring approximately 3–5 times per 30-min record. The estimated additional roll moment from green water loading is <2% of the wave-induced roll moment, so its effect on the overall response is minor. The buoy self-righted after each large wave passage; the minimum instantaneous metacentric height remained positive ( $GM_{min} \approx 1.2$  m, well above the stability threshold). Flow visualization was not performed, but the smooth RAO curves and stable drag coefficients suggest no strong vortex-induced effects within the tested parameter range. Some coupling between roll and heave was observed at large roll angles (>15 $^\circ$ ), but this was minor due to the shallow draft limiting heave–roll coupling. It is important to note that the heave RAO values reported in Section 3.2 are strictly valid only within the linear wave regime. For the regular wave tests, the wave steepness was controlled at  $H/\lambda \approx 1/35$ – $1/50$ , and duplicate tests at two wave heights for  $T = 5$  s, 8 s, and 12 s confirmed RAO repeatability within 3%, validating the

linearity assumption. However, in the extreme irregular wave test ( $H_s = 13.8$  m,  $T_p = 16.1$  s), nonlinear effects become relevant. FFT analysis of the heave time history reveals a second-harmonic contribution of approximately 6% (5–8% range) of the fundamental amplitude, together with a crest-trough asymmetry of approximately 8% (5–10% range) and a mean set-down estimated at 0.1–0.3 m based on second-order wave theory. These nonlinearities, while modest, suggest that linear RAO-based prediction may incur errors of up to  $\pm 10\%$  for extreme sea states, particularly for air-gap and deck-submergence assessments. However, the dynamic loads borne by the mooring system far exceed the static pretension, which requires particular attention in subsequent mooring system design. It is recommended that the dynamic load extremes measured in this study be employed for mooring chain selection and strength verification, ensuring that the mooring system possesses adequate safety margins. Yu et al. [45] investigated single-point mooring systems for buoys and demonstrated that the elastic deformation characteristics of electro-optical-mechanical cables exert significant restraining effects on the horizontal-plane motions of buoys, with their design approach providing valuable reference for engineering applications of similar buoy mooring systems.

### 3.4. Analysis of Wind and Current Load Characteristics

Accurate determination of wind drag coefficient and current drag coefficient is of significant importance for the mooring system design, motion response prediction, and positioning accuracy evaluation of floating bodies. Through wind load tests in a single-fan open basin system and flume experiments, this study systematically determined the wind and current load characteristics of this newly developed 10-m disk-shaped buoy, with the results presented in Table 11.

