

Article

Numerical Simulation of Hydrodynamic Performance of an Offshore Oscillating Water Column Wave Energy Converter Device

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Abstract: Wave energy, as a renewable energy source, plays a significant role in sustainable energy development. This study focuses on a dual-chamber offshore oscillating water column (OWC) wave energy device and performs numerical simulations to analyze the influence of chamber geometry on hydrodynamic characteristics and wave energy conversion efficiency. Unlike existing studies primarily focused on single-chamber configurations, the hydrodynamic characteristics of dual-chamber OWCs are relatively underexplored, especially regarding the impact of critical design parameters on performance. In this study, STAR-CCM+ V2302 software (Version 2410, Siemens Digital Industrial Software, Plano, TX, USA) is utilized to systematically evaluate the effects of key design parameters (including turbine configuration, mid-wall draught depth, and wall angles) on the hydrodynamic performance, wave energy capture efficiency, and wave reflection and loading characteristics of the device. The findings aim to provide a reference framework for the optimal design of dual-chamber OWC systems. The results show that the dual-chamber, dual-turbine (2C2T) configuration offers a 31.32% improvement in efficiency compared to the single-chamber, single-turbine (1C1T) configuration at low wave frequencies. In terms of reducing wave reflection and transmission, the 2C2T configuration outperforms the dual-chamber, single-turbine configuration. When the wall angle increases from 0° to 40°, the total efficiency increases by 166.37%, and the horizontal load decreases by 20.05%. Additionally, optimizing the mid-wall draught depth results in a 9.6% improvement in efficiency and a reduction of vertical load by 11.69%.

Keywords: numerical modelling; oscillating water column; hydrodynamic properties; wave energy conversion rate; wave loads



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1. Introduction

With the growing emphasis on ocean development and environmental sustainability, wave energy is increasingly recognized as a viable solution to the global energy crisis. In line with global efforts toward sustainability, diverse renewable energy solutions have been thoroughly investigated and implemented [1]. Among the options, wave energy is particularly notable for its substantial potential in coastal regions, where natural phenomena such as refraction and shoaling amplify wave energy, making coastal regions ideal locations for wave energy converters (WECs) [2]. The oscillating water column (OWC) has been identified as a pivotal technology in wave energy conversion resulting from its superior efficiency in capturing energy, robust design, and operational simplicity. Increasing global interest has spurred substantial research and development efforts focused on OWC systems, establishing them among the leading researched and dependable approaches

for wave energy utilization [3,4]. A wealth of research highlights the capability of OWC devices to capture wave energy [5], with a range of prototypes already constructed, further demonstrating their practical viability for renewable energy applications [6].

Extensive investigations have explored the hydrodynamic behavior of oscillating water column (OWC) devices, utilizing a wide range of methodologies, including analytical, numerical, and experimental approaches [7–11]. Most of these investigations have centered on single-chamber OWC systems. Evans [12] employed matched asymptotic expansions to derive approximate solutions for capturing wave energy in a floating OWC system linked to a spring-dashpot mechanism, with the assumption of minimal spatial variation in the air chamber's free surface. Similarly, Malara and Arena [13] constructed a theoretical framework for analyzing a 2-D U-shaped OWC under random wave conditions. Their results revealed that the inner and outer chambers predominantly play a role in wave energy conversion efficiency across distinct frequency ranges. However, the applicability of analytical methods is largely constrained to OWCs with simple geometries.

For more complex OWC configurations, numerical analysis becomes indispensable. Cong et al. [14] formulated a frequency-domain 3-D boundary element approach to assess the hydrodynamic behavior of floating OWC devices under wave motion. Their findings demonstrated that floating OWCs broaden the spectrum of efficient energy conversion, enhancing their adaptability to diverse oceanic conditions. Cheng et al. [15] used a higher-order boundary element method to examine the hydrodynamic characteristics of a hybrid wave energy converter, integrating an OWC with an oscillating buoy into a π -type floating breakwater.

The advent of computational fluid dynamics (CFD) has further revolutionized the exploration of OWCs, enabling more comprehensive analyses of the complex hydrodynamic interactions within the chamber and its surrounding environment compared to traditional potential theory-based models. Opoku et al. [16] offered an in-depth analysis of state-of-the-art numerical tools for estimating OWC hydrodynamic performance, offering a comparative analysis of various CFD software and solvers that highlights their advantages and limitations in capturing the intricate dynamics of OWC systems.

The dual-chamber oscillating water column (OWC) device has been proposed as an innovative solution to enhance wave energy conversion efficiency and enable broader energy capture across varying oceanic conditions. This design is applicable to both onshore and offshore installations, offering significant versatility in wave energy extraction. Extensive studies have explored the hydrodynamic behavior of dual-chamber systems under multiple wave conditions and air chamber geometries. Research by Wang et al. [17], Ning et al. [18,19], Elhanafi et al. [20], and Yu et al. [21] demonstrated that the dual-chamber configuration outperforms single-chamber devices, particularly near the resonant frequency, emphasizing the pivotal role of the center wall in enhancing hydrodynamic efficiency.

Further investigations into multi-OWC platforms by Zheng et al. [22] and Simeon et al. [23] employed semi-analytical models to address wave radiation and diffraction phenomena, revealing that multi-OWC systems achieve superior efficiency compared to single OWCs or closely spaced OWC arrays. Hsieh et al. [24] performed experimental evaluations of a floating dual-chamber OWC device, concluding that this design not only improves power generation but also stabilizes energy output, a key advantage for practical applications. Fenu et al. [25] proposed a novel concept of a hybrid platform, which integrates a spar buoy wind turbine with three OWCs. The study conducted experimental investigations to analyze the hydrodynamic behavior of the OWCs' air chambers under both wave and wind loads, as well as the impact of wind action and air chamber damping variation on the dynamic behavior of the floating platform, leading to the power extraction.

In recent years, a growing attention on experimental apparatus designed to optimize system performance under varying environmental conditions [26]. Zhao et al. [27] experimentally analyzed the hydrodynamic characteristics of single-, dual-, and triple-chamber OWC-breakwater systems. Their analysis, focusing on parameters such as capture width ratio, wave reflection and transmission metrics, energy dissipation efficiency, and opera-

tional frequency range, confirmed that multi-chamber designs enhance energy extraction, reduce long-wave energy, and improve performance by broadening operational bandwidths. These findings underscore the potential of dual-chamber OWCs in advancing marine renewable energy technologies and expanding the operational viability of wave energy converters. Howe et al. [28] investigated the energy extraction performance of multiple OWCs integrated within a floating breakwater. They emphasized the significance of OWC device spacing in the design of multi-device structures and proposed new insights for the development of floating offshore multi-purpose structures.

The application of the wave-to-wire (WtW) model in the oscillating water column (OWC) system is an important direction in current wave energy conversion technology research. This model provides a valuable tool for designing an effective OWC-WEC-WtW system by fully coupling the energy capture chamber, pulse turbine, and permanent magnet synchronous generator (PMSG) in the OWC system. Further investigations into the dynamics and control of air turbines in OWC wave energy converters were conducted by Henriques et al. [29], employing a detailed analysis and case study of the Mutriku wave power plant, revealing the significant impact of turbine control parameters on the performance of Wells and biradial turbines. Ciappi et al. [30] applied an integrated wave-to-wire modelling approach for the preliminary design of oscillating water column systems, demonstrating through analytical wave-to-wire models that multi-OWC systems achieve higher efficiency in moderate wave climates compared to single OWCs or closely spaced arrays. Fenu et al. [31] developed a novel real-time simulator for the Wells turbine used in OWC, providing a new high-precision simulation method for experimental testing of wave energy converters.

The aforementioned studies have substantially advanced the optimization of oscillating water column (OWC) devices. However, most existing studies have predominantly concentrated on single-chamber configurations, leaving the hydrodynamic characteristics of dual-chamber OWCs relatively underexplored, especially concerning the influence of critical design parameters. Naik [32] identified that the angles of slanted front walls and sloping bottoms are pivotal factors for maximizing power extraction in dual-chamber OWC devices.

In this study, the STAR-CCM+ code is utilized to perform a comprehensive analysis of the influence of critical structural parameters on the hydrodynamic behavior and wave energy capture efficiency of dual-chamber OWC devices. Specifically, this investigation seeks to evaluate the dual-turbine configuration's ability to mitigate horizontal wave loads and wave reflections, along with the impact of mid-wall draught depth and wall angles on the structural bearing capacity of the device. The findings aim to establish a reference framework for the optimal design and application of dual-chamber OWC systems.

The remainder of this article is structured as follows: Section 2 presents the mathematical framework and numerical model setup used in the study. Section 3 discusses the validation of the numerical model against experimental and analytical benchmarks. Section 4 presents a detailed multi-parameter evaluation of the hydrodynamic behavior of dual-chamber OWCs. Finally, Section 5 summarizes the key findings and provides insights for future research and practical application.

2. Numerical Method

2.1. Governing Equations

The viscous flow model utilized in this study was developed using the STAR-CCM+ software and is based on the continuity and Reynolds-Averaged Navier-Stokes (RANS) equations to represent the flow behavior of an incompressible fluid. The model assumes the fluid within the OWC chamber to be incompressible, as the influence of air compressibility is negligible for small-scale OWC devices [33]. The numerical calculations are performed using the RANS-VOF solver, with the finite volume method applied to discretize the integral form of the RANS equations. A predictor-corrector algorithm is employed to ensure the coupling between the continuity and momentum governing equations. This

study uses the improved K-Omega model variant SST K-Omega (shear stress transport model), which can solve the transport equations for turbulent kinetic energy and specific dissipation rate to determine turbulent eddy viscosity.

