

Efficiency enhancement of a unidirectional impulse turbine for dual-chamber OWC wave energy converters

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Abstract

To address the efficiency limitations of conventional bidirectional turbines in oscillating water column (OWC) wave energy conversion systems, this study proposes a novel dual-chamber OWC configuration coupled with unidirectional impulse turbines. A steady-state Computational Fluid Dynamics (CFD) model based on viscous fluid theory was established and validated against experimental data. Using this model, a comprehensive parametric optimization was performed on rotor blade number, guide vane number, and blade installation angles to enhance aerodynamic performance. The optimized unidirectional turbine achieved a 59.89% increase in average efficiency and a 67.97% improvement in peak efficiency compared to a reference bidirectional turbine. Furthermore, the total number of rotor blades and guide vanes was reduced by 26.67% and 42.31%, respectively, significantly lowering material requirements and manufacturing costs. Flow field analyses revealed improved pressure distribution, reduced separation zones, and enhanced wake uniformity. This study demonstrates the potential of integrating unidirectional turbines into dual-chamber OWC systems to improve energy conversion performance and reduce structural complexity. The findings provide valuable design insights for wave energy converters. Future work will extend to transient simulations and experimental validation under oscillatory flow conditions.

Keywords: Wave energy conversion; Unidirectional impulse turbine; OWC; CFD;

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1.Introduction

Wave energy is being increasingly regarded as a major and promising resource [1]. Among the various wave energy converters (WECs) deployed in the ocean, the oscillating water column (OWC) stands out as one of the most promising device for harnessing wave energy [2]. The OWC device converts wave energy through a partially submerged hollow air chamber, which is open to the ocean at one end and connected to an air turbine at the other. The water column inside the chamber oscillates vertically due to the propagation of incident waves, generating reciprocating airflow. This airflow drives the air turbine, which is connected to an electrical generator, ultimately producing electricity [3].

The concept of OWC was first introduced in the 1940s, and since then, extensive research and development efforts have been conducted globally [4]. In the 1980s and 1990s, full-scale shoreline prototypes were constructed, including the Kværner multiresonant OWC in Norway [5], the Pico plant in Azores, Portugal [6], and the LIMPET plant in Islay, Scotland [7]. These early prototypes successfully demonstrated the principle of operation and energy extraction under real sea conditions.

In recent years, research has increasingly focused on improving the hydrodynamic performance and energy capture efficiency of OWC systems through innovative chamber and turbine designs. Ning, Zhou [8] analyzed a floating dual chamber OWC device and found that it exhibits improved hydrodynamic properties at specific wave frequencies compared to a single-chamber converter. Building on this, Ning, Wang [9] assessed the hydrodynamic performance of a land-based dual-chamber OWC device, highlighting its superior efficiency and broader effective frequency bandwidth compared to a typical single-chamber OWC device. Similarly, Haghighi, Nikseresht [10] reported that dual-chamber OWCs generally exhibit superior hydrodynamic efficiency, with this advantage being more pronounced at shorter wavelengths.

Further studies have explored the interaction between wave conditions and OWC performance. Li, Zhao [11] developed a semi-analytical model based on potential flow theory and matching eigenfunction method, demonstrating that a dual-chamber OWC array has a

broader capture bandwidth than its single-chamber counterpart for both normal and oblique waves, making it more effective in complex wave conditions. Wang, Xu [12] extended this work by conducting a hydrodynamic study on a three-unit OWC array deployed in various coastal scenarios, revealing significant performance variations depending on wave direction and intensity. These studies provide valuable insights into the real-world applicability of OWC arrays in diverse marine environments.

Technological advancements have further improved OWC energy capture through multi-chamber and multi-turbine configurations [13-15]. Cong, Ning [16] proposed a multi-chamber multi-turbine (MCMT) OWC system, demonstrating that subdividing the chamber into multiple modules converts sloshing-mode free-surface motion into separate piston-mode movements, thereby significantly boosting wave energy capture efficiency. Similarly, Wang, Zhang [17] and Wang, Zhang [18] investigated a three-unit OWC array integrated with an inclined breakwater, finding substantial improvements in energy capture efficiency and frequency bandwidth, particularly in the medium-wavelength range, compared to traditional single-unit OWC devices. Furthermore, Chen, Xie [19] explored the performance of a floating backward bent duct buoy (BBDB)-OWC system enhanced with dual chambers and a novel enhancement plate, reporting significantly improvements in both energy capture efficiency and overall hydrodynamic performance.

Most OWC systems utilize bidirectional turbines, such as the Wells turbine and the impulse turbine, which operate under the reciprocating airflow generated by wave-induced oscillations. These turbines eliminate the need for airflow rectification mechanisms and have been widely studied for their efficiency and operational stability. However, bidirectional turbines often suffer from lower peak efficiency and stall issues at high flow rates, limiting their overall performance [20-21]. In contrast, unidirectional turbines have the potential to achieve higher efficiency by maintaining optimal aerodynamic conditions, yet their application in OWCs remains relatively limited. Mala, Jayaraj [22] presented the initial results of the experimental validation of the twin unidirectional impulse turbine topology, demonstrating that the configuration could achieve over 50% efficiency across a wide range of flow coefficients. Pereiras, Valdez [23] extended this work using numerical modeling, identifying the rotor and guide vanes as the primary sources of energy loss in the twin turbine configuration.

Recent studies have further advanced turbine optimization methodologies. López, Carballo [24] proposed a method for determining the optimal turbine diameter for a given chamber, showing that this approach can significantly enhance the capture width ratio of the OWC, with improvements of up to 100% under varying sea conditions. Wang, Zhang [25] investigated multi-stage wave energy conversion in a chamber-breakwater integrated system equipped with a U-shaped impulse turbine, illustrating how integrated configurations can enhance overall conversion efficiency. Amini, Nasiri [26] introduced a bi-level metaheuristic approach to optimize the geometric design of an oscillating surge WEC, showing that their method outperforms existing metaheuristic approaches in terms of computational efficiency.

While research on impulse turbines has primarily focused on optimizing bidirectional turbines, studies on unidirectional turbines remain limited. Experimental studies have been conducted, but optimization efforts related to their transient response characteristics and dynamic behavior during critical operating conditions are scarce. Furthermore, most studies on unidirectional turbines have concentrated on single-chamber systems coupled with dual unidirectional turbines, whereas investigations into the interaction between dual chambers and dual unidirectional turbines remain insufficient.

To address these limitations, this study proposes a novel coupled system integrating air chambers and unidirectional impulse turbines, as illustrated in Fig. 1. In this system, two air chambers share one turbine that operates during the inhalation phase and another that operates during the exhalation phase. Due to the installation of one-way valves, when one side of the system generates electricity, the other side will not have air leakage. This configuration aims to optimize airflow utilization and improve energy conversion efficiency. Computational Fluid Dynamics (CFD) models of the impulse turbine are established based on viscous fluid theory and validated against existing experimental data. Using these validated models, we investigate the energy conversion mechanism of the unidirectional impulse turbine and evaluate its performance under a range of steady airflow conditions.

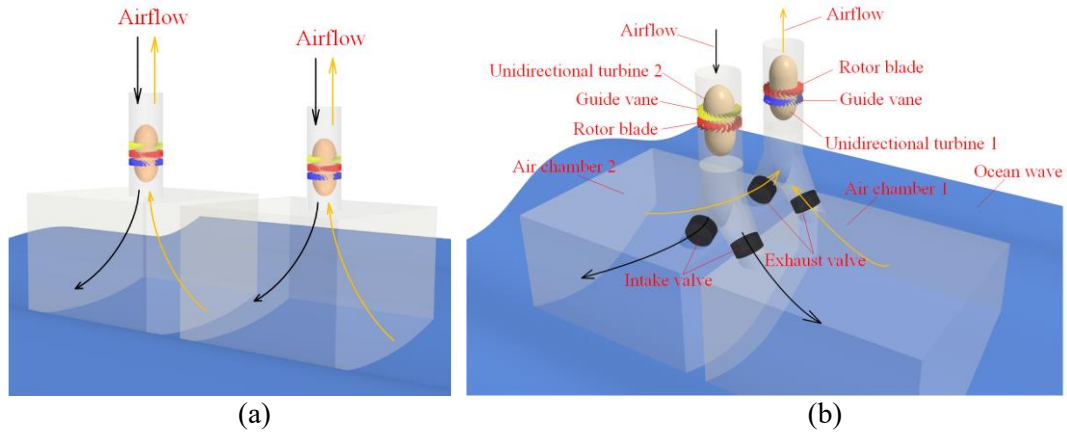


Fig. 1. Schematic of the coupling between the turbine and the air chamber. (a) Conventional configuration: dual bidirectional turbines and dual air chambers. (b) Proposed configuration: dual unidirectional turbines and dual air chambers.

