Effect of the blade arc angle on the performance of a Savonius wind turbine

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Abstract
Savonius wind turbine is a common vertical axis wind turbine which simply comprises two or three arc-type blades and can generate power under poor wind conditions. With the aim of increasing the turbine's power efficiency, the effect of the blade arc angle on the performance of a typical two-bladed Savonius wind turbine is investigated with a transient computational fluid dynamics method. Simulations were based on the Reynolds Averaged Navier–Stokes equations, and the renormalization group $k-\varepsilon$ turbulent model was utilized. The numerical method was validated with existing experimental data. The results indicate that the turbine with a blade arc angle of 160° generates the maximum power coefficient, 0.2836, which is 8.37% higher than that from a conventional Savonius turbine.

Keywords
Vertical axis wind turbine, Savonius, blade arc angle, computational fluid dynamics

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Introduction
Wind turbines are generally divided into two categories, horizontal axis wind turbines (HAWTs) and vertical axis wind turbines (VAWTs), based on the relative position between their rotation axis and the wind direction. VAWTs are more favored in small-scale power generations because they respond to flow from any direction and allow generating equipment to be located on the ground shaft, and the maintenance costs can be reduced.

Savonius wind turbine is a common VAWT which generates torque through the combined effects of drag and inside forces and typically comprises two or three arc-type blades. Savonius turbine has the following advantages over other types of wind turbines: (1) ability to operate under complex turbulent flows; (2) low rotation speed and noise emission; (3) simple structure, low cost.

However, the Savonius turbine has a relatively lower power efficiency, with a maximum efficiency of 0.25, compared with the lift-type wind turbines, such as HAWTs and Darrieus wind turbines. In order to increase the efficiency, comprehensive studies have been done to examine the effects of various design parameters, such as the rotor aspect ratio, the overlap, the number of blades and the endplates, on the performance of Savonius turbines with experimental and numerical methods.

In addition, some researchers worked to enhance the Savonius turbine performance by changing the structure of the turbine. Gupta et al. studied the aerodynamic characteristics of a modified Savonius turbine.
with helical blades. Golecha et al.\textsuperscript{9} and Altan and Atilgan\textsuperscript{10} placed a guide vane in front of the turbine to deflect flow for the returning blade. As for the novel blade shape design, McTavish et al.\textsuperscript{11} proposed a modified blade shape and carried out both steady and transient computational fluid dynamics (CFD) simulations. Kamoji et al.\textsuperscript{12} and Kacprzak et al.\textsuperscript{13} studied the performance of modified turbines with spline-type and Bach-type blades, and an increment of 16\% in the efficiency was found in the case of using spline-type blades.

This article aims to increase the efficiency of Savonius wind turbines by analyzing the effect of the blade arc angle on the turbine performance and to find the optimal arc angle corresponding to the maximum efficiency. The analysis has been performed by solving numerically the incompressible Navier–Stokes equations with the aid of the CFD code Fluent 13.0.\textsuperscript{14}

### Parameters definition

The two-dimensional schematic view and geometrical parameters of a two-bladed Savonius wind turbine are presented in Figure 1, where \( U \) is the wind velocity, \( \theta \) is the azimuth angle of the blade, \( \varphi \) is the blade arc angle, \( \omega \) is the rotation velocity of the turbine, \( r \) is the blade radius, and \( D \) is the turbine diameter.

In order to investigate the effect of the blade arc angle, \( \varphi \), on the performance of the turbine, we performed simulations on turbines with \( \varphi \) varying from 150\(^\circ\) to 200\(^\circ\). The specific values of the turbine geometrical parameters for each case are shown in Table 1.

### Numerical method

Because the straight blades have the same cross section in the span direction, the blade span effect can be ignored and two-dimensional transient simulations are carried out with the aid of the commercial CFD code Fluent 13.0. A sliding mesh model was applied to realize the rotation motion of the rotor.

### Computation domains and boundary settings

In order to allow a full development of the flow as well as decrease the blockage effect, the computational domain was a rectangle of \( 18D \times 12D \). The rotor was placed in the symmetry axis of the top and bottom boundary and at a distance of \( 6D \) from the left boundary (see Figure 2). The overall domain is split into two subdomains, including an external station domain and an internal rotation domain containing the rotor. In the simulations, the internal rotation domain rotates with the rotor angular velocity \( \omega \).

