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To cite this article: Yongchao Zhang et al 2019 IOP Conf. Ser.: Earth Environ. Sci. 240 042009

View the article online for updates and enhancements.
Numerical investigation of performance and flow patterns of a modified Savonius hydraulic rotor

Yongchao Zhang¹, Can Kang¹, Chen Pan¹ and Wisdom Opare¹,²

1. School of Energy and Power Engineering, Jiangsu University, Zhenjiang 21213, China
2. Faculty of Engineering, Takoradi Technical University, Takoradi P.O. BOX 256, Ghana

Corresponding author: Can Kang, e-mail: kangcan@ujs.edu.cn

Abstract. The Savonius hydraulic turbine is effective in harnessing the energy carried by flowing water. The advantages of the Savonius hydraulic turbine, such as simple in structure and good start-up characteristics, have been acknowledged. Nevertheless, the lack of a detailed investigation of the flows around the Savonius hydraulic rotor hampers the technical advancement of the Savonius hydraulic turbine. Here, a modified Savonius hydraulic rotor based on conventional Savonius rotors was designed to investigate the performance and flow characteristics of the Savonius hydraulic rotor. Computational fluid dynamics techniques were used. The influence of the rotor blade geometry and the tip speed ratio was considered. The results indicate that the ratio of the length of the straight edge to the radius of the circular arc of the blade has an important impact on the performance of the Savonius hydraulic rotor. The maximum power coefficient of the rotor with such a ratio of 0.81 is 58% higher than the rotor with the ratio of 0.46. The tip speed ratio corresponding to the maximum power coefficient is related to the ratio. Comparative analysis of the velocity and pressure distributions for different rotor setting angles was conducted to describe the interaction between the flow medium and the rotor. The torque coefficient of the rotor depends considerably on the area and position of high upstream pressure; meanwhile, downstream vortices impose a negative effect on the rotation of the rotor. The results provide a sound support for the optimal design of the Savonius hydraulic turbine.

1. Introduction

The increasing demand for energy consumption and serious environmental problems caused by conventional energy sources are forcing mankind to develop renewable and green energy sources. The hydrokinetic energy, as an emerging energy source, attracts extensive attentions from researchers due to its advantages of high energy density, minimal environmental effects and good predictability [1]. The hydraulic turbine is an effective tool for harnessing the hydrokinetic energy from the incident flow with low head and velocity. It bears unique features like simple structures, low installation and maintenance cost and little impact on the environment [2]. In 2003, the world’s first water current turbine was installed by Marine Current Turbines (MCT) Ltd. and IT-Power in an offshore location with a rated power of 300 kW [3]. The hydraulic turbines are classified into two categories according to the rotor axis orientation with respect to the direction of the incident flow, namely the horizontal-axis turbine and the vertical-axis turbine [4]. Compared to the horizontal-axis turbine which has high efficiency and self-start ability, the vertical-axis turbine is more suitable for the occasions where water flow is limited [5].
At present, the Savonius hydraulic turbine is more popular for small-scale power generation among the vertical-axis turbines.

The Savonius rotor was invented in the 1920s for converting the wind energy into usable energy forms [6]. In recent years, with successful applications of the Savonius rotor in wind turbines, researchers attempted to apply the Savonius rotor in the flowing water in consideration of some distinct advantages of water like higher specific weight and momentum than air at the same velocity [7]. Nevertheless, the efficiency of the Savonius hydraulic turbine is low, necessitating investigations aiming to enhance the performance of the Savonius hydraulic turbine.