The Volume of Fluid (VOF) method is implemented to accurately track and capture the displacement of the free surface, which is crucial for simulating the interaction between water waves and the OWC structure [34]. The governing equations for incompressible viscous flow are expressed as follows:

$$\nabla \cdot \vec{U} = 0, \quad (1)$$

$$\frac{\partial \rho \vec{U}}{\partial t} + \nabla \cdot (\rho \vec{U} \vec{U}) - \nabla \cdot (\mu_{eff} \nabla \vec{U}) = -\nabla p^* - \vec{g} \cdot \vec{X} \cdot \nabla \rho + \nabla \vec{U} \cdot \nabla \mu_{eff} + \sigma \kappa \nabla \alpha, \quad (2)$$

where the velocity vector $\vec{U}$ is defined with two components corresponding to the longitudinal and vertical axes for a two-dimensional problem. The symbol ρ represents the fluid density, which is 1000 kg/m^3 , and μ_{eff} denotes the effective dynamic viscosity. The hydrodynamic pressure is denoted by p^* , while g represents the gravitational acceleration. The final term in Equation (2) accounts for surface tension effects, where σ represents the surface tension coefficient, κ denotes the interface curvature, and α is the phase indicator function.

In this study, the first-order Stokes wave theory is applied to produce regular waves, specifying the free surface elevation and wave velocity components as follows:

$$\eta = \frac{H}{2} \cos(kx - \omega t), \quad (3)$$

$$u_x = \frac{H\omega \cosh k(z+h)}{2\sinh kh} \cos(kx - \omega t), \quad (4)$$

$$u_z = \frac{H\omega \sinh k(z+h)}{2\sinh kh} \sin(kx - \omega t), \quad (5)$$

where η represents the free surface elevation of the incoming waves and H indicates the height of the incident wave. The parameters ω and k correspond to the angular frequency and wave number of the waves, respectively. The velocities of water particles in the x and z directions are represented by u_x and u_z , with z representing the vertical distance from the still water surface, with h signifying the water depth.

The hydrodynamic efficiency is assessed through the capture width ratio (CWR), described as the ratio of power absorbed by the wave energy converter (WEC) to the wave power arriving at the device [35]. Mathematically, the hydrodynamic efficiency ζ is expressed as:

$$\zeta = \frac{P_T}{P_w a}, \quad (6)$$

where P_T signifies the power absorbed by the wave energy converter (WEC) and P_w indicates the wave power per unit width along the wave crest direction. The variable a stands for the characteristic dimension of the WEC, usually identified as the device's width. The wave power, derived from the first-order Stokes wave theory, is calculated as:

$$P_w = \frac{1}{2} \rho g A_i^2 c_g, \quad (7)$$

$$c_g = \frac{w}{2k} \left(1 + \frac{2kh}{\sinh kh} \right), \quad (8)$$

where A_i denotes the amplitude of the incoming wave, C_g the group velocity of the wave, and ρ and g the density of water and the acceleration due to gravity, respectively. The power captured by an air turbine is calculated as follows:

$$P_T = QP, \quad (9)$$

where P represents the pressure of the chamber and Q denotes the air volume flow rate passing through the turbine.

Therefore, the overall efficiency of the dual-chamber OWC system was determined by combining the efficiencies of the front and rear chambers, as follows:

$$\zeta = \zeta_1 + \zeta_2, \quad (10)$$

where ζ_1 and ζ_2 represent the energy conversion efficiencies of the front and rear chambers of the dual-chamber OWC device, respectively.

2.2. Numerical Model Setup

Figure 1 illustrates a schematic representation of the OWC device. For clarity, the OWC configuration is denoted as nCmT, where n and m indicate the number of chambers and turbines, respectively. For instance, “2C2T” represents a dual-chamber, dual-turbine OWC device. Numerical modeling at a scale of 1:50 (based on Froude’s similitude law) is calculated under the assumption of incompressible air. The numerical wave tank (NWT) utilized in this study has a length of $10L$ (L is the incident wavelength) and a height of $2h = 2.4$ m (h is the water depth). For the two-dimensional problem analyzed here, the width of the wave tank is set to 1 m. A wave damping zone, with a length of $1.5L$, is applied at the pressure outlet boundary to absorb reflected waves from both the OWC structure and the outlet boundary.

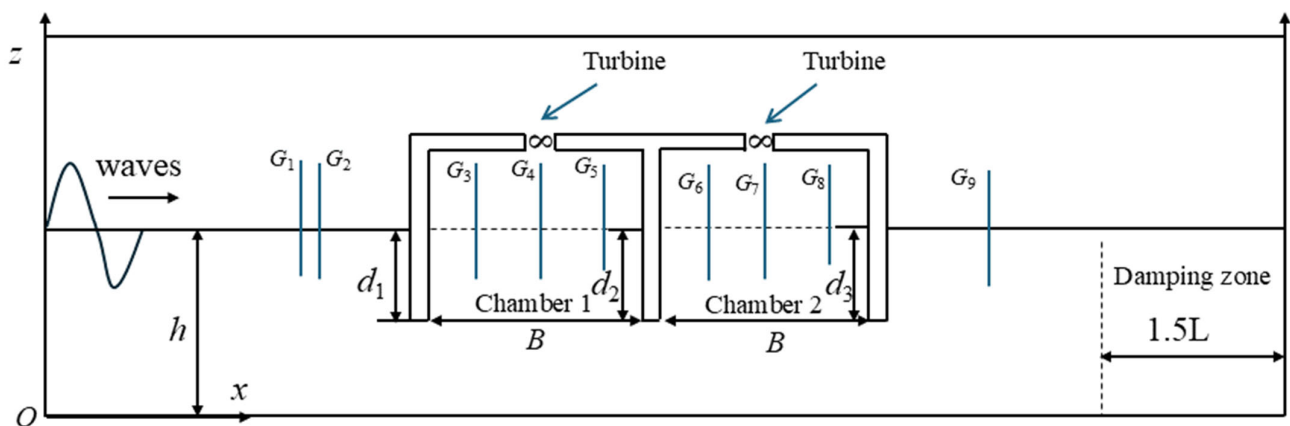


Figure 1. Schematic diagram of a two-dimensional OWC device.

The OWC model is located $5L$ away from the inlet boundary on the left side of the NWT. The boundary conditions are established as follows: the left boundary of the NWT serves as the velocity inlet, while the top and right boundaries function as pressure outlets. The bottom boundary is designated as a sliding wall, and the front and rear boundaries are defined as symmetric planes.

Three wave gauges were positioned at the center of Chamber 1 and Chamber 2, as well as on either side at intervals of $1/3B$. The free surface elevation inside Chamber 1 (η_{OWC1}) and Chamber 2 (η_{OWC2}) was observed at these three points, and the findings were averaged to obtain a representative value. Two wave gauges are placed at L and $0.9L$ from the front wall of the chamber to separate the incident and reflected waves, and one wave gauge is placed at L from the rear wall of the chamber to measure the transmitted wave height. According to the “two-point method” proposed by Goda et al. [36] to get the

incident wave height H_i , reflected wave height H_r , and transmitted wave height H_t , the reflection coefficient C_r and transmission coefficient C_t can be calculated, and the related equations are as follows:

$$C_r = H_r / H_i \quad (11)$$

$$C_t = H_t / H_i \quad (12)$$

The differential air pressure (P) inside the chamber was determined from readings at four specific locations: two within the pneumatic chamber (equidistantly situated on both sides of the power take-off system, halfway between the center of the orifice and the chamber edge) and two outside the chamber along the upper boundary. The mean pressure difference between the interior and exterior monitoring points was then computed as the value of P .

Figures 1 and 2 illustrates the wave tank and the schematic configurations of the two-chamber OWC device, with the chambers designated as Chamber 1 (front chamber) and Chamber 2 (rear chamber). The numerical model setup in this paper is consistent with the experimental arrangement of Elhanafi [37]. The depth of water is set to $h = 1.2$ m, and the wave tank height is 2.4 m. The internal heights of the OWC chambers are $h_c = 0.2$ m, and the wall thickness is $C = 0.02$ m. The diameter of the upper orifice is $e = 0.048$ m, corresponding to an orifice opening ratio of $e/B = 1.5\%$. The chamber width is $B = 0.32$ m. The draft depths of the front and rear walls are $d_1 = 0.15$ m and $d_3 = 0.35$ m, respectively.

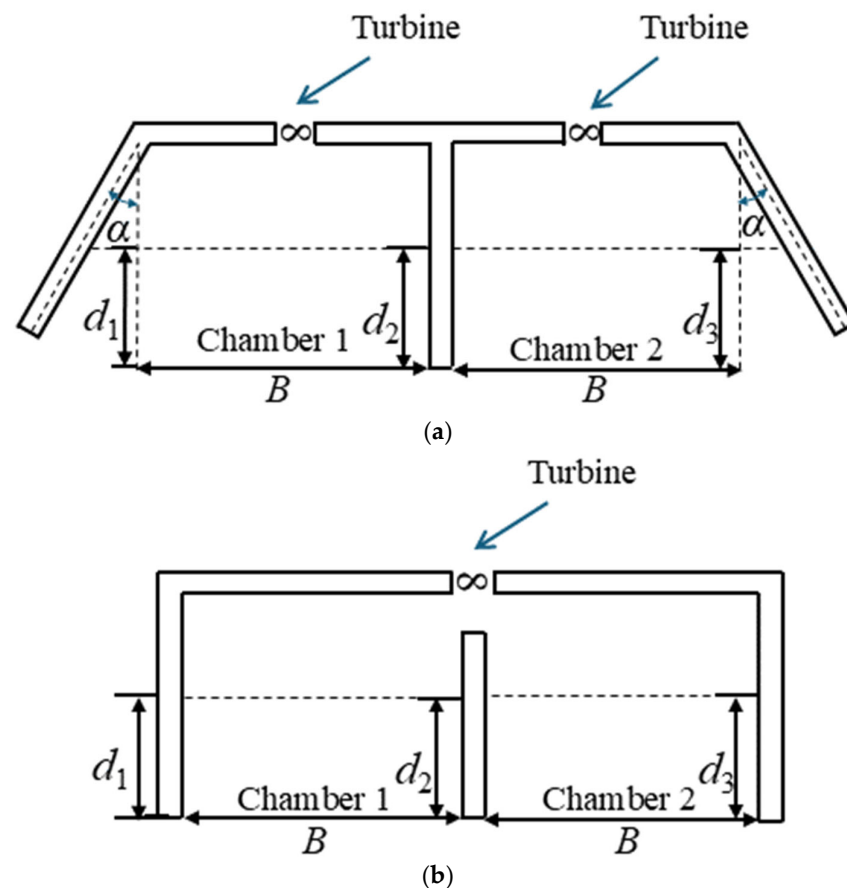


Figure 2. Diagram of the offshore-stationary OWC devices. (a) Dual chambers dual turbines OWC with sloping wall (2C2T); (b) Dual chambers single turbine OWC (2C1T).