The boundary conditions employed consist of a constant velocity inlet (7 m/s) on the left side, a pressure outlet on right, and two symmetry boundary conditions on top and bottom. No-slip boundary conditions were imposed at the surface of the blades. Sliding interfaces exist between the external stationary domain and the internal rotational domain, allowing the transport of the flow properties.

### Grids generation

The computation grids were generated using the MESH tool in ANSYS 13.0. Quadrilateral elements are
less memory-occupying than triangular elements (the number of quadrilateral elements is only a half of the triangular elements considering the same grid nodes number) in two-dimensional simulations and more suitable for simulation of the boundary layers flow. Therefore, the two subdomains were discretized with quadrilateral elements (see Figure 3(a)). As can be seen in Figure 3(b), grids closest to the profiles of the blades were refined with boundary layer elements to describe with sufficient precision the boundary layer flow. The Reynolds number was computed based on the wind speed and the blade chord length, resulting in a value of about $4 \times 10^5$. The first element height of the mesh above the surface was set as 0.04 mm, and the $y^+$ value for the first node from the wall was in the order of 10. There were eight layers in total, and a layer growth rate of 1.2 was chosen. Moreover, grid node density was higher in the rotational subdomain than in the stationary subdomains.

**Turbulence model and solution sets**

The renormalization group (RNG) $k-\varepsilon$ turbulence model was employed. This model is known to give highly accurate predictions of rotating machinery because the effect of swirl on turbulence is included. Moreover, the RNG $k-\varepsilon$ model provides an analytical-derived differential formula for effective viscosity that accounts for low-Reynolds-number effect. Besides, a standard wall treatment is applied to increase the solution accuracy under low Reynolds numbers.

For each case listed in Table 1, several simulations were carried out with the tip speed ratio varying from 0.6 to 1.4. Tip speed ratio represents the ratio of the blade tip speed to the wind speed and has the following expression

$$\lambda = \frac{D\omega}{2U}$$

where $\omega$ is the angular velocity of the blade, $D$ is the blade diameter, and $U$ is the wind speed. Each simulation lasted for three revolutions. The time step used was set as $1^\circ$/step, that is, the rotor turned $1^\circ$ in each time step, and each time step takes 100 iterations. Convergence was determined by the order of magnitude of the residuals. The drop of all scaled residuals below $10^{-5}$ was employed as the convergence criterion.

**Numerical method verification and validation**

**Verification**

A grid convergence study was performed to evaluate the influence of grid density on the torque of the rotor. The simulations were conducted on a conventional Savonius turbine with $\phi = 180^\circ$ (case 4 turbine in Table 1) at a tip speed ratio of 1. Figure 4 shows the dynamic torque coefficients on the turbine for a whole cycle with three different grid densities, with approximately 60,000, 80,000, and 120,000 elements, respectively.

Torque coefficient is defined as follows

$$C_m = \frac{M}{0.25pS U^2 D}$$

where $M$ is the generated torque, and $S$ is the cross section area, given by the relationship $S = DH$ with $H$ being the height of the blade. Since only two-dimensional simulations were performed, the unit height $H = 1$ m was used. It can be seen that grids with approximately 80,000 and 120,000 elements gave substantially the same results. Considering the time economy in the simulation, the grid with 80,000 elements (24,000 elements in the stationary domain and 56,000 elements in the rotation domain) was chosen for the following simulations.

**Validation**

Validation studies were conducted using a conventional two-bladed Savonius turbine, that is, the case 4 turbine. The simulation results are then compared with the
experimental data from the SANDIA lab. Figure 5 shows the comparison between the coefficients of averaged torque on the rotor at a range of different tip speed ratios.

It can be seen that the simulation results coincided well with the experiment data, especially at lower tip speed ratios. Therefore, it is acceptable to use the RNG $k-e$ turbulence together with a standard wall treatment to predict the performance of a low-Re wind rotor.