Kamoji et al. [8] investigated experimentally the influence of the aspect ratio on the performance of the Savonius rotor. The results indicate that the rotor with the aspect ratio of 0.7 shows better performance in both power coefficient and torque coefficient. Patel et al. [9] tested the performance of rotors with different aspect ratios to study the effect of aspect ratio. The optimum value of aspect ratio was recommended as 1.8 according to the experimental results. In order to investigate the influence of the overlap ratio, Yaakob et al. [10] tested the performance of Savonius type vertical-axis marine current turbines with overlap ratios ranging from 0.1 to 0.6 using computational fluid dynamics (CFD) technique. The results justified that the optimum value of the tip speed ratio related to the maximum torque coefficient was 0.21. An experiment was carried out by Kamoji et al. [11] to study the influence of the overlap ratio on the helical Savonius rotors. The authors proved that the helical Savonius rotor with an overlap ratio of zero has the maximum power coefficient. Apart from this, rotor geometry and the number of the rotor stages are important factors which affect the rotor performance significantly. Kumar et al. [12, 13] investigated numerically the effects of the blade arc angle, the blade shape factor and the twist angle of the blade on the performance of the Savonius hydrokinetic turbine. It was observed that the maximum power coefficient value of 0.426 was obtained for the blade arc angle of 150°, the blade shape factor of 0.6 and the twist angle of 12.5°. Wahyudi et al. [14] proposed a new design of Tandem Blade Savonius (TBS) rotor and calculated the pressure field of three types of TBS rotors. The results indicated that the convergent TBS rotor had a higher pressure drop and better performance than other types. Nakajima et al. [15] investigated the effect of the phase difference in the Savonius rotor blades on the power performance. It was proved that the maximum power coefficient of the rotor with 90 degrees of phase difference in the blades was enhanced by about 10%. Mahmoud et al. [16] tested the Savonius rotors with different numbers of blades and found that the Savonius rotor with two blades has better performance than the rotors with three and four blades. For the same purpose, Wenehenubun et al. [17] performed experimental and numerical studies and found that the Savonius rotor with three blades has the best performance at a high tip speed ratio (TSR).

Apart from the geometry parameters of the rotor itself, some adjunct devices, such as deflector plate, guide vanes, end plates and installation parameters, also affect the performance of the Savonius turbine significantly. Golecha et al. [4, 18] experimentally investigated the influence of the deflector plate on the performance of a modified Savonius water turbine. It proves that the deflector plate placed at its optimal position increases the power coefficient of the rotor by 50%. Sivasegaram [19] conducted an experiment to reveal the influence of the end plate size on the performance of the Savonius rotor. The author reported that the optimum end plate diameter is 1.1 times of the rotor diameter. Elbatran et al. [20] designed the ducted nozzle Savonius water turbine by adding a duct around the turbine. The results presented that the efficiency of the ducted nozzle turbine is 78% higher than that of the conventional modified Savonius rotor. Nakajima et al. [21] investigated experimentally the influence of the installation parameters on the performance of the Savonius hydraulic turbine. The results indicated that the distance between the rotor and bottom wall of the tunnel, and the rotating direction of the rotor affect the rotor performance significantly.

Generally, a substantial amount of investigations have been conducted to improve the performance of Savonius hydraulic turbine. However, no consistent conclusions about the optimal design of the Savonius rotors have been proposed, and even some conclusions about the optimal parameters are conflicting. Therefore, it is necessary to make a thorough investigation on the performance and flow patterns for the Savonius hydraulic turbine. In the present study, a modified Savonius hydraulic rotor
based on conventional Savonius rotors was designed to investigate the performance and flow characteristics of the Savonius hydraulic rotor. The transient power coefficient, the torque coefficient and detailed flow patterns for various rotor rotation angles were obtained using unsteady CFD technique. Meanwhile, the blade shape factor and the tip speed ratio were considered.

2. Physical model
A modified Savonius hydraulic rotor based on conventional Savonius rotors was proposed, as shown in figure 1. The diameter of the rotor is 60 mm and the height is 48 mm; therefore, the aspect ratio is 0.8. The blade shape factor \((L/R_1)\) is defined as the ratio of the length of the straight edge to the radius of the blade. The rotation angle \(\theta\) is the angle between the line connecting the tip of two blades and the direction of the incident flow. As indicated in Fig. 1, the incident flow comes from the left and the rotor rotates counterclockwise. Three Savonius rotors with the same diameter and similar shape but different shape factors were proposed, as shown in figure 2. The geometric parameters of the three rotors are listed in Table 1.