The aforementioned parameters are treated as constant values throughout the simulation. A comprehensive overview of all the parameters examined in this study is presented in Table 1.

Table 1. Test variables and conditions.

Test	Type of Chamber	Barrier Wall Draught d_2 (m)	Wall Angle α ($^\circ$)	T (s)	H (m)	B/L
1	2C2T	0.25	0	1.2~1.8	0.04	0.04~0.14
2	2C2T	0.35	0			
3	2C2T	0.15	0			
4	2C1T	0.35	0			
5	2C2T	0.35	15			
6	2C2T	0.35	30			
7	2C2T	0.35	45			

3. Model Validation

To confirm the accuracy and reliability of the numerical modeling, this subsection develops a numerical model based on the physical experiments conducted by Elhanafi et al. [34]. The model adopts the following parameters: chamber width $B = 0.3$ m, front and rear wall drafts $d_1 = d_2 = 0.2$ m, chamber height $h_c = 0.15$ m, orifice width $e = 0.009$ m, wall thickness $C = 0.012$ m, wave height $H = 0.05$ m, and water depth $h = 1.5$ m.

3.1. Convergence Test for Numerical Computational Grids

To minimize the effects caused by wave reflections from the OWC model and wave attenuation, mesh refinement was applied to the free surface region up to a height of $3H$ for the 2D model. The entire fluid domain was discretized using the “Trimmed Mesher” and surface remesher in the STAR-CCM+ automated meshing framework. A base grid with a cell size of 200 mm was employed in regions distant from the free surface and the OWC structure, with gradual refinement applied near the free surface region and the OWC structure. In the free surface region, 20 and 80 grids per wave height and wavelength were adopted, as recommended by ITTC and CD-Adapco. The grid aspect ratio ($\Delta x/\Delta z$) was maintained at a maximum value of 8, ensuring proper resolution of the computational domain. The mesh refinement levels were further defined up to level 5 ($200 \text{ mm} \times 2^{-5} = 6.25 \text{ mm}$) at the OWC wall and level 8 near the orifice. The time step (Δt) was determined based on the Courant number (C_0) condition, which was constrained to $C_0 \leq 1$. A total of 1000 time steps per wave period were used to ensure the stability and accuracy of the simulation results.

Three grid sizes were selected for validation: wave heights discretized into 10, 20, and 30 grids ($\Delta z = H/10, H/20, H/30$), referred to as mesh1, mesh2, and mesh3, respectively. The grid resolution that provided a balance between computational accuracy and cost was selected as the final configuration. The parameters for these three grid schemes are summarized in Table 2.

Table 2. Grid parameters.

Index	mesh1		mesh2		mesh3	
Wave Refinement (m)	X	Z	X	Z	X	Z
	0.028	0.005	0.02	0.0025	0.013	0.0017
Total Number of Grids	149,188		386,108		671,168	

Figure 3 illustrates the grid structure for the single-chamber configuration, while Figure 4 presents a comparison of wave height and chamber pressure between the numerical simulation outcomes from this study and the relevant experimental data from Elhanafi et al. [34]. The results indicate that all three grid configurations effectively support accurate numerical simulations, with minimal errors observed. The CFD results are quantitatively compared to physical measurements by calculating the normalized root mean square error (NRMSD) and the results are shown in Table 3. Considering both computational efficiency

and accuracy, mesh 2 was selected for the numerical calculations conducted in this study. The red line indicates the mesh refinement region, where the computational grid is densified to improve accuracy in this area.

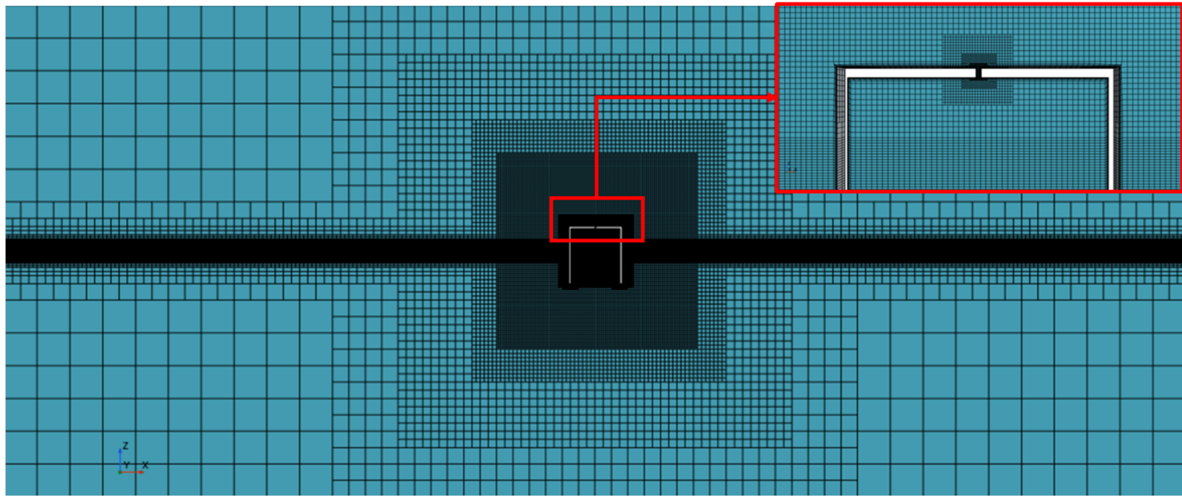


Figure 3. Mesh of single chamber.

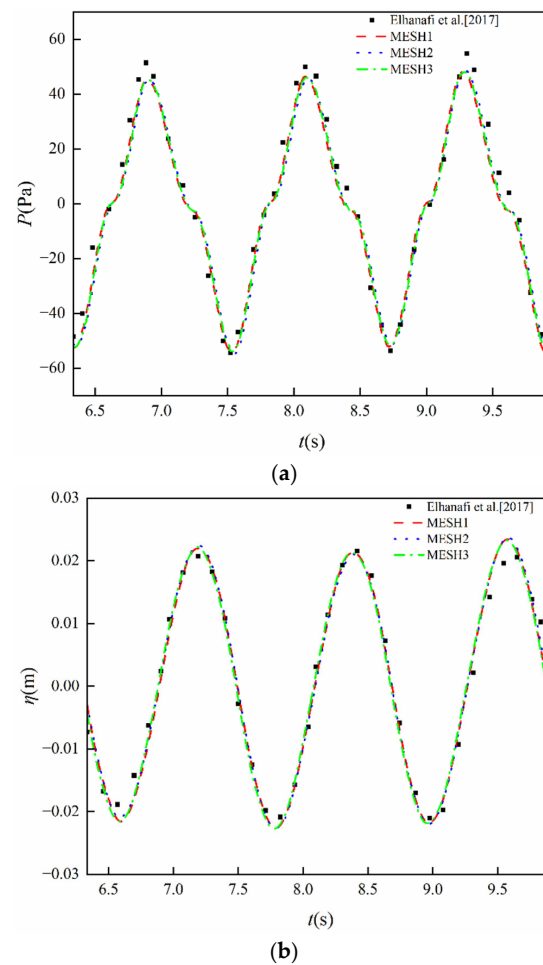


Figure 4. Comparison of numerical simulation results with physical experimental data from Elhanafi et al. [37] for $H = 0.05$ m and $T = 1.2$ s. (a) Wave height at the center of the chamber; (b) Pressure in the chamber.

Table 3. NRMSE and Computing Time.

Index	mesh1	mesh2	mesh3
NRMSE of η_{OWC}	4.43%	4.39%	4.36%
NRMSE of P	4.76%	4.56%	4.56%
Computing Time	4 h 36 min	5 h 02 min	9 h 58 min

3.2. Verification with Single OWC Results

Figures 5 and 6 presents a comparison between the numerical findings of this study and the experimental observations reported by Elhanafi et al. [37,38], illustrating the changes in wave energy conversion efficiency ζ with KB (where $K = \sigma^2/g$ and B is the width of the chamber). The comparison is performed under operational conditions of $T = 0.9$ s to $T = 2.0$ s. The results show good agreement and stability, demonstrating that the numerical model established in this study is reliable and accurate.

$$\vec{F}_s = \rho g h A \cdot \vec{n}, \quad (13)$$

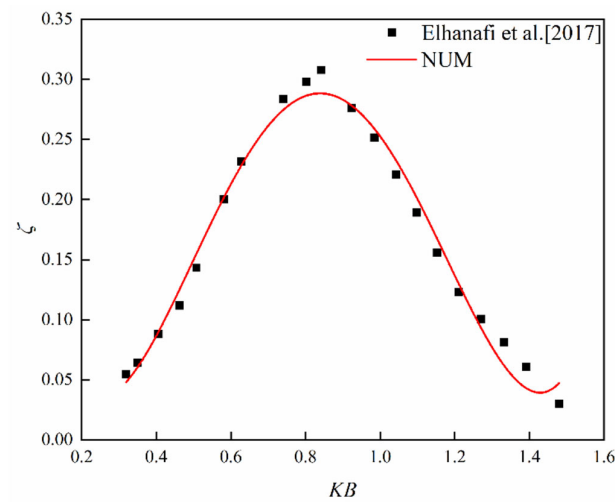


Figure 5. Verification results with physical experimental data from Elhanafi et al. [37] of energy conversion efficiency of single chamber OWC device.

