

Article

Computational Fluid Dynamics Simulation on Blade Geometry of Novel Axial Flow Turbine for Wave Energy Extraction

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Abstract: Wave energy converters (WECs) utilizing the Oscillating Water Column (OWC) principle have gained prominence for harnessing kinetic energy from ocean waves. This study explores an innovative approach by transforming the pivoting Denniss–Auld turbine blades into a fixed configuration, offering a simplified alternative. The fixed-blade design emulates the maximum pivot points during the OWC’s exhalation and inhalation phases. Traditional Denniss–Auld turbines rely on complex hub systems to enable controllable blade rotation for performance optimization. This research examines the turbine’s efficiency without mechanical actuation. The simulations were conducted using ANSYS™ CFX 2023 R2 to solve the three-dimensional, incompressible, steady-state Reynolds-Averaged Navier–Stokes (RANS) equations, employing the $k-\omega$ SST turbulence model to close the system of equations. A grid convergence study was performed, and the numerical results were validated against available experimental and numerical data. An in-depth analysis of the intricate flow field around the turbine blades was also conducted. The modified Denniss–Auld turbine demonstrated a broad operating range, avoiding stalling at high flow coefficients and exhibiting performance characteristics like an impulse turbine. However, the peak efficiency was 12%, significantly lower than that of conventional Denniss–Auld and impulse turbines. Future research should focus on expanding the design space through parametric studies to enhance turbine efficiency and power output.



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Keywords: wave energy; Wells turbine; modified Dennis–Auld turbine; grid independence

1. Introduction

The rapid expansion of the world economy and population is driving a surge in global energy demand. Consequently, greenhouse gas emissions have significantly increased, leading to ecological challenges and negatively impacting the Earth’s climate. Therefore, it is crucial to explore clean energy solutions incorporating renewable energy sources to address the growing energy needs and environmental concerns.

Renewable energy sources have the potential to meet two-thirds of global energy demand and help mitigate global temperature rise by preventing greenhouse gas emissions. Reducing greenhouse gas emissions from now until 2050 is essential to limiting the average global surface temperature increase to 2 °C [1]. Consequently, global leaders and policy-makers have implemented measures to adopt and integrate renewable energy technologies in response to climate change.

Ocean wave energy has emerged as one of the most promising eco-friendly alternatives to fossil fuels among the various renewable energy sources. Over the past two decades, it has garnered significant interest due to its potential to substantially supplement the electricity demand of many countries [2]. Furthermore, the high energy density, minimal ecological impacts, and negligible energy losses of waves traveling long distances have intrigued researchers, driving the development of feasible technologies to harness wave energy [3]. Current research indicates that the potential of ocean wave energy is approximately 30,000 TWh per year. If harvested efficiently, this could meet global electricity

demands [4]. Wave energy converters (WECs) extract energy from waves, converting it into mechanical energy and electrical energy. The commercialization of wave energy hinges on enhancing the performance of WECs [5]. In recent decades, technologies such as the Oscillating Water Column (OWC), floating buoys, point absorbers, and overtopping devices have been extensively explored for wave energy conversion. However, due to their simplicity, OWC WECs are the most widely used devices for converting wave energy [6].

In an OWC, the rotor is the only moving part, rotating at high rpm and connected to an electrical generator to generate electricity. The OWC utilizes the oscillatory motion of ocean waves, where air becomes trapped above the free water surface and undergoes cycles of inhalation and exhalation. This results in the OWC system experiencing unsteady bi-directional airflow, necessitating specialized Power Take-Off (PTO) systems capable of managing this complex flow. According to recent research, the most widely used turbines in OWCs are the Wells [7] and impulse-type concepts [8].

The Wells turbine stands out among axial flow air turbines as one of the most economical and straightforward methods for harnessing wave energy. Proposed by Professor Wells, this self-rectifying axial flow turbine operates within an OWC using compressed air. Its rotor comprises several symmetrical airfoil blades arranged radially and staggered at 90 degrees around the hub (see Figure 1). Due to the symmetric orientation of the blades, the tangential force acting on them remains consistent regardless of the airflow direction, as depicted in Figure 2. Consequently, the Wells turbine consistently rotates in the same direction, even when exposed to bi-directional airflow.

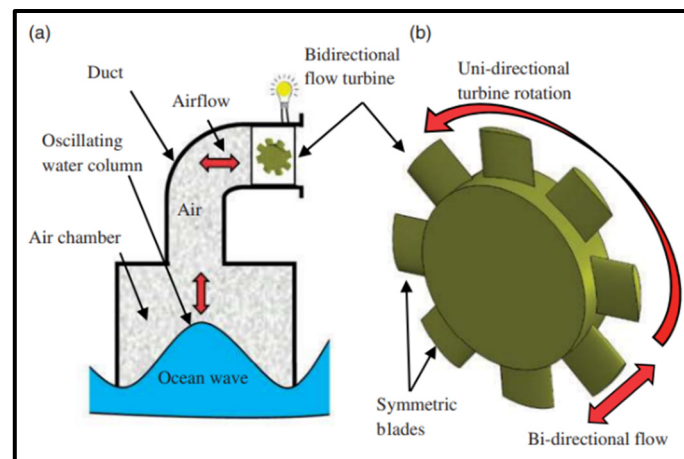


Figure 1. Schematic diagram of OWC and Wells turbine [9]. (a) OWC. (b) Wells turbine.

Despite its advantages, the Wells turbine suffers from inherent shortcomings such as a narrow operating range, a low aerodynamic efficiency, high noise levels, a high axial force coefficient, a low tangential force coefficient, and poor starting characteristics compared to conventional turbines (Figure 3). Consequently, several self-rectifying air turbines have been developed, analyzed, and continuously improved to address these limitations.

Wave energy converters utilizing self-rectifying impulse turbines are gaining popularity as viable alternatives to the Wells turbine [10]. Two sets of guide vanes are symmetrically arranged on both sides of the rotor in bi-directional turbines, whereas uni-directional turbines have only one set.

Setoguchi et al. [11] examined the current advancements in wave energy conversion utilizing self-rectifying air turbines. They conducted numerical simulations to evaluate the overall performance of these turbines under irregular sea wave conditions, focusing on their starting and running characteristics. The study found that impulse turbines demonstrated superior running performance compared to the Wells turbine. Due to their rapid start capability, impulse turbines have a longer electricity generation duration than

Wells turbines. Moreover, they operate at lower rotational speeds due to higher torque coefficients under load-free conditions than Wells turbines.

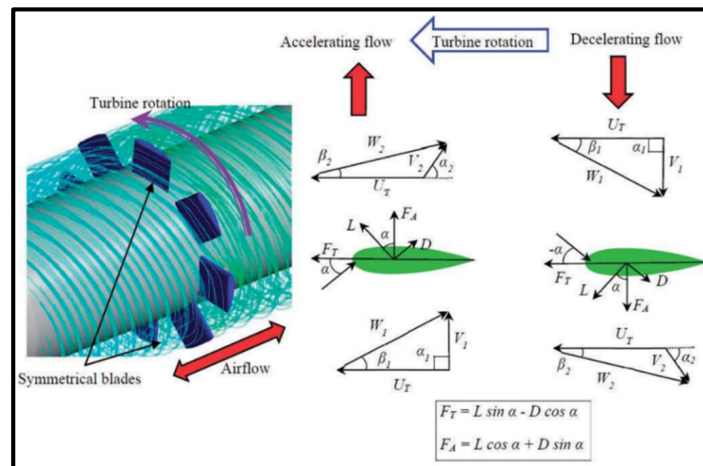


Figure 2. Uni-directional rotation of the Wells turbine blade [12].

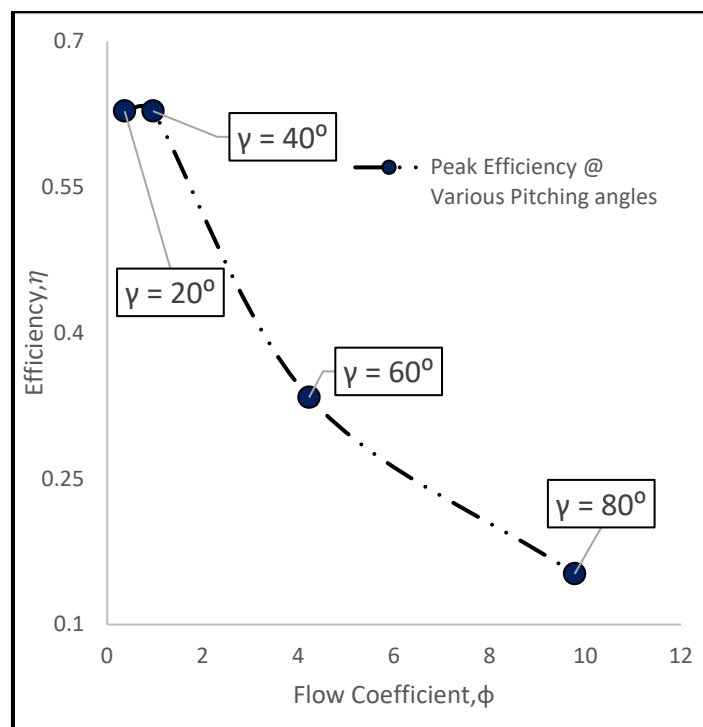


Figure 3. Operating curve of Denniss–Auld turbine [12].

Falcao et al. [13] assessed the performance of different self-rectifying turbines across various ocean wave energy levels and OWC applications. The study revealed that the Wells turbine can achieve a peak efficiency of 75%, a benchmark surpassed only by the most advanced impulse turbines. However, in energetic sea conditions, the average efficiency of the Wells turbine drops significantly compared to that of impulse turbines. It is generally observed that the average efficiency of impulse turbines remains consistent regardless of sea state. However, it typically relies on movable guide vanes for optimal performance in all conditions.

Hu et al. [14] conducted a numerical study on both steady and unsteady flow characteristics in a Wells turbine. They observed a stall near the blade tip under steady flow conditions, while radial flow helped maintain attached boundary layers near the hub. The study investigated two control methods for unsteady flow: constant angular velocity and constant damping moment. Regarding constant angular velocity, the authors noted the hysteresis effect impacting the Wells turbine, particularly at high frequencies of sea waves.

Halder et al. [15] performed a multi-objective optimization of blade sweep for a Wells turbine. As a result, they achieved a 28.28% increase in the peak torque coefficient. However, this improvement came at the cost of a 13.5% decrease in efficiency.

Das et al. [16] experimentally studied a biplane Wells turbine under varying loads and periodic airflows. They observed that during the turbine's acceleration and deceleration phases, the pressure drop and torque coefficients exhibited hysteresis behavior, attributed to the OWC turbine system's capacitive characteristics.

Wang et al. [17] introduced an innovative biomimetic design inspired by hawkmoth wings for the Wells turbine, aiming to enhance its efficiency in energy harvesting. The study found that the biomimetic design modifies the flow field and can achieve higher peak efficiency, especially at high angles of attack.

Folley et al. [18] assessed the performance of the contra-rotating Wells turbine installed in the LIMPET wave power station based on a theoretical analysis and model tests. The study suggests that a biplane or monoplane Wells turbine with guide vanes may offer better performance for OWC applications than the contra-rotating Wells turbine, which exhibited reduced efficiency in their evaluations.

In response to the limitations of Wells and impulse turbines, researchers have developed alternative self-rectifying axial flow turbines such as the HydroAir turbine and the Denniss–Auld turbine. The HydroAir turbine addresses these challenges by increasing the distance between the guide vane rows and rotor blades, reducing the excessive flow angle encountered at the second row of guide vanes [19]. Decreasing the flow velocity at the entrance to the second row of guide vanes can reduce kinetic energy losses.

Another type of self-rectifying axial flow turbine is the Denniss–Auld turbine [20,21]. The Denniss–Auld turbine and the variable-pitch Wells turbine share similar characteristics but differ in the angles at which their blades are set (Figure 4). For instance, the rotor blades are positioned in the range of $\alpha < \gamma < \pi - \alpha$ and $-\theta < \gamma < \theta$ for the Denniss–Auld turbine and the variable-pitch Wells turbine, respectively, where $\alpha \cong 20^\circ - 35^\circ$, $\theta \cong 25^\circ$.

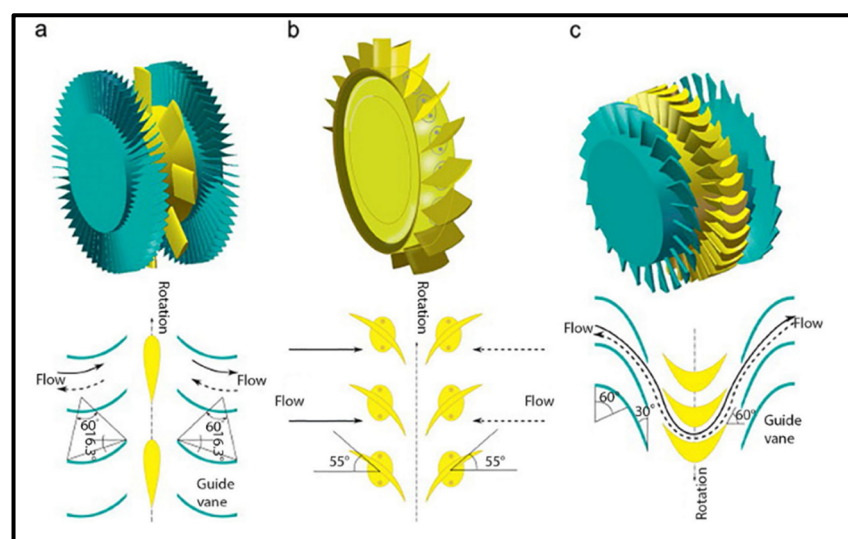


Figure 4. Air turbines: Wells turbine, Denniss–Auld turbine, and impulse turbine [12]. (a) Wells turbine. (b) Denniss–Auld turbine. (c) Impulse turbine.