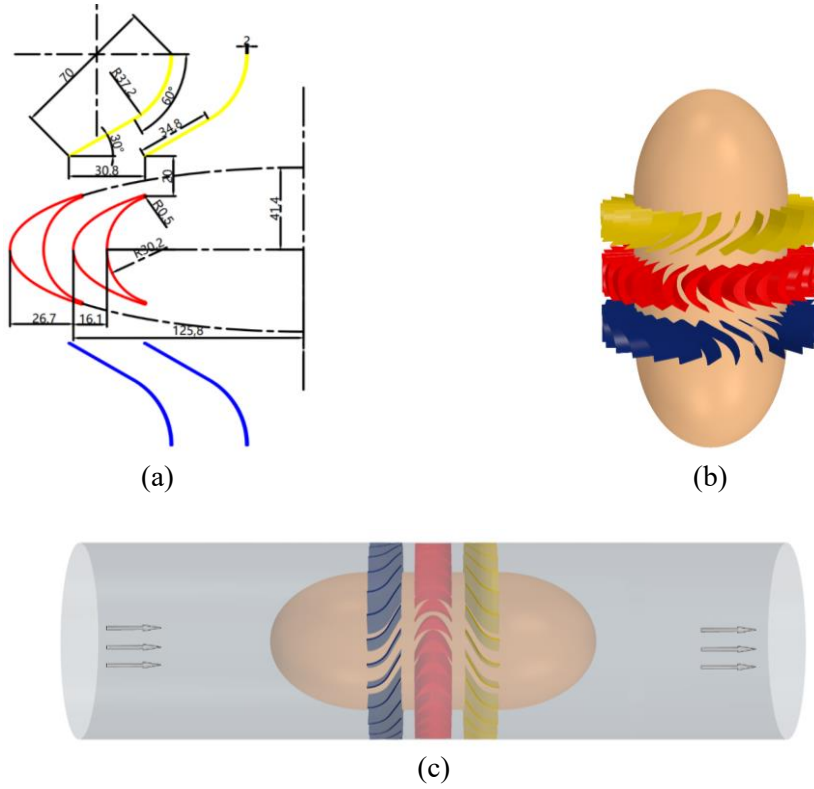
The remainder of this paper is structured as follows. Section 2 introduces the geometric model and numerical setup of the turbines. Section 3 presents the grid-independence study and validation of the CFD model. Section 4 discusses the influence of key parameters on turbine performance and compares the optimized unidirectional design with the conventional bidirectional turbine. Section 5 summarizes the main findings and outlines future research directions.

2. CFD model

2.1 Impulse turbine geometric model

This paper references the design of the axial bidirectional impulse turbine from Setoguchi, Santhakumar [27], as illustrated in Fig. 2. The turbine comprises three primary components: the rotor and two rows of guide vanes. A three-dimensional (3D) schematic of the axial turbine is shown in Fig. 2 (b). In this case study, we conducted calculations for a single turbine, whose geometric parameters are listed in Table 1, serving as the basis for subsequent simulations. For operation under unidirectional flow conditions, the unidirectional impulse turbine is derived from the baseline bidirectional configuration by removing the row of guide vanes located on the non-working side of the flow. The remaining guide-vane row preserves the same geometric dimensions and installation angle as in the bidirectional turbine, ensuring proper aerodynamic guidance in the intended flow direction. All other geometric parameters of the rotor and flow passage remain unchanged, enabling a clear and consistent comparison between the

138 bidirectional and unidirectional configurations.



139 Fig. 2. Axial impulse turbine: (a) schematic diagram of geometric dimensions (Unit: mm); (b)
 140 a 3D schematic of the axial turbine; (c) schematic diagram of the modeling of the axial impulse
 141 turbine.

142 Table 1 Turbine Design Parameters

Parameters	Symbol	Value
Tip Clearance	l_t	1.00mm
Rotor Diameter	D_r	298mm
Hub Diameter	D_h	210mm
Rotor Blade Chord Length	l_r	54mm
Rotor Blade Incidence Angle	β	30°
Rotor Mass	M	4.65kg
Rotor Moment of Inertia	I	0.0595 kg·m ²
Number of Rotor Blades	N_R	30
Number of Guide Vanes	N_G	26
Total Length of Turbine	L_T	499mm

143 The geometric modeling of the flow passage and the axial impulse turbine is conducted at

a 1:1 scale, as illustrated in Fig. 2 (c). The performance indicators of the axial impulse turbine are evaluated using four dimensionless coefficients: input coefficient C_A , torque coefficient C_T , flow coefficient ϕ , and turbine efficiency η , defined by the following formulas [27]:

$$C_A = \frac{2\Delta p Q}{\rho[v_a^2 + U_R^2]b l_r z v_a} \quad (1)$$

$$C_T = \frac{2n_d}{\rho[v_a^2 + U_R^2]b l_r z r_R} \quad (2)$$

$$\phi = \frac{v_a}{U_R} \quad (3)$$

$$\eta = \frac{n_d \omega}{\Delta p Q} = \frac{C_T}{C_A \phi} \quad (4)$$

Where Δp is the total pressure drop of the air passing through the axial impulse turbine; Q is the volumetric flow rate within the flow passage; ρ is the air density; l_r is the chord length of the rotor blade; v_a and U_R are the mean axial flow velocity and the circumferential flow velocity at the rotor blades, respectively; r_R is the rotor radius; n_d is the driving torque generated by the pressure difference between the suction side and pressure side of the rotor blades; ω is the angular velocity of the rotor; b and z are the height and number of the rotor blades, respectively.

In this study, the Reynolds number is defined as follows [28]:

$$Re = \sqrt{v_a^2 + U_R^2} \cdot l_r / (\mu / \rho) \quad (5)$$

Where μ is the dynamic viscosity of air.

2.2 Mathematical model

2.2.1 Governing equations of fluid dynamics

Continuity equation and Reynolds-Averaged Navier-Stokes equations of the fluid are expressed as:

$$\frac{\partial}{\partial x_i}(\rho u_i) = 0 \quad (6)$$

$$\frac{\partial}{\partial x_j}(\rho u_i u_j) = f_i - \frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left[\mu \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3} \delta_{ij} \frac{\partial u_k}{\partial x_k} \right) \right] + \frac{\partial}{\partial x_j}(-\overline{\rho u_i' u_j'}) \quad (7)$$

Where ρ is air density; $i, j, k = 1, 2, 3$; u_i and f_i are the body forces and the average flow velocities, respectively; p is relative pressure; μ is air dynamic viscosity; δ_{ij} is the Kronecker delta function. The final term $-\overline{\rho u_i' u_j'}$ is Reynolds stress, which represents

the effects of turbulence and is expressed in terms of the average velocity gradient as follows [29]:

$$-\rho \overline{u_i' u_j'} = \mu_t \left(\frac{\partial u_j}{\partial x_k} + \frac{\partial u_k}{\partial x_j} \right) - \frac{2}{3} \delta_{jk} \left(\rho K + \mu_t \frac{\partial u_k}{\partial x_j} \right) \quad (8)$$

where K is turbulent kinetic energy and μ_t is turbulent viscosity. The SST K - Ω model is used to calculate turbulent viscosity, as it provides reliable near-wall resolution and robust separation prediction, both of which are essential for accurately capturing the turbulent flow features in impulse turbine passages [30-31]:

$$\frac{\partial}{\partial t}(\rho \psi) + u_j \frac{\partial}{\partial x_j}(\rho \psi) = \frac{\partial}{\partial x_j} \left[(\mu + \sigma_\psi \mu_t) \frac{\partial \psi}{\partial x_j} \right] + G_\psi - Y_\psi \quad (9)$$

Where ψ represents K or Ω ; σ_ψ is a model coefficient; G_ψ is the production term of ψ ; Y_ψ the dissipation term of ψ . The calculation formulas for σ_ψ , G_ψ and Y_ψ can be found in the literature [32].

Studies have shown that if the velocity in the gap above the blade tip exceeds 0.4 Mach, neglecting air compressibility can lead to an overestimation of the device's maximum efficiency by approximately 12% [33-35]. Therefore, it is necessary to account for air compressibility. When air is treated as an ideal gas, the equation of state is given by:

$$p + p_{atm} = \rho_a R T_0 \quad (10)$$

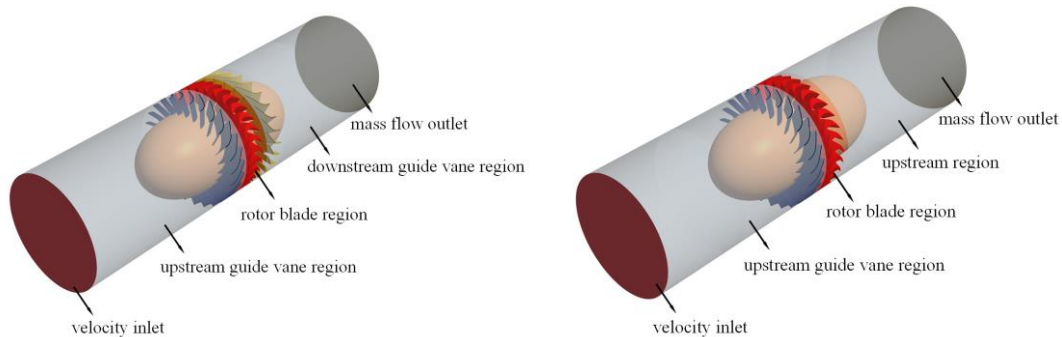
Where p_{atm} is the standard atmospheric pressure; R is the molar gas constant; T_0 is the absolute temperature, assumed to be constant.

2.3 Numerical method

In this study, the CFD software Star-CCM+ is employed to perform the simulations. Fig. 3 illustrates the schematic of the computational domain and the grid for the steady-state simulations of both bidirectional and unidirectional turbine. The entire computational domain is divided into three sub-regions: the upstream region, the rotor region, and the downstream region. The lengths of the upstream and downstream regions are set to twice the rotor diameter. A velocity inlet boundary condition is applied at the flow passage inlet to introduce a specified airflow rate while a mass flow outlet is used to model the outlet, ensuring realistic pipeline flow conditions. These tested velocities are determined based on the target ranges of flow coefficient

ϕ and Reynolds number Re , as well as reference to experimental data. The upstream and downstream guide vane regions are treated as stationary, while the rotor region is modeled using the moving reference frame (MRF) method in the steady-state formulation to account for its relative motion with respect to the guide vane regions. Data exchange between the two guide vane regions and the rotor blade region is facilitated through an internal interface.

The grid is predominantly consisting of hexahedral elements, with trimmed elements employed near the surfaces. To ensure computational accuracy, the grids are refined in critical regions such as turbine surfaces, tip gaps, rotor blades, and guide blades. The finest grid cells are concentrated near the tip gap region. Additionally, to accurately simulate airflow behavior along the diffuser, a boundary layer with a thickness of approximately 0.00864 m, consisting of four layers, is incorporated on the diffuser surface. The governing equations (6) to (9) are discretized using the finite volume method (FVM). For the bidirectional turbine, The total number of grid cells is approximately 2.05 million, with a minimum cell quality of about 0.0472. For the unidirectional turbine, the grid consists of approximately 2.04 million cells, with a minimum cell quality of about 0.0498, primarily located near the tip gap region of the rotor blades. The maximum skewness angle in all regions (rotor, upstream, and downstream ducts) is controlled below 85° . The face validity of the rotor and upstream regions reaches 1.0, while 99.999% of the cells in the downstream region have a face validity ≥ 1.0 . In addition, 96.466% of all cells exhibit a volume change ≥ 0.1 , and no negative-volume cells are present. These indicators confirm that the mesh quality is adequate for ensuring reliable numerical flux calculations.