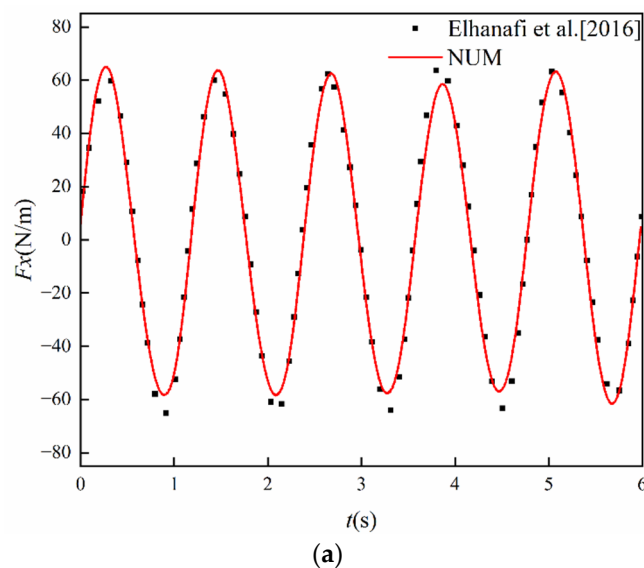


Figure 6. Cont.

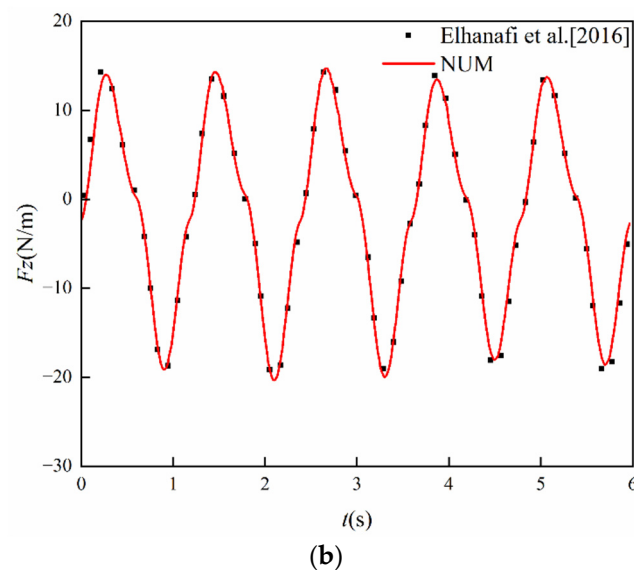


Figure 6. Comparison of numerical simulation of wave loads with physical experiment results of Elhanafi et al. [38]. (a) Horizontal wave loads; (b) Vertical wave loads.

This equation represents the hydrostatic load formula, which must be subtracted when calculating the wave load acting on the OWC device. Based on the experimental data provided by Elhanafi et al. [38], the working condition of $T = 1.2$ s and $H = 0.05$ m was selected for validation. Graphs of the device's single-width horizontal wave load and single-width vertical wave load were generated from the simulation outcomes were compared with the experimental observations. The comparison shows that the simulation results align well with the experimental measurements, confirming the accuracy of the numerical model. This model can therefore be applied for subsequent studies on the wave-induced forces acting on OWC devices.

4. Results and Discussion

In the simulation, the effects of varying wave period (T), incident wave height (H), intermediate wall draft (d_2), wall angle (α), and dual-chamber structural configurations (2C2T and 2C1T) on the hydrodynamic efficiency and wave-induced loads of the dual-chamber OWC wave energy conversion device are analyzed.

4.1. Effects of Types of Chamber Structure

It is well established that increasing the number of chambers can enhance the wave energy conversion efficiency of the system. However, it should be emphasized that a dual-chamber OWC system typically requires two turbines, which inevitably increases the overall cost. Therefore, further investigation into the feasibility of sharing a single turbine between the two chambers is essential.

This section examines two working conditions (Conditions 1 and 4). The single-chamber configuration (with a front wall draft of 0.15 m, a rear wall draft of 0.35 m, and all other parameters unchanged) is simulated for comparison, aiming to evaluate the feasibility of the single-turbine structure.

According to the data presented in Figure 7, the following insights can be derived that within the range of $kh = 1.24$ – 1.79 , the 2C1T structure (dual-chamber, single-turbine) demonstrates higher efficiency compared to both the 1C1T (single-chamber, single-turbine) and 2C2T (dual-chamber, dual-turbine) configurations. However, outside of this range, its efficiency is lower than that of the other two structures. Specifically, at $kh = 1.79$, the efficiency of the 2C1T structure surpasses that of the 1C1T and 2C2T structures by 0.43 and 0.21, corresponding to increases of 78.16% and 27.41%, respectively. This indicates that, at

this frequency, the dual-chamber configuration can effectively capture more wave energy even with a single turbine.

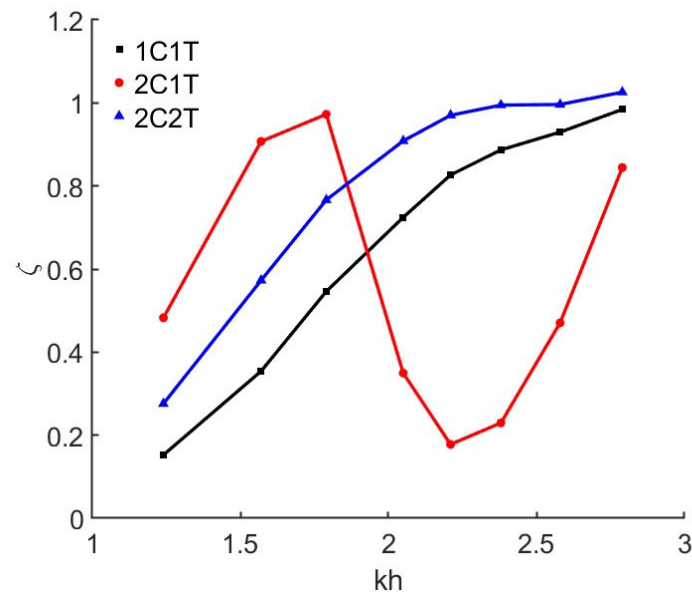


Figure 7. Comparison of wave energy conversion efficiency of different chamber types.

However, as the frequency increases to $kh = 2.21$, the efficiency of the 2C1T structure decreases significantly, being 78.49% lower than the 1C1T structure and 81.67% lower than the 2C2T structure. This suggests that at higher frequencies, the 1C1T and 2C2T configurations exhibit superior hydrodynamic performance.

Therefore, when designing OWC devices, it is crucial to consider not only the number of chambers and turbines but also to refine the device specifications according to the target frequency range to attain the highest wave energy conversion efficiency.

According to Figures 8 and 9, it is observed that although all three structures share the same orifice opening ratio, the orifice area of the 2C1T structure is twice as large as that of the other two structures. This results in significantly lower internal pressure for the 2C1T structure compared to the 1C1T and 2C2T configurations, while its orifice flow rate is markedly higher than that of the other structures.

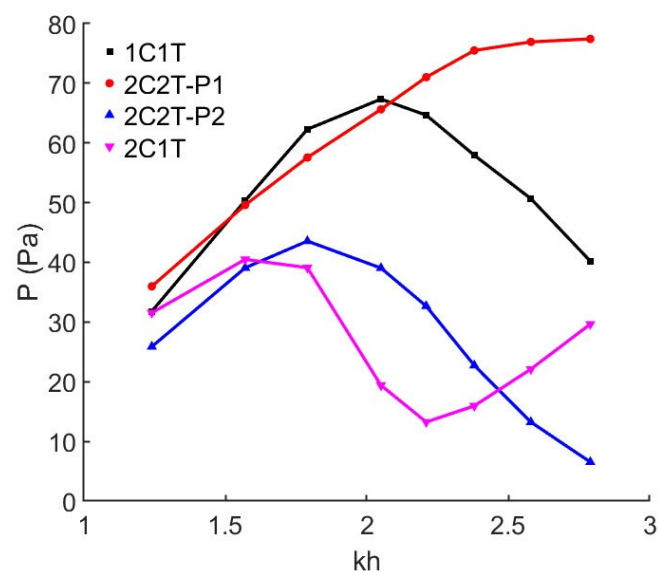


Figure 8. Comparison of Pressure of different chamber types.

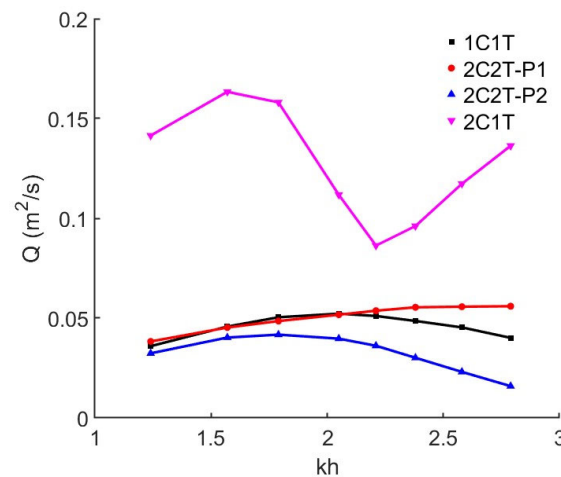


Figure 9. Comparison of orifice flow rates of different chamber types.