**Results and discussion**

**Torque characteristics**

Keeping the wind speed constant, simulations were carried out for cases 1–6 at different tip speed ratios. Figure 6 shows the coefficients of the averaged torque on the turbines with respect to $\lambda$ at different blade arc angles. It can be seen that all the curves of torque look like straight lines and are almost parallel to each other. Take the case $\varphi = 180^\circ$ as reference; we find that as $\varphi$ increases, the torque curves lower down, while as $\varphi$ decreases, the torque curves rise until $\varphi = 160^\circ$ and then lower down. The case $\varphi = 160^\circ$ has the highest curve when $\lambda \geq 1.2$ and that $\varphi = 170^\circ$ has the highest curve when $\lambda < 1.2$. Besides, the averaged torques decrease with the increase of $\lambda$. This can be illustrated by Figure 7.

Figure 7 shows the dynamic torque on a single blade of the case 4 turbine for a whole rotation cycle. The torque curves are plotted with respect to the azimuth angle $\theta$. As $\lambda$ increases, the turbine rotates faster and the blade has a higher linear velocity. Therefore, the relative velocity between the wind and the blade on the downwind direction will reduce, decreasing the thrust force and the positive torque on the blade (the region of positive torque in Figure 7 gets smaller as $\lambda$ increases), while the relative velocity between the wind and the blade on the upwind direction will go up, increasing the drag and the negative torque on the blade (the region of negative torque in Figure 7 gets larger as $\lambda$ increases).

**Power characteristics**

The coefficient of power is defined as

$$C_P = \frac{P}{0.5 \rho SU^3} = \lambda C_m$$

where $P$ is the generated power. Figure 8 shows the coefficients of power generated by the turbines with respect to $\lambda$ at different blade arc angles. It can be seen from the figure that there is a peak value for each curve and the coefficient of power increases with $\lambda$ until it obtains its peak; after that it drops down as $\lambda$ further increases. The $\lambda$ corresponding to the peak $C_P$ on each
Figure 5. Results for numerical method validation.

Figure 6. Coefficient of torque with respect to $\lambda$ for turbines with different blade arc angles.
Figure 7. Blade dynamic torque coefficient for the case 4 turbine at different tip speed ratios.

Figure 8. Coefficient of power with respect to $\lambda$ for turbines with different blade arc angles.
curve becomes larger as the blade arc angle increases. Taking the case \( \varphi = 180^\circ \) as reference, we find that as \( \varphi \) increases, the power curves lower down, while as \( \varphi \) decreases, the power curves rise until \( \varphi = 160^\circ \) and then lower down. The case \( \varphi = 160^\circ \) has the highest curve when \( \lambda \geq 1.2 \) and that \( \varphi = 170^\circ \) has the highest curve when \( \lambda < 1.2 \).

Table 2 lists the maximum coefficient of power for cases 1–6. It can be seen that the turbine efficiency can be obviously enhanced by choosing a proper blade arc angle. The turbine with a blade arc angle of 160° has the highest coefficient of power, 0.2836, obtained at a tip speed ratio of 1.2, which is 8.37% higher than that from a conventional Savonius turbine (the case 4 turbine).

**Figure 9.** Blade dynamic torque coefficient for different blade arc angles at \( \lambda = 1 \).

| Case | Blade angle | \( C_p \text{ max} \) | \( C_p \) gain percentage (relative to case 4) |
|------|-------------|------------------------|-----------------------------------------------|
| 1    | 150         | 0.2687                 | 2.67%                                         |
| 2    | 160         | 0.2836                 | 8.37%                                         |
| 3    | 170         | 0.2835                 | 8.33%                                         |
| 4    | 180         | 0.2617                 | 0.00%                                         |
| 5    | 190         | 0.2521                 | −3.67%                                        |
| 6    | 200         | 0.2271                 | −13.22%                                       |

**Mechanism of the effect of blade arc angle on the turbine performance**

Figure 9 shows the dynamic torque on a single blade with different arc angles for a rotation cycle at \( \lambda = 1 \). The torque curves are plotted with respect to the azimuth angle \( \theta \). The blade torque enters the positive terrain at about \( \theta = 310^\circ \) and gets the maximum at about \( \theta = 20^\circ \). The blade torque enters the negative at about \( \theta = 130^\circ \) and gets the minimum at about \( \theta = 250^\circ \). The blade arc angle has a slight delay-effect on the azimuth angle when the blade torque shifts from negative to positive, and from positive to negative. Besides, the blade arc angle has a regular influence on the peak values of the torque: (1) the peak positive torque decreases with the increase of the blade arc angle, and (2) the peak negative torque increases with the blade arc angle. But these two impacts have opposite contributions to the averaged torque; impact 1 decreases the averaged torque while impact 2 is on the contrary. Therefore, there must be an optimal blade arc angle which makes the turbine to generate the maximum torque, and eventually power.