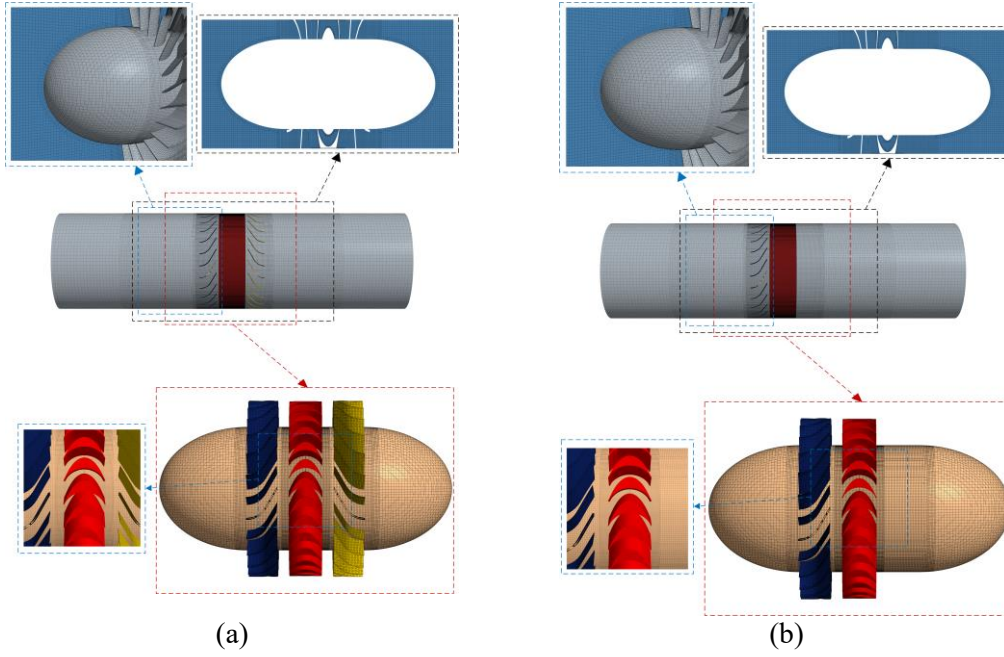
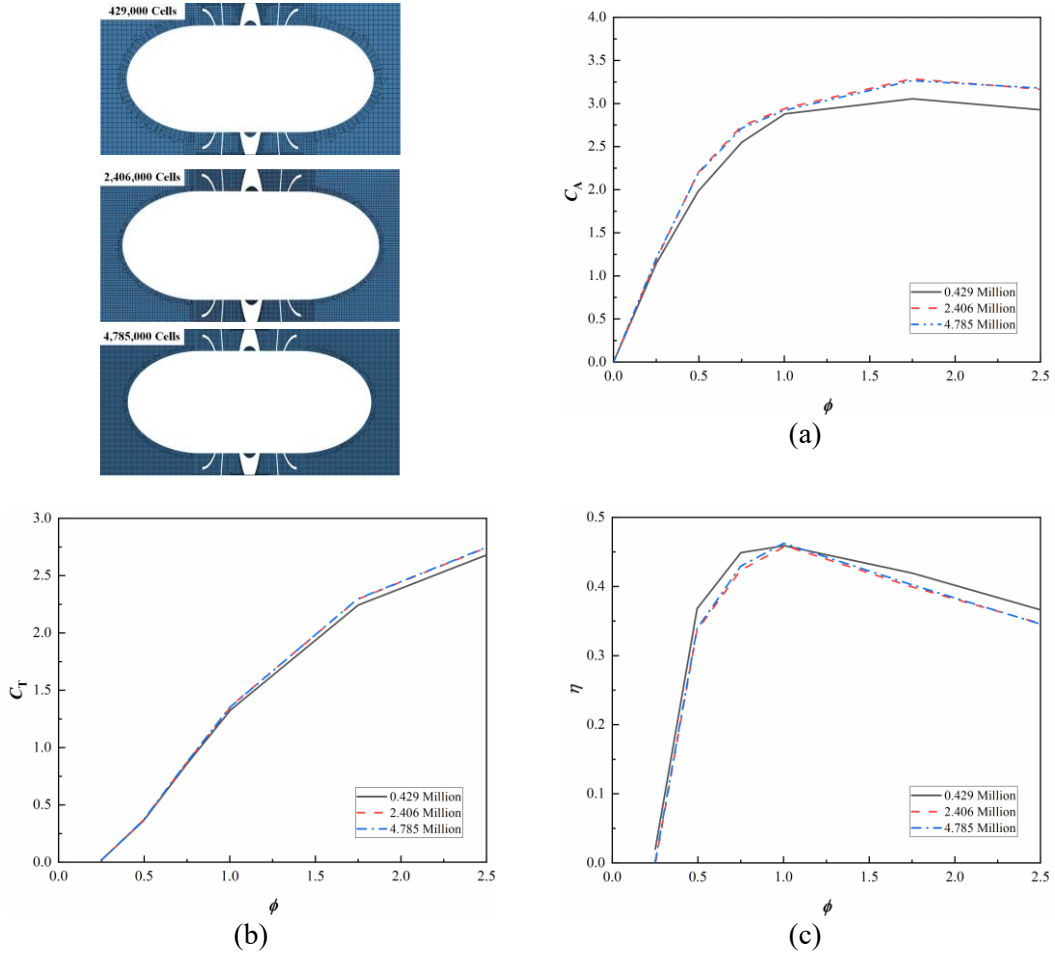


Fig. 3. Schematic of computational domain partitioning and grid for steady-state simulation of turbines: (a) bidirectional turbine; (b) unidirectional turbine.

3. Convergence analysis and model validation

To verify the grid independence of the numerical model, this study utilizes the turbine geometric arrangement shown in Table 1. Three different grid resolutions are considered: 429,000, 2,406,000, and 4,785,000 cells, as illustrated in Fig. 4. Fig. 4 (a)-(c) display the variations of C_A , C_T , and η of the bidirectional turbine as a function of ϕ during the steady-state simulation for different grid resolutions. At this point, with $Re = 5.14 \times 10^4$, six typical flow coefficient data points ($\phi = 0.25$, $\phi = 0.50$, $\phi = 0.75$, $\phi = 1.00$, $\phi = 1.75$ and $\phi = 2.50$) are selected, and the corresponding inlet velocities are 1.85 m/s, 3.38 m/s, 4.57 m/s, 5.39 m/s, 6.61 m/s, and 7.07 m/s respectively. The results obtained with the grid cell number of 429,000 exhibit a significant deviation from those computed with 4,785,000 cells, which is used as the reference benchmark. In contrast, the results for 2,406,000 cells closely align with those of 4,785,000 cells, with both the input coefficient and torque coefficient being identical. Within the investigated range of ϕ , the maximum relative error in turbine efficiency for the 2.406 million-cell grid, calculated relative to the 4.785 million-cell reference, is only 1.35%. This indicates that further increases in grid resolution yield negligible changes in the steady-state simulation results. Therefore, the grid of 2,406,000 cells is adopted for the steady-state

236 simulations in subsequent studies.



237 Fig. 4. Variations of (a) C_A , (b) C_T , and (c) η with ϕ for different grid resolutions.

238 Fig. 5 compares the steady-state simulation results with the experimental data from
 239 Setoguchi, Takao [36]. The comparison shows slight discrepancies between the numerical
 240 simulation and the experimental data, with the maximum relative error in turbine efficiency not
 241 exceeding 6%. This deviation may be attributed to the neglect of energy losses caused by
 242 friction damping from the rotor rotation in the numerical model, as observed in the experiments
 243 [37]. Despite these differences, the overall trends of the two curves are consistent, confirming
 244 that the steady-state model is reliable in predicting turbine performance.

