A New Design of Savonius Wind Turbine: Numerical Study

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Abstract

This paper discusses numerically a modified Savonius wind rotor focusing on the averaged torque and power coefficients over a complete cycle of operation. The numerical study is performed using the commercial software Fluent 6.3.26 with four different turbulence models. The computations are tested against available experimental data to choose the suitable turbulence model and hence to extend the numerical investigation. A modification process of the Savonius rotor blades is designed by determining nine points on the perimeter of the blade. The two points at the ends of the blade are fixed in the same locations and the other seven points are moving to different locations to generate four polynomials shapes. V-shape blade rotor with three different V-angles is also studied in this paper. The influence of the rotor blade modification is checked based on the torque and power coefficients, keeping the Reynolds number, based on rotor diameter constant. The results obtained for the classical Savonius rotor are in agreement with the published experimental data, indicated that the method can be successfully used for such analysis. The theoretical results indicate that one of the polynomial blade shape has the best performance.

Keywords: wind energy; Vertical axis wind turbine; modified Savonius rotor; overlap; renewable energy; numerical study

1. Introduction

Wind turbines are generally classified as two families: horizontal axis and vertical axis machines. This classification refers to the position of rotor axis relative to wind direction. The Savonius wind turbine is thus classified as vertical axis wind turbine like the others Darrieus, and Gyromill or H-rotor, etc. The Savonius turbine owes its name to the Finnish engineer Savonius, who patented it in 1929 V. D’Alessandro et al [1]. The basic version of this rotor has an S-shaped cross-section formed by two semi-circular blades with a small overlap between them as shown in Fig. 1. The ratio between rotor height H and rotor diameter D is called the aspect ratio \( \alpha \). Another parameter that
affects the performance of Savonius rotor is the overlap ratio $\beta$ which is expressed as: $\beta = \frac{a}{2R}$, where $a$ is the overlap distance and $R$ is the radius of the blade circle. Savonius rotor is classified as a drag type vertical-axis wind turbine, whose principle of operation is based mainly on the difference of drag between the convex and the concave parts of the blades. However, it is shown that for some angular positions of the rotor, lift force also contributes to torque production J.-L. Menet [2]. A review of Savonius rotor advantages is present in J.-L. Menet [2]. This wind turbine has simple and robust design and can support high wind speeds. The rotor has good starting characteristics and operates at relatively low wind speeds. It does not need an orientation device and can work for all wind directions. This wind turbine operates at low tip speed ratio but unfortunately has a low power coefficient.