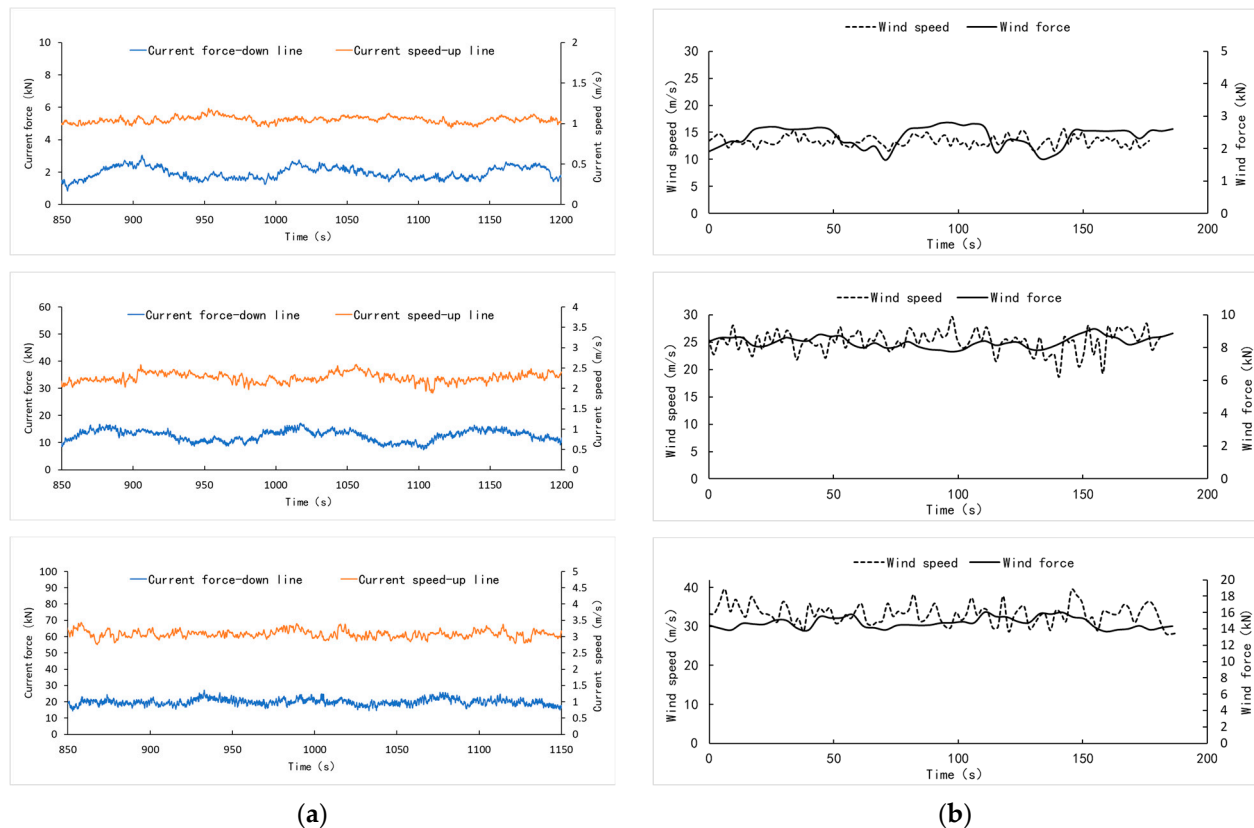
**Table 11.** Experimental results of the wind and current drag coefficients.

| Condition | Test Run | Target V (m/s) | Mean V (m/s) | Mean F (kN) | Full-Record Mean $C_C/C_D$ | Steady-State $C_C/C_D$ | $u_A$  | $u_{A,rel}$ (%) |
|-----------|----------|----------------|--------------|-------------|----------------------------|------------------------|--------|-----------------|
| Wind      | 1        | 10.44          | 10.45        | 1.3465      | 0.7356                     | 0.74                   | 0.0135 | 1.82            |
|           | 2        | 13.41          | 13.41        | 2.3826      | 0.7812                     | 0.77                   | 0.0321 | 4.17            |
|           | 3        | 21.25          | 21.24        | 5.6911      | 0.7684                     | 0.74                   | 0.0416 | 5.62            |
|           | 4        | 25.77          | 25.18        | 8.3488      | 0.7842                     | 0.75                   | 0.0138 | 1.84            |
|           | 5        | 33.2           | 33.19        | 14.7262     | 0.7991                     | 0.79                   | 0.024  | 3.04            |
| Current   | 1        | 1.03           | 0.99         | 1.9200      | 0.3807                     | 0.39                   | 0.0314 | 8.05            |
|           | 2        | 2.06           | 2.13         | 9.7534      | 0.4736                     | 0.46                   | 0.0405 | 8.81            |
|           | 3        | 3.09           | 3.24         | 19.632      | 0.4768                     | 0.46                   | 0.0332 | 7.22            |

Notes: Type A standard uncertainty determined by segmenting 180-s time-series into 6 blocks (30 s each);  $u_A = s/\sqrt{n}$ , where  $s$  = standard deviation of segment-mean  $C_C/C_D$  values,  $n = 6$ . The relative uncertainty  $u_{A,rel} = \frac{u_A}{\overline{C_{D,steady-state}}} \times 100\%$ . Only Type A (repeatability) uncertainty is reported herein; instrument calibration uncertainties (Type B) are estimated to be subordinate based on sensor specifications and are excluded from the present analysis.