Furthermore, in the Wells turbine, the flow attaches to the blade's leading edge. In contrast, the Denniss–Auld rotor blades alternate between acting as leading edges and trailing edges depending on the direction of fluid flow. This characteristic necessitates that the rotor blades of the Denniss–Auld turbine are designed with identical edges.

When the OWC system inhales or exhales, the rotor blades of the Denniss–Auld turbine must pivot rapidly between their extreme positions. In contrast, Wells turbine rotor blades require smaller angular ranges of blade pitch to operate effectively.

The blade pitching angles are critical to the aerodynamic performance of a Denniss–Auld turbine. Figure 3 shows the operating curve of the Denniss–Auld turbine for variable blade pitch angles, γ . According to the analytical model prediction using blade element theory, the peak efficiency occurs at around $\gamma = 40^\circ$ and decreases with an increase in γ . So, the optimum blade angle should be between 20° and 40° . This entails the basis of our design comprising $\gamma = 30^\circ$ (Figure 5) for improved performance.

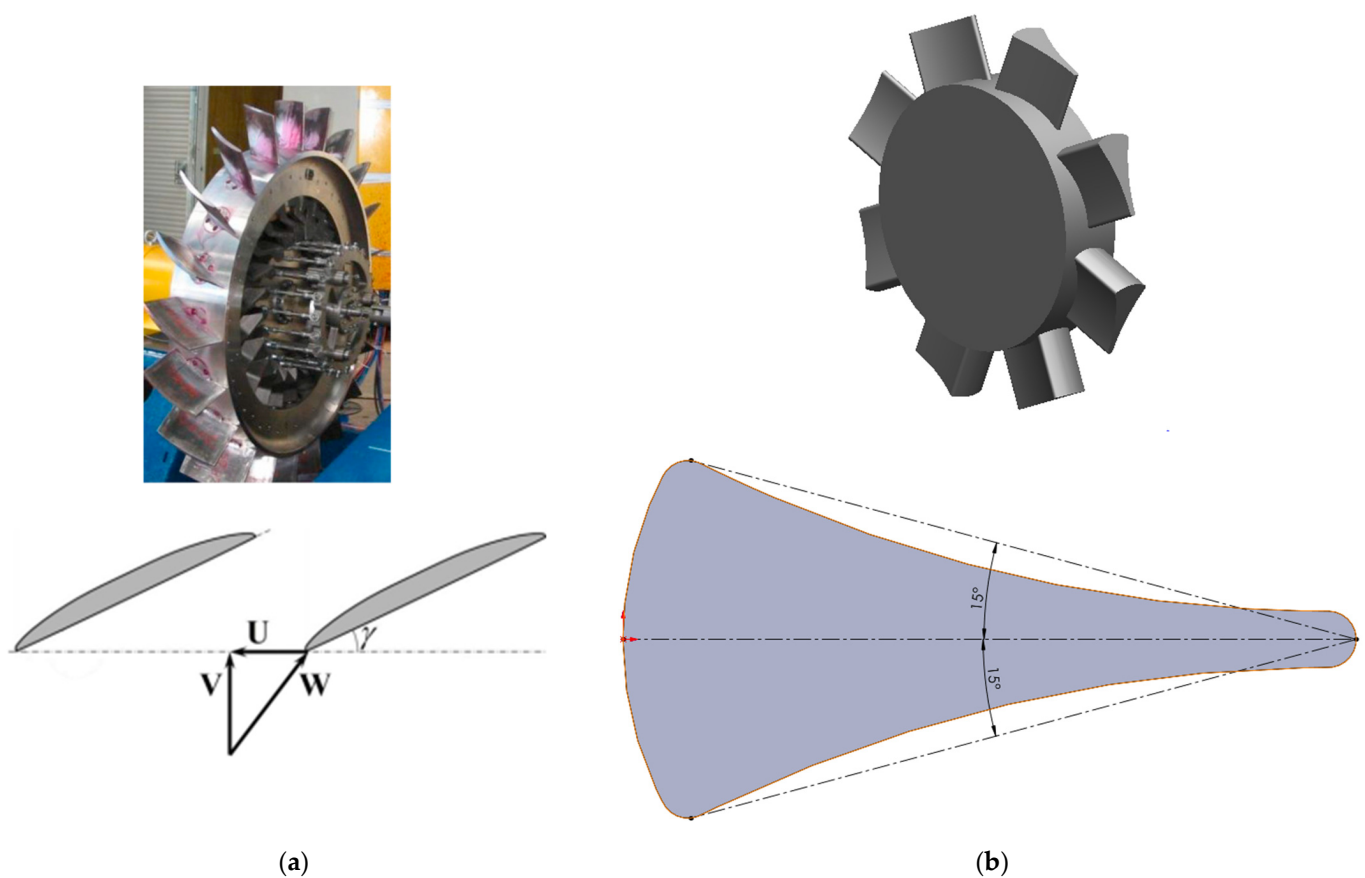


Figure 5. (a) Denniss–Auld turbine and (b) modified Denniss–Auld turbine.

This study addresses the difficulties mentioned above by redesigning the Denniss–Auld turbine blades to reduce mechanical complexity by preventing pivoting. In addition, we have investigated the aerodynamic performance characteristics of a modified Denniss–Auld turbine comprising novel airfoil-shaped blades using ANSYSTM CFX 2023 R2 by solving the steady-state, incompressible, three-dimensional Reynolds-Averaged Navier–Stokes (RANS) equations coupled with the $k-\omega$ SST turbulence model [22–24]. Ultimately, the novelty of this research is the design of a modified Denniss–Auld turbine without pivoting with the goal of mitigating stall at high flow coefficients. Under steady-state flow conditions, we determined the turbine's performance in non-dimensional coefficients, i.e., torque coefficient, efficiency, and input coefficient, for a wide range of flow coefficients. Numerical accuracy was achieved through a grid independence study [4,22,25–29]. Furthermore, we

validated our numerical modeling by comparing the results to available experimental [30] and numerical results [31] for a baseline Wells turbine with a uniform tip clearance.

2. Numerical Methods

Numerical simulations for this study were performed using the pressure-based solver within ANSYSTM CFX, where velocity and pressure equations are solved simultaneously by treating them as a single equation system. In addition, a fully implicit discretization approach was undertaken to discretize the equations for computational modeling. For instance, the advection and diffusion terms in the RANS equations were treated with a second-order scheme and a shape function-based approach, respectively. The details of the equations can be found in reference [32].

When computing turbulent flow, its pressure and velocity field decompose into mean and fluctuating parts. RANS equations, governing mean flow, are derived from averaging the Navier-Stokes equations over time. However, due to the inherently nonlinear nature of the Navier-Stokes equations, the velocity field still experiences fluctuations resulting from the convective acceleration (Equation (1)) [33]. As a result, a nonlinear term, the Reynolds stress, appears, which Boussinesq considers a function of the mean flow component concerning the closure problem (Equation (2)) [33].

This study adopts the $k\text{-}\omega$ SST turbulence model since it predicts flow separation more accurately than other RANS models. In addition, the model can also resolve complex boundary layer flows with adverse pressure gradients (e.g., Turbomachinery).

$$\frac{\partial(\rho U)}{\partial t} + \nabla \cdot (\rho U U) = -\nabla P + \nabla \cdot \left[\mu \left(\nabla U + (\nabla U)^T \right) \right] + \rho g - \nabla \cdot \left(\frac{2}{3} \mu (\nabla \cdot U) \right) - \underbrace{\nabla \cdot (\rho U' U')}_{\text{ReynoldsStress}}, \quad (1)$$

$$-\rho U' U' = \nu_t \underbrace{\left(\nabla U + (\nabla U)^T \right)}_{\text{MeanVelocityGradients}} - \frac{2}{3} \rho k \delta_{ij}, \quad (2)$$

where δ_{ij} is the Kronecker delta.

2.1. Numerical Modeling

The performance characteristics of a Wells turbine comprising symmetrical NACA0015 blades, fixed at a 90-degree staggered angle with uniform tip clearance, were assessed numerically and compared to existing data. Because tip leakage is among the most significant secondary fluid flow phenomena found in Turbomachinery, tip clearance has been used to predict turbine performance accurately. The specifications of the simulated turbine are given in Table 1.

When performing CFD analyses, it is necessary to discretize the computational domain to solve the governing equations for fluid flow. Therefore, the CFD results are prone to discretization errors. The quality of the computational grid can often become the primary cause of the error. Figure 6 depicts the unstructured mesh used in this study.

Within ANSYS meshing, we can specify a cell size within a volume. Even though the mesh is unstructured and the cell sizes vary slightly, we selected four grid sizes for the mesh resolution within our control volume: coarse (10 mm), medium (5 mm), fine (2.5 mm), and extra-fine (1.25 mm).

It is necessary to resolve the boundary layer flow adequately around the rotor blades to determine the dynamic stall point of the turbine accurately. For this reason, hexahedron inflation layers were densely placed around the blade surface to resolve the boundary layer. The computational domain extends four and six times the axial chord length in the leading- and trailing-edge directions, respectively. The boundary condition at the inlet is set to a uniform axial velocity with a specified turbulence intensity of 5%. A uniform static pressure boundary condition is imposed at the outlet of the computational domain. No-slip wall boundary conditions are set on the blade surface, hub, and tip.

Considering that the Wells turbine has a symmetrical geometry, the computations are limited to one blade-to-blade passage along with periodic boundary conditions, as shown in Figure 7. The details of the meshing and boundary conditions are provided in Table 2. From the definition, the dimensionless wall distance, y^+ , is given by the following [31]:

$$y^+ = \frac{yu_T}{\nu}, \quad (3)$$

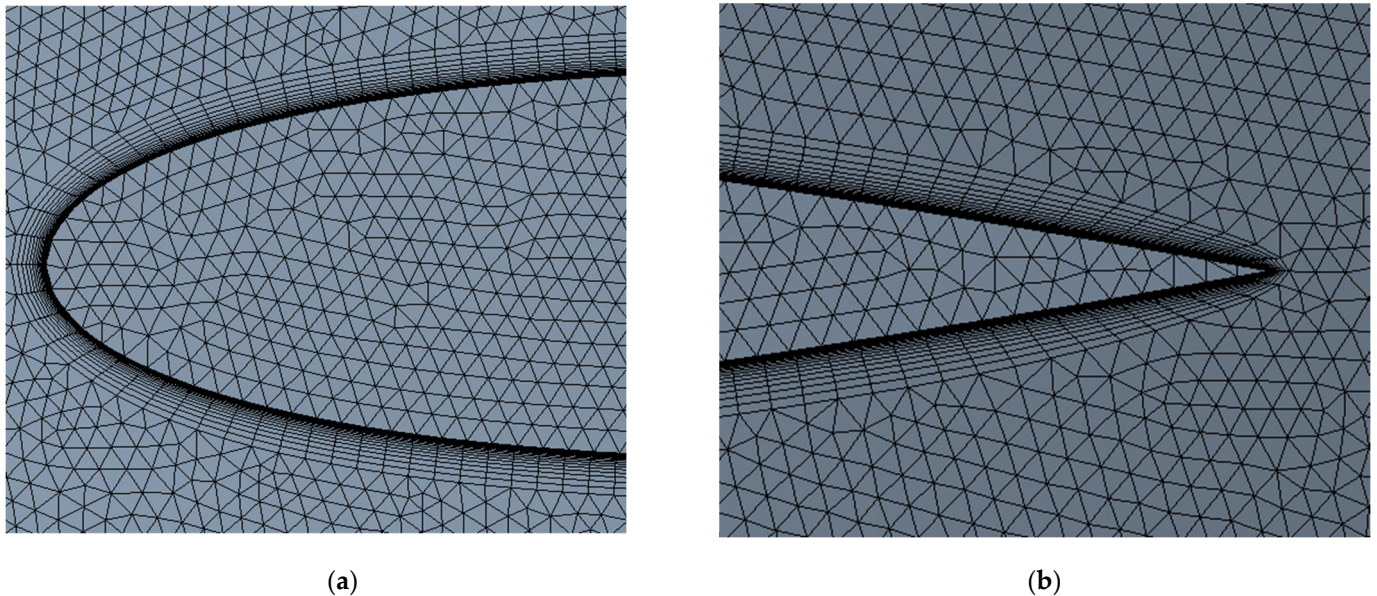


Figure 6. Computational mesh generation around the blade: (a) mesh near leading edge and (b) mesh near trailing edge.

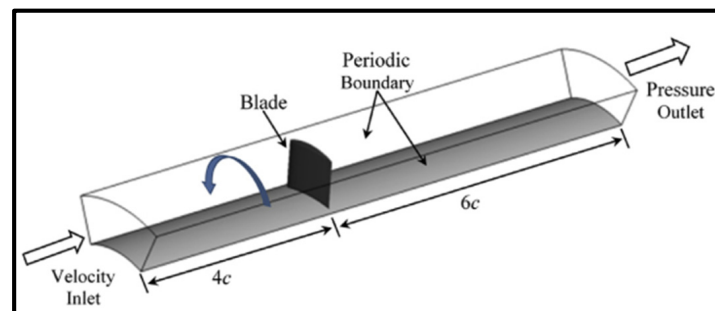


Figure 7. Computational domain and boundary conditions.