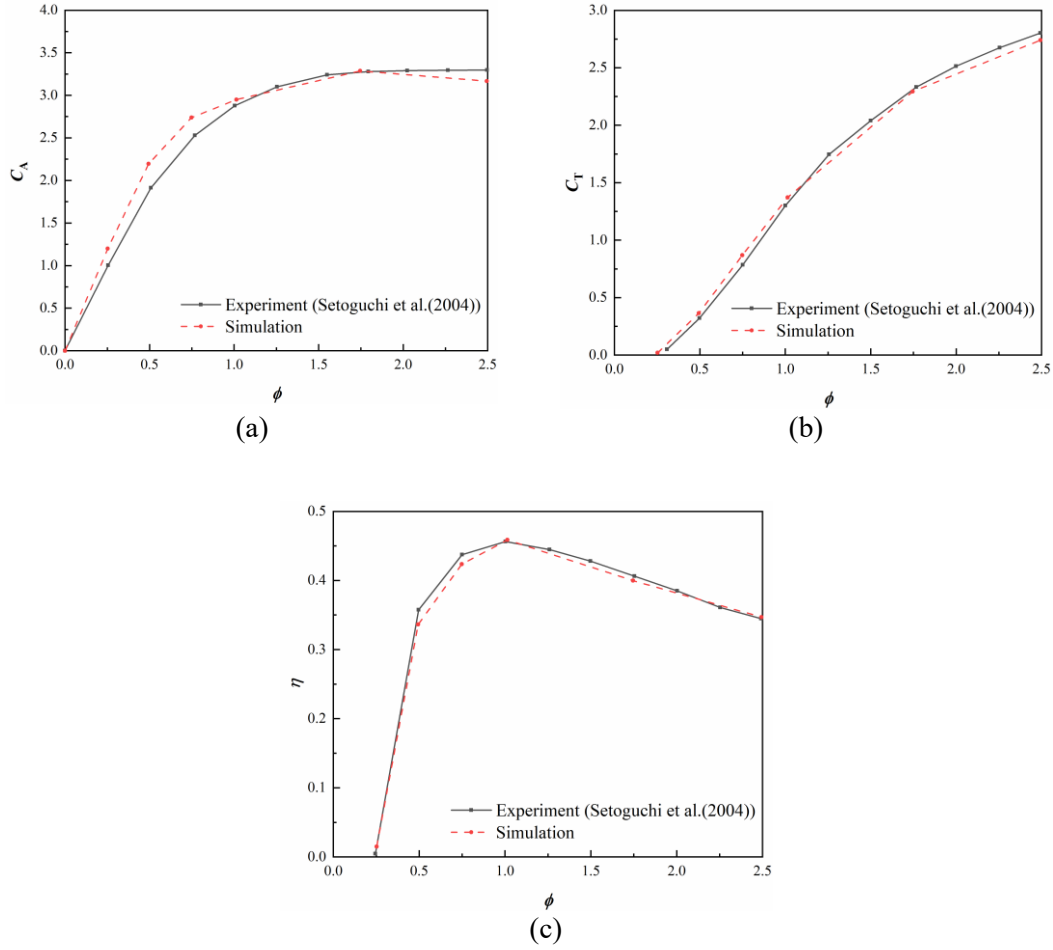


Fig. 5. Comparison of numerical simulation results with experimental results for various ϕ : (a) C_A , (b) C_T , and (c) η .

4. Results and discussion

4.1 Effect of Re

The effect of Re on the performance of the unidirectional turbine under constant airflow conditions is investigated. Re is defined by Eq.(5). The turbine's physical parameter is listed in Table 1. Fig. 6 shows the variation of C_A , C_T , and η with Re for different ϕ . As shown in Fig. 6 (a), C_A increases with ϕ for all Re . When $Re < 1.714 \times 10^4$, C_A gradually increases as Re decreases, meaning more input power is required for the same flow condition. When Re exceeds 1.714×10^5 , the variation of ϕ with C_A tends to converge, suggesting a reduced influence of Re on C_A within the examined range of ϕ . Fig. 6 (b) shows that C_T also increases with ϕ . However, unlike C_A , C_T remains almost unchanged for different Re , indicating that the aerodynamic torque generated by the turbine remains relatively stable across different Re . Fig. 6 (c) shows

that η initially increases with ϕ , reaches a peak, and then gradually decreases. Additionally, higher Re values correspond to slightly higher η across the entire range of ϕ , but the rate of improvement diminishes as Re increases, which is consistent with the findings of the experimental study [36]. When Re exceeds 1.714×10^5 , η tends to exhibit consistent variation with ϕ . This behavior can be explained as follows: At lower Re ($Re < 1.714 \times 10^4$), C_A is relatively high, which increases the input power according to Eq. (4), leading to lower efficiency. As Re increases, C_A decreases slightly, reducing the input power (denominator in Eq. (4)), which, for a given turbine output, results in an improvement in η . However, once Re reaches a sufficiently high value, further reductions in C_A become negligible, causing η to level off. Therefore, the Reynolds number is maintained at $Re = 1.714 \times 10^5$ in the subsequent simulations.

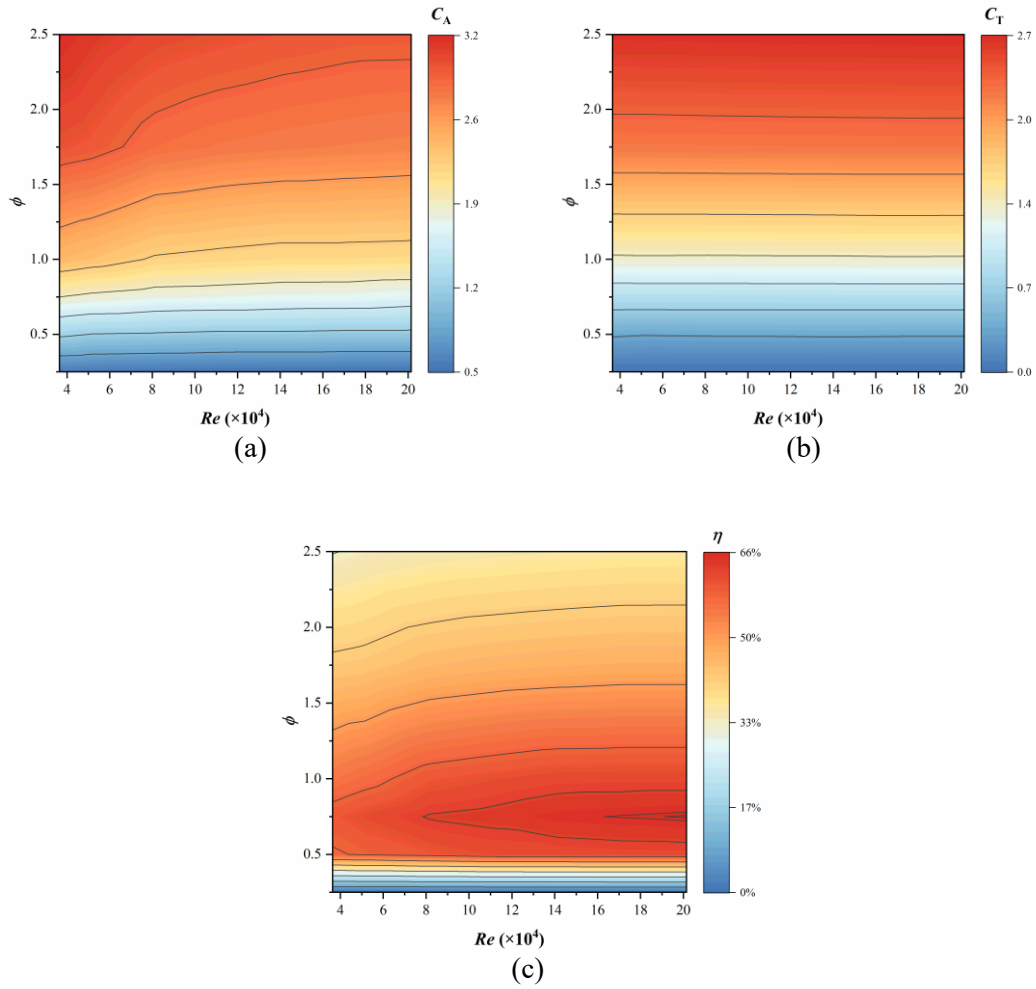
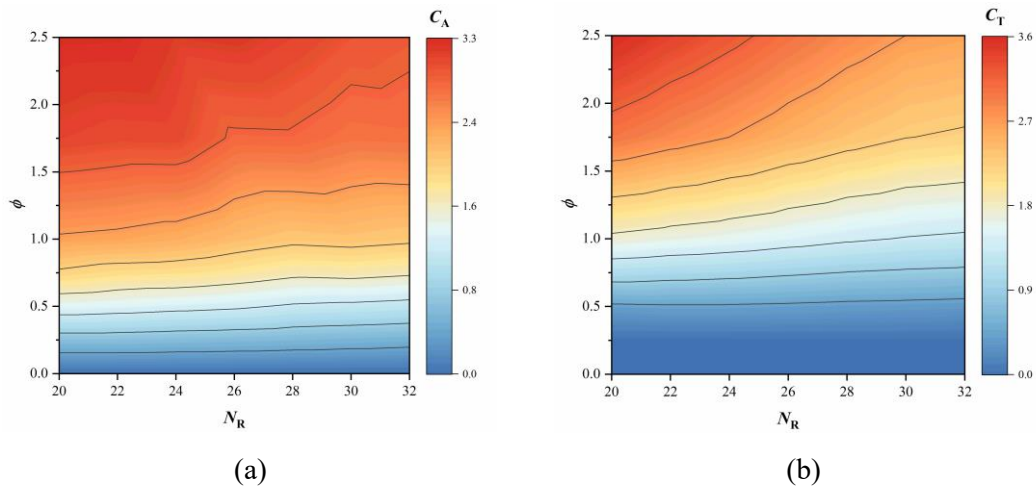
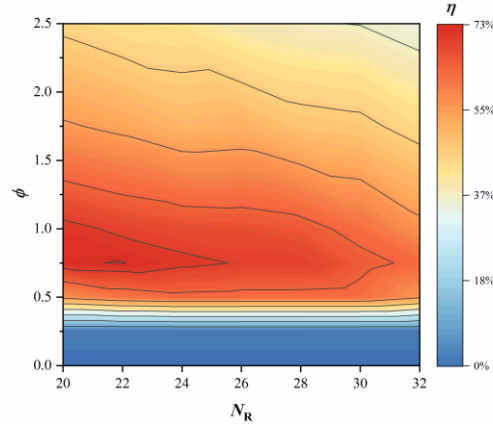


Fig. 6. Variations of (a) C_A , (b) C_T , and (c) η with Re for different ϕ .

4.2 Effect of rotor blade number

Fig. 7 illustrates the variations of C_A , C_T , and η with rotor blade number (N_R) for different ϕ . As shown in Fig. 7 (a), C_A increases monotonically with ϕ for any given N_R , indicating that higher flow rates require greater aerodynamic input. For a given ϕ , C_A slightly decreases as N_R increases, suggesting that increasing the number of blades can reduce the aerodynamic input requirement under the same flow condition, likely due to improved aerodynamic guidance and reduced flow separation. Fig. 7 (b) shows that C_T also increases monotonically with ϕ , indicating higher torque generation at higher flow conditions. However, C_T tends to decrease with increasing N_R . This suggests that adding more blades does not necessarily enhance torque output; instead, it may induce additional flow resistance or interfere with optimal flow interaction, thereby reducing the aerodynamic torque. Fig. 7 (c) reveals that η increases significantly with ϕ at low values, peaks around $\phi = 0.75$, and then gradually decreases. Additionally, η tends to be higher at moderate blade numbers (typically around $N_R = 22$), representing an significant increase relative to the baseline case of $N_R = 30$, as recommended by Setoguchi, Santhakumar [27] (see Table 1). These results demonstrate that a moderate reduction in rotor blade number can enhance turbine performance, which may be attributed to reduced flow blockage and diminished aerodynamic interactions between adjacent blades. Accordingly, the rotor blade number is fixed at $N_R = 22$ in the subsequent simulations, unless otherwise specified.