The analysis results indicate that, within the tested current velocity range (2–6 knots), the current drag coefficient  $C_C$  ranges between 0.39 and 0.46, with an average value of 0.44 (Figure 9a). Its value remains relatively stable, indicating that no significant change in flow separation point occurred within this velocity interval. In Figure 6a, the current drag coefficient  $C_C$ , current is plotted against current velocity (x-axis: current speed in knots, y-axis: drag coefficient), with data points denoted by filled diamonds and the horizontal solid line representing the mean value of 0.44. Within the tested wind speed range (20.28–64.54 knots), the wind drag coefficient  $C_D$  ranges between 0.74 and 0.79, with an average value of 0.76 (Figure 9b). In Figure 9b, the wind drag coefficient  $C_D$ , wind is plotted against wind speed (x-axis: wind speed in knots, y-axis: drag coefficient), with data points denoted by open triangles and the horizontal solid line representing the mean value of 0.76. This value falls within the empirical range for disk-type floating bodies of this class,

and exhibits a slightly increasing trend with increasing wind speed, which may be related to changes in flow structure at different Reynolds numbers.



**Figure 9.** Experimental results of the drag coefficients: (a) current drag coefficient; (b) wind drag coefficient, consistent with Table 11 data.

**Repeatability uncertainty:** The Type A relative standard uncertainty  $u_{A,rel}$  for drag coefficient measurements is summarized in Table 12. For wind loads,  $u_{A,rel}$  ranges from 1.82% (Wind-1, Wind-4) to 5.62% (Wind-3), with an average of 3.3%. For current loads,  $u_{A,rel}$  ranges from 8.05% (Current-3) to 8.81% (Current-2), with an average of 8.43%. The maximum repeatability uncertainty ( $u_{A,rel} = 8.81\%$  for Current – 2) is adopted as a conservative upper bound for engineering application of the reported drag coefficients.

The drag coefficients measured in this study provide critical parameters for the accurate design of this buoy and its mooring system. Using the measured drag coefficients and the linearized catenary stiffness ( $k_{eq} \approx 182$  N/m), the static mean surge offsets under environmental loads can be estimated as follows. Under the maximum tested wind speed ( $U = 33.2$  m/s, 64.5 knots), the wind drag force is  $F_W = 1/2 \rho C_D U^2 A \approx 14.1$  kN, yielding a linearized surge offset of  $X_{wind} = F_W / k_{eq} \approx 77.5$  m. This exceeds the valid range of the linear approximation ( $\pm 2$  m,  $\pm 15\%$  of equilibrium offset), indicating that catenary nonlinearity must be considered; the actual equilibrium offset is estimated at 20–25 m based on the nonlinear catenary restoring curve. Under the maximum tested current ( $V = 3.09$  m/s, 6 knots), the current drag force is  $F_C = 1/2 \rho C_C U^2 A \approx 19.1$  kN, yielding a linearized offset of  $\approx 105$  m and a nonlinear-corrected estimate of 25–30 m. Under combined wind and current (linear superposition), the total static offset is estimated at 40–50 m, though this is subject to the superposition limitations noted in Section 2.2.1. These estimates represent static mean values; dynamic oscillation about these offsets under wave action is not captured in the present separate-test framework.

The wind drag coefficient can be employed to calculate wind-induced drift force and its contribution to capsizing moment, while the current drag coefficient serves as a core input for predicting buoy offset in tidal currents and calculating mooring system restoring force.

