

Original Article

Power Coefficient Variation of an H-Darrieus Turbine with S1046 Airfoils at Different Tip Speed Ratios

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Received: 25 June 2026

Revised: 18 September 2026

Accepted: 22 September 2026

Published: 26 October 2026

Abstract - Vertical Axis Wind Turbine (VAWT) efficiency is influenced by the airfoils' geometry and its rotational speed. Previous research has been conducted to increase the Power Coefficient (C_p) using mainly NACA airfoils and reporting cycle-averaged performance parameters. However, the effect of the Tip Speed Ratio (TSR) on the instantaneous evolution of C_p and the Torque Coefficient (C_t) per cycle, as well as its relationship with the flow field in contact with the blades, has not been fully characterized. This paper presents a numerical study of the effect of TSR on the C_p and C_t of an H-Darrieus turbine using S1046 airfoils to identify the conditions associated with maximum C_p and explain the aerodynamic mechanisms governing turbine performance throughout a cycle. This evaluation was conducted using two-dimensional transient Computational Fluid Dynamics (CFD) in ANSYS Fluent. The numerical model's reliability was evaluated by comparing the predicted C_p -TSR values with those reported by Hashem and Mohamed, yielding relative differences ranging from 1.22% to 5.96% across the TSR ranges evaluated. The aerodynamic analysis included: turbulent Kinetic Energy (k) contours, Pressure Coefficient (P_c), and velocity trajectory lines at key azimuthal positions of the rotation. It was found that C_p increases with TSR, reaching a maximum of 0.342 at a TSR of 2.25; as TSR continues to increase, C_p decreases. Flow field analysis showed that at a TSR of 2.25, there was better pressure distribution around the blades and a more uniform flow passage through the turbine, whereas at higher TSR values, the interaction between the blades and the flow was less efficient. The results show the effect of TSR on the performance of the rotating turbine throughout the aerodynamic analysis and serve as a reference for selecting the operating regime of H-Darrieus turbines.

Keywords - C_p , C_t , H-Darrieus, S1046, TSR.

1. Introduction

The utilization of wind power using wind turbines is one of the most important alternatives for the transition to sustainable energy systems [1]. Among the diverse configurations of wind turbines, Vertical Axis Wind Turbines (VAWTs) are characterized by their omnidirectional operation capacity and their minor visual and acoustic impact; this makes them key for their application in urban environments and systems of distributed generation [2-7]. Among the different configurations of VAWTs, the H-Darrieus turbine reaches the highest Power Coefficient (C_p) among VAWTs, along with its simple structure [5-8].

Despite its many advantages, the aerodynamics of the H-Darrieus turbine are more complex than those of Horizontal-Axis Wind Turbines (HAWT). This is because turbine rotation causes each blade to be subject to continuous variations in its angle of attack, causing highly transient effects such as flow

separation, wake-blade interaction, and dynamic stall [4, 9-12]. These phenomena generate periodic fluctuations in aerodynamic loads that affect torque generation and limit turbine performance [13, 14].

The C_p of a Darrieus turbine is influenced by geometric and operational parameters, such as the aspect ratio, the number of blades, the airfoil geometry, and TSR [15-20]. Airfoil geometry is an important factor affecting turbine performance because of its direct impact on the lift-to-drag ratio. Consequently, much of the research documented in the literature has evaluated various airfoil geometries, mainly employing symmetrical airfoils from the NACA family (series 0012, 0015, 0018, and 0021), to maximize aerodynamic efficiency and improve self-starting capability [4, 10, 21-23]. On the other hand, some studies on H-Darrieus turbines have used airfoils from the Selig airfoil series, demonstrating favorable C_p characteristics under operating conditions



typical of low-power VAWTs [24, 25]. In particular, the S1046 airfoil has demonstrated promising aerodynamic performance in low-speed applications, thanks to its ability to delay flow separation and maintain high lift-to-drag ratios [24, 26].

Although previous studies have evaluated the effect of TSR on the performance of H-Darrieus turbines, their analysis typically focuses on average performance parameters per cycle when modifying the turbines' profiles [27, 28] or geometric characteristics [29, 30]. This approach minimizes the analysis of instantaneous variations in torque and power per cycle, and it is important to conduct a detailed analysis of the evolution of C_p and C_t throughout the turbines' rotation to explain the aerodynamic mechanisms affecting turbine performance as the TSR varies.

Computational Fluid Dynamics (CFD) is one of the most widely used tools for analyzing VAWTs, as it allows for a detailed study of the temporal evolution of the flow in contact with the turbine, the torque-generation mechanisms, and the vortices associated with dynamic losses [31-33]. Various studies confirm that two-dimensional transient process simulations based on URANS and the SST $k - \omega$ turbulence model reach an adequate balance between accuracy and computational cost for predicting the aerodynamic performance of H-Darrieus turbines [34-36]. However, the accuracy of these simulations depends on factors such as the size of the computational domain, mesh quality, time resolution, and the rotation strategy used to define the turbine's rotation.

The purpose of this research was to investigate the effect of the TSR on C_p and C_t per cycle of an H-Darrieus wind turbine with three S1046 airfoils, using 2-D time-dependent CFD simulations in ANSYS Fluent at a wind speed of 8 m/s and TSR values from 0.6 to 2.5. The evolution of C_p and C_t per cycle is analyzed along with k contours, P_c contours, and velocity pathlines at representative azimuthal positions to identify the aerodynamic mechanisms associated with their variations.

2. Methodology

2.1. Geometric Model of VAWT

The geometry used in this investigation is based on a three-bladed H-Darrieus turbine with a diameter and height of 0.29 m, using the S1046 airfoil with a chord length of 0.05 m. These dimensions are based on previous studies reported in the literature by Sengupta et al. [26, 37].