![Figure 1. Geometry of the Savonius rotor.](image1)

![Figure 2. Schematic of three rotors.](image2)

![Figure 3. Dimensions of the computational domain.](image3)

Figure 3. shows the model of computational domain. The computational domain is divided into two zones, the stationary zone and rotational zone, and the data transfer between the two regions was realized via the interface. The diameter of the rotational zone \(D_r\) is 100 mm, which is larger than the diameter of the rotor to ensure the continuity of the flow field around the rotor. Meanwhile, the computational domain extended \(5D_r\) upstream and \(10D_r\) downstream with respect to the rotor to realize the integrity of the flow structures around the rotor.

| Geometric parameters | Rotor 1 | Rotor 2 | Rotor 3 |
|----------------------|---------|---------|---------|
| Diameter, \(D\) (mm) | 60      | 60      | 60      |
| Height, \(H\) (mm)   | 48      | 48      | 48      |
| Aspect ratio         | 0.8     | 0.8     | 0.8     |
| Length of straight edge, \(L\) (mm) | 8 | 10 | 12 |
| Ratio of blade, \(R_1\) (mm) | 17.4 | 16.1 | 14.8 |
| Blade shape factor   | 0.46    | 0.62    | 0.81    |

3. Numerical scheme
3.1. Mesh generation
The quality of the grids has a significant influence on the accuracy of the numerical results. Structured hexahedral grids were adopted in the present work in view of small truncation error and helpful convergence characteristics associated with structured grids. The commercial code of ANSYS ICEM was employed for the mesh generation and the final grid scheme was illustrated in figure 4. The grids near the blade surface were refined to ensure that the distance from the centroid of the first grid cell to the blade surface ($y^+$) is less than 1, which satisfies the requirements of capturing the near-wall flow phenomena.

![Figure 4. Cross-sectional view of structural grids.](image4.png)

![Figure 5. Mesh sensitivity examination.](image5.png)

Grid number is another important factor that affects the accuracy of the numerical simulation results. Therefore, the grid sensitivity examination was performed as a preliminary step. Five grid schemes with different grid numbers were designed and the simulations were carried out at the same tip speed ratio. Figure 5 shows the variation of the torque coefficient of Rotor 1 with the grid number. It is noted that at first the torque coefficient exhibits a significant change as the grid number increases. However, the difference of the torque coefficient between different grid schemes decreases rapidly as the grid number exceeds 2.6 million. The difference of torque coefficient between the grid schemes with 3.8 and 4.7 million grids is even less than 0.1%. Therefore, the grid scheme with 3.8 million grids was applied in subsequent simulation.

3.2. Turbulence model and boundary conditions
ANSYS Fluent software was employed for the numerical simulation, and the transient simulation was conducted based on the Reynolds averaged Navier-Stokes equations. The SIMPLE pressure-velocity coupling solver was used. Meanwhile, second-order upwind scheme was applied to discretize the convective terms to improve the accuracy of the calculation. The shear stress transport (SST) $k$-$\omega$ model was selected as the turbulence model for its unique advantages of combining the $k$-$\omega$ and the $k$-$\varepsilon$ turbulence models for treating the near-wall flows and the free-stream flows respectively. Previous work [20, 22-24] has proven that the SST $k$-$\omega$ turbulence model was suitable for predicting the performance and flow field of the Savonius rotor. Nasef et al. [25] analyzed the influence of four two-equation turbulence models on the performance of Savonius rotor by comparing the numerical results with the experimental results. The result showed that the SST $k$-$\omega$ turbulence model gave rise to more accurate results compared with other three two-equation turbulence models.

Pure water of 20 °C, with the density of 1000 kg/m$^3$ and the dynamic viscosity of $1.0 \times 10^{-3}$ m$^2$/s, was used as the working fluid. The velocity inlet boundary condition was adopted in the flow area and the incident flow was assumed to be uniform. Velocity magnitudes ranging from 0.8 m/s to 3.8 m/s were set at the inlet under conditions of different Reynolds numbers. The outflow boundary condition was set at the outlet of the computational domain. The side surfaces and the upper and lower surfaces of the computational domain was set as stationary wall. In the unsteady calculation process, the coupling of the rotational and the stationary domains was solved by the mesh motion model. The rotational speed of the rotor was set to 160 rpm; meanwhile, the time step was set to $1.042 \times 10^{-3}$ s, which corresponded to the period with which the rotor rotates by 1 degree. Besides, the convergence criterion was set to $9 \times$...
10⁻⁶ to guarantee the simulation accuracy. Surface roughness of the blade surface was taken into consideration and the roughness value was set to 0.025 mm.