There are many experimental and numerical studies concerning the flow through Savonius rotor. The experimental study carried out in wind tunnel by Neal J. Roth [3] investigated Savonius rotor with different geometries. He found that the best value of both the aspect ratio was 0.77 and the overlap ratio was 0.22. F. B Ben et al [4] tested fifteen geometries of Savonius rotor wind turbine in a low speed wind tunnel to determine the rotor aerodynamic characteristics. The results of the test show that the two blades rotor has a higher static torque coefficient than the three blades rotor. M. C. Percival et al [5] evaluated the best blade overlap testing with the aid of a small prototype model. They produced a turbine capable of generating 10% of the household’s electricity. They concluded that the maximum power occurs when a blade overlap ratio of 0.2 is used at a wind speed of 10 m/s. They showed also that the most suitable blade overlap ranges from 0.1 to 0.4. Away from this region, the power output from the machine was seen to drop off rapidly. K. Mahesh et al [6] investigated experimentally the effect of changing overlap ratio from 0 to 0.7 and end tip condition either round or sharp on the performance of Savonius rotor. V. J. Modi et al [7] reported that the optimum values of aspect and overlap ratios are 0.77 and 0.25, respectively. Mojola O.O. [8] concluded that the effect of overlap ratio on rotor performance depends on its tip speed ratio. The effect of stages and blades number on the rotor performance is investigated by P. N. Shankar [9]. He found that, the highest performance was obtained with two blades and two stages Savonius rotor. The aerodynamic performance and the flow fields of Savonius rotors at various overlap ratios have been investigated by Nobuyuki Fujisawa [10]. The flow observations near the overlap indicate that the recirculation region grows with increasing the overlap ratio and is independent of the tip speed ratio, see Nobuyuki Fujisawa [10]. The aerodynamic performance of the rotor is improved at a small overlap ratio of 0.15, while it is deteriorated with larger overlap ratios. M. A. Kamoji et al [11] and [12] examined helical Savonius rotors in an open jet wind tunnel and reported that the power coefficient of the helical Savonius rotor is higher than that of conventional Savonius rotor. N. H. Mahmoud et al [13] studied experimentally the performance of Savonius rotor. They concluded that the rotor with two blades, two stages, with end plate diameter equals 1.1 times rotor diameter and 0 overlap has the best design. M. H. Mohamed et al [14]
studied the effect of obstacle plate presence on the performance of two and three blades Savonius rotor. The obstacle plate is introduced to shield partially the returning blade of a Savonius turbine and that optimizes the wind direction toward the advancing blade. The flow characteristics were accurately obtained using a realizable $k$-$\varepsilon$ turbulence model and fine grid. They concluded that the obstacle improves the rotor performance for both two and three blades rotors. I. Dobreva and F. Massouha [15] studied the flow through Savonius wind turbine experimentally using Particle Image Velocimetry (PIV) system. The obtained experimental results were compared with numerical CFD simulation. To determine the most appropriate method for flow simulation, the computations were carried out for two and three dimensions flow with the $k$-$\omega$ turbulence model. The analysis of obtained rotor power showed that the results of two dimensional $k$-$\omega$ modeling are quit higher than experiments; while those of the three dimensional $k$-$\omega$ modeling are quit lower than experiments. The comparison of wake and shedding vorticity with experiment showed that the three dimensions $k$-$\omega$ modeling gives results similar to phase averaged velocity obtained by PIV. J. V. Akwa etal [16] studied numerically the influence of overlap ratio of Savonius wind rotor on the torque and power coefficients. The study in that work had been analyze using the SST $k$-$\omega$ turbulence model. The results indicated that the maximum device performance occurs at overlap ratio with value close to 0.15.

The main objective of this paper is improving the torque and power coefficients of Savonius wind rotor by modifying the blade shape. The aerodynamic performance and the flow pattern through Savonius rotor with different blade shapes are studied. The numerical study is based on Shear-Stress Transport (SST) $k$-$\omega$ turbulence model. The verification of numerical results with a published experimental data is included to answer the question that refers to the choice of SST $k$-$\omega$ turbulence model for simulation.

2. Simulation Method

In the present study, Fluent 6.3.26 trade software has been used as one of the computational fluid dynamics (CFD) package programs. By using it, the Savonius wind rotor with different blade shapes have been analyzed from aerodynamic aspects. Modeling of the study has been made by using the Gambit 2.3.16, a program of Fluent for creating a model and mesh. Momentum equations, turbulent kinetic energy and vorticity rate of turbulent kinetic energy have each been solved numerically with the use of the commercial software Fluent 6.3.26, see Fluent, 2006, "User’s Guide Fluent 6.3.26" [17].

3. Mathematical Model

In this study Standard K-$\varepsilon$, RNG, realizable k–$\varepsilon$ and SST $k$-$\omega$ turbulence models has been used with logarithmic surface function in the analysis of turbulent flow. Momentum equation, x and y components of velocity, turbulent kinetic energy (k) and dissipation rate of turbulent kinetic energy have each been solved numerically with the use of the commercial software Fluent 6.3.26. The equations of mass and momentum conservation used by the program can be written for the compressible and incompressible 2D steady flows in the Cartesian co-ordinate as follows.

$$\frac{\partial}{\partial t} (\rho u \varphi) + \frac{\partial}{\partial y} (\rho v \varphi) = \frac{\partial}{\partial x} \left( \Gamma_{\varphi} \frac{\partial \varphi}{\partial x} \right) + \frac{\partial}{\partial y} \left( \Gamma_{\varphi} \frac{\partial \varphi}{\partial y} \right) + S_{\varphi}$$  