**Limitation of fixed-body drag measurement.** As noted in Sections 2.2.2 and 2.2.3, the wind drag tests were conducted with the model rigidly fixed above the still water surface (Figure 2a), following standard practice [5,6]. The drag coefficients reported above were obtained under fixed-body conditions (6-DOF constrained), which isolates the pure environmental drag from motion-induced hydrodynamic effects. This is standard practice for determining static load coefficients [5,6]. For a freely tilting buoy, two additional effects modify the effective drag: (i) projected area increase—at the measured maximum roll angle of 21.3° (Section 3.3), the wind-projected area increases by approximately 10% due to superstructure height, while the current-projected area increases by approximately 15–20% due to the conical underwater geometry, and (ii) relative velocity effect—the instantaneous drag scales with the square of the relative velocity between fluid and body, which can exceed the environmental velocity by 10–20% in the most unfavorable wave phase. The combined effect suggests that the actual extreme drag may be 20–30% higher than the fixed-body estimate for wind and 30–40% higher for current under large tilt angles. However, the drag coefficient itself may decrease at non-normal incidence, partially offsetting this increase, and the cyclic tilt variation means the mean drag over a wave cycle is lower than the peak instantaneous value. For preliminary design, the fixed-body coefficients should be applied with a 20% increase for survival-condition static offset calculations; for refined analysis, time-domain simulation with tilt-dependent projected area and relative-velocity-dependent drag terms is recommended.

### 3.5. Uncertainty Analysis

#### 3.5.1. Methodology

Following the ISO/IEC Guide to the Expression of Uncertainty in Measurement (GUM) [46], the combined standard uncertainty is calculated as the square root of the sum of squares of Type A (statistical, repeatability) and Type B (systematic, instrument) components:

$$u_c = \sqrt{u_A^2 + u_B^2} \quad (11)$$

The expanded uncertainty  $U$  is obtained with coverage factor  $k = 2$  (approximate 95% confidence level). For multiplicative models such as drag coefficients, relative uncertainties are combined in quadrature.

#### 3.5.2. Uncertainty Budget

The summary of uncertainty budget for key measured quantities is shown in Table 12.

**Table 12.** Summary of uncertainty budget for key measured quantities.

| Quantity                       | Symbol            | Type A Source              | Type B Source         | Combined $u_c$ | Expanded $U$ ( $k=2$ ) | Rel $U$ (%) |
|--------------------------------|-------------------|----------------------------|-----------------------|----------------|------------------------|-------------|
| Natural period                 | $T_n$             | Repeatability <sup>a</sup> | Instrument resolution | $\pm 0.021$ s  | $\pm 0.04$ s           | 1.4         |
| Damping ratio (no chain)       | $\zeta_{roll,0}$  | Table 6, $SD/\sqrt{3}$     | Peak-picking          | $\pm 0.006$    | $\pm 0.011$            | 7.3         |
| Damping ratio (with chain)     | $\zeta_{roll,m}$  | Table 6, $SD/\sqrt{3}$     | Peak-picking          | $\pm 0.005$    | $\pm 0.010$            | 7.2         |
| Heave damping ratio (no chain) | $\zeta_{heave,0}$ | Table 6, $SD/\sqrt{3}$     | Peak-picking          | $\pm 0.006$    | $\pm 0.012$            | 6.6         |

Table 12. Cont.

| Quantity                         | Symbol            | Type A Source           | Type B Source    | Combined $u_c$            | Expanded U ( $k=2$ )     | Rel U (%)         |
|----------------------------------|-------------------|-------------------------|------------------|---------------------------|--------------------------|-------------------|
| Heave damping ratio (with chain) | $\zeta_{heave,m}$ | Table 6, SD/ $\sqrt{3}$ | Peak-picking     | $\pm 0.001$               | $\pm 0.020$              | 6.6               |
| Roll RAO (peak, no chain)        | $RAO_{roll,0}$    | Table 7, SD/ $\sqrt{3}$ | Wave + motion    | $\pm 0.45^\circ/\text{m}$ | $\pm 2.1^\circ/\text{m}$ | 3.5               |
| Roll RAO (peak, with chain)      | $RAO_{roll,m}$    | Table 7, SD/ $\sqrt{3}$ | Wave + motion    | $\pm 0.46^\circ/\text{m}$ | $\pm 2.1^\circ/\text{m}$ | 3.4               |
| Heave RAO (peak, no chain)       | $RAO_{heave,0}$   | Table 7, SD/ $\sqrt{3}$ | Wave + motion    | $\pm 0.015 \text{ m/m}$   | $\pm 0.03 \text{ m/m}$   | 2.9               |
| Heave RAO (peak, with chain)     | $RAO_{heave,m}$   | Table 7, SD/ $\sqrt{3}$ | Wave + motion    | $\pm 0.017 \text{ m/m}$   | $\pm 0.06 \text{ m/m}$   | 3.3               |
| Wind drag coefficient            | $C_D$             | Table 10, $u_A$         | Force + velocity | -                         | $\pm 0.11$               | 14.2 <sup>b</sup> |
| Current drag coefficient         | $C_C$             | Table 10, $u_A$         | Force + velocity | -                         | $\pm 0.09$               | 18.8 <sup>c</sup> |
| Max tension (moderate)           | $T_{max,m}$       | Table 9, SD/ $\sqrt{3}$ | Sensor           | $\pm 0.50 \text{ kN}$     | $\pm 1.0 \text{ kN}$     | 4.1               |
| Max tension (extreme)            | $T_{max,e}$       | Table 9, SD/ $\sqrt{3}$ | Sensor + wave    | $\pm 4.45 \text{ kN}$     | $\pm 8.9 \text{ kN}$     | 16.6              |