Additionally, the results indicate that for the 1C1T and 2C1T configurations, the efficiency of the device increases substantially with rising internal pressure. However, the corresponding increase in orifice flow rate does not improve efficiency to the same extent. This suggests that variations in pressure exert a more pronounced influence on device efficiency during wave energy conversion. Increasing the flow rate, on the other hand, is not always effective in enhancing wave energy conversion efficiency and may lead to energy losses and reduced overall performance.

In summary, while increasing the number of chambers and turbines generally improves the energy conversion efficiency of OWC devices, an excessive number of turbines may negatively impact efficiency in certain frequency conditions.

To further investigate the reasons behind the reduced efficiency of the 1C1T structure, a comparison of vortex distributions for the three configurations—1C1T, 2C2T, and 2C1T—was conducted. The vortex distributions around the 2C1T and 2C2T structures at $kh = 2.38$ and $kh = 1.57$ ($T = 1.3$ s and $T = 1.6$ s) are shown in Figure 10. The results indicate that the 2C2T structure adapts better to the rapidly fluctuating airflow conditions associated with short-period waves ($T = 1.3$ s). The dual-turbine design contributes to stabilizing the airflow, reducing vortex intensity at specific points, and minimizing energy dissipation. This, in turn, enhances the efficiency of energy conversion by enhancing the effective energy of the airflow entering the turbines.

In contrast, under long-period waves ($T = 1.6$ s), the airflow becomes more stable. The single-turbine centralized airflow characteristic of the 2C1T structure moderates vortex intensity, facilitating the efficient transfer of low-frequency wave energy and achieving relatively high conversion efficiency despite a certain degree of energy loss.

The differences in wave energy conversion efficiency between the 2C1T and 2C2T structures under varying wave periods are primarily attributed to the distinct adaptations in airflow concentration, vortex strength, and airflow distribution. It can also be observed that the 2C1T structure demonstrates advantages in wave energy capture under lower-frequency conditions, while the 2C2T structure is better suited for high-frequency wave energy capture.

The impact of various structural designs on the device is analyzed in terms of wave loads. The horizontal load and vertical load of dual-chamber configurations (2C2T and 2C1T) are examined based on the results shown in Figure 11. The horizontal load of the 2C2T structure is consistently smaller than that of the 2C1T structure across the entire frequency range of the experimental incident wave. Similarly, the vertical load of the 2C2T structure is lower than that of the 2C1T structure within the interval $kh = 1.24$ – 1.79 . However, outside this range, the vertical load of the 2C2T structure exceeds that of the 2C1T structure.

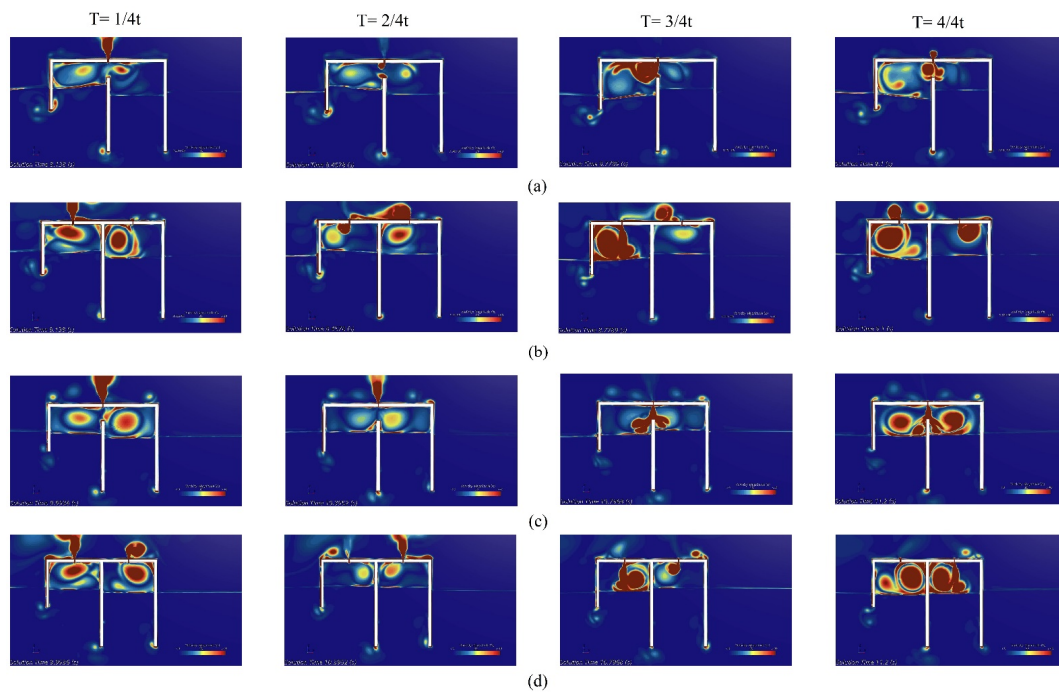


Figure 10. Vortex distribution of different structures. (a) Vortex distribution on 2C1T at $T = 1.3$ s. (b) Vortex distribution on 2C2T at $T = 1.3$ s. (c) Vortex distribution on 2C1T at $T = 1.6$ s. (d) Vortex distribution on 2C2T at $T = 1.6$ s.

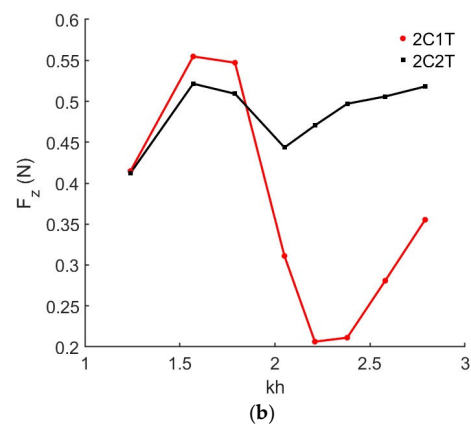
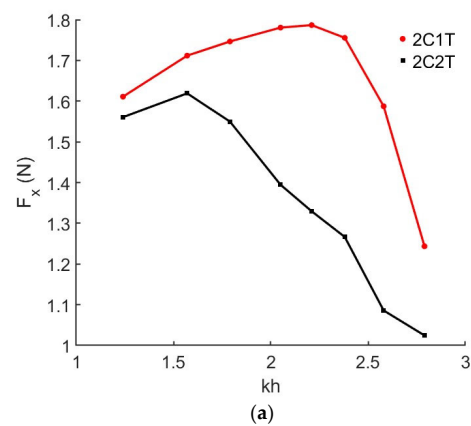


Figure 11. Comparison of loads of different chamber types. (a) Horizontal wave loads; (b) Vertical wave loads.

For example, at $kh = 2.38$, the horizontal load of the 2C2T structure is 1.27 N, while its vertical load is 0.50 N. Compared to the 2C1T structure, the horizontal load of the 2C2T configuration is reduced by 27.87%, whereas the vertical load increases by approximately 135%.

Figures 12 and 13 compare the reflection and transmission coefficients of different structural configurations. As shown in the figures, the reflection coefficient of the 2C2T structure remains relatively stable across the range of kh , while the reflection coefficient of the 2C1T structure exhibits greater variability, reaching a peak at $kh = 2.38$. Overall, the reflection coefficients of the 2C2T structure are generally lower than those of the 2C1T structure throughout the kh interval studied in the experiments, particularly in the $kh > 1.79$ range, where the peak reflection coefficient of the 2C2T structure is approximately half of that of the 2C1T structure.

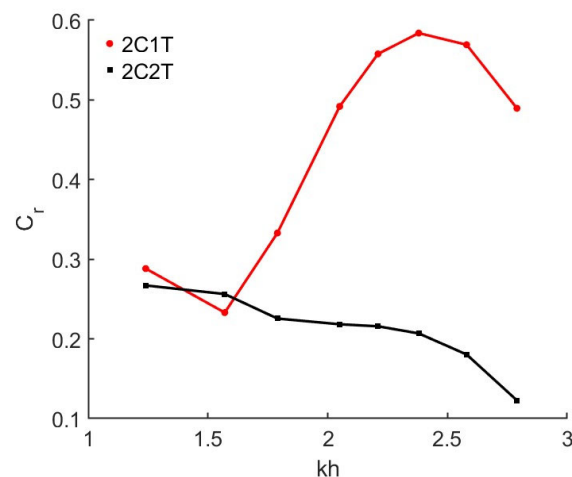


Figure 12. Comparison of reflection coefficients of different chamber types.

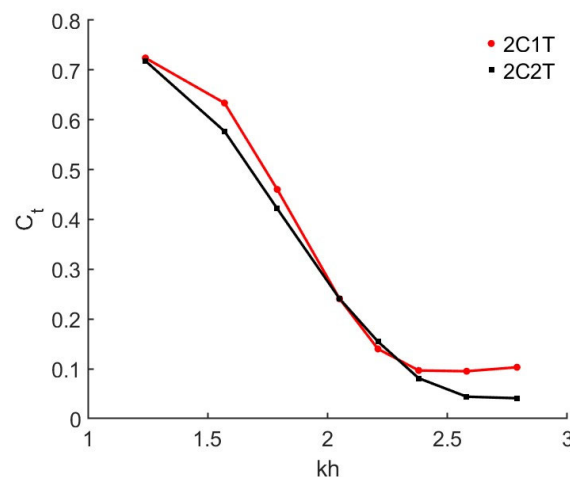


Figure 13. Comparison of transmission coefficients of different chamber types.

This observation suggests that the 2C2T structure has a notable advantage in suppressing wave reflection, allowing a larger proportion of wave energy to enter the interior of the chamber for effective utilization, rather than being dissipated through reflection. Additionally, the transmission coefficient of the 2C2T structure is slightly lower than that of the 2C1T structure, indicating that the 2C2T configuration is marginally more effective in preventing wave energy from passing through the device.