The pitch angle was kept fixed at 0 degrees, and rotation was defined as counterclockwise. Only the central shaft of the turbine, with a diameter of 0.012 m and located at the origin of the coordinate system, was included. Figure 1 shows a detailed view of the turbine model study.

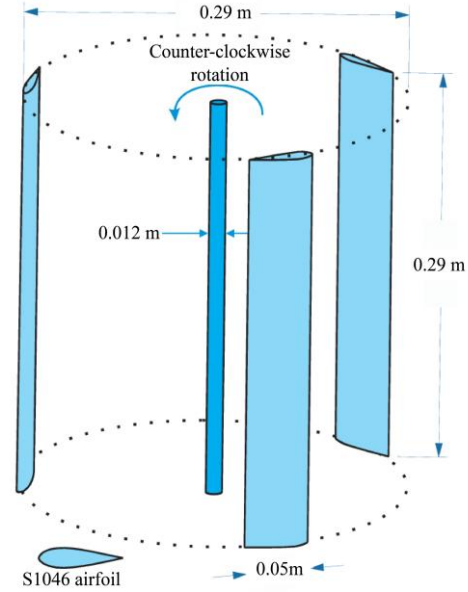


Fig. 1 Geometric model of the H-Darrieus turbine with three blades and an S1046 airfoil

2.2. Boundary Conditions and Computational Domain

The analysis was carried out in 2D (Figure 2), considering two computational domains: one static and one rotating, both with dimensions based on the turbine Diameter (D). The rotating domain was defined as a diameter of $2D$ [10], while the outer domain had dimensions of $15D \times 10D$. The turbine was centered along the outer domain, with $5D$ upstream and $10D$ downstream, to allow for proper wake development and minimize impact with the side walls [10, 38, 39]. The interaction between the rotating and stationary domains was solved using a sliding mesh interface. The working fluid is air under standard conditions. The boundaries were as follows (Figure 3): the inlet was considered as a velocity inlet (8 m/s in the x -direction); the outlet was configured as an outflow; the blade surfaces and the shaft were defined as walls with a no-slip condition; and the lateral sides of the domain were defined as velocity inlet (as in [27, 39]) with a component tangential to this side (x -direction in this case), avoiding in such case the use of a wall condition.

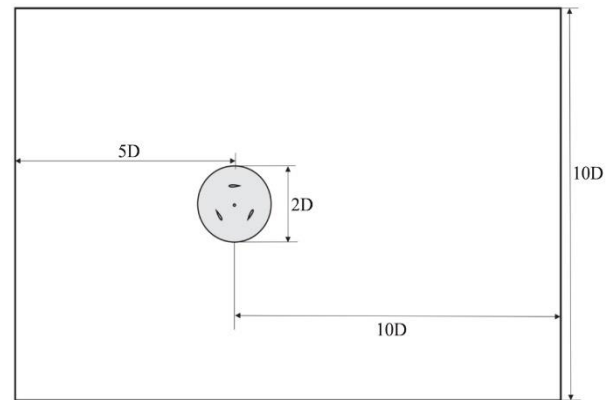


Fig. 2 Dimensions of the computational domain

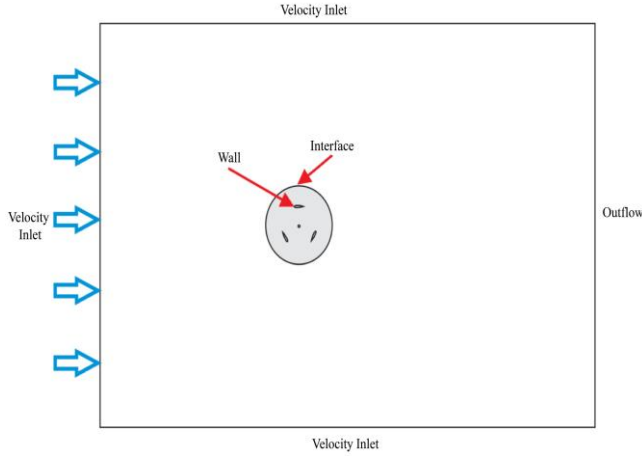


Fig. 3 Boundary conditions

3. Numerical Model

3.1. Governing Equations and Turbulence Model

The flow was considered incompressible, transient, and turbulent. The CFD model solves the Reynolds-Averaged Navier-Stokes in transient configuration (URANS) equations. Mass conservation for incompressible flow is expressed by Equation (1):

$$\frac{\partial u_i}{\partial t} + \frac{\partial u_i}{\partial x_i} = 0 \quad (1)$$

Where u_i represents the components of the average velocity in the x_i direction.

Momentum conservation was solved using Equation (2).

$$\rho \left(\frac{\partial u_i}{\partial t} + u_j \frac{\partial u_i}{\partial x_j} \right) = -\frac{\partial \bar{p}}{\partial x_i} + \mu \frac{\partial^2 u_i}{\partial x_j \partial x_j} - \frac{\partial}{\partial x_j} (\rho \overline{u_i u_j}) \quad (2)$$

Where the density is ρ , the static pressure is $\bar{p}$, the dynamic viscosity is μ , and $-\rho \overline{u_i u_j}$ represent the Reynolds stresses associated with turbulent fluctuations.