4. Results and discussions

4.1. Performance characteristics

The torque coefficient, $C_t$, and the power coefficient, $C_p$, are two quantities that directly reflect the performance characteristics of the Savonius rotor. The power coefficient is the ratio of the output power of the rotor to the actual hydro kinetic power available in the blade swept area, as expressed in equation (1).

$$C_p = \frac{M\omega}{\frac{1}{2}\rho AU^3}$$

Here, $M$ stands for the torque produced by the rotor, $\omega$ is the angular velocity of the rotor, $\rho$ is the density of the water, $A$ is the blade swept area and $U$ represents the velocity of the flow stream.

The torque coefficient is defined as:

$$C_t = \frac{M\omega}{\frac{1}{2}\rho AU^2}$$

where $D$ represents the diameter of the rotor.

Tip speed ratio ($TSR$) is the ratio of the tangential velocity at the rotor tip to the flow stream velocity:

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

Figure 6. shows the variation of the power coefficient with the tip speed ratio for the three Savonius hydraulic rotors considered. It is evident that the overall performance of Rotor 3 is better than that of Rotor 2 and Rotor 1. With the increase in the blade shape factor $L/R_1$, the rotor performance is enhanced. The maximum power coefficient of Rotors 1, 2 and 3 are 0.104, 0.14 and 0.164, respectively. Compared with Rotor 1, Rotor 3 is superior in the maximum power coefficient by about 58%. Moreover, it is indicated in Fig. 6 that the tip speed ratio corresponding to the maximum power coefficient increases with the blade shape factor $L/R_1$. The maximum power coefficient of Rotor 1 corresponds to a tip speed ratio of 0.32, while the maximum power coefficient of Rotor 3 arises at the tip speed ratio of 0.5.

The dynamic torque coefficient of the three rotors at $TSR=0.42$ is illustrated in figure 7. It is seen that the three curves of the dynamic torque coefficient are similar over the whole rotation cycle. From the starting position ($\theta=0^\circ$), the torque coefficient increases gradually with the rotation of the rotor, and arrives at its maximum at about $\theta=45^\circ$. Then it drops with the increase in the rotating angle and approaches zero near the rotation angle of 110°. After that, the torque coefficient becomes negative and the smallest value is reached near the rotation angle of 125°. Then the torque coefficient increases again and reaches zero at about $\theta=146^\circ$. Subsequently, the torque coefficient becomes positive and increases gradually with the rotating angle until $\theta=180^\circ$. Within one rotation cycle, the torque coefficient changes with the period of 180°, which is ascribed to that the rotor is composed of two identical blades. In one period, negative torque coefficients cover about one-twelfth of the period.

Figure 8. shows the variation of the instantaneous torque coefficient with the rotation angle for Rotor 3 at $TSR=0.5$, at this $TSR$ value the Rotor 3 has the maximum power coefficient of 0.164. Within one rotation cycle, the maximum torque coefficient is 0.571 at $\theta=38^\circ$. After that, the torque coefficient decreases with the increase in the rotation angle and reaches zero at $\theta=108^\circ$. As the rotation angle continues to increases, the torque coefficient becomes negative, that means the rotation is hindered by the fluid. At the rotation angle of 128°, the negative torque reaches the largest of -0.182. Subsequently, the negative torque coefficient diminishes gradually and arrives at zero at $\theta=146^\circ$. In the remaining part of the rotation cycle, the torque coefficient becomes positive and increases gradually.
4.2. Flow patterns

In figure 9., the pressure and velocity distributions for Rotor 3 at different rotation angles at TSR=0.5 are displayed. It is inferable from the pressure distribution that pressure drop occurs across the rotor from the upstream to the downstream side. This phenomenon justifies that the pressure energy is extracted by the rotor and then converted to the mechanical energy of the rotor and causes the rotation of the rotor. At $\theta=0^\circ$, a high pressure zone forms at the tip of the advancing blade, as shown in figure 9. (a). This high pressure area is related to the velocity direction perpendicular to blade surface near the tip of the advancing blade. In this context, a stagnation region forms near the blade surface, and all the kinetic energy is converted to pressure energy. Hence, the pressure difference between the concave and convex sides of the blade is high for the advancing blade, and a positive torque is produced to drive the rotation of the rotor. For the velocity distribution, it is seen that two large-scale swirling flow structures, vortex A and B marked with red ellipses, are generated near the rotor. In the downstream region, there is also a vortex structure, vortex C, which is shed from the surface of the rotor.