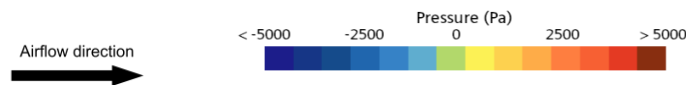




(c)

Fig. 7. Variations of (a) C_A , (b) C_T , and (c) η with N_R for different ϕ .

Given the periodic arrangement of the rotor blades, a representative blade is selected for detailed pressure distribution analysis, as shown in Fig. 8. The figure illustrates the pressure distribution on both the pressure and suction sides of the selected blade, with airflow directed from left to right. As N_R decreases from 30 to 26, the high-pressure region on the pressure side becomes more extensive and notably more uniform, suggesting improved aerodynamic loading and reduced risk of local flow separation. When N_R is further reduced to 22 and 20, a slight decline in pressure-side uniformity is observed. Concurrently, the negative-pressure region near the leading edge of the pressure side diminishes, which effectively reduces the energy loss due to airflow backflow. On the suction side, a reduction in N_R leads to a decrease in the peak pressure value at the leading edge of the high-pressure zone, which helps mitigate reverse torque. Furthermore, the area of the mid-chord negative pressure zone increases. This indicates enhanced aerodynamic suction and improved blade loading, likely due to reduced interference between adjacent blades and better utilization of the incoming flow. Overall, the pressure distributions suggest that a moderate reduction in rotor blade number can enhance aerodynamic loading by deepening the suction effect while maintaining favorable pressure recovery on the pressure side.



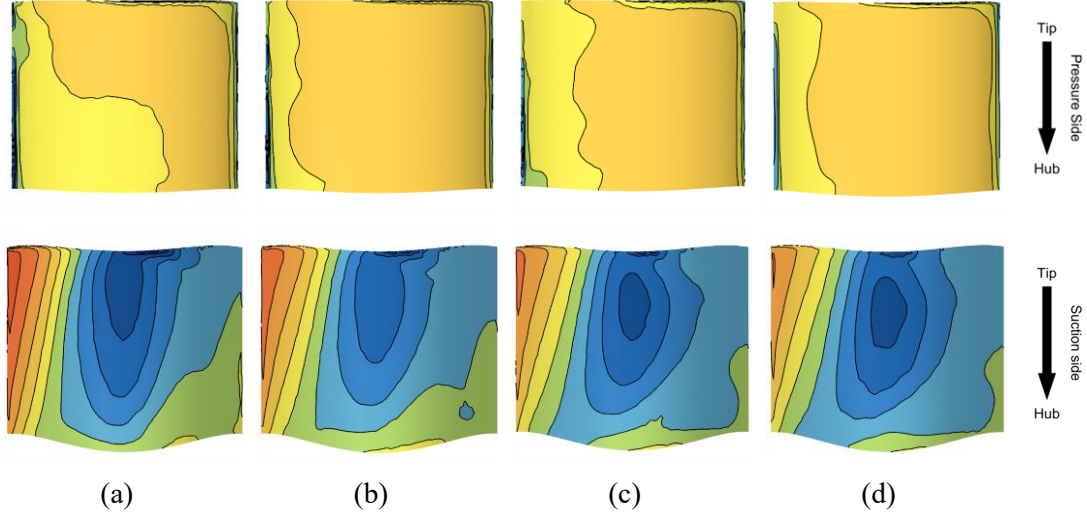


Fig. 8. Pressure distributions on the pressure and suction sides of a blade for different N_R at $\phi = 0.75$: (a) $N_R = 30$; (b) $N_R = 26$; (c) $N_R = 22$; (d) $N_R = 20$.

Fig. 9 presents the velocity distributions at the blade-height cross section for different N_R , highlighting the influence of blade number on downstream wake characteristics and flow uniformity. In all cases, a high-velocity jet forms near the suction side of each blade, followed by a relatively low-velocity wake region downstream. When $N_R = 30$, the rotor blades are densely arranged, resulting in a narrow, well-organized wake structure. This indicates strong aerodynamic control with minimal flow loss, though such dense configuration may introduce additional flow resistance. Reducing N_R to 26 causes slight wake broadening, but the overall flow field remains coherent and uniform. At $N_R = 22$, the wake becomes significantly wider, and low-velocity regions begin to develop, indicating diminished aerodynamic guidance and the emergence of flow separation. When the blade number is further reduced to $N_R = 20$, the wake broadens considerably and exhibits prominent low-speed zones, suggesting increased flow losses and potential flow instability.

In conjunction with the pressure analysis in Fig. 8, the results in Fig. 9 confirm that a moderate number of blades (especially $N_R = 22$) achieves a favorable balance between aerodynamic interaction and wake control. It enhances suction performance on the blade surface while maintaining coherent downstream velocity structures, thereby improving turbine performance. The findings are consistent with the efficiency trends shown in Fig. 7 (c).

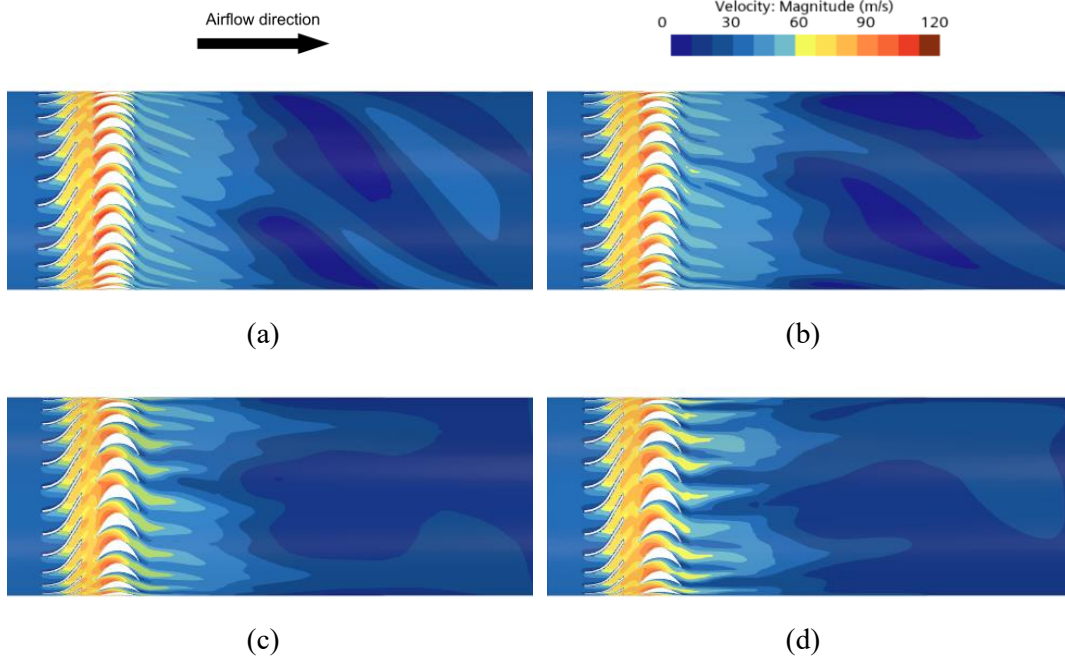


Fig. 9. Velocity distribution at the blade-height cross section for different N_R at $\phi = 0.75$: (a) $N_R = 30$; (b) $N_R = 26$; (c) $N_R = 22$; (d) $N_R = 20$.

4.3 Effect of guide vane number

Fig. 10 shows the variations of C_A , C_T , and η with guide vane number (N_G) for different ϕ . As shown in Fig. 10 (a), C_A increases monotonically with ϕ , indicating that higher flow rates demand greater aerodynamic input. For any fixed ϕ , C_A also increases with N_G . For example, when $\phi = 2.5$, C_A increases by approximately 28.5% from $N_G = 22$ ($C_A = 2.949$) to $N_G = 32$ ($C_A = 3.129$). This implies that a higher number of guide vanes introduces additional flow resistance or passage losses, thereby requiring more input power. Fig. 10 (b) reveals a similar increasing trend in C_T with ϕ , suggesting enhanced torque generation under high-flow conditions; however, its dependence on N_G is relatively weak (C_T increases by 6.43% as N_G increases from 22 to 32 for $\phi = 2.5$) and nearly saturates beyond a certain point, indicating limited improvement due to marginal gains in flow guidance or increased resistance. In Fig. 10 (c), turbine efficiency η rises rapidly at low ϕ , peaks at approximately $\phi = 0.75$, and then slightly declines. The influence of N_G on η is non-monotonic. At $N_G = 24$, relatively high η is maintained over a broad range of ϕ (approximately 0.7 – 1.1), suggesting an effective balance between aerodynamic guidance and flow resistance. However, as N_G increases to 26 – 28, the high-efficiency region becomes more narrower ($\phi = 0.7 – 1.0$), indicating reduced adaptability to flow variations or increased

aerodynamic losses due to vane blockage. Interestingly, at $N_G = 30$, the efficiency improves again across a broader range of ϕ (approximately 0.6 – 1.2), highlighting enhanced aerodynamic performance and robustness under varying flow conditions. When N_G reaches 32, the efficiency slightly declines, likely due to excessive resistance introduced by the surplus vanes. Overall, $N_G = 30$ appears to provide the most robust and efficient configuration, offering consistently high performance across a wide range of operating conditions. Therefore, unless otherwise specified, the guide vane number is fixed at $N_G = 30$ in the subsequent simulations.