The effects of turbulence were solved using the SST $k - \omega$ turbulence model. This model solves the following transport equations: one for k (Turbulent Kinetic Energy) and another for ω (the specific dissipation frequency). The equation for k is expressed by Equation (3) [40].

$$\frac{\partial(\rho k)}{\partial t} + \frac{\partial(\rho u_i k)}{\partial x_i} = \frac{\partial}{\partial x_j} \left[(\mu + \sigma_k \mu_t) \frac{\partial k}{\partial x_j} \right] + P_k - \beta^* \rho k \omega \quad (3)$$

Where P_k represents the k production, β^* and σ_k are model constants, and μ_t corresponds to the turbulent viscosity.

The specific dissipation frequency ω is given by Equation (4).

$$\frac{\partial(\rho \omega)}{\partial t} + \frac{\partial(\rho u_i \omega)}{\partial x_i} = \frac{\partial}{\partial x_j} \left[(\mu + \sigma_\omega \mu_t) \frac{\partial \omega}{\partial x_j} \right] + \frac{\rho \gamma}{\mu_t} S^2 - \beta \rho \omega^2 + 2\rho(1 - F_1) \frac{1}{\omega} \frac{1}{\sigma_{\omega,2}} \frac{\partial k}{\partial x_j} \frac{\partial \omega}{\partial x_j} \quad (4)$$

In this expression, γ and β are model coefficients, σ_ω and $\sigma_{\omega,2}$ are turbulent diffusion coefficients, S is the mean strain-rate tensor, and F_1 is the SST blending function. The last term represents the cross-diffusion between k and ω . The μ_t is calculated by Equation (5).

$$\mu_t = \frac{\rho a_1 k}{\max(a_1 \omega, S F_2)} \quad (5)$$

Where the magnitude of the mean strain tensor is S , a_1 is constant, and the mixing function is F_2 . The SST $k - \omega$ formulation connects the nearby-wall achievement of the $k - \omega$ model with a transformed $k - \varepsilon$ formulation in the exterior region of the boundary layer. The blending functions provide a gradual transition between both formulations, while the modified turbulent-viscosity formulation improves the prediction of flows subjected to adverse pressure gradients and separation.

3.2. Model Assumptions

The numerical computations were based on the following assumptions:

- A 2D profile was located at the middle of the blade height, and the effect of vorticity at the blade ends was neglected.
- The flow was considered incompressible because the Mach number is less than 0.3.
- Wind velocity was considered in a steady state and normal to the inlet domain.
- CFD computations were transient, but because angular speed and wind speed were constant for every TSR, the case of a wind turbine startup was not studied in this research.

For each TSR, the turbine performance parameters, such as C_p and C_t , were calculated at time steps of 1° of turbine rotation using Equation (6).

$$\text{Time step} = \frac{1}{\omega \frac{180}{\pi}} \quad (6)$$

Where ω is the angular velocity of the turbine.

3.3. Operating and Performance Parameters

The aerodynamic variables used to evaluate the turbine's performance were calculated based on the average torque obtained during the last simulated cycle. The TSR was determined as the relation of the tangential speed at the blade tip to the incident flow velocity. It is calculated using Equation (7).

$$TSR = \frac{\omega R}{V} \quad (7)$$

Where the turbine radius is R , and V is the inlet flow velocity.

Equation (8) is used to compute the mechanical power extracted by the turbine:

$$P = T\omega \quad (8)$$

Where the mechanical power is P , and T is the average torque of the turbine. Furthermore, C_p and C_t were determined using Equations (9) and (10), respectively.

$$C_p = \frac{P}{\frac{1}{2}\rho AV^3} \quad (9)$$

$$C_t = \frac{T}{\frac{1}{2}\rho ARV^2} \quad (10)$$

Similarly, the turbine's strength was calculated using Equation (11).

$$\sigma = N \frac{c}{D} \quad (11)$$

Where the number of blades is N , the chord length of the airfoil is c , and the turbine diameter is D .

Finally, the operating conditions were characterized by the Reynolds number using Equation (12).

$$Re_c = \frac{Vc}{\nu} \quad (12)$$

Considering air at standard conditions, with $\nu \approx 1.5 \times 10^{-5} \text{ m}^2/\text{s}$, $V = 8 \text{ m/s}$, and $c=0.05 \text{ m}$, the following results were obtained $Re_c = 2.67 \times 10^4$

3.4. Solver Configuration and Grid Independence

The simulations were performed in ANSYS Fluent using a transient scheme with a sliding mesh and a prescribed angular velocity. The turbulence was solved using the SST $k-\omega$ model, which is suitable for modeling near-wall regions and separation effects on airfoils [41]. The default y^+ insensitive near-wall treatment implemented in ANSYS Fluent was employed, with a $y^+ \approx 1$ to resolve the viscous sublayer [42]. The algorithm SIMPLE was used to link pressure and velocity, and a second-order scheme was applied during numerical approximation, using a convergence criterion of 1×10^{-6} .

To discretize the model, the effect of the computed results against the mesh size was carried out using the average torque and C_p per cycle as the study variables, calculating the relative error in C_p for each mesh size with Equation (13). Figure 4

shows the variation of torque with the mesh size, and Table 1 shows the mesh convergence for C_p . Mesh 2 was selected for the study, which exhibited a relative error of 0.0255% versus the mesh with the highest number of cells. The selected mesh has 347,854 elements and a refinement near the wall consisting of 20 layers of inflation around the blades and the axis of rotation, with a first-cell size of $4 \times 10^{-5} \text{ m}$, which corresponds to $y^+ \approx 1$, as shown in Figure 5. For each TSR , 15 cycles were simulated, and the average C_p was calculated using the values from the last cycle, once the periodic performance had stabilized.

$$Error\% = \left| \frac{C_{p_{fine}} - C_{p_{coarse}}}{C_{p_{fine}}} \right| \times 100 \quad (13)$$