With the increase in the rotation angle, the stagnation region shifts to the concave side of the advancing blade. The area covered by high upstream pressure expands gradually and high pressure is concentrated at the concave side of the advancing blade. Meanwhile, the pressure at the convex side of the advancing blade decreases with the rotation angle. Furthermore, the pressure difference on both sides of the advancing blade is intensified and the positive torque coefficient of the rotor increases accordingly. At $\theta=38^\circ$, as shown in figure 9. (b), the area covered by high pressure reaches its maximum and high pressure is completely concentrated at the concave side of the advancing blade, while the torque coefficient reaches as high as 0.571. The vortex B near the convex side of the returning blade sheds from the blade, and vortex A in the concave side of the returning blade migrates to the rotor center gradually.

As the rotation angle continues to increase, the area of the returning blade that faces the incident flow is increased. The high pressure area in upstream becomes smaller and the high-pressure zone migrates to the convex side of the returning blade slowly. Hence, the pressure difference between the two sides of the returning blade is reinforced and the negative torque produced by the returning blade is strengthened accordingly. Concurrently, pressure at the concave side of the advancing blade drops and the pressure difference over the advancing blade decreases as well, leading to the decrease in the positive torque generated on the advancing blade. As a result, the torque coefficient of the rotor decreases as the rotation angle increases. At the rotation angle of 108 $^\circ$, it is illustrated form figure 9. (c) that the area with high upstream pressure resides in the central part of the convex side of the returning blade. The negative torque caused by the returning blade equals to the positive torque produced by the advancing blade and the resultant torque tends to zero. At this time, a swirling flow forms above the convex side of the advancing blade, vortex D, and a low-pressure area occurs near the tip of the advancing blade.

As the rotating angle exceeds 108 $^\circ$, the area with high upstream pressure migrates to the outer part of the returning blade and the negative torque caused by the returning blade increases accordingly. However, the low-pressure area near the convex side of the advancing blade sheds off from the advancing blade gradually and the pressure difference between the two sides of the advancing blade
attenuates simultaneously. The torque becomes negative due to the negative torque produced by the returning blade excels the positive torque produced by the advancing blade. The negative resultant torque increases with the rotation of the rotor. At the rotation angle of 128 °, the negative torque produced by the returning blade reaches the largest as the area with high upstream pressure moves towards the tip of the returning blade, as presented in figure 9. (d). In this case, the highest negative torque coefficient of -0.182 is obtained. In the velocity field, the vortex intensity increases gradually and the vortex area expands as well. The large-scale swirling flow imposes resistance for the rotation of the rotor.
Figure 9. Pressure and velocity distributions for Rotor 3 at different rotation angles at TSR=0.5 (left: pressure distribution; right: velocity distribution) (a) $\theta=0^\circ$, $C_t=0.416$; (b) $\theta=38^\circ$, $C_{t_{\text{max}}}=0.571$; (c) $\theta=108^\circ$, $C_t=0$; (d) $\theta=128^\circ$, $C_{t_{\text{min}}}=-0.182$; (e) $\theta=146^\circ$, $C_t=0$.

As the rotating angle increases, the area with high upstream pressure transfers to the tip side of the returning blade. Meanwhile, the transverse dimension of the high-pressure area over the returning blade decreases and the acting position gradually approaches the axis of the rotor. On the other hand, the low-pressure area at the convex side of the advancing blade has shed off from the blade surface. So the pressure on both sides of the advancing blade tends to be balanced and the pressure difference is minimized. As a result, the resultant negative torque coefficient of the rotor decreases with the rotation of the rotor. It is seen from figure 9. (e) that at the rotation angle of $146^\circ$, the distance between the acting position of the high upstream pressure on the returning blade and the axis approaches zero. Meanwhile, the pressure difference on both sides of the advancing blade diminishes due to the low-pressure area deviating considerably from the advancing blade. So the resultant torque coefficient is zero at the rotating angle of $146^\circ$. At this rotation angle, the downstream swirling flow evolve into a set of symmetrical vortices with opposite directions of rotation. Subsequently, the pressure at the concave side of the returning blade increases gradually. On the contrary, the pressure at the convex side of the returning blade decreases by degrees, except from the high-pressure area near the tip side. Hence, the returning blade turns into advancing blade and the torque coefficient increases again. One rotation cycle is covered as the rotation angle is increased to $180^\circ$.