Where y , u_T , and ν signify the absolute first layer distance from the wall, friction velocity, and kinematic viscosity of the fluid, respectively. The $k-\omega$ SST turbulence model, because it is sensitive to the mesh, requires high resolution with a value of $y^+ < 1$ to model the viscous sublayer region. Accounting for the required y^+ value, the first prism layer height obtained was 1.1×10^{-5} m. There was a total of 20 inflation layers, with a growth factor of 1.2.

In the fluid zone, the Moving Reference Frame (MRF) was used, which rotates at the same speed as the turbine rotor. Hence, the following conservation equations were solved:

I. The conservation of mass

$$\frac{\partial \rho}{\partial t} + \nabla \cdot \rho \vec{v}_r = 0, \quad (4)$$

II. The conservation of momentum

$$\frac{\partial(\rho \vec{v})}{\partial t} + \nabla \cdot (\rho \vec{v} \vec{v}) + \rho [(\omega \hat{a}) \times (\vec{v} - \vec{v}_t)] = -\nabla p + \nabla \cdot \underline{\tau} \quad (5)$$

Here, $\vec{v}_t$, $\vec{v}$, ω , $\underline{\tau}$, and $\hat{a}$ indicate the transitional velocity, absolute velocity, angular velocity, viscous stress, and a unit vector defining the axis of rotation, respectively.

The performance characteristics of the turbine were assessed for various flow coefficients, ϕ , where

$$\phi = \frac{v}{U_{tip}}, \quad (6)$$

Since we used pseudo-transient simulations in ANSYS CFXTM, time step values were required even with steady flow. Therefore, for the first few flow configurations ($\phi < 0.2$) far from the blade stall, the time step size, Δt , was set as 0.05 s, selected from the Auto Timescale option [32]. However, the results were oscillatory and unsteady for the pre- and post-stall conditions ($\phi > 0.2$). So, we reduced to $\Delta t = 0.005$ s; therefore, the fluctuations decreased, leading to convergence.

Table 1. Main features of the turbine models.

Features	Parameters
Number of blades	8
Blade chord length	125 mm
Hub diameter	400 mm
Tip diameter	600 mm
Solidity at mean radius	0.64
Tip clearance	1% of the chord length
Hub-to-tip ratio	0.67
Aspect ratio	0.75

ANSYS CFX uses second-order accurate approximations. The second-order backward Euler scheme achieves time discretization with implicit and conservative time stepping. For additional details, please see reference [34].

Table 2. Meshing and appropriate boundary conditions.

Parameter	Single Turbine
Mesh/nature	Unstructured
No. of inflation layers	20
y^+	<1
Fluid	Air at 25 °C
Turbulence model	k- ω SST
Inlet	Uniform inlet velocity
Outlet	Pressure outlet
Hub, tip, and blade	No-slip wall
Number of revolutions	2000 rpm
Convergence criterion	1×10^{-5}
Mass imbalance	0.001

The variation in the flow coefficient is achieved by changing the axial velocity, v , whereas the circumferential velocity at the tip radius, U_{tip} , remains constant. The root mean square (RMS) residuals of the governing equations are monitored to determine convergence. As part of the convergence verification, torque output and pressure drop across the turbine are also observed. Sometimes, set values of the convergence criteria of each residual are lowered to guarantee that the monitored quantities remain constant. The net mass imbalance in this study was less than 0.058 percent.

2.2. Validation of the Numerical Model

Our research considers seven steady-state flow conditions, each characterized by an associated flow coefficient, ϕ . The turbine operating range is chosen as $0.075 \leq \phi \leq 0.275$, or equivalently $4^\circ \leq \alpha \leq 15^\circ$, where α is the incident angle of attack.

According to the literature [3], the following non-dimensional coefficients have been used to describe turbine performance:

The torque coefficient, C_T ,

$$C_T = \frac{T}{\rho \omega^2 R^5}, \quad (7)$$

The pressure drop coefficient, ΔP_0^* ,

$$\Delta P_0^* = \frac{\Delta p_0}{\rho \omega^2 R^2}, \quad (8)$$

The input coefficient, C_A ,

$$C_A = \frac{2\Delta p_0 Q}{\rho (v^2 + U_R^2) b c z v}, \quad (9)$$

The efficiency, η ,

$$\eta = \frac{T \omega}{\Delta p_0 Q}, \quad (10)$$

where Q , ρ , T , Δp_0 , ω , z , U_R , c , b , and v represent the volumetric flow rate, air density, blade torque, static pressure drops across the turbine, angular velocity, number of blades, circumferential velocity at blade mean radius, blade chord length, blade height, and axial inlet velocity, respectively.

The Reynolds number is defined as follows:

$$Re = \frac{\rho c \sqrt{(v^2 + U_{tip}^2)}}{\mu}, \quad (11)$$

A grid independence study is essential for obtaining accurate results in numerical simulations. Therefore, we picked four different cell sizes to study the effect of grid resolution on the results. Then, we compared the numerical and experimental results of the computed torque coefficients for each mesh at $\phi = 0.225$.

As evident in Figure 8, there is a substantial drop in the torque coefficient below the medium grid. The discrepancy between the medium, fine, and extra-fine grids was ~24%, ~4%, and ~0.5%, respectively. Due to the negligible variation in the torque coefficient between the fine and extra-fine grid, the fine grid was used for subsequent simulations. To simulate the flow, the workstation specifications of Intel (R) Xeon (R) Gold 6258 R CPU @ 2.7 GHz and 2.69 GHz (2 processors) with a 56-core system were used. The equipment was sourced from Dell, based in Round Rock, TX, USA. A table listing the cell size statistics and a figure showing the grid independence study are provided in Table 3 and Figure 8, respectively.

We selected the torque coefficient as the parameter evaluated for each mesh to reach grid independence, consistent with the literature [35,36]. However, some studies [37] also employed efficiency too. In either case, we see a negligible change in the results for the extra-fine grid (See Table 4).

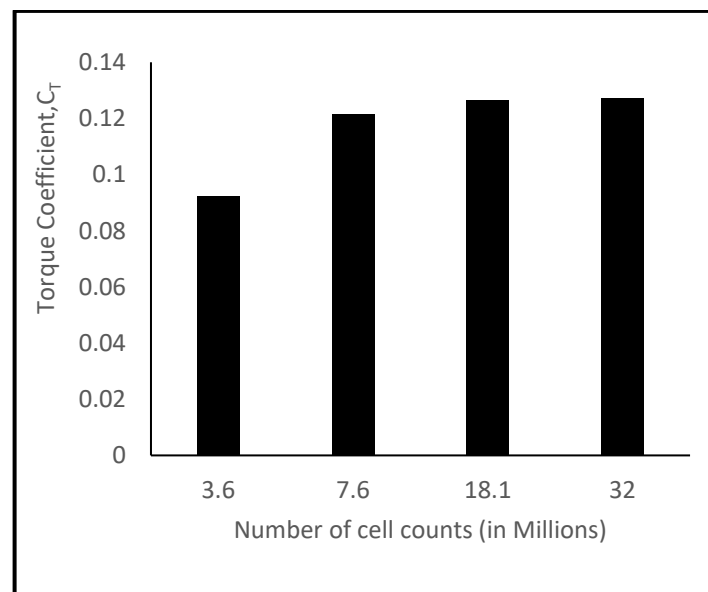


Figure 8. Grid independence study on turbine torque coefficient at $\phi = 0.225$.

Table 3. Grid sizes.

Mesh	No. of Cells (Millions)	Cell Size (mm)	Turbulence Model	Simulation Time
Coarse	3.6	10.0	k- ω SST	8 h
Medium	7.6	5.0		14 h
Fine	18.1	2.5		20 h
Extra-fine	32	1.25		26 h

Table 4. Uncertainty estimation.

Parameters	Numbers	
Grid size (mm)	Extra-fine, fine, medium, and coarse	1.25, 2.5, 5.0, 10.0
Average cell size (h)	h_1, h_2, h_3, h_4	8×10^{-4} m, 0.0017 m, 0.0033 m, 0.0057 m
Grid refinement factor (r)	r_{21}, r_{32}, r_{43}	2.07, 1.9681, 1.7318
Performance parameter (ϕ)	$\phi_1, \phi_2, \phi_3, \phi_4$	0.1273, 0.1267, 0.1216, 0.0925
Apparent order	p	2.83
Approximate relative error	$e_a^{21}, e_a^{32}, e_a^{43}$	0.0050, 0.0403, 0.2393
Extrapolated values	$\phi_{ext}^{21}, \phi_{ext}^{32}, \phi_{ext}^{43}$	0.1279, 0.1319, 0.1613
Extrapolated relative error	$e_{ext}^{21}, e_{ext}^{32}, e_{ext}^{43}$	0.0096, 0.0787, 0.4268
GCI	$GCI_{extrafine}^{21}, GCI_{fine}^{32}, GCI_{medium}^{43}$	0.0058, 0.0520, 0.4088

3. Results and Discussion

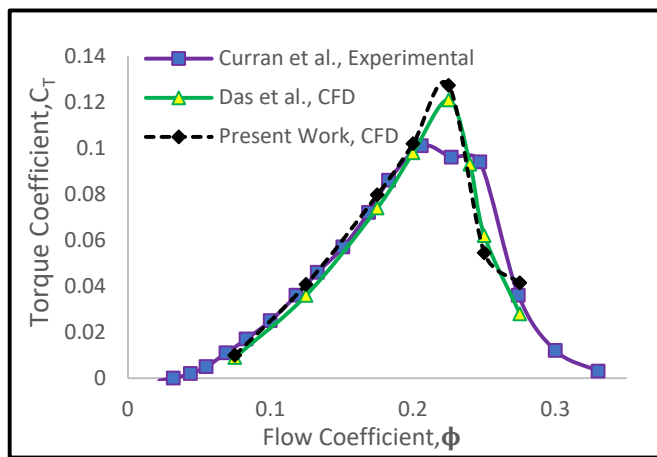
This study examined turbine performance for a constant rotational speed of 2000 rev/min at various inlet velocities. The flow coefficient ϕ captures the operating range of the turbine, where $0.075 \leq \phi \leq 0.275$. Detailed information on the range of the inlet velocity, flow coefficient, and Reynolds number is shown in Table 5.

In the current CFD simulations, we compare the experimental and numerical data. Comparing the computed results to the experiment [30] and CFD [31] data, Figure 9 illustrates a high level of agreement up to $\phi = 0.2$ for the pressure drop coefficient (ΔP_0^*), torque coefficient (C_T), and efficiency (η). But for $\phi > 0.2$, i.e., at a high angle of attack, deep stalling is evident from the rapid drop in blade torque and turbine efficiency. This is a result of leading-edge flow separation. Furthermore, the RANS simulations overestimate the

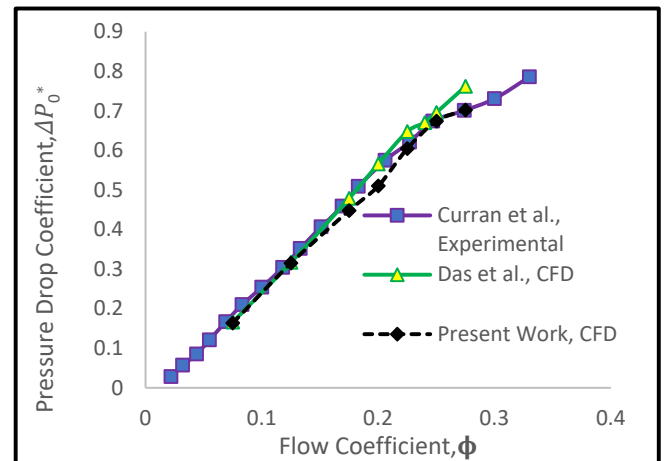
torque. This is typical of even well-resolved RANS computations; they are often inaccurate past the blade stall point [38]. Solving complex three-dimensional separated flow is a principal factor in the resulting deviation in the numerical and experimental results. Yet, under the same geometrical conditions, the present work is consistent with existing CFD results [31].

Table 5. Inlet axial velocity and flow coefficient ranges.

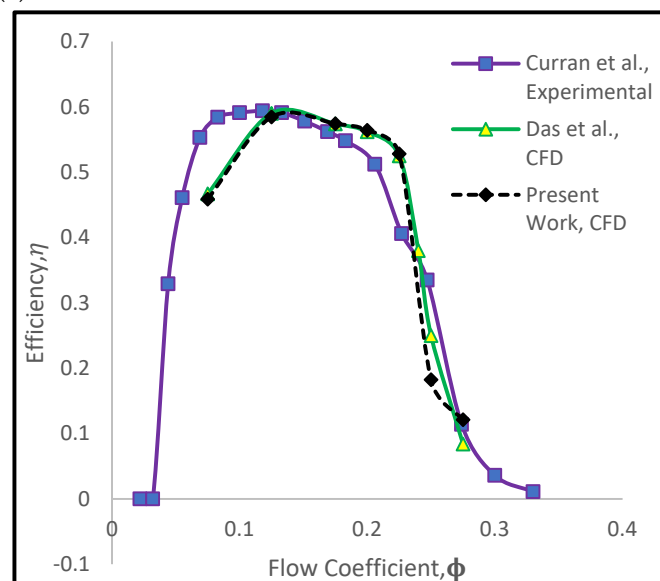
Flow Coefficient, ϕ	Inlet Axial Velocity, v (m/s)	Reynolds Number, Re
0.075	4.71	5.25×10^5
0.125	7.85	5.27×10^5
0.175	10.99	5.31×10^5
0.20	12.94	5.34×10^5
0.225	14.14	5.36×10^5
0.25	15.52	5.39×10^5
0.275	17.28	5.43×10^5



(a)



(b)



(c)

Figure 9. Validation of current CFD results with experimental [30] and other CFD [31] results in non-dimensional parameters: (a) torque coefficient; (b) pressure drop coefficient; and (c) efficiency.