Table 1. Mesh independence study for average C_p

Mesh	Cells number	C_p	Relative error (%)
1	489645	0.15668	-
2	347854	0.15664	0.0255
3	285675	0.15654	0.0893
4	198465	0.15460	1.3275
5	174648	0.15286	2.4380

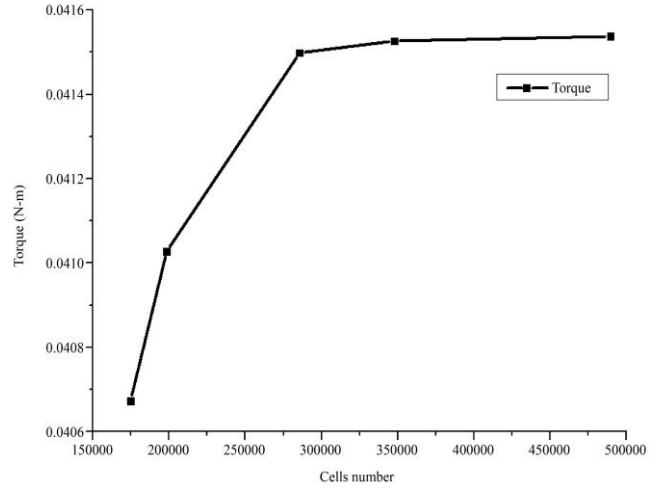


Fig. 4 Mesh independence study for average torque

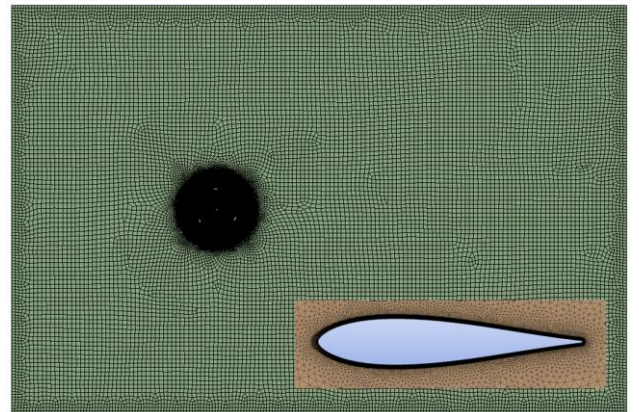


Fig. 5 Mesh for simulation

3.5. Verification of the Numerical Model

To estimate the reliability of the computations used in this study, the C_p values obtained were compared with the numerical results reported by I. Hashem and M.H. Mohamed [36]. In their work, they applied a two-dimensional CFD model to evaluate 24 airfoil profiles, including the S1046 profile analyzed in this study. Before comparing the profiles, I. Hashem and M.H. Mohamed validated their numerical model using the NACA 0021 profile, comparing their predictions with previously published experimental and three-dimensional CFD results. Therefore, the results for the S1046 profile reported in [36] are considered an appropriate reference for evaluating the predictions obtained with the numerical model used in this study. Figure 6 shows a comparison of the two sets of data.

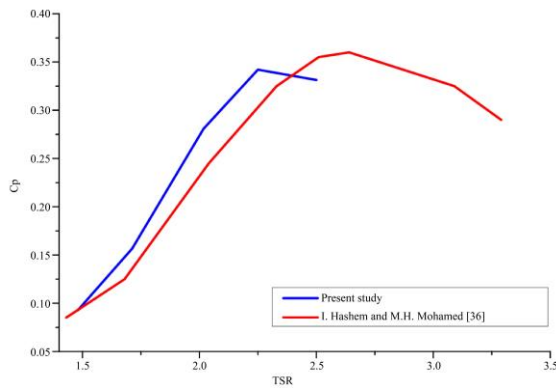


Fig. 6 Comparison of the predicted C_p -TSR curve with the S1046 numerical results reported by Hashem and Mohamed [36]

The CFD model successfully reproduces the general trend of the C_p -TSR curve, including the gradual increase in turbine performance with increasing TSR and the location of the maximum C_p . With a TSR of 1.43, the relative error was 1.22%. This increased to 5.96% near the upper end of the TSR range compared. Bianchini et al. [43], through comparisons with experimental measurements and three-dimensional simulations, proved that appropriately configured two-dimensional CFD models can reproduce the main performance trends and relevant flow characteristics of H-Darrieus turbines at a substantially lower computational cost. However, the lack of tip losses and secondary flow in the spanwise direction can lead to differences in the magnitude of the predicted aerodynamic coefficients. Considering these characteristics, the 2D approach adopted in this work provides a suitable framework for comparing the transient response of the same turbine within the evaluated TSR range.

4. Results and Discussion

4.1. Effect of TSR on the Average C_p and C_t per Cycle

Table 2 and Figure 7 show the average C_p and C_t per cycle for each TSR considered. At low TSR values (0.60–1.075), both coefficients remain below 0.02, indicating that the turbine's net contribution is practically zero per cycle. As

the TSR continues to increase, a notable change is observed: beginning at a TSR of 1.48, C_p and C_t increase to 0.092 and 0.062, respectively. This indicates a non-linear performance of the turbine coefficients with respect to the increase in TSR ; they rise to a maximum at a TSR of 2.25, where the values obtained per cycle were $C_p=0.342$ and $C_t=0.152$. When the TSR increases from 2.25 to 2.50, the previous trend is reversed. C_p decreases to 0.331, while C_t decreases to 0.132. Therefore, C_t shows a substantially greater reduction than C_p . This is significant because it demonstrates that a small change in the average C_p does not account for the total change in the turbine's aerodynamic response. Next, the instantaneous distributions of C_p and C_t are examined to determine how these differences evolve per cycle.