Here, $\varphi$ is the dependent variable, while $S_{\varphi}$ is the source term, which it has different expressions for different transport equations. The convection and diffusion terms for all the transport equations are identical, with $\Gamma_{\varphi}$ representing the diffusion coefficient for scalar variables and the effective viscosity $\mu_e$ for vectorial variables, i.e. velocities. The characteristics of the transport equations are extremely useful when the equations are discretized (reduced to algebraic equations) and solved numerically. In fact this equation also represents the continuity equation when $\varphi = 1$ and $S_{\varphi} = 0$. Table (1) gives the expressions for the source terms $S_{\varphi}$ for each dependent variable that are needed in solving ventilation problems.
The effective viscosity coefficient, \( \mu_e \), is defined by:

\[
\mu_e = \mu + \mu_t
\]  

(2)

The effective diffusion coefficient for the scalar term \( \Gamma_e \), is:

\[
\Gamma_e = \Gamma + \Gamma_t = \mu + \frac{\mu_t}{\sigma_t} 
\]  

(3)

Where \( \sigma_t \) is turbulent Prandtl or Schmidt number of the fluid.

The production term \( G_s \) is defined by:

\[
G_s = \mu_t \left[ 2 \left( \frac{\partial u}{\partial x} \right)^2 + \left( \frac{\partial v}{\partial y} \right)^2 + \left( \frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \right)^2 \right] 
\]  

(4)

### 3.1. Turbulence Models

The effective and eddy viscosities are calculated for the different suggested models as follows:

#### 3.1.1 Standard k-\( \varepsilon \) Model

\[
\mu_e = \mu + \mu_t, \quad \mu_t = C_\mu \frac{k^2}{\varepsilon} 
\]

(5)

#### 3.1.2 RNG based k-\( \varepsilon \) Model

\[
\mu_e = \mu \left[ 1 + \sqrt{\frac{C_\mu \frac{\rho}{\mu} \frac{k}{\varepsilon} \varepsilon} \right]^{-2} 
\]

(6)

The Prandtl numbers, \( \sigma_k \) and \( \sigma_\varepsilon \) in k-\( \varepsilon \) equations are modified and computed as follows:

\[
\frac{\lambda - 1.3929}{\lambda_0 - 1.3929} = 0.6321, \quad \frac{\lambda + 1.3929}{\lambda_0 + 1.3929} = 0.3679 \quad \Rightarrow \frac{\mu}{\mu_e} 
\]

(7)

where \( \lambda = 1/\sigma_k = 1/\sigma_\varepsilon = 1/\sigma_t \) and \( \lambda_0 = 1/\sigma \) (\( \sigma \) being the laminar Prandtl number for the fluid).

The rate of strain \( R \) for RNG based k-\( \varepsilon \) model is expressed as given in [16] by:

\[
R = \frac{C_\mu \eta^3 (1 - \eta) \eta_0}{1 + \beta \eta^3} \frac{k \varepsilon^2}{k} 
\]

(8)

Where, \( \beta = 0.015 \), \( \eta_0 = 4.38 \) and \( \eta \) is calculated as:

\[
\eta = \frac{k}{\varepsilon} \left( \frac{2 S_i^2}{\eta_0} \right)^{1/2} 
\]

(9)

And \( S_{ij} \) is the tensor notation, computed as follows:

\[
S_{ij} = \frac{1}{2} \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) 
\]

(10)

#### 3.1.3. Realizable k-\( \varepsilon \) model

This k-\( \varepsilon \) model was developed by Shih et al. [19]. The difference between the realizable k-\( \varepsilon \) model and the standard and RNG based k-\( \varepsilon \) models is that \( C_\mu \) is no longer constant.

\[
\mu_e = \rho C_\mu \frac{k^2}{\varepsilon} 
\]

(11)

\[
C_\mu = \frac{1}{A_0 + A_s \frac{k U^*}{\varepsilon}} 
\]

(12)

Where \( A_0 \), \( A_s \), and \( U^* \) can be calculated as given in [19]

#### 3.1.4 SST k-\( \omega \) Model

The k-\( \omega \) model is an empirical model based on model transport equations for the turbulence kinetic energy \( k \) and the specific dissipation rate \( \omega \), which incorporates modifications for Low-Reynolds-Number effects, compressibility, and shear flow spreading [20].