After validating the numerical procedure, we simulated our modified Denniss–Auld turbine. The simulations were carried out for a range of axial inlet flow velocities and rotational speeds corresponding to a flow coefficient range of $0.41 \leq \phi \leq 2.25$. Figures 10–12 show that the turbine fails to produce positive torque at the lower end of the flow coefficient range, resulting in negative efficiency, suggesting that it cannot generate power (i.e., torque) corresponding to the input rotational speed (rpm). Thus, the negative efficiency implies that the given rpm and inlet velocity do not physically allow the turbine to rotate. Due to its inability to produce positive torque at a low flow coefficient, $\phi < 0.44$, we simulated the turbine at high flow coefficients, which correspond essentially to flow conditions at a high angle of attack.

This study could have considered reporting the results in terms of different Tip Speed Ratios (TSRs), λ . TSR is defined as follows:

$$\lambda = \frac{U_{\text{tip}}}{v}, \quad (12)$$

By comparing Equations (6) and (12), we can deduce

$$\phi = \frac{1}{\lambda}, \quad (13)$$

Consequently, the results we demonstrated in terms of ϕ should be similar if expressed through λ .

At higher flow coefficients, the modified Denniss–Auld turbine has similarities to the impulse turbine. For instance, the torque coefficient increases with the flow coefficient; this increase is similar in behavior to the impulse turbine. The turbine blades do not reach stall at higher flow coefficients, contrary to Wells turbines, as shown in Figure 10. In addition, the slopes of the torque coefficient curves are milder at low flow coefficients but become steeper at higher flow coefficients.

Figure 11 shows that the modified Denniss–Auld turbine maintains a relatively constant efficiency for a wide range of flow coefficients, like the impulse turbine. Nonetheless, it has a lower efficiency compared to the impulse turbine with fixed guide vanes. Moreover, this figure shows an increase in turbine efficiency with increasing axial inlet velocity.

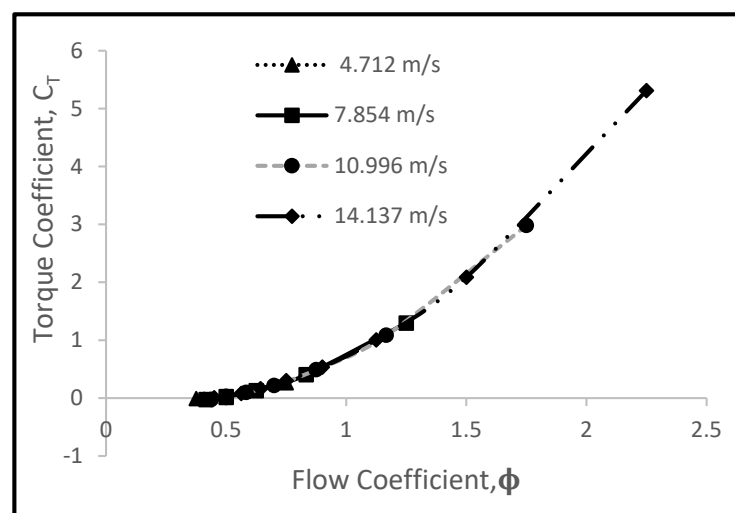


Figure 10. Torque coefficient vs. flow coefficient for various inlet axial flow velocities.

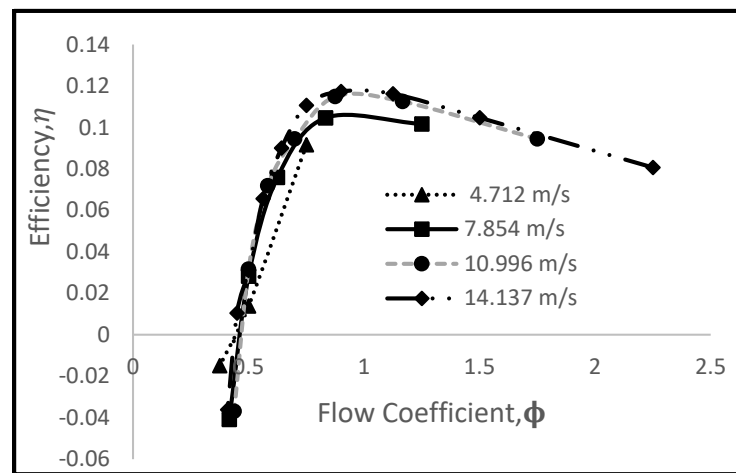


Figure 11. Efficiency vs. flow coefficient for various inlet axial flow velocities.

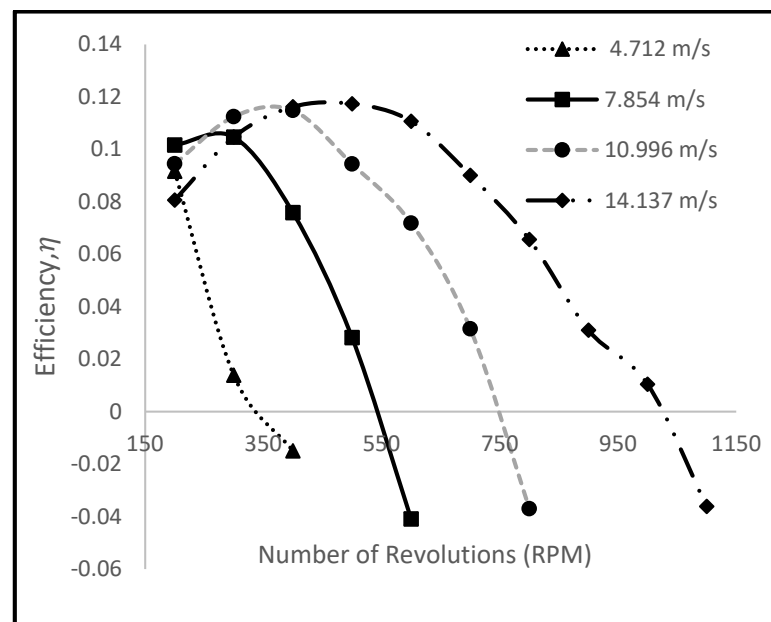


Figure 12. Efficiency vs. number of revolutions (RPM) for various inlet axial flow velocities.

While Wells turbines have a higher peak efficiency than other turbines, their operating range is generally lower. However, this modified Denniss–Auld turbine has a wider operating range depending on the incident flow velocity (Figure 12). In other words, the turbines can achieve high rotational speeds if the axial flow velocity is high enough. The results are comparable to the article by Finnigan et al. [21], according to Figure 13 below. It shows that the maximum lift coefficient increases with the Reynolds number; the high lift coefficient causes the rise in turbine torque.

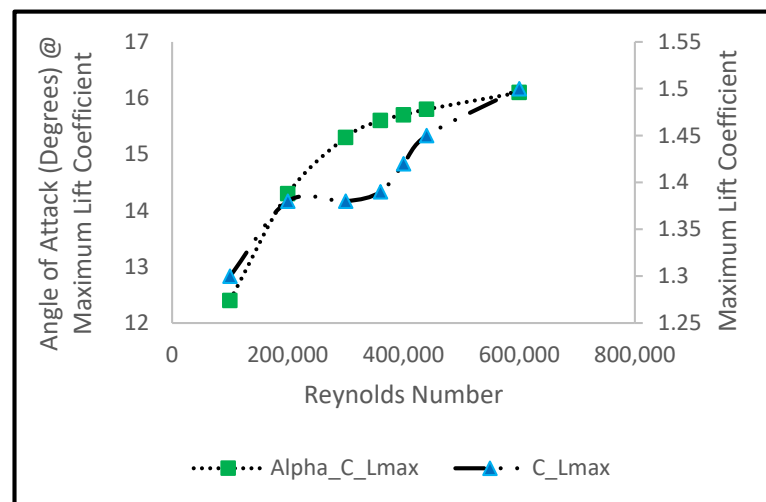


Figure 13. Angle of attack at maximum lift coefficient and maximum lift coefficient vs. Reynolds number.

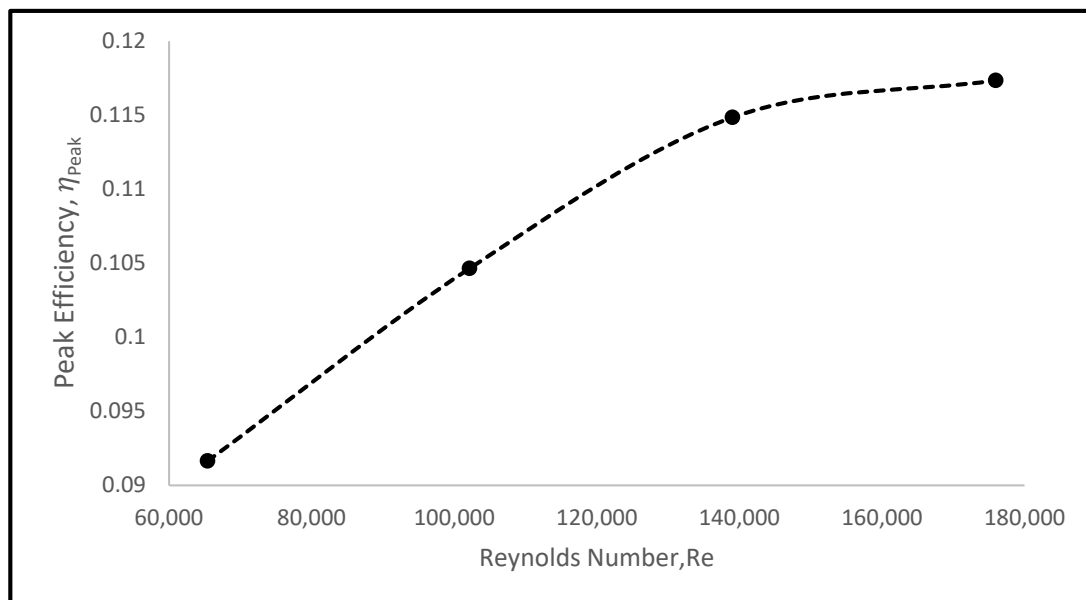


Figure 14. Peak efficiency vs. Reynolds number.

Figure 14 illustrates how the Reynolds number affects the turbine's peak efficiency. Variations in the Reynolds number ranged from 6.5×10^4 to 3.1×10^5 . We observed a substantially linear increase in peak efficiency with various flow coefficients.

In Figure 15, the torque coefficient is plotted as a function of turbine rotational speed at a given inlet axial flow velocity. It is evident from this figure that increasing the turbine's rotational speed for a certain axial velocity reduces the torque coefficient drastically. A steep slope is observed at an incident speed of 4.7 m/s, but it becomes more gradual with increasing rotational speeds and axial velocities. In addition, the curves shift rightward, accommodating the high rotational speed associated with an increase in inlet velocity, which suggests the turbine, in turn, can operate within a broader range. Thus, the turbine can experience higher rotational speeds when subjected to greater axial velocity at the inlet.

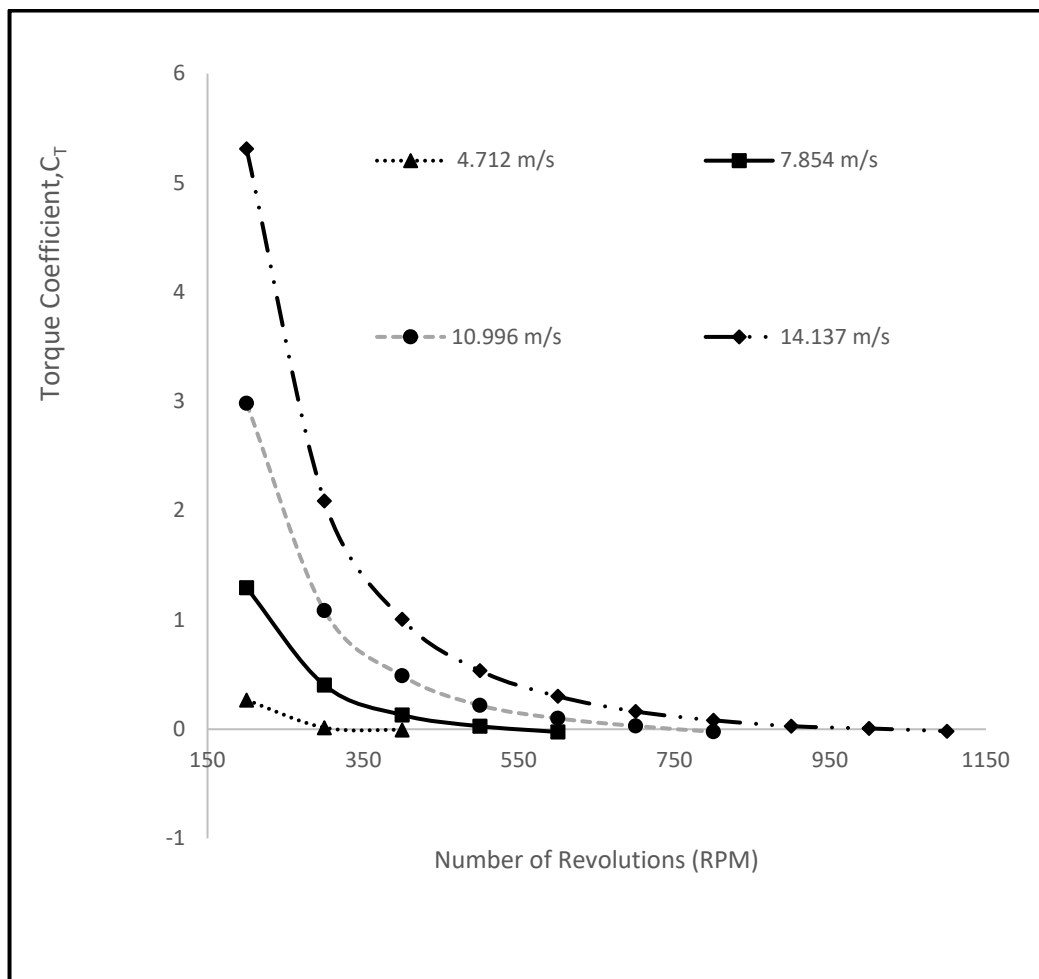


Figure 15. Torque coefficient vs. number of revolutions (RPM) for various inlet axial flow velocities.