Notes: <sup>a</sup> Estimated from instrument timing resolution (100 Hz); <sup>b</sup> Wind-3 condition; <sup>c</sup> Current-2 condition. Type A for damping, RAO, and tension is computed from the standard deviation of three repeated tests divided by  $\sqrt{3}$ . Type A for drag coefficients is taken directly from Table 10. Type B is derived from instrument specifications in Table 3. The RAO Type A is based on the measured repeatability of regular wave tests (CV = 2.9–3.0% for roll, 2.4–2.9% for heave). The conservative upper bound for drag coefficients is the expanded uncertainty of Current-2 at approximately 19%.

### 3.5.3. Discussion and Implications

The uncertainty analysis reveals a hierarchy of confidence across the measured quantities. The natural period and damping ratio are the most reliably determined, with relative expanded uncertainties below 8%. The damping ratio uncertainty is dominated by repeatability (Type A), reflecting the inherent variability of free-decay tests. The close agreement between the two damping values ( $0.150 \pm 0.011$  vs.  $0.140 \pm 0.010$ ) confirms that the mooring chain introduces only a marginal increase in total damping.

RAO measurements exhibit excellent repeatability, with relative expanded uncertainties of approximately 3–4% for roll and 3% for heave. The low coefficient of variation (CV = 2.9–3.0% for roll, 2.4–2.9% for heave) reflects the high quality of the wave generation systems and the precision of the optical motion capture system. The facility-to-facility transition at  $T \approx 5.5 \text{ s}$  does not introduce detectable systematic bias, as the RAO curves exhibit smooth continuity (Figure 6).

Drag coefficients exhibit the highest relative uncertainty (14–19%), driven primarily by Type A repeatability in the open-basin and flume environments. The single-fan wind generation system (Section 2.1.4) and the flume circulation system (Section 2.1.5) introduce flow-field variability that is difficult to eliminate through short-duration tests. The measured coefficients ( $C_{D,wind} \approx 0.76 \pm 0.11$ ,  $C_{C,current} \approx 0.44 \pm 0.09$ ) should be applied with appropriate caution for full-scale design, particularly for the current drag coefficient where the relative uncertainty approaches 19%.

Extreme mooring tension has significant uncertainty (16.6%) due to the stochastic nature of extreme wave events in irregular seas. The dynamic amplification factor of  $3.0 \times$  (maximum tension 61.56 kN vs. static pretension 20.5 kN) is based on the largest observed value across three repetitions. For engineering design, it is recommended to apply a safety factor of at least 4–5 on the static pretension to account for both the dynamic amplification and its measurement uncertainty.

It is important to note that the present uncertainty analysis addresses only the quantifiable sources of error. Systematic effects such as Reynolds-number dissimilarity (Section 2.2.4), facility-specific flow characteristics (Sections 2.1.4 and 2.1.5), and the

absence of simultaneous wind–wave–current loading (Section 2.2.1) represent additional, non-quantified uncertainties that should be considered when applying the present results to prototype design.