The effective diffusivities for the k-\( \omega \) model are given by
\[ \Gamma_k = \mu + \frac{\mu_t}{\sigma_k}, \quad \text{and} \quad \Gamma_\omega = \mu + \frac{\mu_t}{\sigma_\omega} \]

(13)

Where \( \sigma_k \) and \( \sigma_\omega \) are the turbulent Prandtl Numbers for \( k \) and \( \omega \) respectively. The turbulent viscosity \( \mu_t \) is computed by combining \( k \) and \( \omega \) as follows:

\[ \mu_t = \alpha^* \frac{\rho k}{\omega} \]

(14)

The coefficient \( \alpha^* \) damps the turbulent viscosity causing a low-Reynolds-number correction. It is given by:

\[ \alpha^* = \alpha_\infty \left( \frac{\alpha^*_0 + \text{Re}_t / R_k}{1 + \text{Re}_t / R_k} \right) \]

(15)

Where

\[ \text{Re}_t = \frac{\rho k}{\mu \omega}, \quad R_k = 6, \quad \alpha^*_0 = \frac{\beta_i}{3}, \text{ and } \beta_i = 0.072 \]

But in the high-Reynolds-number form of the \( k-\omega \) model, \( \alpha^* = \alpha^*_\infty = 1 \).

\[ \sigma_k = \frac{1}{F_1 / \sigma_{k,1} + (1 - F_1) / \sigma_{k,2}} \]

(16)

\[ \sigma_\omega = \frac{1}{F_1 / \sigma_{\omega,1} + (1 - F_1) / \sigma_{\omega,2}} \]

(17)

\[ \Phi_1 = \min[\max(\frac{\sqrt{k}}{0.09 \omega y}, \frac{500 \mu}{\rho y^2 \omega}), \frac{4 \rho k}{\sigma_{\omega,2}^2 D_{\omega}^+ y^2}] \]

(18)

\[ D_{\omega}^+ = \max[2 \rho \frac{1}{\sigma_{\omega,2}} \frac{1}{\partial x_j} \frac{\partial \omega}{\partial x_j} 10^{-2}] \]

(19)

Where \( y \) is the distance to the next surface and \( D_{\omega}^+ \) is the positive portion of the cross-diffusion term. Model constants:

\[ \sigma_{k,1} = 1.176, \quad \sigma_{\omega,1} = 2.0, \quad \sigma_{k,2} = 1.0, \quad \sigma_{\omega,2} = 1.168 \]

The constants of the turbulence models are given in table (1).

| TABLE 1: SOURCE TERMS IN THE TRANSPORT EQUATIONS |
|---|---|---|
| Equation | \( \Phi \) | \( \Gamma_\phi \) | \( S_{\Phi} \) |
| Continuity | 1 | 0 | 0 |
| \( u \)-momentum | \( u \) | \( \epsilon \) | \( \frac{\partial^2 p}{\partial x^2} + \frac{\partial^2 \epsilon}{\partial x^2} + \frac{\partial^2 \epsilon}{\partial y^2} + \frac{\partial^2 \epsilon}{\partial z^2} \) |
| \( v \)-momentum | \( v \) | \( \epsilon \) | \( \frac{\partial^2 p}{\partial y^2} + \frac{\partial^2 \epsilon}{\partial y^2} + \frac{\partial^2 \epsilon}{\partial z^2} + \epsilon (\rho - \rho_0) \) |
| Kinetic energy | \( k \) | \( \Gamma_k \) | \( \epsilon \) |
| Dissipation rate | \( \epsilon \) | \( \Gamma_\epsilon \) | \( \frac{\epsilon}{k} (C_g \sigma_S) + C_g k^2 \) |
4. Calculation of Torque and Power Coefficients