Figure 16 shows a plot of the input coefficient versus the flow coefficient. The input coefficients are lower when the flow coefficients are smaller; therefore, the turbine harnesses less power at the lower range of flow coefficients. At a lower coefficient range ($\phi < 0.9$), the slope of the input coefficient is steeper, so the rate of change in power production is sharper than at a flow coefficient range of $\phi > 0.9$. Inlets of varying axial velocities show similar trends.

According to Figure 17, the input coefficient drops exponentially with increasing turbine rotational speed. Also, a decline in the slope of the curves is associated with an increase in the incident axial velocity.

The torque coefficient steadily increases with the Reynolds number at peak efficiency, as shown in Figure 18. It is evident that the input coefficients increase until a Reynolds number of 1×10^5 and then remain constant. This results in a gradual increase in peak efficiency.

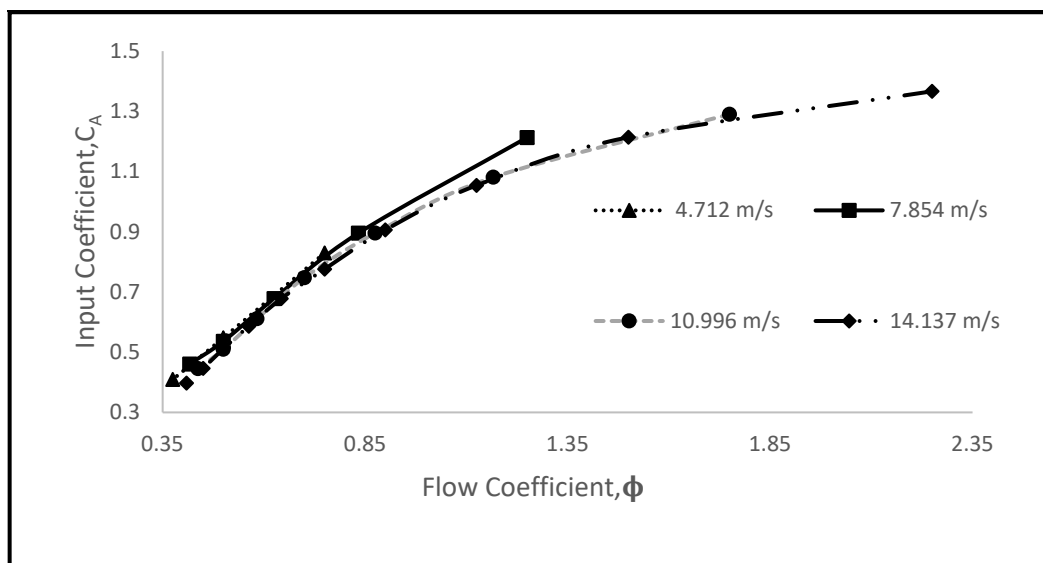


Figure 16. Input coefficient vs. flow coefficient for various inlet axial flow velocities.

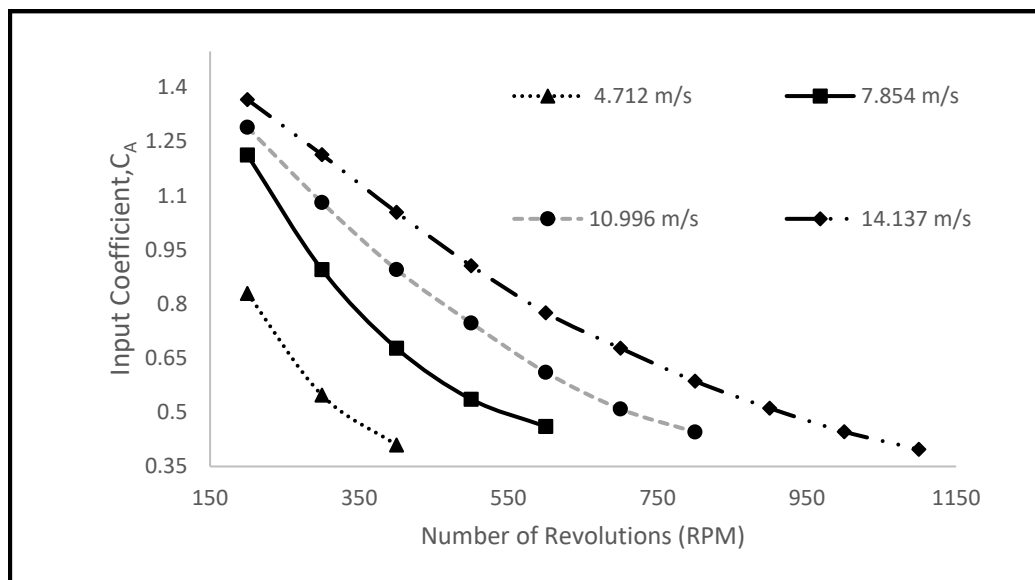


Figure 17. Input coefficient vs. number of revolutions (RPM) for various inlet axial flow velocities.

Figure 19 provides a comparison of the modified Denniss–Auld turbine with the standard Denniss–Auld turbine in terms of non-dimensional coefficients for a wide range of flow conditions. For $\gamma = 20^\circ$ and 40° , the C_T and C_A increase almost linearly with ϕ , and the slope decreases with increasing γ .

On the contrary, the C_T rises exponentially with ϕ , which means that the modified Denniss–Auld turbine produces much higher torque than the standard design at higher ϕ . C_A demonstrates logarithmic growth; the sharper increase occurs in the lower range of ϕ , and the gains become flattened out for higher ϕ .

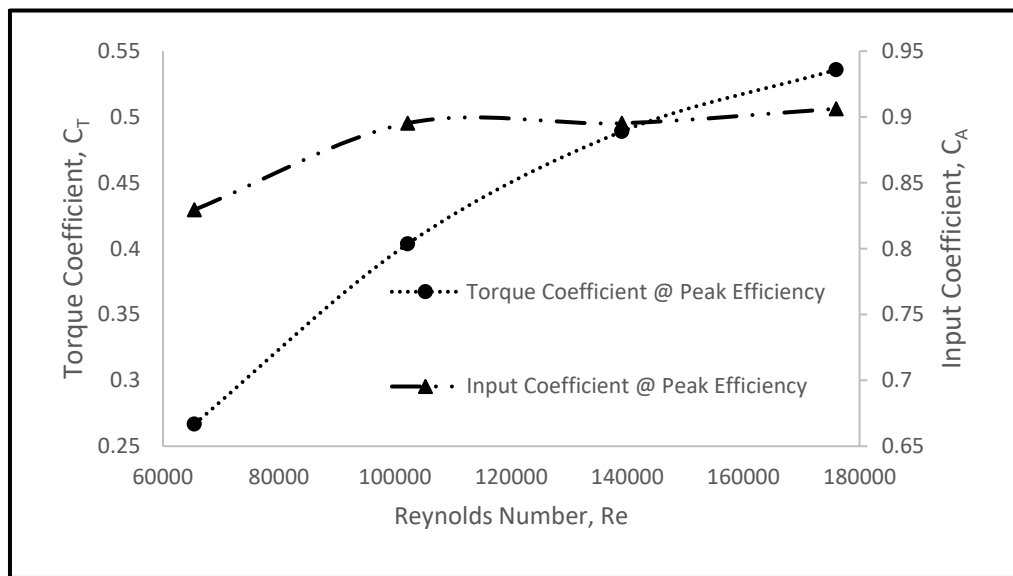


Figure 18. Torque coefficient and input coefficient vs. Reynolds number at peak efficiency.

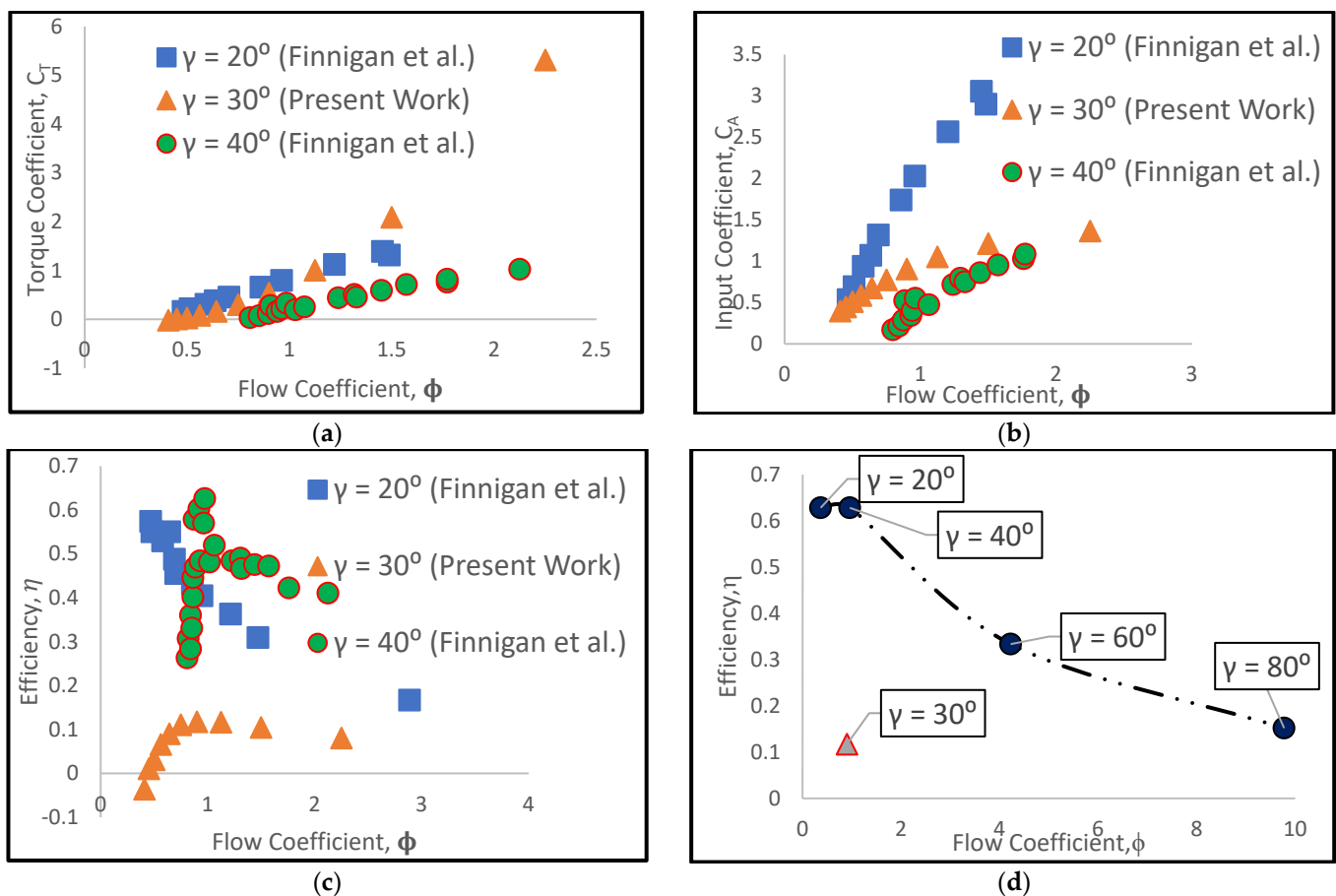


Figure 19. Comparison with standard Dennis–Auld turbine [21]: (a) torque coefficient; (b) input coefficient; (c) efficiency; and (d) peak efficiency at various blade pitching angles.

The trend of the efficiency plots is similar for all the γ , except for $\gamma = 20^\circ$. The reasoning behind this could be attributed to the shortcomings associated with the existing test rig to resolve the peak efficiency. The modified Dennis–Auld turbine produces the lowest aerodynamic efficiency over the entire range of ϕ . The peak efficiency of the standard

Denniss–Auld turbine is 63% and occurs for $\gamma = 40^\circ$ at $\phi = 1$. However, the modified Denniss–Auld turbine has a peak efficiency of 11.73% that occurs at $\phi = 0.9$.