Table 2. Average C_p and C_t per cycle for each TSR

TSR	C_p	C_t
0.60	0.002	0.003
1.075	0.015	0.014
1.48	0.092	0.062
1.712	0.156	0.091
2.017	0.280	0.139
2.25	0.342	0.152
2.50	0.331	0.132

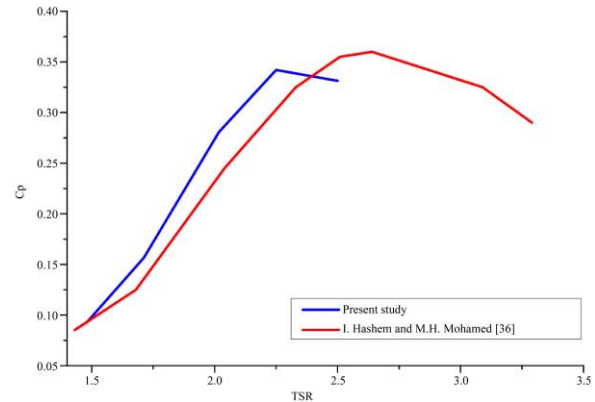


Fig. 7 Curve of the average C_p and C_t per cycle for each TSR

4.2. Instantaneous Power Coefficient Evolution per Cycle

The instantaneous C_p curve in Figure 8 shows three main peaks per cycle, separated by 120° . This periodicity shows how the individual contributions are reflected in the total C_p . The magnitude and azimuthal position of these peaks change as TSR increases, indicating that TSR affects not only the cycle-average C_p but also its distribution throughout the cycle. At low TSR values, the amplitude of the fluctuations is relatively small, which explains the very low average C_p value. As TSR increases, so does the magnitude of fluctuations. The highest instantaneous values are observed for a TSR of 2.25, where the maximum C_p value is 0.982. Under these conditions, the blades spend a greater fraction of the cycle generating positive torque, which explains the maximum C_p value obtained for the turbine. Negative C_p values are also observed at specific azimuthal positions. The

most notable value (-0.375) was obtained for $TSR=2.017$ at 22° . These instantaneous variations in C_p also enable identification of the regions of the turbine cycle where the contributions to C_p are greatest.

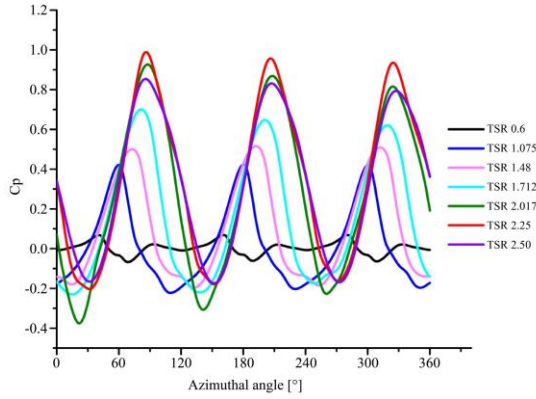


Fig. 8 Instantaneous C_p per cycle for the TSRs evaluated

4.3. Instantaneous Torque Coefficient Evolution per Cycle

Figure 9 shows the distributions of the instantaneous C_t per cycle for the evaluated TSR values. As with the C_p , three dominant peaks are observed per cycle. For a TSR of 0.6, the amplitude oscillates between 0.114 and -0.113, so its contribution to rotation is practically zero. As the TSR increases, the regions producing positive torque become larger than those producing negative C_t until reaching a TSR of 2.25, where the highest value (0.433) is recorded, with a minimum value of -0.088. For a TSR of 2.017, the maximum values of instantaneous C_t are close to those obtained for a TSR of 2.25. However, when examining the minimum values, there is a greater prevalence of negative C_t (-0.152). In the case of the TSR of 2.50, the highest value decreases to 0.34, a magnitude compared to those obtained for the $TSRs$ of 2.0175 and 2.25. However, for the minimum values, it yielded the most negative value (-0.069). In addition to the analysis of the C_t values per cycle, as the TSR increases, the peaks are displaced to the right, clustering around 90° , 210° , and 330° for the highest TSR values. This is congruent with the performance of the C_p .

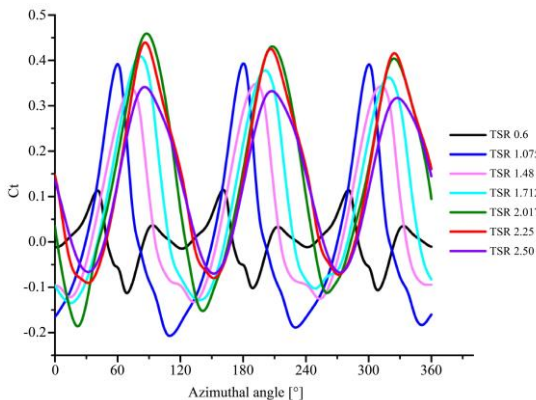


Fig. 9 Instantaneous C_t per cycle for the TSRs evaluated

Figures 10 and 11 show the instantaneous C_t contribution for each turbine blade per cycle for $TSR=2.25$ and $TSR=2.50$, respectively. For $TSR=2.25$, the individual maximum value is 0.345, while for $TSR=2.50$, they decrease to 0.296. This reduction is the reason for the decrease in the average turbine's C_t when the TSR increases from 2.25 to 2.50.