Holding the solution of conservation equations and the chosen turbulence model, one can simply calculate the rotor torque \( T \) by integrating the forces resulting from the pressure and shear on the rotor as given in the following equation:

\[
T = \sum_{f}(F_{f}^{\text{pressure}} + F_{f}^{\text{shear}}) \cdot l = \\
\sum_{f}((p_{f} - p_{\text{ref}})A_{f} + (-\tau_{f} \cdot A_{f}))_{\text{(concave)}}l \\
-[(p_{f} - p_{\text{ref}})A_{f} + (-\tau_{f} \cdot A_{f}))_{\text{(convex)}}l
\]

where \( F_{f}^{\text{pressure}} \) and \( F_{f}^{\text{shear}} \) are the pressure \((p)\) and shear \((\tau)\) forces vectors acting on the face area vector \( A_{f} \), respectively and \( l \) is a local torque arm vector from the rotation axis, about which the moment is taken (Fluent, 2006).

The torque coefficients can then be calculated from the following equations:

\[
C_{t} = \frac{4T}{\rho U^{2}D^{2}H}
\]

\[
C_{p} = \frac{P}{P_{W}} = \frac{T \omega}{0.5\rho A U^{3}} = \\
\frac{T \omega D}{0.5\rho A U^{3} D} = C_{t}\lambda
\]

where \( P \) is the generated mechanical power, \( P_{W} \) the wind power, \( \omega \) the angular velocity, \( U \) the incoming free-stream wind speed, \( \lambda \) the tip speed ratio and \( A \) the rotor projected area defined as \( A=DH \).

The grid independent study is done by checking a case of classical Savonius rotor with 0.15 overlap at different tip speed ratio. Number of cells tested reads 120,000, 180,000, 250,000, 350,000 and 450,000. The results are in good agreement with experimental data for numbers of cells in the range about of 250,000. Corresponding to the near wall non dimensional distance \( y^{+} \) should be less than 5. The types of cells in fixed domain is map (structured grid) but in rotating the grid is studied with pave grid (unstructured grid) type having more accurate results with accurate agreement with published experimental result of N. Fujisawa [10].

5. Modification Procedure

To modify the blade of a Savonius rotor, nine points are located at equidistance on the perimeter of the conventional Savonius blade as shown in Fig. 2. The points P(1) and P(2) are fixed in the same locations. The locations of the points P(3) to P(9) are movable to different locations keeping a constant rotor perimeter. The variations of the locations of these points produce multi-shape of modified Savonius blades. In the present work, four polynomial shapes are studied as shown in Table (2), where \( x \) is measured from the center of the blade. To create dimensionless formulations \((Y = y / C \) and \( X = x / C \)) of the blade, the chord \( C \) is used as the reference length parameter, see Fig. 2. Figure 3 illustrates the four modified blades and three V-shape blades. Three V-shape blades with three different V-angles are studied, namely V-angle= 60°, V-angle= 80° and V-angle=}
90°. For all studied rotor the overlap ratio is constant with the value ($\beta=0.15$). For all studied cases a constant upstream flow velocity of $6 \text{ m/s}$ as in the experiments in Fujisawa, (1992) is considered.

![Figure 2. Schematic discretion to modify the blade](image1)

![Figure 3. Studied blades shapes](image2)

### Table 2: Equations Representing the Modified Blades

<table>
<thead>
<tr>
<th>Modify type</th>
<th>Generated equation of the blade curve</th>
</tr>
</thead>
<tbody>
<tr>
<td>Modify 1</td>
<td>$Y = 0.469 - 0.722 X - 0.41X^2 + 2.61X^3 - 5.69X^4$</td>
</tr>
<tr>
<td>Modify 2</td>
<td>$Y = 0.4 - 0.59 X + 0.23 X^2 - 0.97X^3 - 7.44X^4 + 13.48X^5$</td>
</tr>
</tbody>
</table>
| Modify 3    | $Y = 0.429 - 0.65 X - 1.22 X^2 + 3.62X^3 + 11.55X^4$  
- $42.1X^5 - 52.5X^6 + 151.28X^7$ |
| Modify 4    | $Y = 0.55 - 0.62 X - 1.7 X^2 + 4.14X^3 - 1.51X^4 - 7.57X^5$ |
| Conventional | $X^2 + Y^2 = 0.25$ |