Figures 20 and 21 show the coefficient of pressure for a varying flow coefficient at an inlet velocity of 7.854 m/s. The results show that with the increasing flow coefficient, the peak pressure point moves downstream on the pressure side of the turbine. The peak efficiency of the turbine is at a flow coefficient of 0.83. This is a result of the flow attachment point no longer being on the leading edge of the turbine. This is evident by the inflection point and decreases in pressure at $x/c = 0.042$ at a flow coefficient of 1.25 and 2.5. Increasing the flow coefficient to 2.5 increases the inflection point and causes a further decrease in pressure. This results in the decrease in turbine efficiency discussed earlier. Furthermore, the rounded shape of the trailing edge of the turbine causes the pressure side to experience a severe decrease in pressure leading to lower pressure compared to the suction side.

Figure 22 shows a plot of C_p at a flow coefficient of 0.42 at 40% and 80% span of the turbine. The plot at 40% span shows a slightly higher pressure difference indicating better aerodynamic performance compared to the 80% span location. However, the trends of both plots are similar.

Figures 23 and 24 show the pressure with increasing flow coefficient at 40% and 80% span of the turbine. The contours at 40% span illustrate the change in pressure distribution on the turbine at various flow coefficients (Figure 23). These results show that at a flow coefficient below 1.25 the peak pressure occurs on the leading edge. Above $\phi = 1.25$, the peak surface pressure is downstream of the leading edge (pressure side of the airfoil), leading to a decrease in turbine efficiency. Similarly, the same behavior is displayed at the 80% span position (Figure 24).

Due to the surface curvature of the modified Denniss–Auld turbine, the torque coefficient increases exponentially throughout the flow coefficient range investigated in this study. However, a penalty occurs in the corresponding pressure drops.

A comparison of the rotational speed and torque versus the flow coefficient and various inlet velocities is shown in Figure 25. The results show that the torque increases sharply at a flow coefficient of between 0.5 and 1.0. From there, the torque increases but less substantially. The intersection between the rotational speed and the torque also nearly represents the peak power output, as shown in Figure 26. As expected, the power output increases the inlet velocity up to a flow coefficient of ~ 1.0 ; then, the power decreases significantly with the flow coefficient. Figures 25 and 26 represent a performance map that can be used to determine the power output for a range of inlet velocities, rotational speed, and flow coefficients that can be used for future optimization of the studied turbine blades.

Figure 27 shows the wall shear distribution at the 40% and 80% span locations. Both locations show similar trends where the flow attaches at the leading edge ($x/c = 0$). The attachment location moves downstream on the pressure side of the turbine with an increasing flow coefficient. In the cases shown, there is a sharp increase in shear stress moving towards the suction side downstream of the incipient leading-edge attach point. The increase in shear stress is reduced with increasing flow coefficient at $x/c = 0.1$. The boundary layer separates from the suction side until it reattaches close to the trailing edge. On the pressure side, the flow displays different behavior. For flow coefficients $\phi = 0.125$ and $\phi = 0.83$, the boundary layer is attached along the chord of the turbine. However, at $\phi = 0.063$, the flow shows an increasing downstream negative shear stress that is typically representative of an adverse pressure gradient but not so much as to cause separation until very close to the trailing edge at $x/c = 0.98$.

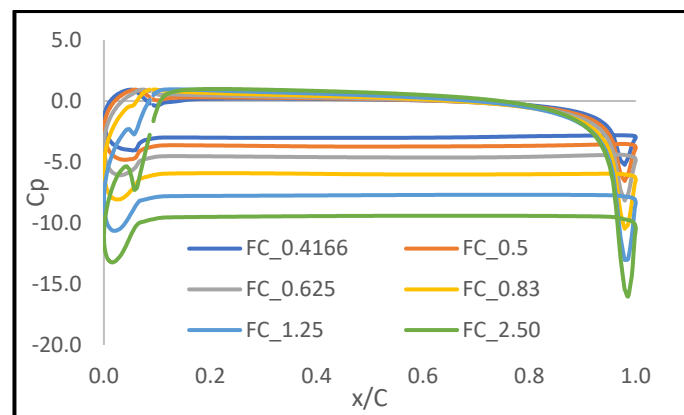


Figure 20. C_p comparison at a plane through 40% from hub at different flow coefficients.

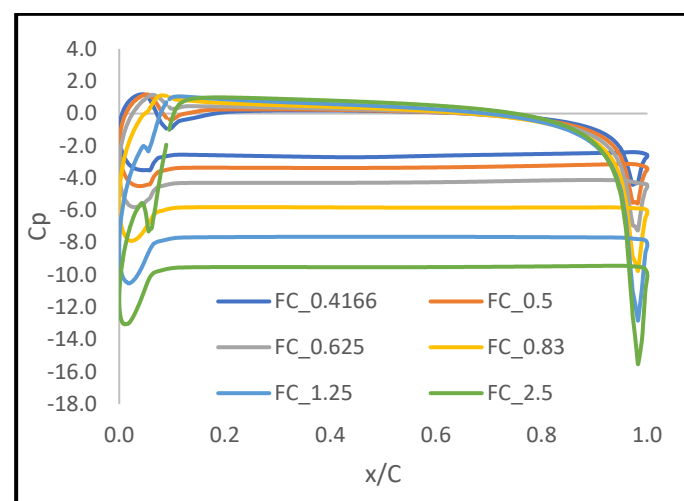


Figure 21. C_p comparison at a plane through 80% from hub at different flow coefficients.

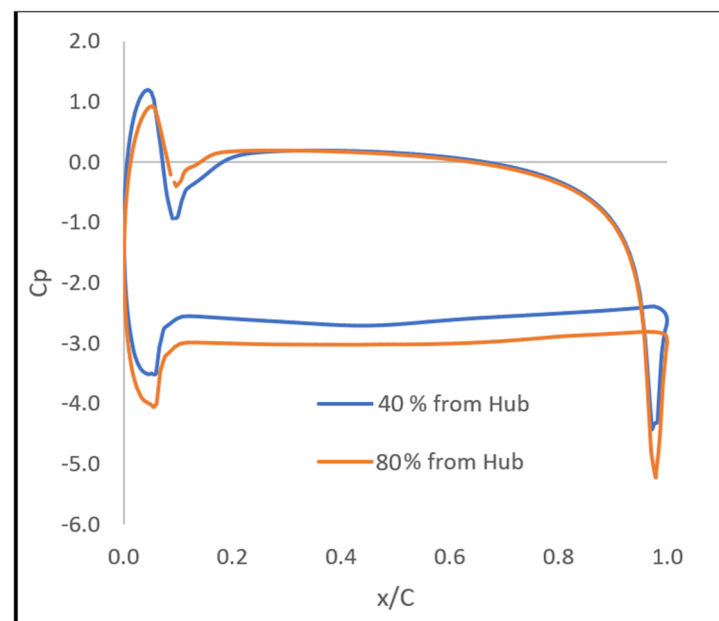


Figure 22. C_p comparison at a plane through 80% and 40% from hub at flow coefficient of 0.42.

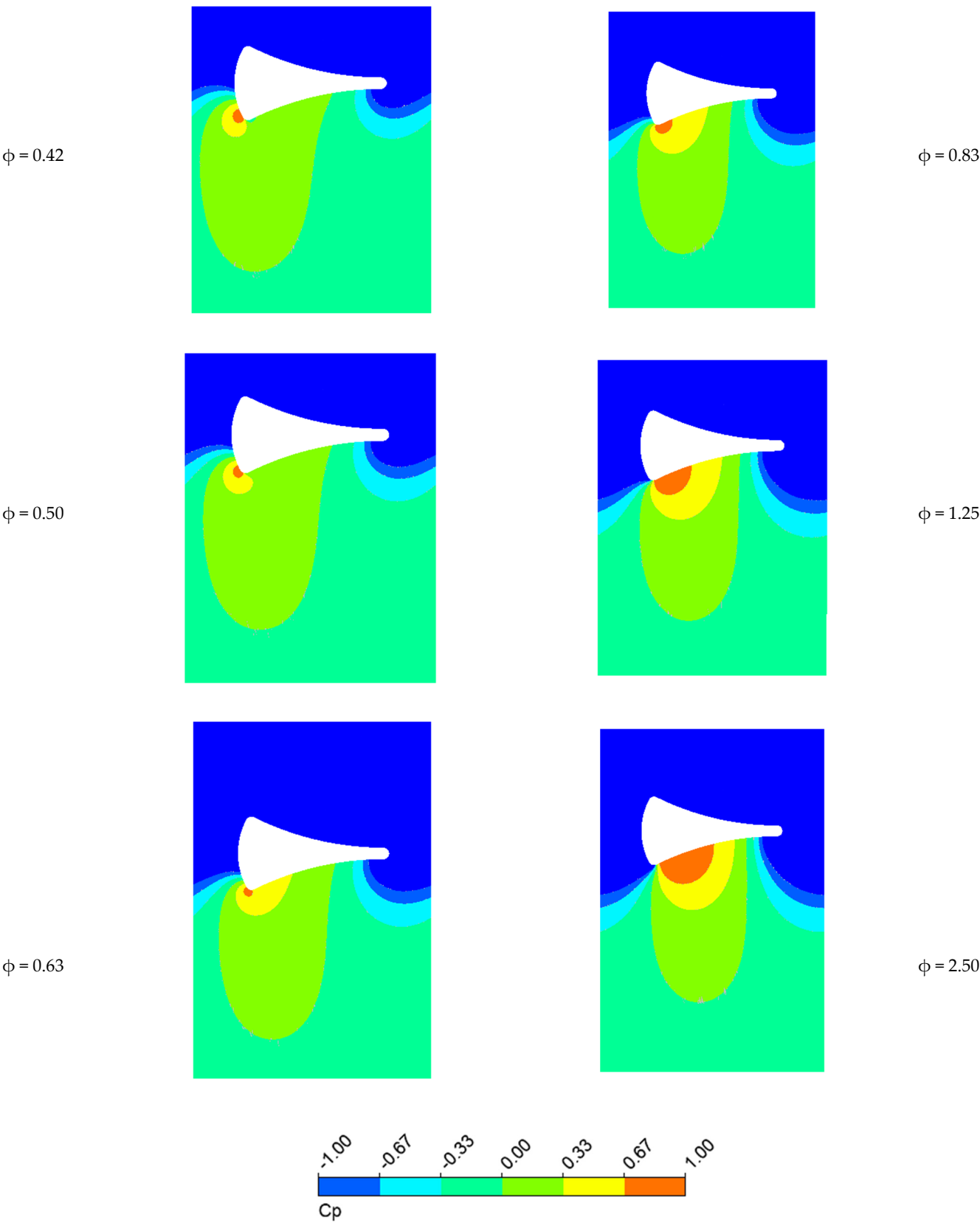


Figure 23. C_p at a plane through 40% from hub at various flow coefficients.

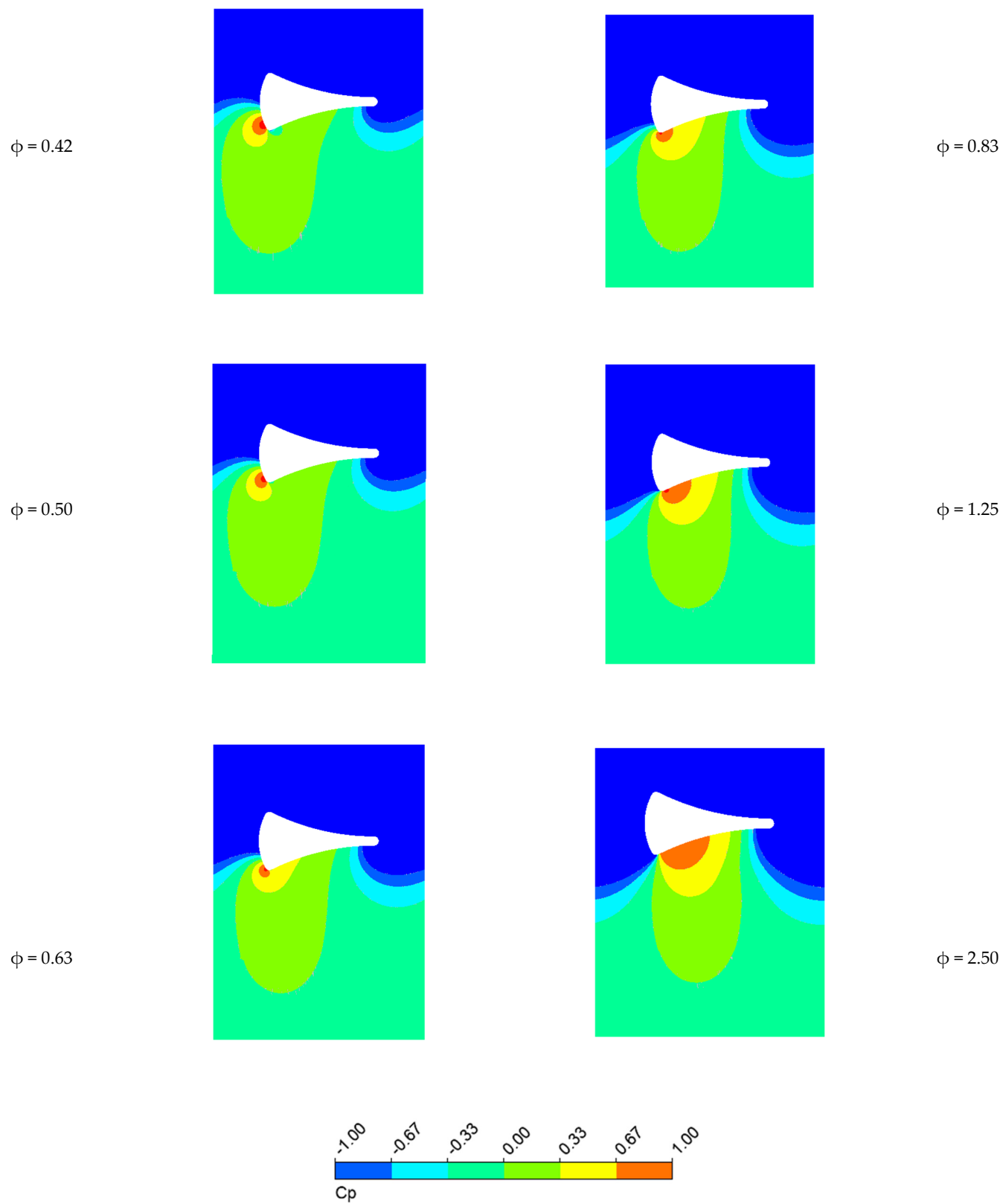


Figure 24. C_p at a plane through 80% from hub at various flow coefficients.