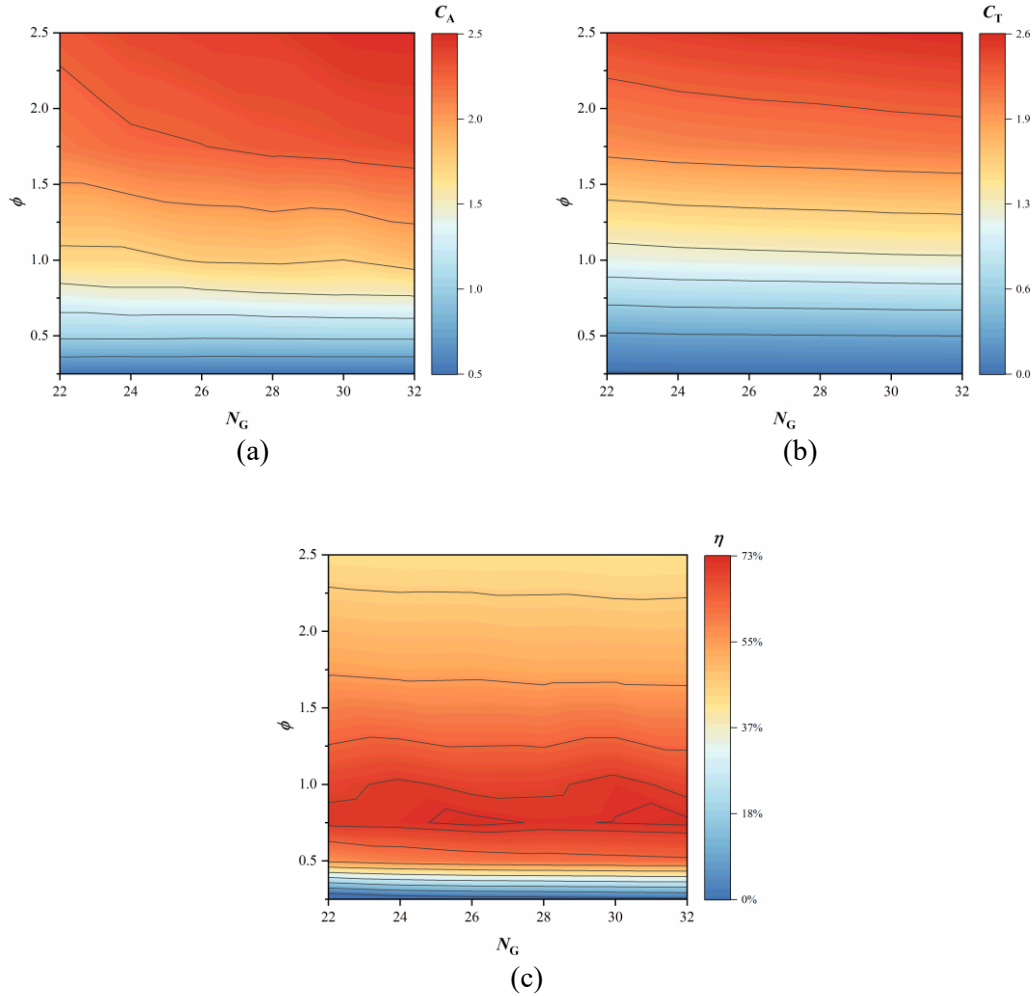
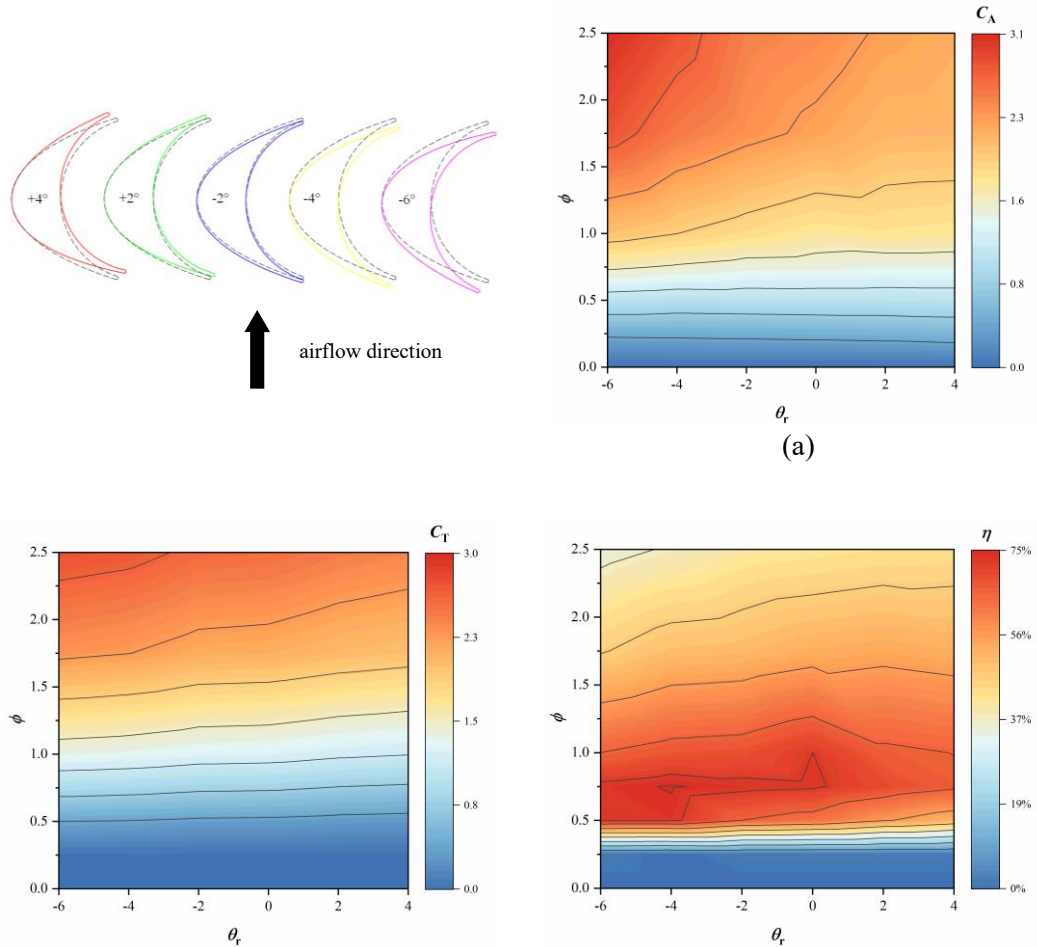


Fig. 10. Variations of (a) C_A , (b) C_T , and (c) η with N_G for different ϕ .

4.4 Effect of rotor blade installation angle

Fig. 11 illustrates the variations of C_A , C_T , and η with respect to rotor blade installation angle θ_r and ϕ . The schematic in the top-left of the figure shows different rotor profiles corresponding to varying θ_r , which alters the relative alignment between incoming airflow and rotor blades. The dashed lines denote the baseline profile at $\theta_r = 0$. In Fig. 11 (a), C_A increases

monotonically with ϕ , indicating that greater flow rates require higher aerodynamic input. The dependence of C_A on θ_r is moderate, with slightly elevated values observed when θ_r lies in the range of approximately -4° to 0° , suggesting reduced flow loss or better alignment with the airflow in this region. Fig. 11 (b) reveals that C_T also increases with ϕ . The highest torque output is generally achieved when $\theta_r \approx -4^\circ$ with an average increase of 26.76% in C_T compared to the 0° baseline, indicating that slight negative rotor angles enhance aerodynamic torque transfer. η , shown in Fig. 11 (c), exhibits a clear peak near $\theta_r = -4^\circ$, particularly for moderate flow conditions ($\phi = 0.6 - 1.2$). At this angle, the balance between input power and torque output is optimized, leading to enhanced energy conversion performance. Efficiency decreases noticeably when θ_r exceeds 0° or becomes too negative (e.g., $\theta_r < -6^\circ$), likely due to increased flow separation or misalignment between the rotor and airflow. Overall, a rotor angle of approximately $\theta_r = -4^\circ$ offers the best aerodynamic performance across a broad range of flow conditions, and is therefore selected as the baseline configuration in the following sections unless otherwise specified.



(b)

(c)

Fig. 11. Variations of (a) C_A , (b) C_T , and (c) η with θ_r for different ϕ .

4.5 Effect of the guide vane installation angle

Fig. 12 illustrates the effects of guide vane installation angle θ_g on turbine aerodynamic performance under varying flow coefficient ϕ . The schematic in the top-left corner shows different guide vane profiles corresponding to various θ_g values, where the dashed line indicates the baseline case of $\theta_g = 0^\circ$. Fig. 12 (a) shows that as ϕ increases, C_A also rises significantly, especially at large positive θ_g , indicating that greater guide vane angles lead to increased flow resistance and higher input energy demands. This suggests that overly positive vane angles may be energetically unfavorable due to excessive aerodynamic blockage. In Fig. 12 (b), C_T increases with ϕ , indicating enhanced torque generation under stronger airflow. The influence of θ_g is more moderate: C_T gradually increases with θ_g (C_T increases by 36.52% as θ_g increases from -4° to $+6^\circ$ for $\phi = 2.5$), reflecting improved flow redirection, though the marginal gain diminishes beyond $\theta_g > 4^\circ$. According to Nguyen [38], as the guide vane angle decreases, the flow becomes steeper, increasing the angle of attack, which can cause flow separation and higher losses. Conversely, a positive guide vane angle helps align the flow with the rotor blades, reducing flow separation and pressure drop, thus improving torque transfer, particularly at high-flow conditions. Fig. 12 (c) presents that η peaks at 77% when θ_g is around 2° , particularly in the ϕ range of 0.5 – 1.0. This indicates that small positive guide vane angles achieve a good balance between effective flow guidance and minimized aerodynamic losses. In contrast, both excessively negative and overly positive θ_g values lead to efficiency drops due to either insufficient guidance or excessive resistance. Overall, the results suggest that a moderate installation angle of $\theta_g = 2^\circ$ is optimal for maintaining high efficiency across a wide range of flow conditions.

