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NUMERICAL INVESTIGATION OF MODIFIED SAVONIUS WIND TURBINE WITH VARIOUS STRAIGHT BLADE ANGLE

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ABSTRACT

The present paper aims to numerically investigate the two-dimensional flow analysis of modified Savonius wind turbine using computational fluid dynamics. The effects of the straight blade angle on the turbine performance were studied. Simulations based on the RANS equations and the SST-k-ω turbulence model are used to simulate the airflow over the turbine blades. Both the static and dynamic simulations were performed. In the static simulation, the drag and lift coefficient on the Savonius turbine were directly calculated at every angular position, and the time-averaged moment and power coefficients were computed in each of the dynamic simulations. From the results, it can be concluded that the turbine with a straight blade angle of 100° model gives the better performance at higher Tip Sped Ratio (TSR) than other models.

KEYWORDS

VAWT, Savonius wind turbine, Bach type, Computational Fluid Dynamics, Straight blade angle, Power Coefficient.

1. INTRODUCTION

The Savonius wind turbine is a mechanical power-generating device that has been studied by many researchers since the 1920s. This turbine has some advantages such as relatively simple construction, low manufacturing cost, wind reception from any direction with low operating speed, good starting ability and has many rotor configuration options. However, the efficiency of the Savonius wind turbines is lower than the other horizontal axis wind turbines due to the negative torque generated by the returning blade. Savonius wind turbines are usually applied as wind pump for irrigation or agricultural purposes, street lighting systems, and driving electrical generators [1-4].

Numerous experimental and numerical studies of the Savonius wind turbine have been found in many technical and scientific literatures. Different configurations and arrangements of Savonius rotor show that the rotor performance is influenced by operational conditions, geometric and airflow parameters. The performance of this rotor can be improved by changes in design parameters, including blade arc angle, aspect ratio, overlap size, and gap size [5-10]. The overlap and gap region between Savonius blades, allowing the fluid was entering the concave side of a blade to flow to the side of the other blades and produce additional pressure. Several studies aboutSavonius rotor have been carried out on the number of blades and stages of the rotor [11-16]. Some authors also have reported that augmentation and wind guide improve on power coefficient that made the construction of the turbine system more com... vonius rotor focus on the modification of bl... ce than the previous model. Published resu... otor performance is affected by model and ... [23-28].

The ... into three geometric shapes, classical or s... and Bach type. Bach type has a blade shap... ut there is a section with a straight blad... nd that, the Bach-type rotor is better than ... g blade geometry has experimentally been ... ance of Bach type.Modi and Fernando conducted an experimental study and flow visualization on this Bach type blade to get the best performance on the optimal configuration of the Savoniusrotor associated with the blade geometry, gap size (a/d) =0; blade overlap (b/d) = 0; aspect ratio (A) =0.77; blade shape parameters (p/q) =0.2; blade arc angle (ϕ) =135° [29]. A group researchers performed the experiments by modifying the Bach-type blades without using a shaft, where Cp_{max} was obtained on the blade with an overlap ratio of 0, the angle of the blade curve of 124° and the aspect ratio 0.7, compared with the semi-circular model [30]. Additionally, compared the new profile blade with modified Bach type, semi-circular blade, semi-elliptic blade, and Benesh type models to determine the optimum values of power coefficient. The optimized results obtained a new profile that performs better with Cp_{max} 0.31 than modified Bach, Benesh type, semi-elliptic and semi-circular of 0.3; 0.29; 0.26 and 0.23 respectively.

Computational approaches have also been used to efficiently predict the flow past a Bach-type turbine and calculate its power coefficient. Wang and Yeung simulated the flow past a micro-scale Bach-type vertical-axis wind using a viscous Discrete-Vortex Method for use as micro-scale energy harvesters that can be applied to power [31]. A studied the flow physics around a modified Bach and Benesh profiles of Savonius rotorusing multi-physics CFD solver to determine drag and lift characteristics acting on the rotors [32]. The result shows that the drag coefficient (CD) for the modified Bach profile is higher than the Benesh profile. Also, numerically analyzed three geometries of Savonius wind turbine rotors (semi-circular, elliptic and Bach type) [33]. All geometries attain maximum Cp at TSR 0:8. In that region, the Bach-type geometry produces the highest power output. Furthermore, the numerical investigation to asses an optimal blade profile in the modified design of Bach-type blades has been done, the results showed that the maximum power coefficient with blade arc angle of 135° [34].