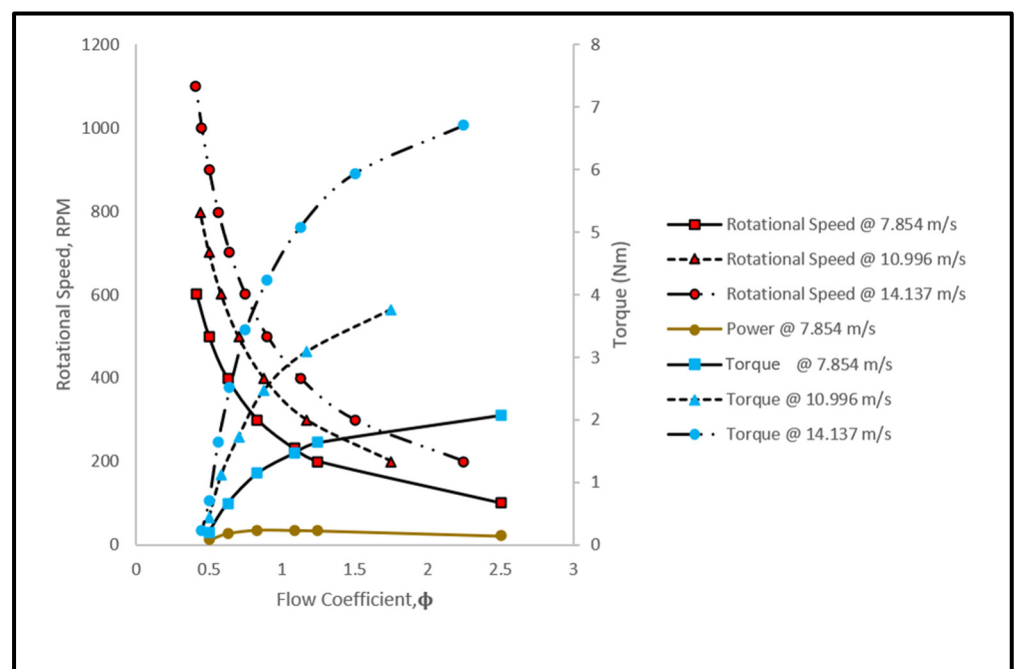


Figure 25. Rotational speed and torque at various flow coefficients.

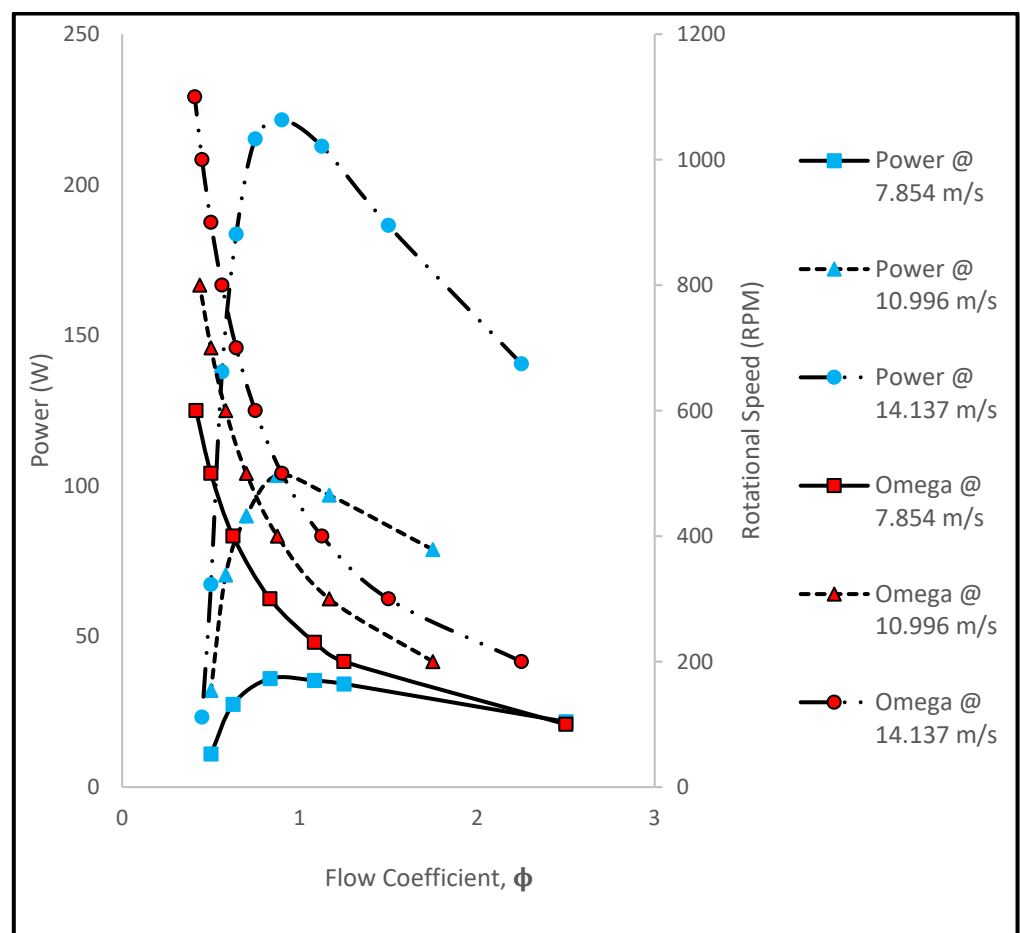
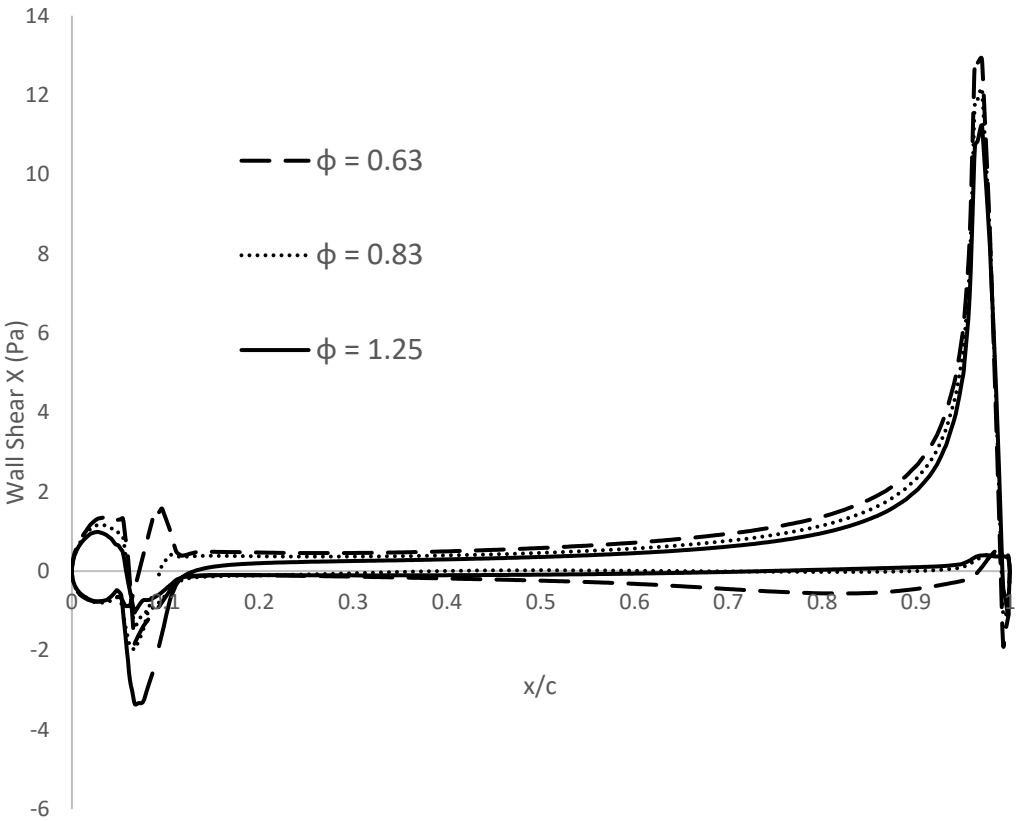


Figure 26. Power and rotational speed at various flow coefficients.

Span Location

40%



80%

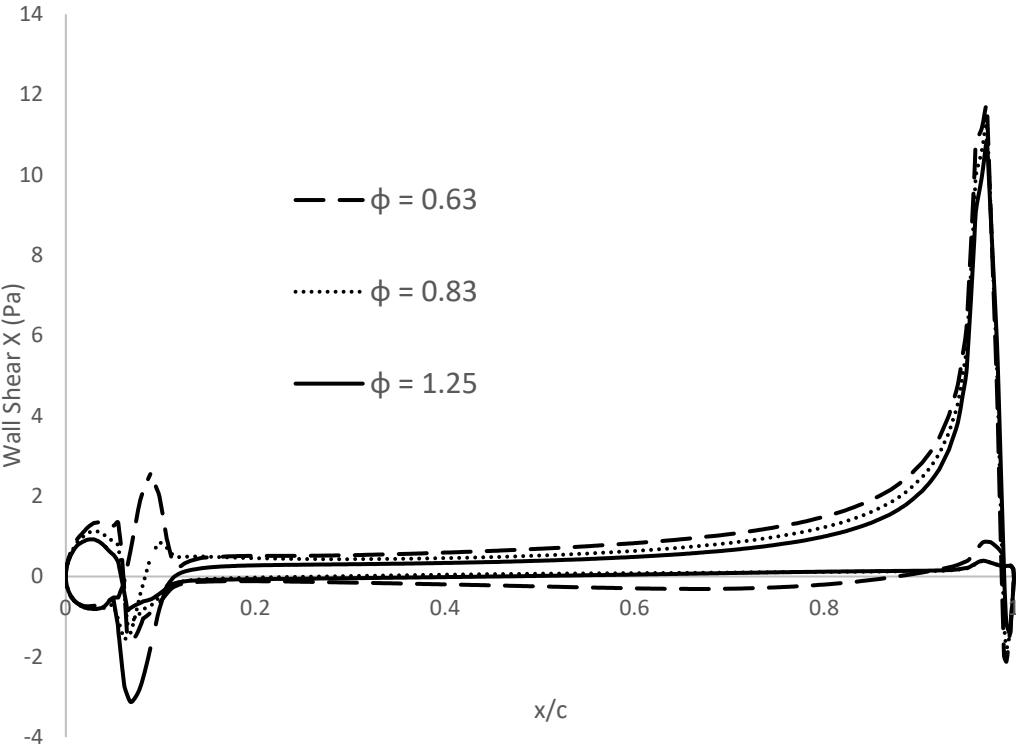


Figure 27. Wall shear for various flow coefficients at 40% and 80% span locations.

Figures 28 and 29 show that the flow field streamlines at the 40% and 80% span locations, respectively. The flow field is highly three-dimensional, mostly separated on the suction side. At the 40% span location, beginning at a flow coefficient of 0.42, two large vortices are seen on the suction side of the turbine. As the flow coefficient increases, the vortices convect away from the blade. We observe a similar highly separated flow at the 80% span location. The comparison of the 80% and 40% spans reveals that the vortices have already moved away from the blade at the lowest flow coefficient ($\phi = 0.42$) and advect away from the blade with an increasing flow coefficient. Additionally, we predicted a decrease in the velocity magnitude upstream of the leading edge. This decrease leads to an increase in the static pressure, thus increasing the pressure drop.