#### 4. Conclusions

Through systematic physical model tests, this study conducted an in-depth investigation into the hydrodynamic performance of a newly developed 10-m disk-shaped buoy under the actions of wind, waves, and currents. All hydrodynamic parameters reported below are at prototype scale ( $\lambda = 10$ , Froude scaling). The contributions of this study are threefold: (i) the hybrid experimental strategy combining a large-scale wave flume and harbor basin demonstrated feasibility for broad-period buoy hydrodynamic characterization, with internal consistency verified across the facility transition (Section 2.2.1); (ii) the comprehensive dataset provides uncertainty-quantified design parameters for shallow-draft disc buoys ( $D/T \approx 10$ ) at 1:10 scale, with model-scale Reynolds numbers approaching self-similarity for most conditions (Section 2.2.4, Table 4); and (iii) the high intrinsic roll damping ( $\zeta = 0.14$ – $0.15$ , expanded uncertainty  $\pm 0.010$ – $0.011$ ) achievable without appendages offers a passive stability solution for ocean observation platforms, with estimated radiation-damping contribution of 65–75% (Section 3.1, Table 7). The following main conclusions are drawn:

(1) In terms of experimental methodology, the hybrid testing strategy combining a large-scale wave flume and a harbor basin demonstrates practical feasibility for achieving broad-period (2.5–15 s) hydrodynamic characterization of a buoy. The internal consistency across the facility transition is supported by the qualitative continuity of RAO curves (Figure 6), as no overlapping wave-period conditions were available for direct quantitative cross-facility validation. Dedicated overlapping tests or benchmark inter-facility calibration campaigns are therefore required to rigorously confirm the equivalence of wave generation quality and linear response regimes between the two facilities. This limitation should be considered when applying the present RAO dataset to numerical model validation or design optimization. The separate testing strategy additionally precludes direct measurement of nonlinear wind–wave–current interaction effects, which represents a complementary methodological constraint discussed in Section 2.2.1.

(2) In terms of buoy dynamic characteristics, the inherent dynamic parameters of the newly developed shallow-draft disc buoy were experimentally identified through calm-water decay tests. The natural roll period of the buoy is approximately 3.0 s, the natural heave period is approximately 3.6 s, and the non-dimensional roll damping ratio is 0.14–0.15. The high damping level, achieved through the  $D/T \approx 10$  hull geometry without appendages, provides quantified design parameters for shallow-draft disc buoys of this class, complementing prior studies on appendage-dependent damping systems. The tests demonstrate that, under the single-point mooring configuration tested, the mooring system has limited influence on these natural characteristics, which are primarily determined by the mass distribution and hydrodynamic geometry of the buoy itself. This observation may not apply to spread-mooring or taut-mooring configurations, which could introduce significant rotational and vertical stiffness coupling, potentially altering the natural frequencies and dynamic response. The measured roll damping ( $\zeta = 0.14$ – $0.15$ , expanded uncertainty  $\pm 0.01$ – $0.011$ ) is favorable for motion stability within the tested conditions ( $Re_D = 2 \times 10^5$ – $6 \times 10^5$  for current,  $1.1 \times 10^5$ – $3.5 \times 10^5$  for wind). This damping level, which exceeds that of typical ship-type buoys ( $\zeta < 0.1$ ) and Spar-type buoys ( $\zeta = 0.08$ – $0.12$ ), suggests potential benefits for platform stability, though full-scale validation is recommended to confirm the Reynolds-number scaling of viscous damping contributions.

(3) In terms of motion response laws, the regular wave tests reveal pronounced characteristics of the buoy motion. Pronounced resonant peak occurs in roll motion near the wave period of 3.0 s, with RAO peak values exceeding  $26^\circ/\text{m}$ , which is in excellent agreement with the natural period; this period should be avoided in operational design. The heave motion exhibits favorable wave-following regime for periods greater than 5 s, with RAO values approaching 1.0.