The literature review reported presents that changes in the blade geometry design of Savonius wind turbines could improve the power coefficient. Many Inventive design enhancements have been done on Bach-type blades, but the simplest modification is to modify the shape of the blade. Most of the previous research workcarried out on modifications of the curved blade, relating to the arc angle of the blade for efficiency improvement, but the effect of straight blade angles on the performance of the rotor has not been investigated so far. This straight blade is necessary to the movement of the airflow after passing through the curved blade leading to overlap areas which potentially reduces the negative torque of the returning blade. This section can be further investigated by bending the blade at the connection of the curved blade and the straight blade, inward and outward to form a specific angle.

In this paper, the main purpose of this study is to numerically investigate different modified Bach-type of savonius wind turbine configurations that differ for the bend angle between curved blade and straight blade to obtain optimum performance. Both dynamic and static simulations are executed. The static analyses are carried out to compute aerodynamic performance associated with lift and drag coefficients and the dynamic simulations to obtain turbine performance in terms of moment coefficient (C_M) and power coefficient (C_P) during operative cycles.

2. METHODOLOGY

2.1 Blade Model

This study focuses on the modification of the geometry of the Savonius wind turbine of Bach type, especially on the straight blade. The blade of the Savonius-Bach type as the baseline model used refers to Roy and Saha [34]. The optimum configuration is obtained at the angle of the blade curve ϕ = 135°, the length of the straight blade (S1) and the radius of the blade (S2) of 42% of the length of the blade chord, the blade overlap, e = 40% of the blade chord length and the distance between the blades, a = 10% of the blade chord length. Modify geometric shapes in which the straight blades are bent inwards and outward to form an angle (β) varied between of 70°-110° at intervals of 10° (in this case, β = 90° is baseline or without being bent). The constant parameters are the rotor diameter (D), the blade angle (ϕ), the length of the straight blade (S1), the radius of the blade (S2) and, and the distance between the blades (a). The diagram of the modified design under test is shown in Fig. 1.

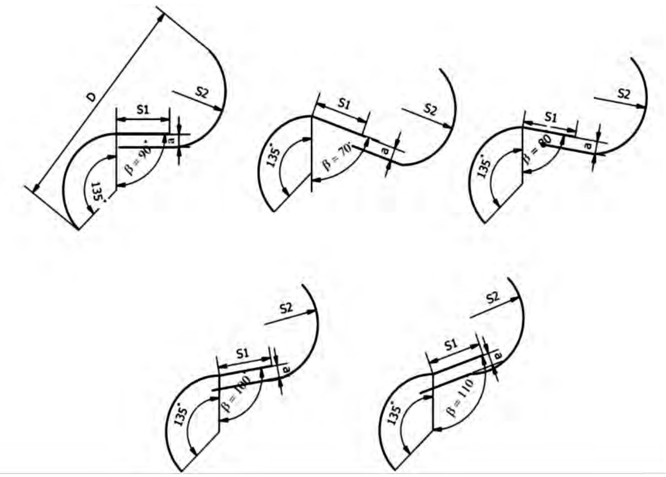


Figure 1: Geometry changes of Bach type blade at the different angle of β

2.2 Solution Methodology

In this study, the performance of Savonius wind turbine with straight blade modification was performed by numerical simulation using 2 (two) Dimensional Computational Fluid Dynamicsfor static and dynamic simulations. The static simulation is used to analyzethe fluid flow characteristics across the blade at various angular positions, meanwhile dynamic simulation with rotating blades is used to predict the turbine performance with the parameters including power coefficient (C_P) and torque or moment coefficient (C_M). A total of 35 dynamic simulations were perf