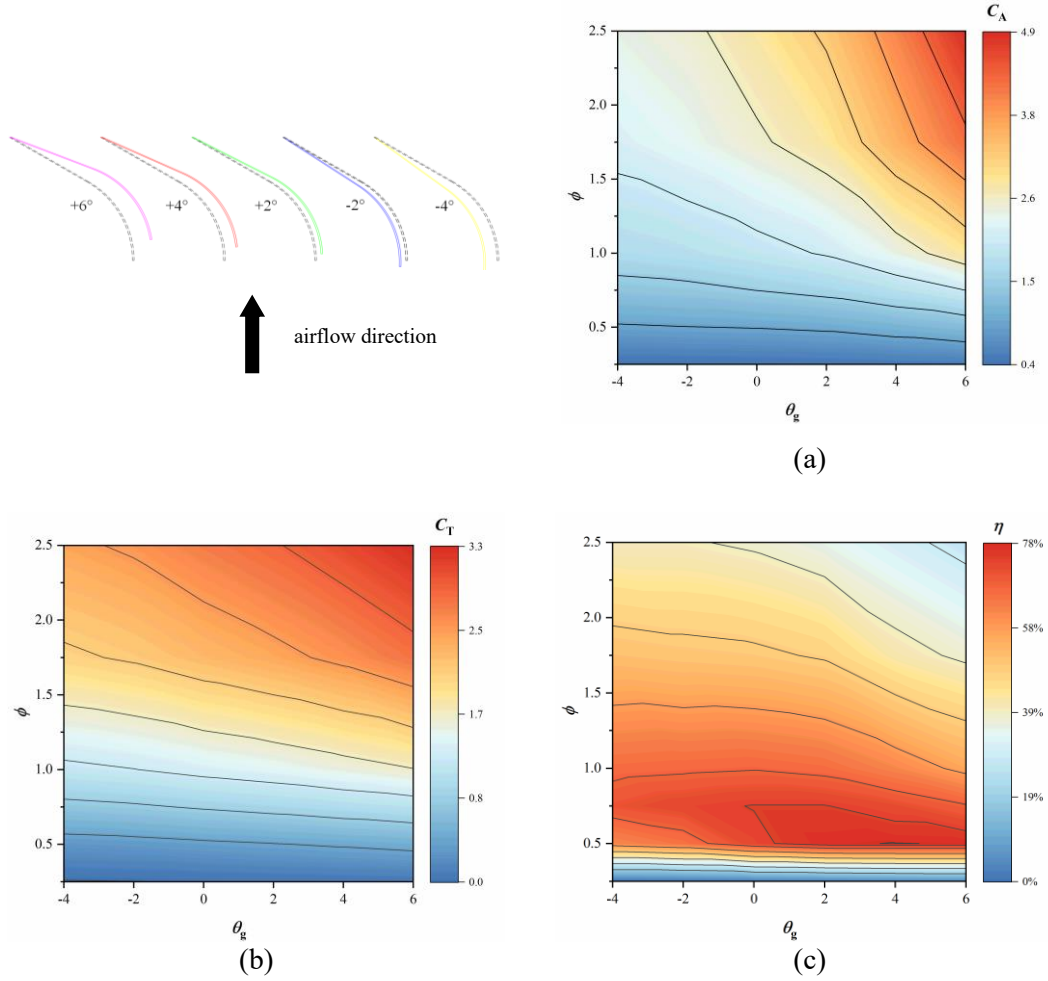


Fig. 12. Variations of (a) C_A , (b) C_T , and (c) η with θ_g for different ϕ .

4.6 Comparison between the conventional turbine and the optimised unidirectional turbine

After optimising the design of unidirectional turbine, the number of rotor blades $N_R = 22$, the number of guide vanes $N_G = 30$, the rotational angle of rotor blades $\theta_r = -4^\circ$, and the rotational angle of guide vanes $\theta_g = 2^\circ$. Compared to the baseline case recommended by Setoguchi, Santhakumar [27] (see Table 1), the total number of blades and guide vanes is reduced by 26.67% and 42.31%, respectively, which significantly reduces the manufacturing material cost.

Fig. 13 demonstrates the performance comparison curves of the optimised design of the unidirectional turbine with that of the bidirectional turbine recommended by Setoguchi, Santhakumar [27]. From the figure, it can be seen that the unidirectional optimised input coefficients are lower than the bidirectional ones in the range of flow coefficients $\phi = 0 - 2.3$,

while the opposite trend is shown at $\phi = 2.3-2.5$; the optimised torque coefficients of the unidirectional turbine are higher than those of the bidirectional ones throughout the range of flow coefficients, with a maximum increase of 9.89% relative to the bidirectional case; moreover, the efficiency of the unidirectional turbine is higher than the bidirectional one in the range of the full range of flow coefficients ($\phi = 0.2-2.5$), with the peak efficiency reaching 77.1%, which is higher than the bidirectional one ($\eta = 45.9\%$), with an increase of 67.97%, and the average efficiency after optimisation increased to 52.8%, which increased by 59.89% compared to the bidirectional turbine ($\bar{\eta} = 33\%$). The maximum growth rate of efficiency under the condition of the same ϕ reaches 129.38%. The results show that the optimised design of the unidirectional turbine significantly improves the performance of the turbine. It should be noted that while the unidirectional turbine is expected to outperform the bidirectional turbine under ideal unidirectional flow, the actual flow generated by the OWC may not be perfectly unidirectional. This limitation arises from the dynamics of the one-way valves and the variability of incoming waves. As a result, the performance advantage of the unidirectional turbine could be reduced in practice. Further studies, both numerical and experimental, are needed to quantify the degree of unidirectionality achievable in realistic operating conditions.

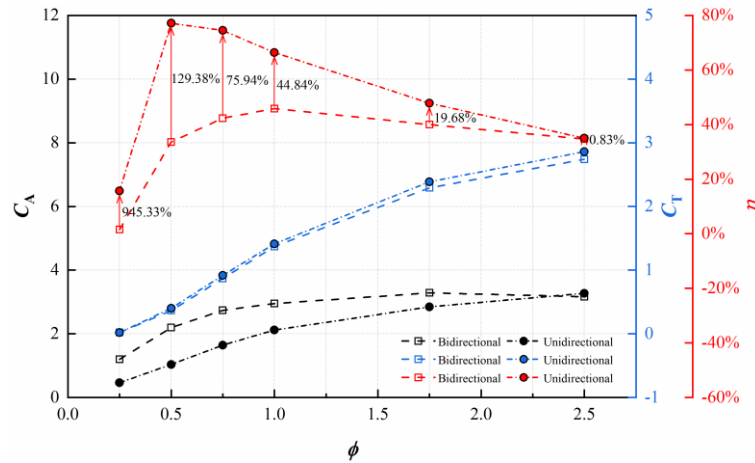


Fig. 13. Comparison of the performance between the traditional bidirectional turbine and the optimized unidirectional turbine in terms of C_A , C_T , and η as a function of ϕ .

Fig. 14 demonstrates pressure distribution on both the pressure and suction sides of the traditional bidirectional turbine and the optimized unidirectional turbine for $\phi = 0.5$. It can be observed that, on the pressure side of the unidirectional turbine, the pressure distribution becomes more uniform along the airflow direction compared to that of the traditional design.

Furthermore, the negative pressure region near the leading edge is significantly reduced in both area and intensity, as evidenced by the shrinkage of the corresponding range. This change effectively reduces the energy loss caused by the return flow on the pressure side.

On the suction side, both the area and peak value of the high-pressure zone at the leading edge are reduced in the unidirectional turbine. Additionally, the negative pressure distribution on the suction side is also more uniform, which helps to alleviate excessive pressure gradients and reduce the risk of blade damage. Moreover, the optimised negative pressure zone on the suction side is shifted toward the rear edge, aligning with the high-pressure zone on the pressure side. This rearrangement enhances the driving torque and contributes to improved overall mechanical efficiency of the turbine.