(4) In terms of extreme sea-state performance, the test results indicate that the buoy survived the tested extreme sea state ( $H_s = 13.8\text{ m}$ ,  $T_p = 16.1\text{ s}$ ) without capsizing tendency. However, the dynamic amplification factor of  $3.0\times$  (maximum tension  $61.56 \pm 8.9\text{ kN}$ ) suggests that the mooring system design should incorporate adequate safety margins for more severe conditions. Under extreme sea states ( $H_s = 13.8\text{ m}$ ,  $T_p = 16.1\text{ s}$ ), the maximum roll angle is  $21.30^\circ$  and the maximum heave amplitude is 8.96 m, with no tendency toward capsizing observed. The mooring system exhibits a significant dynamic amplification effect, with the maximum dynamic line tension reaching 61.56 kN, approximately 3.0 times the initial pretension; this dynamic factor serves as the controlling basis for mooring system strength design.

(5) In terms of environmental load characteristics, preliminary load design parameters were obtained through tests. The mean wind drag coefficient of the buoy is 0.76, and the mean current drag coefficient is 0.44, providing preliminary design parameters for load coefficients of this type of floating body, with the caveat that the drag coefficient uncertainty (14–19%) warrants conservative application in full-scale design. It is additionally noted that these coefficients were measured under fixed-body conditions and should be applied as conservative estimates for freely moving buoys, with potential corrections for relative-velocity effects in refined analyses.

Additionally, the measured drag coefficients enable preliminary estimation of static mean surge offsets under wind and current loading (20–30 m for extreme individual loads, 40–50 m for combined loading with nonlinear catenary correction). However, dynamic surge RAOs and coupled drift–wave interaction were not measured, representing a limitation for applications requiring directional wave spectrum estimation or precise mooring offset prediction. Future work should include dedicated surge characterization in the experimental matrix and coupled wind–wave–current basin tests to validate the superposition-based drift estimates.

This study not only provides complete data support and theoretical basis for the design optimization and safe operation of this specific buoy, but its systematic experimental methodology, comprehensive database, and in-depth understanding of dynamic characteristics also offer valuable reference for the research and development of similar ocean buoys. It is recommended that subsequent research focus on the nonlinear dynamic optimization of mooring systems to further enhance the survival capability of buoys under extreme sea states, and conduct comprehensive uncertainty analyses to quantify the effects of measurement errors, scaling effects, and statistical variability on the reliability of experimental results. It is important to acknowledge that simultaneous wind–wave–current loading was not tested in this study; instead, wind, wave, and current effects were investigated separately and then comprehensively assessed. Future work should address the nonlinear interactions under simultaneous multi-environmental loading. Future work should also include detailed phase analysis of the heave response relative to the incident wave, which would provide additional validation of the linear hydrodynamic assumptions and enhance the completeness of the motion response dataset. It is explicitly clarified that the disc-shaped hull form itself is not claimed as novel; rather, the novelty resides in the hybrid testing methodology, the uncertainty-quantified hydrodynamic dataset for this specific

$D/T \approx 10$  geometry at large scale, and the demonstrated feasibility of achieving high roll damping through extreme shallow-draft design without appendages.

Finally, it is important to reiterate that the comprehensive assessment of separate wind, wave, and current effects relies on a linear superposition assumption. While this approach provides practical preliminary design parameters, the unquantified nonlinear interactions under simultaneous loading represent a key limitation. Future work should prioritize coupled wind–wave–current basin tests or high-fidelity coupled numerical simulations to validate and, if necessary, correct the superposition-based estimates for operational and survival conditions.

**Take-home message:** This study demonstrates that shallow-draft disc buoys ( $D/T \approx 10$ ) can achieve high intrinsic roll damping without appendages through large-scale hybrid testing, providing directly applicable design parameters for ocean observation platforms and a validated methodology for broad-period hydrodynamic characterization.

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**Data Availability Statement:** The data presented in this study are available upon reasonable request from the corresponding author. Restrictions apply to the availability of raw test data due to institutional policy.

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