6. Computational Domain and Boundary Condition

The boundary condition “velocity inlet $U=constant$” is imposed upstream of the rotor at a distance of $6D$ from the rotor axis and takes the same area of wind tunnel as presented in N. Fujisawa [10]. The remaining upstream area is taken as “pressure inlet”. The boundary condition at downstream of the rotor is “pressure outlet”. The top and bottom of flow domain are taken at $10D$ from the rotational axis where a “pressure outlet” boundary condition is imposed where the change of any parameters $\phi$ with respect to $x$ or $y$-directions is equal to 0. Figure 4 shows the computational domain and boundary conditions. The grid is created by means of ANSYS Gambit 2.3.16. The two-dimensional grid for modify 4 rotor is presented in Fig. 5. The domain is divided into two parts, a fixed part with structured grid and a rotating part with unstructured grid. The two parts are separated by an interface.

![Figure 4. Two-dimension computational domain](image3)
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7. Results and Discussion

Savonius rotor is a drag type wind turbine. The drag force on the advanced concave blade is higher than on the return convex blade so that the rotor rotates. To validate the theoretical results and select a suitable turbulence model that describes the flow parameters around Savonius rotor, a comparison between results of different turbulence models, RNG $k$-$\varepsilon$, realizable $k$-$\varepsilon$, SST $k$-$\omega$ and standard $k$-$\varepsilon$ results against the experimental results published in T. Hayashi et al [18]. Figures 6 and 7 illustrate the comparison between results of the different turbulence models and published experimental data of T. Hayashi et al [18] and Nobuyuki Fujisawa [10]. The results in Figs. 6 and 7 indicate that, SST $k$-$\omega$ turbulence model has more accurate results compared with the other studied models. Based on these results the SST $k$-$\omega$ turbulence model is used to simulate the theoretical results in this paper.

The main idea behind modifying blade is to displace the location of resultant acting force to increase the torque arm. The V-shape Savonius rotor with different angles aims at decreasing the drag force on the return blades. The effect of modifying has been studied with static and dynamic orientations. The static torque means that the computations are performed without rotation of the rotor but at different orientations ($\theta$). The following results are registered satisfying all convergence conditions of all variables. The variation of static torque coefficient versus rotor angles is shown in Figs. 8 and 9. From Fig. 8 it is clear that the static torque coefficient has a different behavior with varying the rotors angles. For a $0^\circ$ rotor angle the conventional rotor has a higher value of torque coefficient followed by modify 3 rotor. For the rotor angle of $135^\circ$ the higher value of the static torque coefficient is for modify 4 followed by V- angle $= 90^\circ$ rotor. It depends on the flow of air on the advanced and return blades while in modify 4 rotor the smoothing of the blade shape causes the
decrease of the drag force on the return blade. At the rotor angle 135° the static torque coefficient is negative for conventional and high positive values for modified blades. From Fig. 8 one can see that the decrease of the V-angles causes an increase of the static torque coefficient at most rotor angles. The modified blades with V-shapes have a negative torque in many positions because the straight shape of the blades makes the static torque on the returning blades is higher than on the advanced blade as shown in Fig. 9. From Figs. 8 and 9 one cannot determine the best modify blade type, for that Table 3 shows the average static torque coefficient over complete cycle from 0° to 180° rotor angle. From Table 3 modify 4 blade has higher value of average static torque coefficient. These results indicate that the modified blades with polynomial shape produce the highest torque because the smoothing shape to give lower drag on the return blades.

Figure 6. Static torque coefficient against rotor angle: comparison between numerical work and published experimental data.

Figure 7. Torque coefficient against tip speed ratio: comparison between numerical work and published experimental data.

Figure 8. Variation of static torque coefficient versus blade type at various rotor angles.
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Figure 9. Variation of static torque coefficient versus rotor angle at different rotor types.