Based on these results, it can be further speculated that the lower reflection coefficient of the 2C2T structure contributes to more wave energy entering the chamber's interior, potentially increasing the vertical load and pressure. The 2C2T structure not only reduces

reflection but also suppresses wave transmission through the device, enhancing wave energy utilization. Furthermore, it exhibits better performance in controlling horizontal loads. However, additional measures may be required to manage the increased vertical load under high-frequency conditions.

4.2. Effects of Intermediate Wall Draught

For the two-dimensional dual-chamber numerical model, wave propagation from the front chamber to the rear chamber occurs only through the bottom opening of the intermediate wall. The draft of the intermediate wall (d_2) significantly influences wave energy distribution and the overall efficiency of the device. This subsection simulates and analyzes three types of intermediate wall drafts ($d_2 = 0.15$ m, 0.25 m, and 0.35 m), corresponding to conditions 1, 2, and 3.

As shown in Figure 14, with an increase in d_2 , the efficiency of the front chamber improves with increasing kh values. A larger intermediate wall draft (e.g., 0.35 m) significantly enhances the efficiency of the front chamber. For the rear chamber, efficiency initially increases and then decreases as kh increases, with a smaller draft (e.g., 0.15 m) proving more effective at higher kh values.

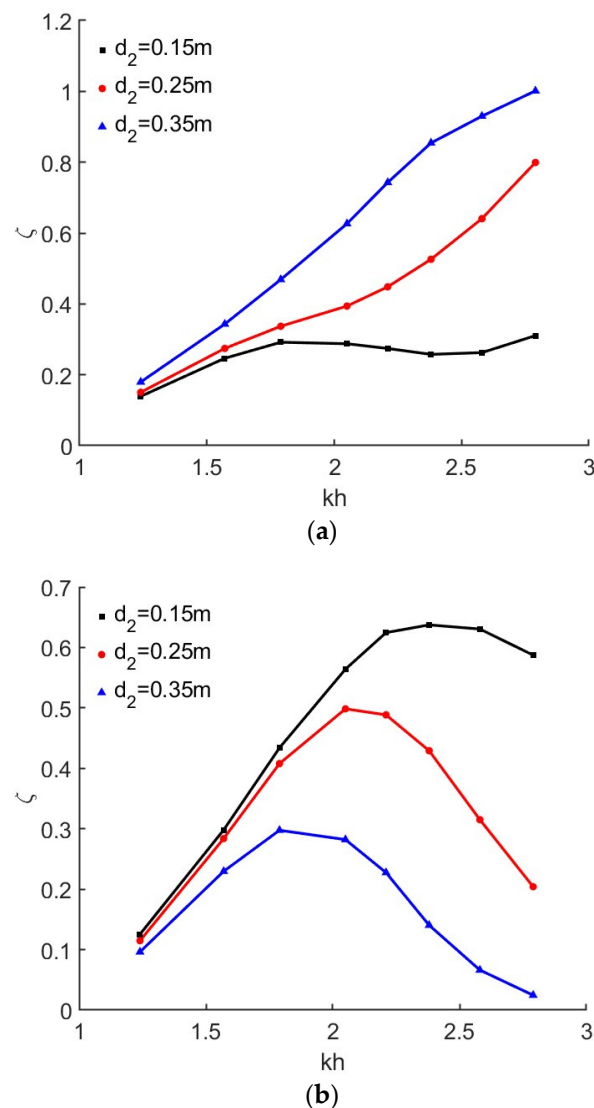


Figure 14. Cont.

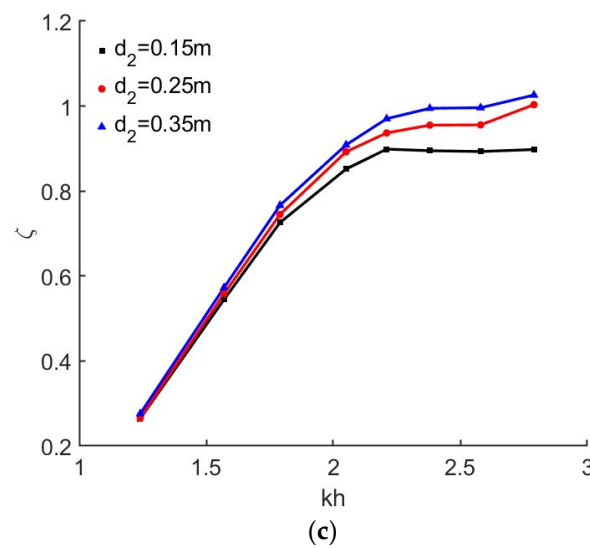


Figure 14. Comparison of the efficiency of different intermediate wall drafts. (a) The front chamber efficiency; (b) The rear chamber efficiency; (c) Total efficiency.

Although different drafts have a pronounced effect on the efficiencies of the front and rear chambers, their impact on the total efficiency of the device is relatively minor, with larger drafts only slightly improving overall efficiency. For instance, at $kh = 2.79$, increasing the draft from 0.15 m to 0.35 m raises the efficiency of the front chamber from 0.31 to 1.00, an increase of 223.6%. In contrast, the efficiency of the rear chamber decreases sharply from 0.59 to 0.02, a reduction of 95.8%. Consequently, the total efficiency of the device increases from 0.90 to 1.03, representing an improvement of 14.3%.

This analysis indicates that the design of the intermediate wall draft must balance the efficiency of the front and rear chambers to achieve optimal wave energy conversion under varying wave conditions.

As illustrated in Figure 15, the horizontal load of the device exhibits an overall increasing trend with the gradual increase in the draft of the intermediate wall. For instance, at $kh = 2.05$, when the intermediate wall draft increases from 0.15 m to 0.35 m, the horizontal load of the device rises from 1.12 N to 1.39 N, representing an increase of 25.44%. Simultaneously, the vertical load decreases from 0.63 N to 0.44 N, corresponding to a reduction of 29.34%.

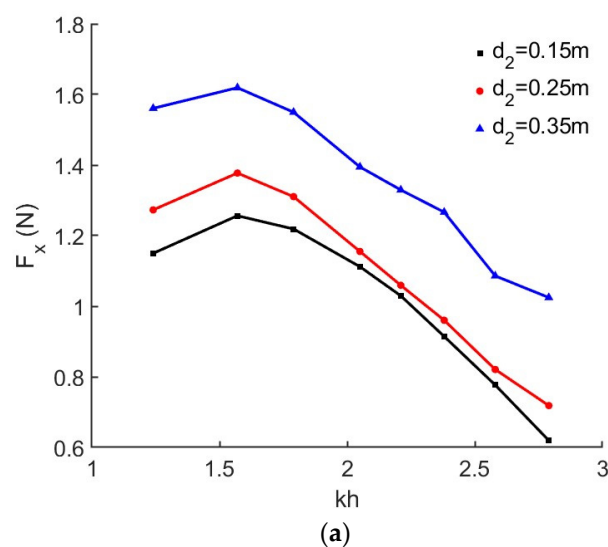


Figure 15. Cont.

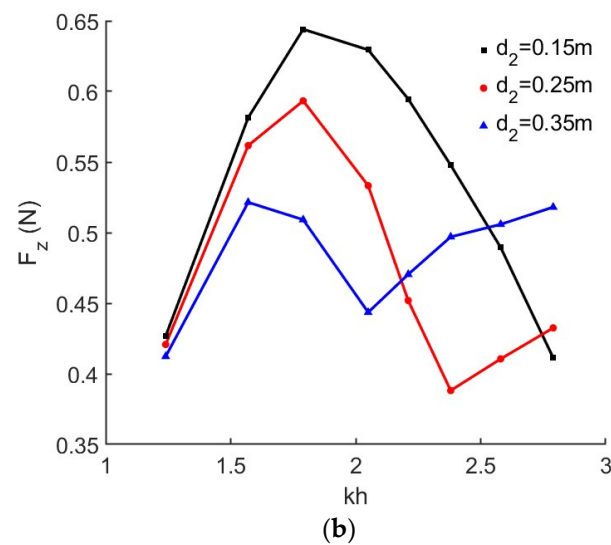


Figure 15. Comparison of loads of different intermediate wall draughts. (a) Horizontal wave loads; (b) Vertical wave loads.

However, at $kh = 2.79$, both horizontal and vertical loads demonstrate an increasing trend with the draft. Specifically, when the draft is increased to 0.35 m, the horizontal load rises by 65.24%, while the vertical load increases by 26.08%. These results indicate that increasing the draft at higher kh values leads to elevated loads on the device. Consequently, structural stability and safety considerations must be prioritized during the design of the device.

Figures 16 and 17 illustrate the influence of intermediate wall draughts on the reflection and transmission coefficients under varying conditions. When the draft is 0.15 m or 0.25 m, the reflection coefficient exhibits a similar trend with kh , initially decreasing and then increasing. For a larger draft (0.35 m), the reflection coefficient shows a consistent increasing trend with kh , indicating that deeper drafts enhance wave reflection. Conversely, the transmission coefficient decreases with increasing draft and diminishes further as kh increases.

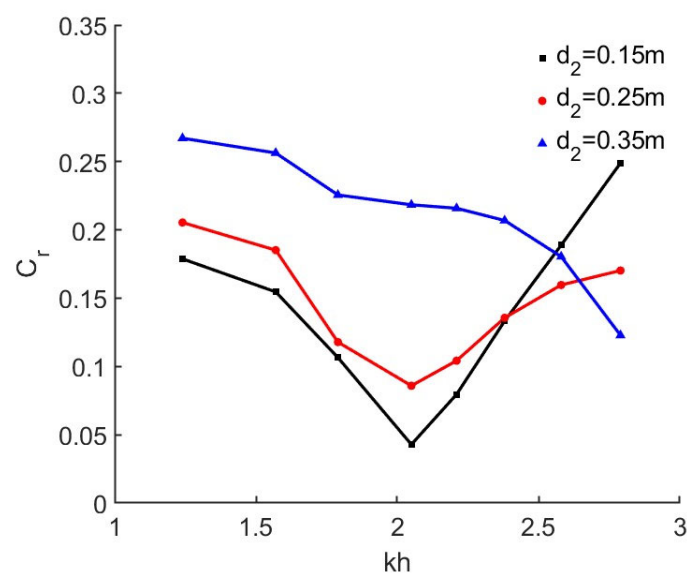


Figure 16. Comparison of reflection coefficients of different intermediate wall draughts.