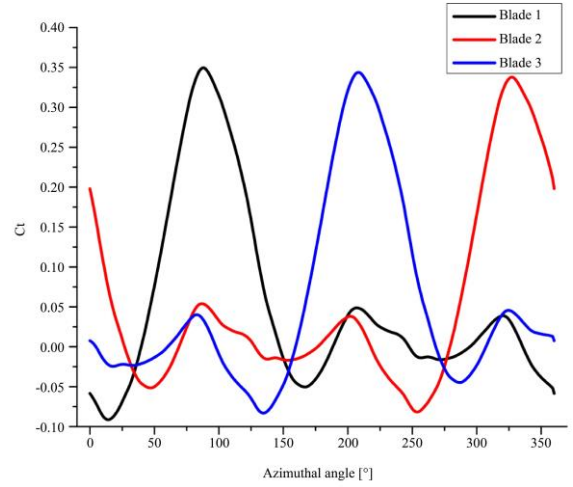


Fig. 10 Instantaneous C_t for each blade per cycle at $TSR=2.25$

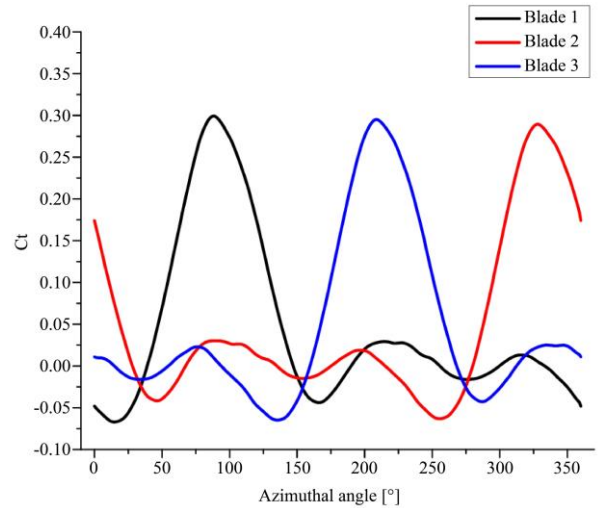


Fig. 11 Instantaneous C_t for each blade per cycle at $TSR=2.50$

4.4. Flow Field Analysis

Flow field variables were examined to explain the mechanisms responsible for the C_p and C_t values observed in the previous sections. The flow field was analyzed at representative azimuthal positions that account for the minimum and maximum energy extraction conditions for the 2.25 and 2.50 $TSRs$. For this purpose, azimuthal positions were considered at 30° intervals, evaluating Turbulent Kinetic Energy (k), pressure coefficients, and velocity pathlines.

4.4.1. Turbulent Kinetic Energy (k) Contours

Figures 12 and 13 show the distribution of k for $TSR=2.25$ and 2.50, respectively, at azimuthal positions of 0° , 30° , 60° ,

and 90° . Under both conditions, the highest concentrations of k are located around the blades and extend downstream along the turbine's rotational path. When comparing the two TSR values, the differences observed are the area and magnitude of the k regions. For $TSR=2.50$, the k regions are more extensive, particularly noticeable at the 30° , 60° , and 90° positions, with a greater prevalence of regions with values of $20 \text{ m}^2/\text{s}^2$. In contrast, for $TSR=2.25$, the concentrations of k are less extensive, remaining localized around the blades and with values of $20 \text{ m}^2/\text{s}^2$ only at the 30° position.

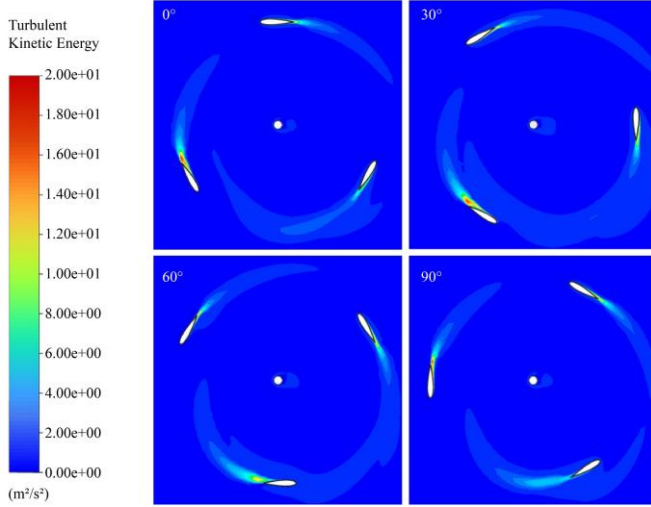


Fig. 12 k contours for different azimuthal positions at $TSR=2.25$

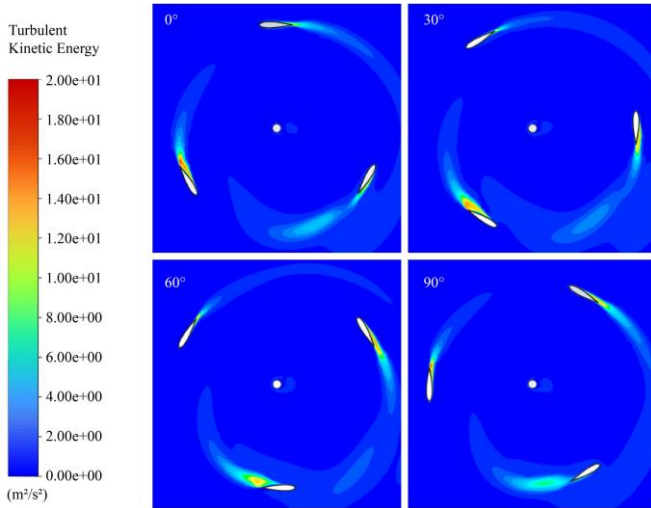


Fig. 13 k contours for different azimuthal positions at $TSR=2.50$

4.4.2. Pressure Coefficient (P_c) Contours

Figures 14 and 15 show the distribution of the P_c around the turbine for $TSR=2.25$ and 2.50 , respectively. Azimuthal positions of 0° , 30° , 60° , and 90° are considered to compare the distribution of P_c between the two TSR values. The differences are that, for $TSR=2.25$ at the 30° position, the blades located both at the top and perpendicular to the flow (left side) exhibit a high-pressure region of lesser magnitude

at their leading edge compared to the same blades with a TSR of 2.50 ; given that the rotation is counterclockwise, this pressure generates a braking effect on the turbine.