Pressure distribution for the three rotors at different rotation angles and at TSR=0.42 are displayed in Fig. 10. The rotation angle related to the maximum torque coefficient of the three rotors are $44^\circ$, $44^\circ$ and $42^\circ$ respectively. The area of the blade concave side increases with the shape factor, $L/R_1$, and the ability of collecting fluid with the concave side of the advancing blade is improved accordingly. More fluid gathers at the concave side of the advancing blade and more kinetic energy is converted to pressure energy. As a result, the pressure at the advancing blade concave side of Rotor 3 is the highest and the area of high upstream pressure for Rotor 3 is larger than that for Rotors 1 and 2, as shown in figure 10. (a). Hence, the maximum torque coefficient of Rotor 3 is the largest and is reached at $42^\circ$, earlier than $44^\circ$ for Rotors 1 and 2.

The torque coefficient reaches zero for the first time in the rotation cycle, an area with high upstream pressure forms at the convex side of the returning blade and an area with low downstream pressure at the convex side of the advancing blade, as presented in figure 10. (b). The positions of the two areas are symmetrical about the axis of the rotor. Meanwhile, the area covered by low-pressure downstream of the returning blade differs with the shape factor as the low-pressure area for Rotor 3 reaches its smallest.

The torque coefficient is negative as the area with high upstream pressure migrates towards the convex side of the returning blade. The $L$ of Rotor 3 is the largest, so the contact area between the convex side of the returning blade and the incident flow for Rotor 3 is the largest. Hence, the resistance to the returning blade of Rotor 3 is the largest. It is indicated from figure 10. (c) that high upstream pressure acts on the middle part of the convex side of the returning blade of Rotor 3. However, for Rotors 1 and 2, high upstream pressure acts on the tip side of the convex side of the returning blade. Therefore, the negative torque coefficient of Rotor 3 is the largest value of 0.189, while for Rotors 1 and 2, 0.156 and
0.184, respectively. Although the negative torque coefficient of Rotor 3 is the highest, it does not affect the overall performance of Rotor 3. Figure 10. (d) illustrates the pressure distribution for three rotors at the rotation angle which the torque coefficient reaches zero for the second time. The torque coefficient of Rotor 3 reaches zero at the rotation angle of 142°, which is earlier than the 148° of Rotor 1 and 146° of Rotor 2. Therefore, the rotation angle range in which the torque coefficient is negative is smallest for Rotor 3. This is also one of the reasons for the optimal performance of Rotor 3.

5. Conclusions
Unsteady numerical simulation was conducted to investigate the performance and flow characteristics of modified Savonius hydraulic rotors. Both the rotor blade geometry and tip speed ratio were taken into
consideration. Detailed instantaneous pressure distributions and local flow patterns around the rotors were obtained. The influence of the blade shape factor on the performance and flow field near the rotor was analyzed. The results are helpful for devising methods of improving the efficiency of the Savonius hydraulic rotor. The following conclusions are drawn from the results:

1. The ratio of the length of the straight edge to the radius of the circular arc of the blade affects the performance of the rotor significantly. The rotor with the ratio of 0.81 has the maximum power coefficient of 0.164, which is 58% higher than the rotor with the ratio of 0.46.

2. The torque coefficient of the rotor depends considerably on the area and position of the high upstream pressure. The larger of high-pressure area at the concave side of the advancing blade and the closer the position of high pressure to the outside of the blade, the higher the torque coefficient is.

3. The large-scale vortices downstream of the returning blade hinder the rotation of the rotor and produce negative torque. As the large-scale vortices approaches the returning blade, the resistance to the rotation of the rotor is reinforced.

Acknowledgement
The authors are grateful to the financial support from the Six Talents Peak Project of Jiangsu Province of China under Grant 2015-ZBZZ-018.

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