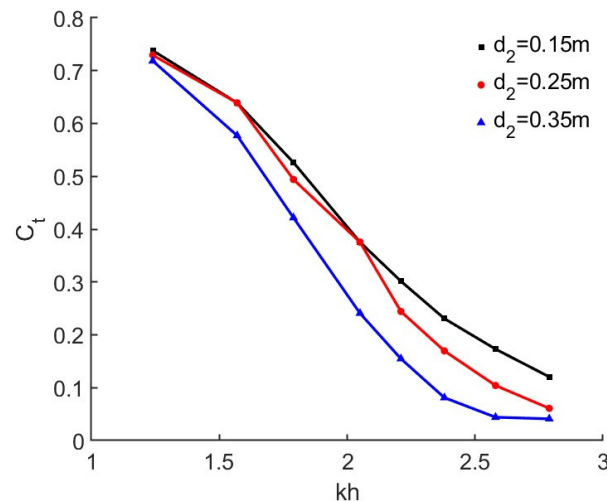


Figure 17. Comparison of transmission coefficients of different intermediate wall draughts.

Under smaller draft conditions (e.g., 0.15 m), waves pass through the device more easily, resulting in relatively high transmission coefficients. However, as the draft increases, the device's resistance to wave transmission grows, significantly reducing the transmission coefficient, particularly in the higher kh range. This suggests that larger intermediate wall drafts reflect more wave energy back into the front chamber rather than allowing it to transmit through the entire device.

These findings highlight the dual impact of intermediate wall drafts on the performance of OWC devices. At lower kh values, appropriately increasing the intermediate wall draft improves wave energy conversion efficiency (as shown in Figure 16) by reducing vertical loads while accommodating larger horizontal loads. However, at higher kh values, deeper drafts subject the device to significantly increased horizontal and vertical loads, potentially challenging its structural load-bearing capacity. Therefore, in the design of OWC devices, it is critical to optimize the balance between energy capture efficiency and structural load distribution to ensure the stability and safety of the device under varying wave conditions.

4.3. Effects of Wall Angles

The wall angle is recognized as a critical factor influencing the hydrodynamic performance of the OWC. To examine its impact, simulations corresponding to conditions 5, 6, and 7 were conducted to investigate the effects of varying wall angles ($\alpha = 0^\circ, 15^\circ, 30^\circ$, and 45°) on the hydrodynamic performance and loading characteristics of the dual-chamber OWC.

The trend of device efficiency with wave frequency (kh) for different wall angles is analyzed in Figure 18. As the wall angle increases, the efficiency of the front chamber gradually improves, contributing to enhanced wave energy conversion in the front chamber. For example, at $kh = 2.79$, when the wall angle is gradually increased from 0° to $15^\circ, 30^\circ$, and 45° , the efficiency of the front chamber increases by 1.82%, 3.56%, and 6.15%, respectively.

In contrast, the efficiency of the rear chamber exhibits different behavior. Within the interval $kh = 1.24\sim 1.79$, the efficiency of the rear chamber decreases with increasing wall angles, while it increases within $kh = 2.05\sim 2.79$. For instance, at $kh = 1.24$, the efficiency of the rear chamber decreases from 0.096 at $\alpha = 0^\circ$ to 0.067 at $\alpha = 45^\circ$, representing a reduction of 29.74%. Conversely, at $kh = 2.79$, the efficiency of the rear chamber increases from 0.025 at $\alpha = 0^\circ$ to 0.081 at $\alpha = 45^\circ$, an improvement of 229.98%.

Additionally, the peak wave energy conversion rate of the rear chamber shifts to higher frequencies as the wall angle increases. Overall, the total efficiency of the dual-chamber system shows a positive trend with increasing wall angles. Larger wall angles (e.g., 45°) yield more significant improvements in efficiency, with the total efficiency at $\alpha = 45^\circ$ surpassing that of other angles under almost all test conditions.

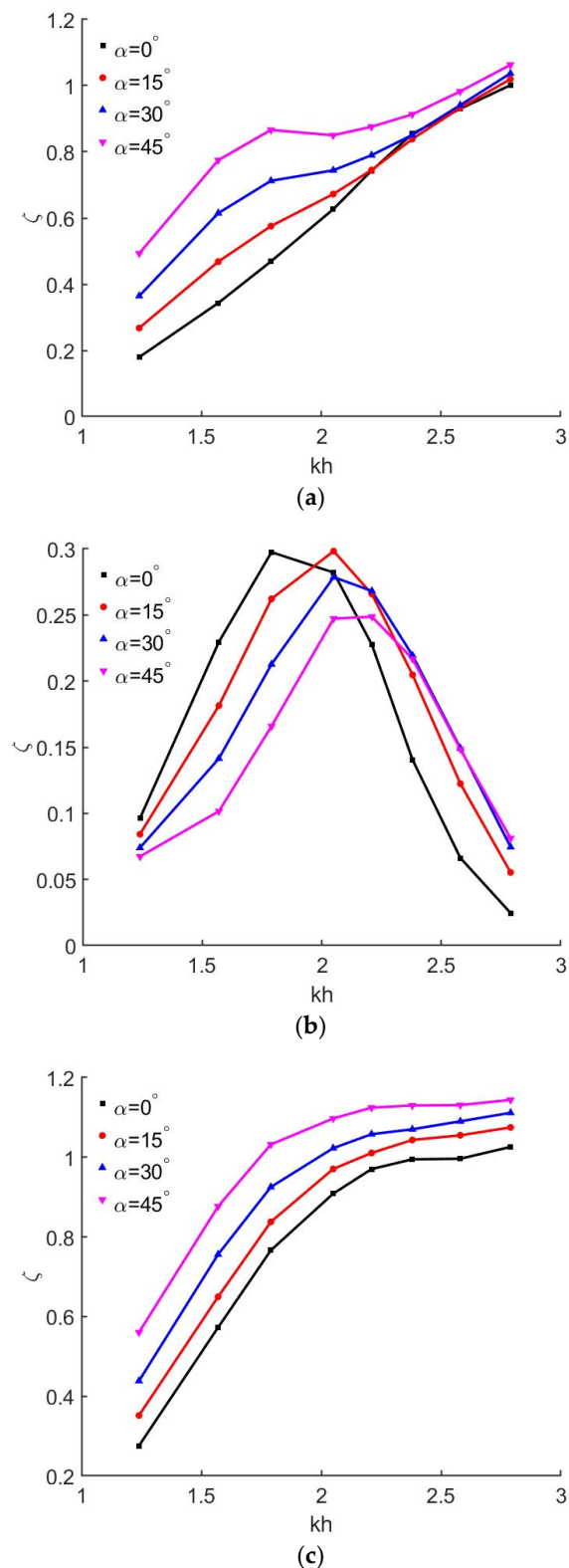


Figure 18. Comparison of the efficiency of different intermediate wall drafts. (a) The front chamber efficiency; (b) The rear chamber efficiency; (c) Total efficiency.

These results suggest that optimizing the wall angle can significantly enhance overall system performance. Therefore, when designing OWC devices, the effects of the wall angle on the efficiencies of both the front and rear chambers should be carefully considered.

The optimal wall angle should be selected based on specific wave conditions to maximize system performance.

Figure 19 illustrates the influence of wall angle on the horizontal and vertical loads acting on the device. A general trend of decreasing horizontal loads is observed as the wall angle increases, with the most pronounced reduction occurring at $\alpha = 45^\circ$. For example, at $kh = 1.24$, the horizontal load decreases by 14.79% as the wall angle increases from 0° to 45° . In contrast, the vertical load demonstrates a more complex behavior, generally increasing with larger wall angles within the $kh < 2.38$ interval. For instance, at $kh = 2.58$, the vertical load increases by 15.55% as the angle increases from 0° to 45° . However, at $kh = 2.38$, the vertical load decreases by 18.63% when the angle increases from 0° to 15° .

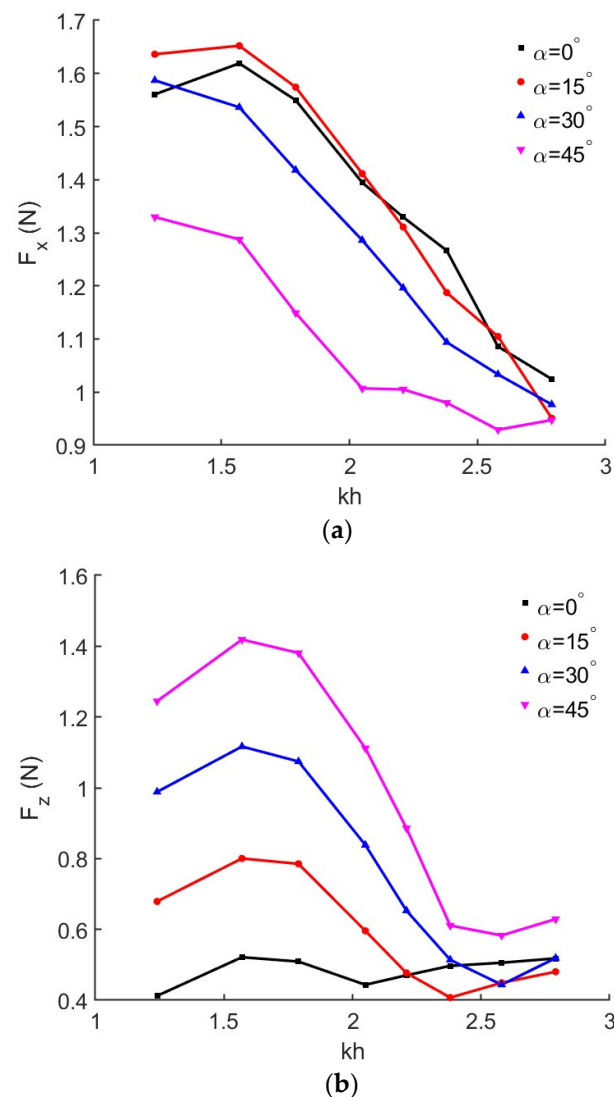


Figure 19. Comparison of loads of different intermediate wall draughts. (a) Horizontal wave loads; (b) Vertical wave loads.