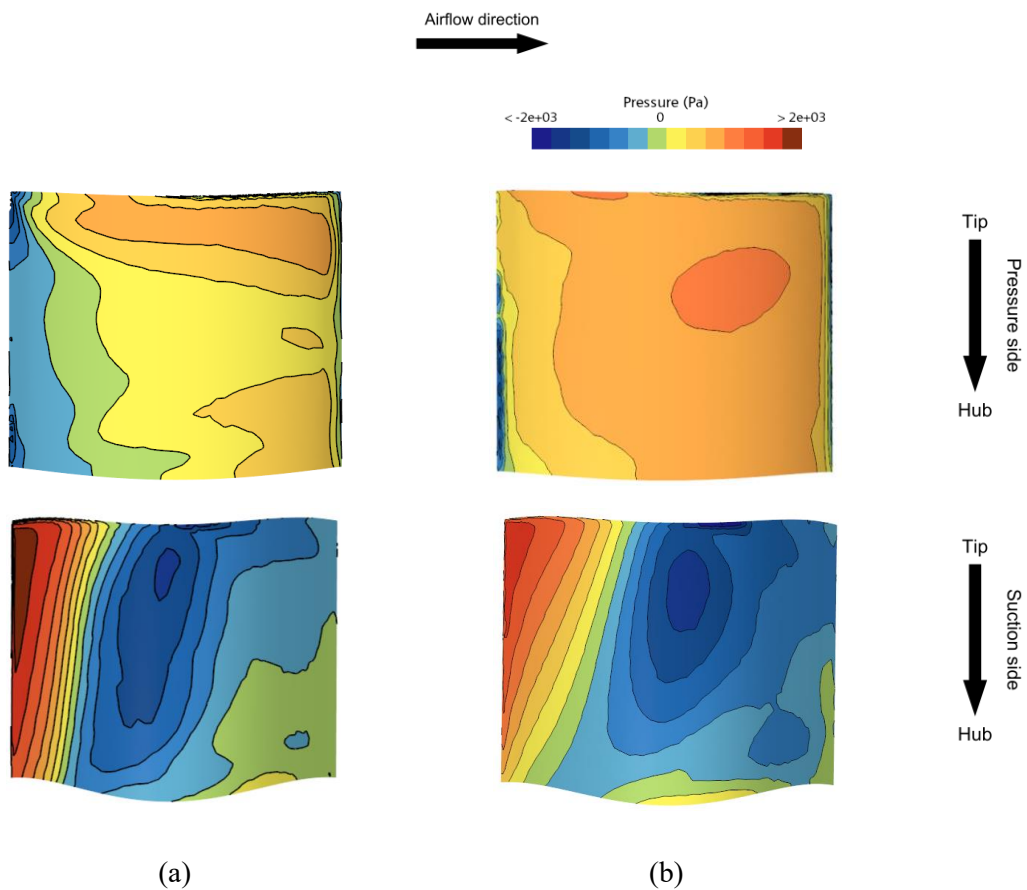
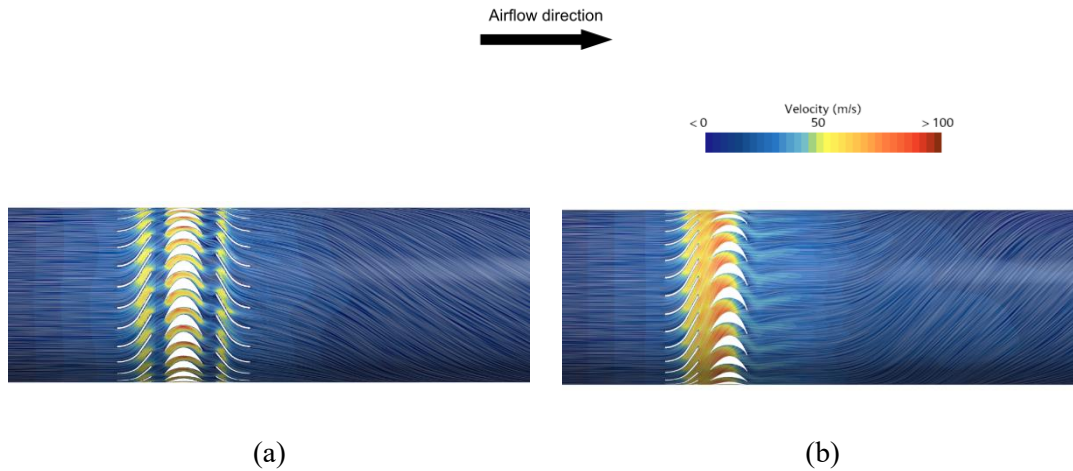


Fig. 14. Pressure distribution on both the pressure and suction sides of a blade for $\phi = 0.5$: (a) traditional bidirectional turbine; (b) optimized unidirectional turbine.

错误!未找到引用源。 compares the streamline and velocity distribution at the blade-height $H/b = 0.5$ cross-section between the traditional bidirectional turbine and the optimized unidirectional turbine. In the traditional turbine (错误!未找到引用源。 (a)), the flow exiting

445 the guide-vane passage shows noticeable high-velocity streaks concentrated near the rotor-
 446 blade leading edges. These streaks transition abruptly into the main flow, producing visible
 447 discontinuities in the streamline pattern and indicating stronger velocity gradients. In contrast,
 448 the optimized unidirectional turbine (错误!未找到引用源。 (b)) presents a significantly
 449 smoother flow field. The high-velocity region at the rotor inlet becomes narrower and more
 450 uniformly distributed along the blade span, reducing sharp transitions between adjacent flow
 451 channels. The streamlines in 错误!未找到引用源。 (b) remain more coherent as they pass
 452 through the blade passages, suggesting reduced shear interaction and weakened flow mixing.
 453 Additionally, in the downstream region, the optimized turbine exhibits a more uniform medium-
 454 velocity zone, while the traditional turbine shows clear patches of low-velocity and high-
 455 velocity fluctuation. This broader and more stable wake structure in Fig. 15(b) indicates that
 456 the optimized design effectively mitigates flow separation and suppresses large-scale
 457 unsteadiness, thereby reducing energy loss.



458 Fig. 15. Streamline and velocity distribution at blade height cross-section $H/b = 0.5$ ($\phi = 0.5$):
 459 (a) traditional bidirectional turbine; (b) optimized unidirectional turbine.

460 Fig. 16 compares the vorticity distribution derived from the Q-criterion isosurfaces ($Q =$
 461 $1.0 \times 10^4 \text{ s}^{-2}$) and the blade-height cross-section for the traditional bidirectional turbine and the
 462 optimized unidirectional turbine. The quantitative statistics demonstrate a clear improvement
 463 in the vortex characteristics after optimization. Specifically, the surface standard deviation of
 464 vorticity decreases by 12.1% (from $6.20 \times 10^3 \text{ s}^{-1}$ to $5.45 \times 10^3 \text{ s}^{-1}$), while the surface-averaged
 465 vorticity decreases by 9.9%. The concurrent reduction in both fluctuation intensity and mean

vorticity level indicates that the optimized configuration suppresses strong shear layers and mitigates the formation of energetic vortical structures within the flow passage. Consistent with this, the traditional turbine generates large, continuous vortex tubes downstream of the guide-vane trailing edges, whereas the optimized turbine exhibits fewer and smaller coherent vortices that decay more quickly, producing a smoother and more uniform vorticity field. This result confirms the effectiveness of the optimized design in mitigating large-scale coherent vortices and enhancing the aerodynamic performance of the turbine

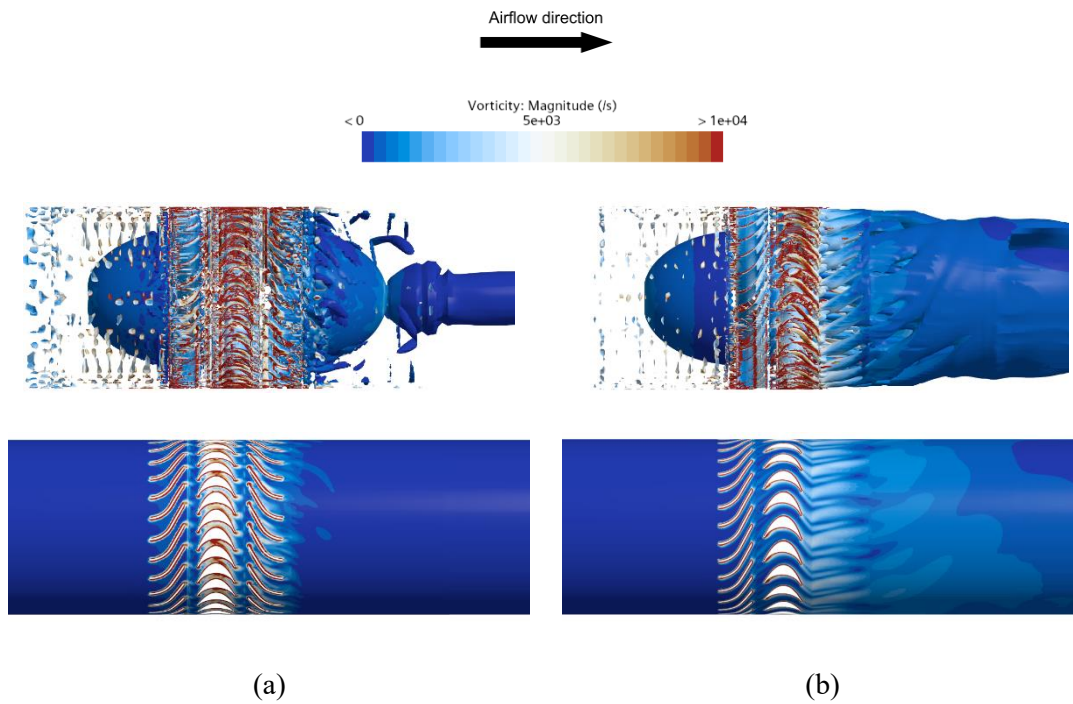


Fig. 16. Vorticity distribution on Q-Criterion isosurfaces and at the specified blade height cross-section $H/b=0.5$ ($\phi=0.5$): (a) traditional bidirectional turbine; (b) optimized unidirectional turbine.

5. Conclusions

This study proposed and numerically investigated a novel unidirectional impulse turbine integrated with a dual-chamber OWC wave energy conversion system. A steady-state CFD model was established based on viscous fluid theory and validated against experimental data. Through a comprehensive parametric analysis, the aerodynamic performance of the unidirectional turbine was significantly improved across a wide range of flow coefficients. The main conclusions are as follows:

(1) The rotor blade number, guide vane number, and blade angles were identified as key parameters affecting turbine performance. A moderate blade number and slight negative rotor angle were found to be optimal for balancing aerodynamic loading and minimizing losses.

(2) Detailed pressure and velocity field analyses revealed more uniform blade loading, reduced flow separation, smoother downstream wake, and mitigated turbulence intensity in the optimized turbine, contributing to higher mechanical and energy conversion efficiency.

(3) The optimized turbine configuration—with 22 rotor blades and 30 guide vanes, rotor angle -4° , and guide vane angle 2° —reduced the total number of blades by 26.67% and guide vanes by 42.31%, leading to lower manufacturing costs and simpler assembly. Compared to the baseline bidirectional turbine, the optimized unidirectional turbine achieved a 59.89% increase in average efficiency (from 33.0% to 52.8%) and a 67.97% improvement in peak efficiency (from 45.9% to 77.1%). The maximum efficiency gain under specific flow conditions reached 129.38%.

Despite these promising results, the present study is limited to steady-state simulations under constant airflow conditions, which do not fully capture the transient and bidirectional nature of airflow in real OWC environments. Therefore, future research will focus on conducting transient simulations to evaluate the unsteady aerodynamic performance of the turbine under oscillating flow conditions, as well as performing experimental validation in wave flume or wind tunnel facilities to confirm the aerodynamic improvements and efficiency gains observed in the numerical analysis.

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