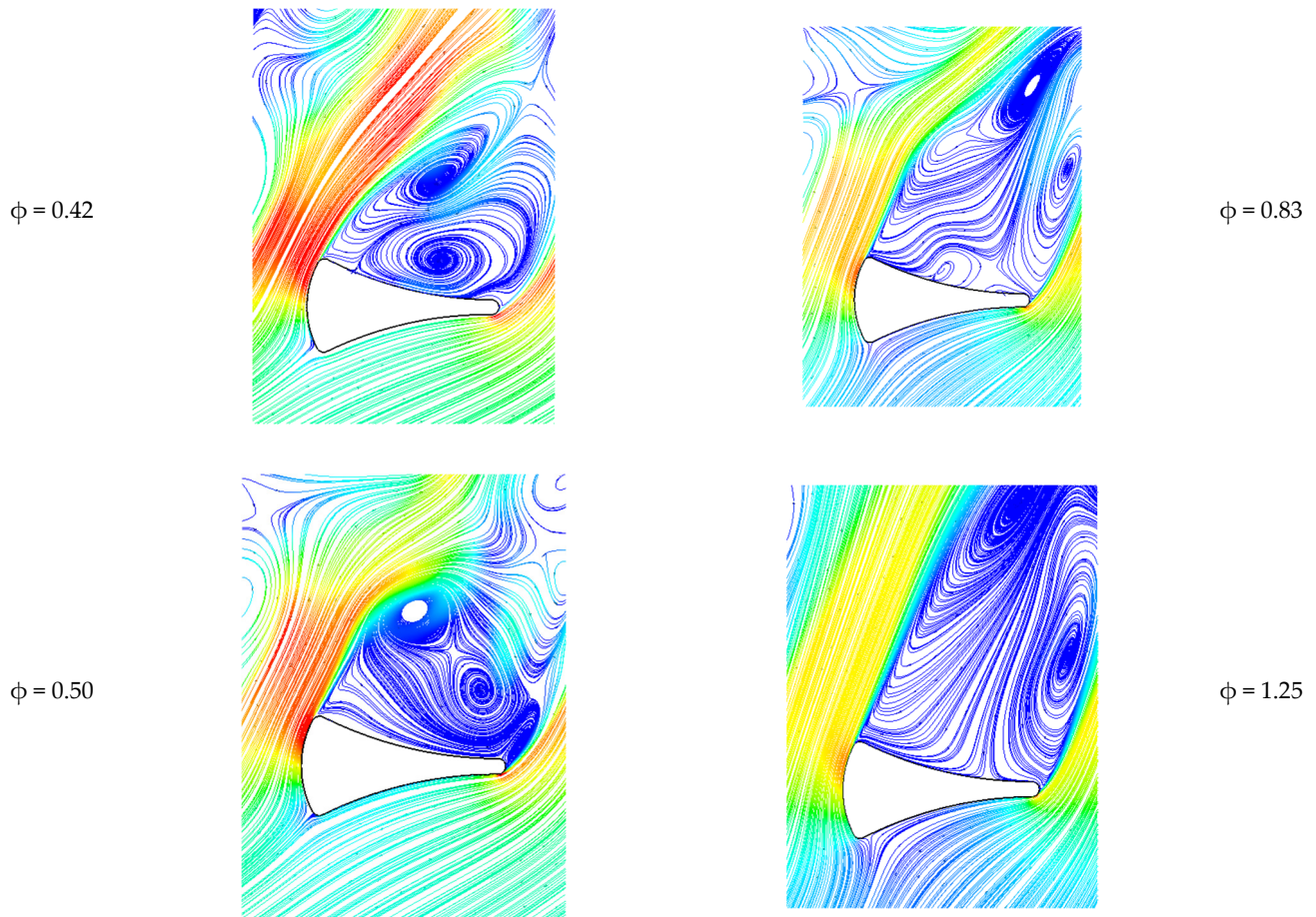


Figure 28. Cont.

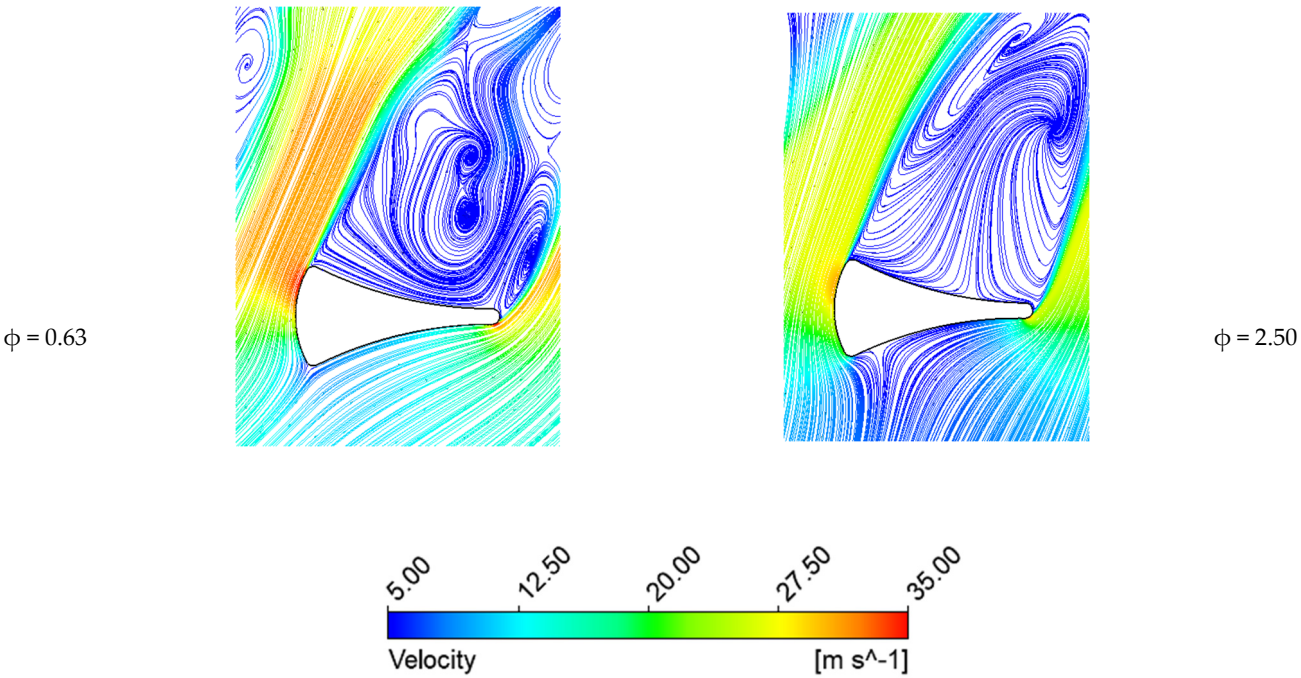


Figure 28. Streamlines at a plane through 40% from hub at various flow coefficients.

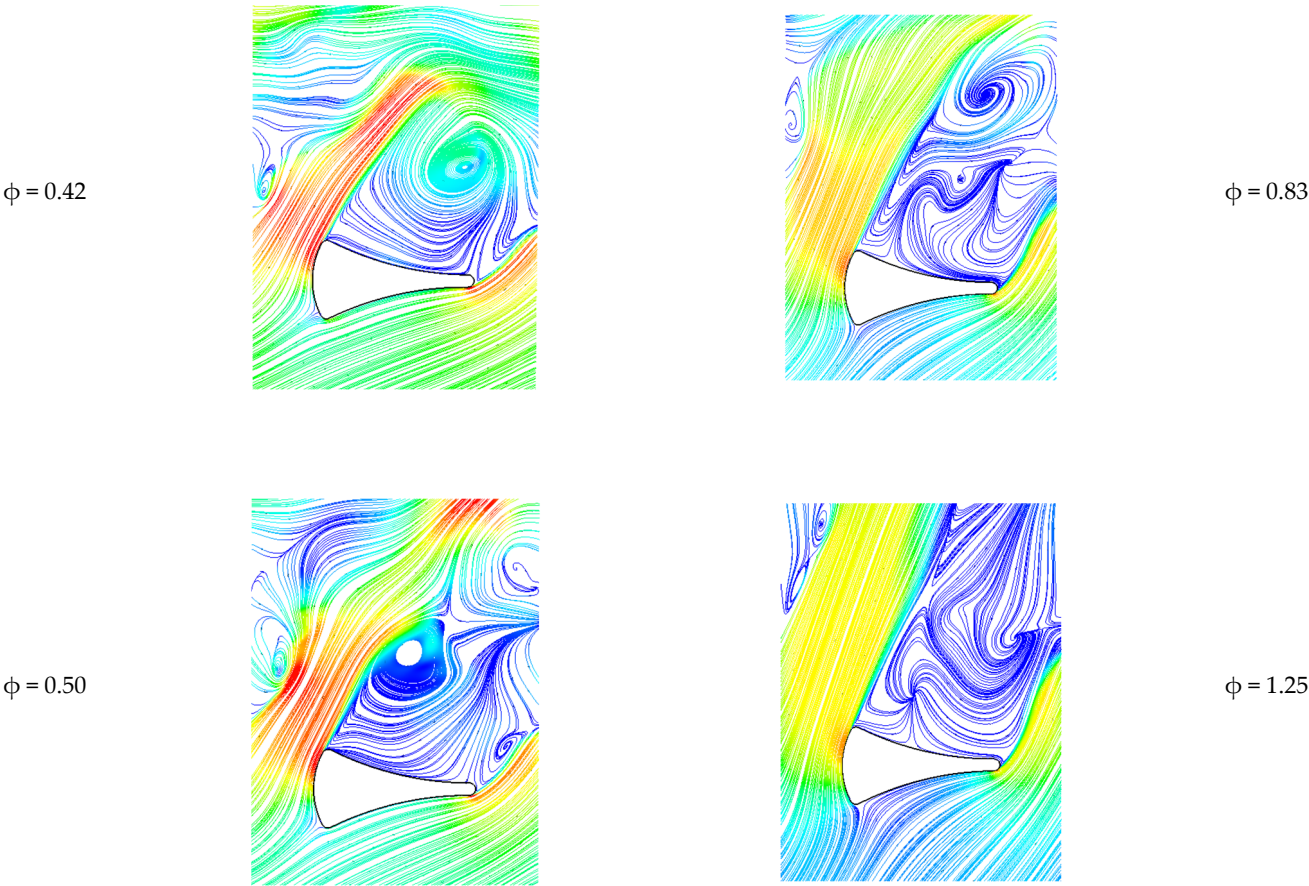


Figure 29. Cont.

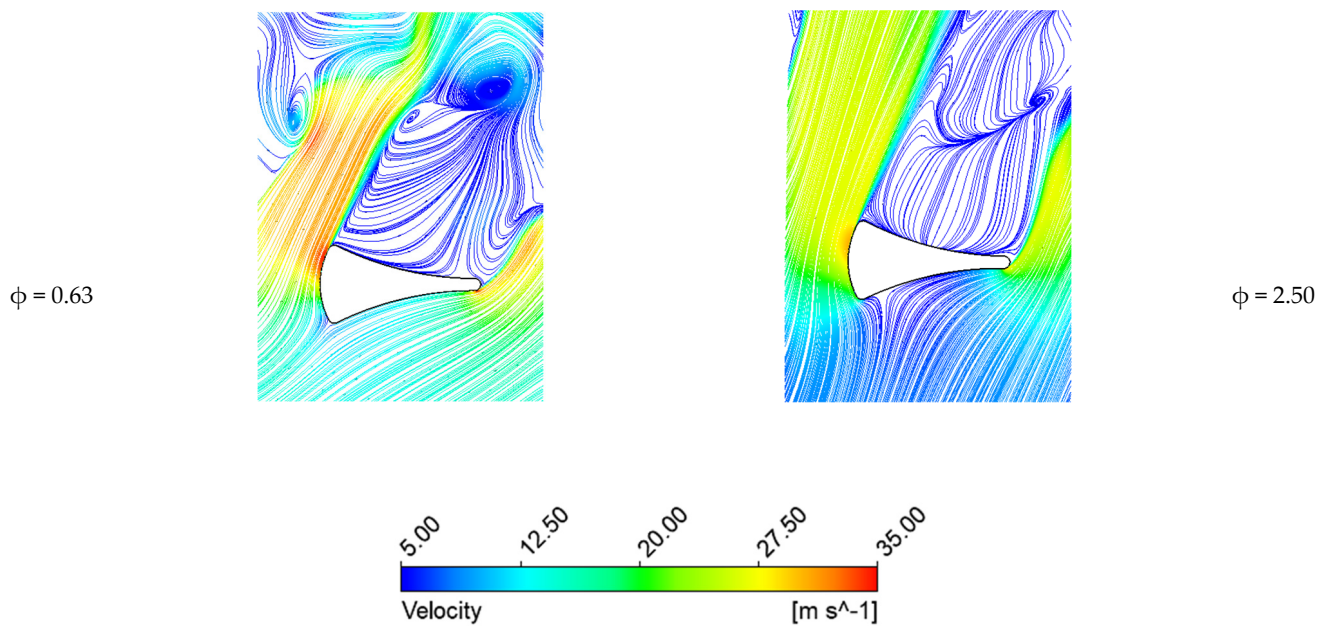


Figure 29. Streamlines at a plane through 80% from hub at various flow coefficients.

4. Conclusions

This study aimed to investigate a novel bi-directional axial flow turbine for the OWC wave energy converter through three-dimensional CFD simulations. Modifying the Denniss–Auld turbine blades, we examined how blade geometry impacts aerodynamic performance. The results demonstrated that the modified Denniss–Auld turbine behaves similarly to impulse turbines at higher flow coefficients. Unlike Wells turbines, the modified Denniss–Auld turbine does not experience blade stall at higher flow coefficients. It maintains fairly constant efficiency across a wide range of flow coefficients, akin to the impulse turbine.

Although increasing axial inlet velocity can enhance efficiency, the modified Denniss–Auld turbine remains significantly less efficient than an impulse turbine. The proposed turbine has the advantage of requiring less mechanical complexity, as it does not need a variable pitching mechanism for the turbine blades. However, its lower efficiency suggests that further optimization is necessary.

Future work should focus on optimizing the turbine blades' geometry, potentially incorporating active flow control to improve aerodynamic performance.

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