Similarly, at the 90° position, the differences are noticeable in the blade normal to the flow (left side) and the upper right blade. For the blade normal to the flow, for a TSR of 2.25 , the blade exhibits both the prominent pressure region on the outward wall of the blade and a minimum pressure region on the inner wall, which have greater amplitude than those obtained with a TSR of 2.50 , in addition to the fact that this pressure occurs in the direction of rotation.

For the upper-right blade, a difference is also present, but in this case, the pressure region at the leading edge is smaller for a TSR of 2.25 than for a TSR of 2.50 . At the 0° and 60° positions, the changes exhibit the same phenomenon, but the difference is smaller.

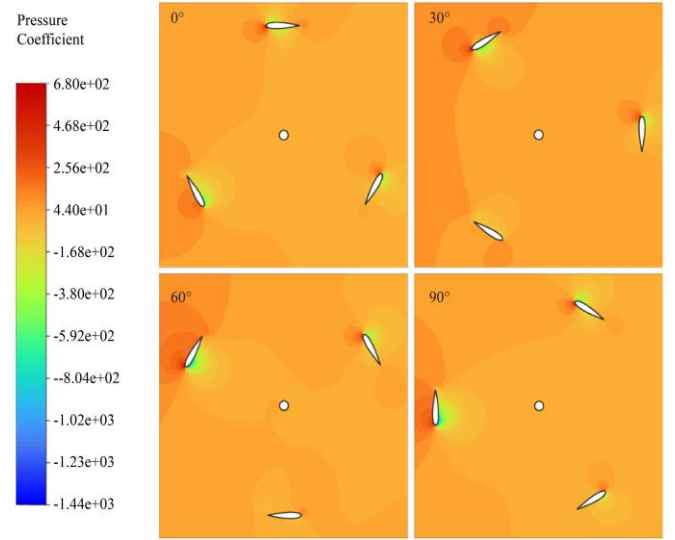


Fig. 14 P_c for various azimuthal positions at $TSR=2.25$

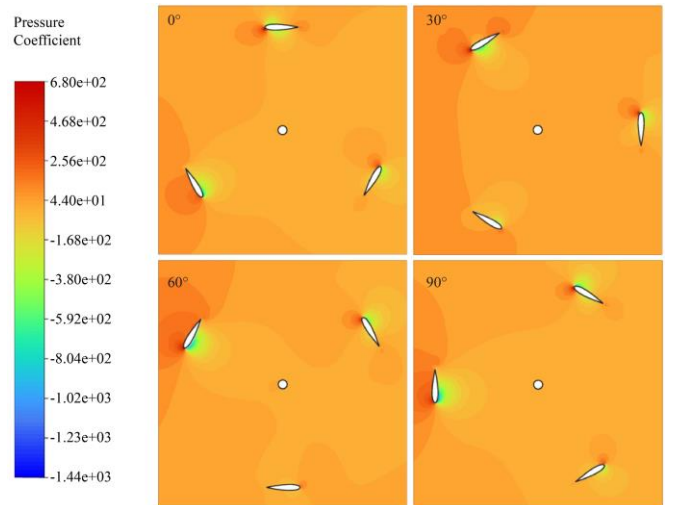


Fig. 15 P_c for various azimuthal positions at $TSR=2.50$

4.4.3. Velocity Pathlines Analysis

Figures 16 and 17 show the velocity pathlines around a blade during one cycle for $TSR=2.25$ and 2.50 , respectively. Because the angle of attack on the blade changes constantly, it is necessary to visualize the performance of the flow in contact with its walls. Between 0° and 60° , the flow remains largely attached to the blade walls. Upon reaching 60° , a region of higher velocity develops at the leading edge; at 90° , the deviation of the flow lines increases, and a region of lower velocity appears on the lower wall of the blade. At 120° , the pathlines separate more noticeably from the blade surface, and a recirculation region forms immediately downstream. Between 150° and 210° , this region decreases, and the pathlines gradually return to a direction closer to that of the

surrounding flow. Between 240° and 300° , a second marked disturbance appears, accompanied by an extensive low-velocity region downstream of the blade. At 330° , the size of this region decreases, and the trajectory lines gradually align with the incoming flow. The differences between $TSR=2.25$ and 2.50 become more evident in regions where the wake exhibits significant disturbance. At $TSR=2.50$, the low-velocity regions downstream of the blade cover a larger area, especially between 90° and 150° and between 240° and 300° . With a TSR of 2.25 , these regions remain comparatively localized and are confined to a shorter azimuthal range. The persistence of the disturbed wake with a TSR of 2.50 coincides with the lower instantaneous torque contribution observed for this operating condition.