These findings indicate that the effect of wall angle adjustments on vertical loads is not always linear and may vary depending on specific wave conditions and wall angles. Overall, while the horizontal load on the oscillating water column device tends to decrease with increasing wall angles, the vertical load generally exhibits an upward trend, particularly for $kh < 2.38$, where the change is more pronounced. This behavior may be attributed to the larger wall angle dispersing wave force in the horizontal direction and amplifying impact forces in the vertical direction.

These observations underscore the importance of considering the wall angle's influence on both horizontal and vertical loads during the design process. By optimizing the wall angle, it is possible to effectively reduce horizontal loading while enhancing energy conversion efficiency and controlling vertical load increases, leading to improved structural integrity and overall device performance.

As illustrated in Figures 20 and 21, the reflection coefficient generally decreases as the wall angle increases, a trend that becomes particularly pronounced at higher frequencies. This observation suggests that increasing the wall angle effectively reduces wave reflection. Concurrently, the transmittance coefficient also decreases with increasing wall angle, with this trend being more prominent in the $kh < 1.79$ interval. At higher frequencies ($kh > 1.79$), the transmittance coefficient remains relatively small regardless of the wall angle, indicating that wave transmission is significantly restricted under such conditions.

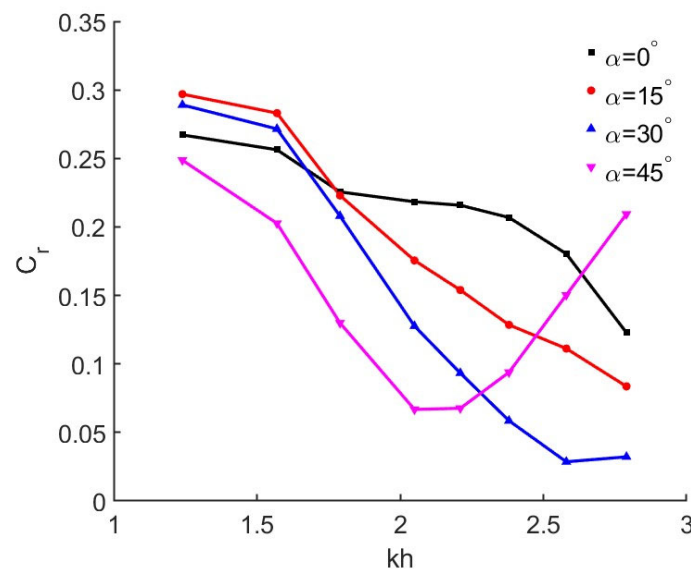


Figure 20. Comparison of reflection coefficients of different wall angles.

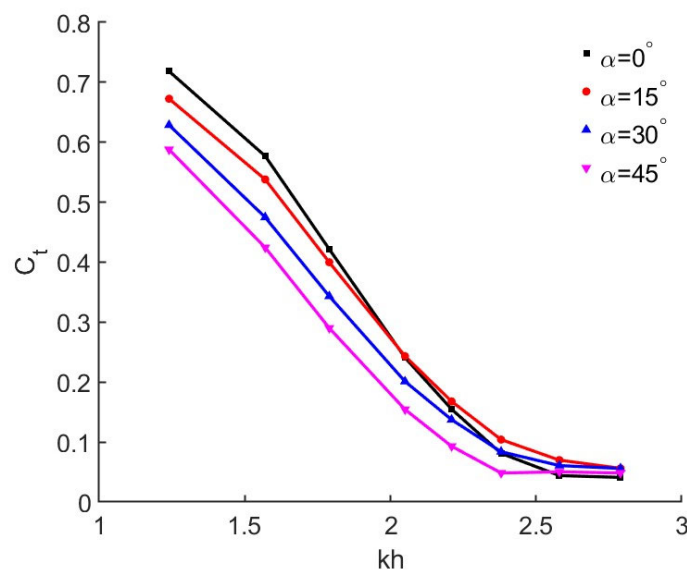


Figure 21. Comparison of transmission coefficients of different wall angles.

Overall, waves are more likely to penetrate the structure when the wall angle is small; however, as the wall angle increases, the structure's resistance to wave transmission intensifies, thereby reducing the proportion of transmitted wave energy. The simultaneous

decrease in both the transmittance and reflection coefficients suggests that increasing the wall angle not only suppresses the reflection of incident waves but also inhibits wave transmission. This dual effect enhances the device's efficiency in capturing wave energy, contributing to improved overall performance.

5. Conclusions

This study employed numerical simulations on a dual-chamber oscillating water column (OWC) device to examine the effects of various design parameters on hydrodynamic behavior and wave loads. The primary findings are outlined as follows:

(1) Compared with the single-chamber system at low wave frequency, the two-chamber system shows a significant efficiency improvement. The 2C2T structure's conversion efficiency is 31.32% higher than the 1C1T structure, which can improve the overall efficiency by improving the efficiency of the rear chamber under long-wave conditions.

(2) Increasing the number of air chambers and turbines can usually improve the energy conversion efficiency, but in some frequency cases, increasing the number of turbines may lead to a decrease in efficiency. When $kh = 1.79$, the energy conversion efficiency of the 2C1T structure is increased by 78.16% compared with that of 1C1T, and 27.41% is increased by 2C2T structure. When the increase to $kh = 2.38$, the efficiency of the 2C2T structure was 315.22% higher than that of the 2C1T structure. Compared with the double-chamber single-turbine structure, the double-chamber double-turbine structure is more effective in reducing the horizontal load, reducing the horizontal load by 27.87% compared with the 2C1T at $kh = 1.79$, but may show a larger vertical load in some frequency ranges. In terms of reducing wave reflection and transmission, the double-chamber twin-turbine structure is superior to the single-turbine structure.

(3) The increase of the wall angle helps to improve the wave energy capture ability of the front air chamber and reduce the horizontal load, but it will lead to the peak value of the rear air chamber efficiency to decrease, while the total efficiency is still increased and the horizontal load is reduced. When the wall angle increased from 0° to 40° , the efficiency increased by 166.37% on average, and the horizontal load decreased by 20.05%. Optimizing the wall angle improves the overall performance of the dual-chamber system and helps to reduce wave reflections.

(4) Increasing the draft of the middle wall (from 0.15 m to 0.35 m) can improve the efficiency of the front room, but may lead to a decrease in the efficiency of the back room; the total efficiency is increased by about 9.6% and the vertical load is reduced by 11.69%.

(5) Future research will extend to the detailed analysis of turbines and generators, aiming to optimize the efficiency of the entire process of converting wave energy to electrical energy.

Appendix A provides a detailed list of Greek and Latin symbols used throughout the study, along with their corresponding definitions. Appendix B presents the energy conversion efficiency results of different tests, highlighting the key parameters and performance metrics under various conditions.

Author Contributions: P.T.: Writing—review & editing, Funding acquisition, Conceptualization. X.L.: Writing—original draft, Software, Investigation, Formal analysis. W.W.: Writing—review & editing, Conceptualization, Methodology, Formal analysis. H.Z.: Conceptualization, Writing—Reviewing and Editing. All authors have read and agreed to the published version of the manuscript.

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Conflicts of Interest: The authors declare no conflicts of interest.

Appendix A. Example Appendix Section

Greek		Latin	
ρ	Fluid density	L	Incident wavelength
σ	Surface tension coefficient	H_i	Incident wave height
η	Wave height	H_r	Transmitted wave height
ω	Angular frequency	C_r	Reflection coefficient
ζ	Energy conversion efficiency	C_t	Transmission coefficient
α	wall angle	h	Water depth
Latin		h_c	Internal heights of the OWC chamber
C_g	Group velocity of the wave	C	Wall thickness of OWC
P	Pressure of the chamber	e	Diameter of the upper orifice
Q	Air volume flow rate	B	Chamber width
A_i	Amplitude of the incoming wave	d	Draft depth
k	Wave number of the waves	H	Wave height
P_T	Extracted pneumatic power	T	Wave period
P_W	Wave power per unit width	F_x	Horizontal wave load
a	Width of tank	F_z	Vertical wave load

Appendix B. Energy Conversion Efficiency of Different Tests

Test	Parameter			Energy Conversion Efficiency							
	Type	d2(m)	$\alpha(^{\circ})$	kh = 2.79	kh = 2.58	kh = 2.38	kh = 2.21	kh = 2.05	kh = 1.79	kh = 1.57	kh = 1.24
1	2C2T	0.25	0	1.003	0.956	0.955	0.937	0.892	0.745	0.558	0.265
2	2C2T	0.35	0	1.026	0.996	0.995	0.970	0.909	0.766	0.573	0.276
3	2C2T	0.15	0	0.898	0.893	0.895	0.899	0.852	0.726	0.545	0.264
4	2C1T	0.35	0	0.844	0.470	0.230	0.178	0.350	0.973	0.907	0.483
5	2C2T	0.35	15	1.075	1.055	1.043	1.011	0.971	0.838	0.650	0.352
6	2C2T	0.35	30	1.112	1.090	1.070	1.058	1.023	0.925	0.756	0.438
7	2C2T	0.35	45	1.144	1.131	1.130	1.124	1.097	1.032	0.876	0.560

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