Figure 10 shows the pressure distribution on the blade of cases 2, 4, and 6 turbines at \( \theta = 0^\circ \) (one typical azimuth position of the positive torque region). As the blade arc angle decreases, the pressure on the concave side of the blade increases slightly, while that on the convex side drops obviously. Therefore, the blade with
a smaller arc angle could obtain a higher positive torque near $\theta = 0^\circ$ (also shown in Figure 9). The discrepancy of the pressure on the blade surface can be explained by the contours of pressure and velocity in Figures 11 and 12.

It can be seen in Figure 11 that a negative pressure area exists on the convex side of the blade, and as blade arc angle increases, this area gets smaller and weaker. This is because that there is a stream with high velocity near the concave side of the blade (see Figure 12), and as the blade arc angle increases, the blade becomes thicker and longer and has a higher aerodynamic drag on the flow and thus reduces the flow velocity (also see Figure 12). It is the drop of flow velocity that decreases the negative pressure. Besides, a slight drop in the pressure on the concave side of the blade is also observed (see Figure 11). This is possibly caused by the shelter-effect of the blade. Blade with a larger arc angle has a longer tip, which impedes the wind to flow from the upstream to the concave surface of the blade and reduces the torque on the blade.

Figure 13 shows the pressure distribution on the blade of cases 2, 4, and 6 turbines at $\theta = 270^\circ$ (one typical azimuth position of the negative torque region).
Distinctions between the pressure on the concave side of the blade are not substantial. But the negative pressure on the convex side decreases as the blade arc angle increases. Therefore, the blade with a larger arc angle could obtain a smaller negative torque near $\theta = 270^\circ$ (also shown in Figure 9). The reason possibly lies in that the blade with a larger arc angle has a smoother shape and results in a smaller drag and thus a smaller negative torque.

Figure 14 shows the pressure contours of three turbines at $\lambda = 1$ and $\theta = 0^\circ$: (a) case 2 turbine, (b) case 4 turbine, and (c) case 6 turbine. And as the arc angle decreases, the stagnation point moves downward, leading to an outward shift of the whole blade pressure center. Table 3 lists the positions of the pressure centers of the three blades. The outward shift of the blade pressure center adds the arm of force with respect to the rotation center of the turbine, which further increases the negative torque.

Figure 13. Pressure distribution on the blade of cases 2, 4, and 6 turbines at $\lambda = 1$ and $\theta = 270^\circ$.

Conclusion
Two-dimensional CFD simulations were performed on Savonius wind turbines. The effect of the blade arc
angle on the turbine’s torque and power performance was studied. The mechanism of how the blade arc angle affects the turbine performance was explained with field contours. We find that the positive peak torque on the blade can be increased by reducing the blade ellipticity, while the negative peak torque is on the contrary. Turbine with a blade arc angle of $\theta = 160^\circ$ has the highest coefficient of power, 0.2836, which is 8.37% higher than that from a conventional Savonius turbine.

### Declaration of conflicting interests

The authors declare that there is no conflict of interest.

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**Table 3.** Centers of pressure of the blades at $\lambda = 1$ and $\theta = 270^\circ$.

| Case | Blade arc angle | Center of pressure |
|------|----------------|--------------------|
| 2    | 160            | $y = -0.2232$      |
| 4    | 180            | $y = -0.2229$      |
| 6    | 200            | $y = -0.2223$      |

**Figure 14.** Pressure contours of cases 2, 4, and 6 turbines at $\lambda = 1$ and $\theta = 270^\circ$: (a) case 2 turbine, (b) case 4 turbine, and (c) case 6 turbine.