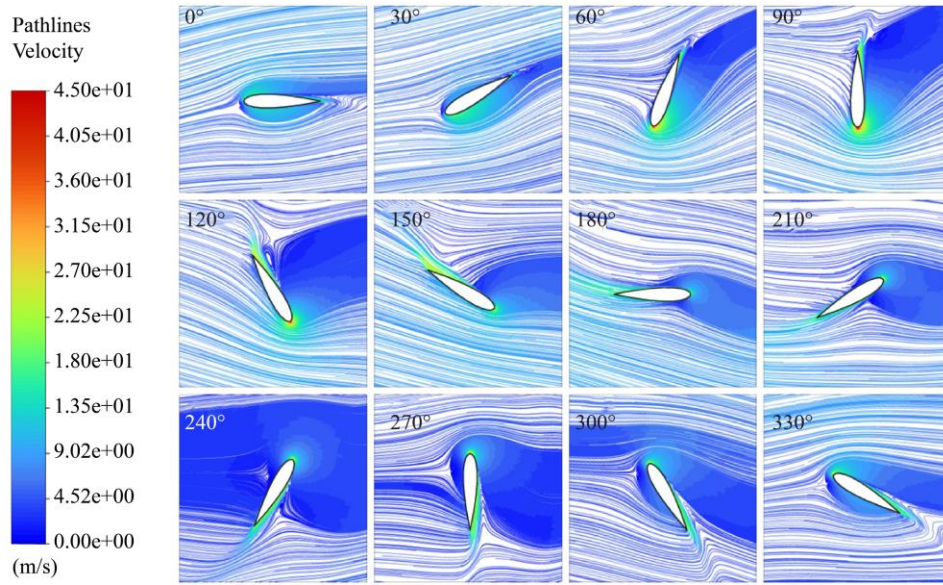


Fig. 16 Pathlines on the blades' surface for different azimuthal positions per cycle at $TSR=2.25$

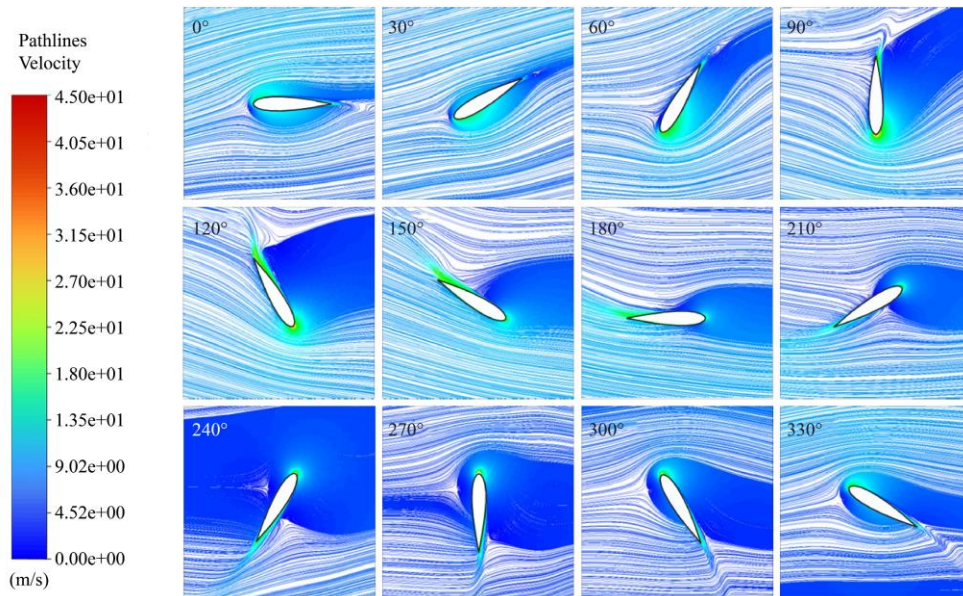


Fig. 17 Pathlines on the blades' surface for different azimuthal positions per cycle at $TSR=2.50$

5. Conclusion

The variation in TSR affects the aerodynamic response of the H-Darrieus turbine from 0.60 to 2.50. Both average coefficients per cycle increased as the TSR increased to 2.25, where the highest values were obtained ($C_p=0.342$ and $C_t=0.152$). When the TSR increased to 2.50, these coefficients decreased to 0.331 and 0.132, respectively. The reduction was more pronounced for C_t , at 13.2%, than for C_p , at 3.2%, indicating that the torque response is more sensitive to changes between these two operating conditions.

The instantaneous curves for both coefficients exhibited periodic behavior with three main peaks per cycle (90°, 210°, and 330°), separated by 120°, corresponding to the blade spacing. Individual analysis of the torque for each blade confirms this sequence, as each blade makes its main positive contribution in a different sector of the revolution. For $TSR=2.25$, the highest instantaneous C_t occurs at the 90° azimuthal position for blade 1, and this is shifted by 120° for blades 2 and 3, with a value of 0.349. For a TSR of 2.50, this value decreases to 0.299. The flow field analysis for $TSR=2.25$

and 2.50 showed differences in the distribution of the pressure coefficient, k , and velocity pathlines. The pressure distribution around the blades differs between the two conditions, particularly at 30° and 90°, which coincide with the differences observed in the instantaneous contribution of C_t . The k contours show regions with a greater extent of high energy for $TSR = 2.50$, particularly at 30°, 60°, and 90°, whereas for $TSR = 2.25$, these regions are less extensive. The pathlines also showed this difference, where $TSR=2.50$ generates regions of low velocity and disturbed flow, mainly between 90°–150° and 240°–300°, which explains the lower instantaneous contribution to C_t observed for $TSR=2.50$.

Conflicts of Interest

The authors report no conflicts of interest related to this work.

Acknowledgments

All authors contributed equally to the completion of this work. The first author thanks SECIHTI for the postdoctoral grant 745525.

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