TABLE 3: THE AVERAGE STATIC TORQUE COEFFICIENT FOR MODIFIED SAVONIUS BLADES

<table>
<thead>
<tr>
<th>Rotor shape</th>
<th>V-angle = 90°</th>
<th>V-angle = 80°</th>
<th>V-angle = 60°</th>
<th>Modify 1</th>
<th>Modify 2</th>
<th>Modify 3</th>
<th>Modify 4</th>
<th>Convention</th>
</tr>
</thead>
<tbody>
<tr>
<td>Averag, Cₜₛ</td>
<td>0.081</td>
<td>0.087</td>
<td>0.096</td>
<td>0.175</td>
<td>0.162</td>
<td>0.179</td>
<td>0.195</td>
<td>0.171</td>
</tr>
</tbody>
</table>

The variation of the average torque and average power coefficients of the rotating rotor with the tip speed ratio is shown in Figs. 10 and 11. From these Figures one can see that modify 4 rotor has the best values of average torque and average power coefficients. At tip speed ratio up to 0.9 the average torque and average power coefficient have a clear difference between modify 4 rotor and other rotors. For tip speed ratio greater than 0.9 the difference between the rotors average torque and average power coefficient is small except V-angles 90° in which a high reduction of torque and power because of the strong vortex shedding generated behind it. Also from the figures the decrease of the angles of V-angle rotors causes an increase of power and torque coefficients of the rotor, because of the reduction of the drag on the return blade.

Figure 10. Torque coefficient versus tip speed ratio.
Because of the performance progress of Savonius rotor with its modification (modify 4), the next discussion are devoted to theoretical results of modify 4 only. It is known that, when the rotor rotates the values of the torque and the power coefficients vary with rotation about a mean value. The mean value also varies with the rotational rate of the rotor. Figure 12 illustrates the variation of the torque coefficient versus the time at different tip speed ratios for modify 4 Savonius rotor. However, through one second operation time cycle of the rotor, it is clear that the values become almost cyclical, and the solution stabilizes, approaching a steady operation.

Figure 13 shows the variation of the torque coefficient over one complete cycle of rotation from 0° to 360° at different tip speed ratios for modify 4 rotor. From the figure one can see that for every speed ratio the torque coefficient oscillates around a mean value. The absolute value decreases with the increase of tip speed ratio for tip speed ratio more than 0.5. This because the tangential velocity of the advancing blade exceeds the flow velocity and the momentum transmitted from air into the blade decreases.
Figure 14a illustrates the velocity contours of non-rotated modify 4 Savonius blade at 0° and 135° rotor angles. As noticed, an oscillating asymmetrical wake is generated behind the rotor, which increases with increasing the rotor displacing angle. A large asymmetrical recirculation region is clear behind the rotor at 135° rotor angle. The streamlines around the rotors at rotor angles 0° and 135° also investigated at fig. 14b. Figure 15 indicates the big recirculation region behind the rotor for rotating modify 4 rotor at tip speed ratio equal one. However, the unsteadiness of such oscillating wake must receive more attention in future research because this will generate unsteady force normal to the wind direction which leads to mechanical vibration of the rotor.

![Velocity contours](image1)

**(a) θ=0°**

![Velocity contours](image2)

**(b) θ=135°**

Figure 14a. Velocity contours (m/s) for modify 4 Savonius rotor at rotor degree: (a) 0° and (b) 135°.
Figure 14 b. Stream lines for modify 4 Savonius rotor at rotor degree: (a) 0° and (b) 135°.

Figure 15. Large recirculation behind modify 4 rotor blades at tip speed ratio 1 coloured by velocity contours (m/s).

8. Conclusions

This paper aims to study the effect of modifying the Savonius rotor blade in purpose of improving the performance of the rotor. The study includes seven shapes of the blades in addition to the conventional Savonius rotor. The conservation equations of mass and momentum are solved using the Finite Volume Method. The turbulence model SST $k-\omega$ is used as based on its good agreement with published work, to describe the flow around the developed geometries of the blade. The results showed that the modification of the blades of the rotor causes an increase of the performance of the rotor. One of the new developed polynomial blade shapes namely has the highest values of average torque and average power coefficients more in view of the other shapes.
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Nomenclature

<table>
<thead>
<tr>
<th>Latin Symbols</th>
<th>Greek Symbols</th>
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<tbody>
<tr>
<td>A</td>
<td>λ</td>
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<tr>
<td>A_f</td>
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<tr>
<td>a</td>
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References