$$= \frac{\omega \cdot R}{V}$$

aver static simulations to determine the lift coef C_D) during the half cycle of the turbine rota ry. For dynamic simulation, a specific rota d to obtain variations in the tip speed ratio es the ratio of blade tip speed to the wind expression:

where: ω is the turbin rotational speed; R is the rotor radius of rotation; V is the freestream velocity. The power coefficient C_P, the moment coefficient C_M, and the tip speed ratio TSR are as follows:

$$C_P = C_M \cdot TSR$$

The inlet velocity is taken 7 m/s for all simulations, with Reynolds number of 1.2×10^5. The flow is assumed to be unsteady and turbulent, operating at a constant rotational rate for each case. Computational models are arranged in Gambit software, while numerical analysis is conducted with CFD Fluent. Figure 2 shows a two-dimensional model prepared in Gambit.

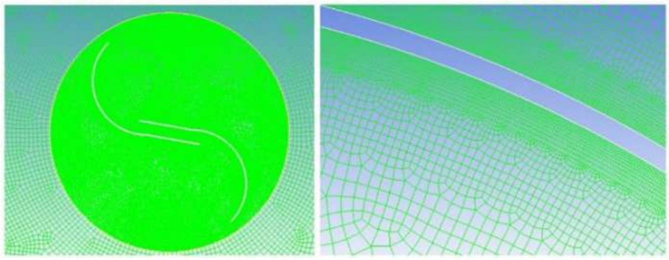


Figure 2: Mesh generation around the Savonius blade

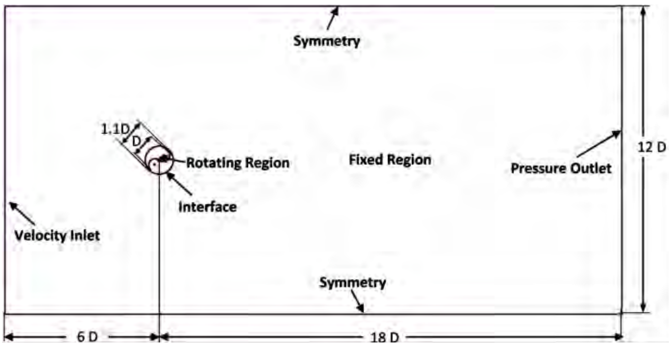


Figure 3: Boundary conditions and computational domain

The computational domain of rectangular shape shown in figure 3, dimensions are given based on turbine diameter. It is expected that the fluid flow passing through the turbine is not disturbed by the upper and lower sides of the domain. There are two regions in the computational domain: the rotating region of the turbine blades and the fixed areas on the outside, connected by the interfaces. The circle interfaces were made between the rotating and fixed region. The interface of the two mesh region boundaries have the same size, move against each other with no gaps to reduce calculation errors, and to achieve faster convergence.

The domain boundaries consist of the inlet velocity(the inlet side), the pressure outlet (the outlet side), and the top and bottom wall as symmetry. The no-slip boundary conditions are applied to the turbine blades. The domain is discretized using quadrilateral mesh. A size function was applicated with the rotor blade to obtain the better computational results near the blade surface, with start size parameter was chosen 0.1 mm, the growth rate of 1.1 and size limit was 1 mm. In order to accurately capture the flow behaviour around the blades, ten boundary layers of the structural mesh were generated. The number of cells for the blade is about 178,000.

For the simulation process used 2D double precision assuming transient flow, the turbulent flow of the fluid is modelled by the Transition k–ω Shear Stress Transport (SST) turbulence model with low Reynolds corrections. The turbulent intensity of 1% and turbulent length scale of 0.01 was applied to approximately account for the incoming flow turbulence. The cell zone condition for dynamic simulation used mesh motion on the rotating domain by input rotational velocity to get the variation of tip speed ratio from 0.2 to 1.4. The solution method includes; (1) the Pressure-Velocity coupling with Semi-Implicit Method for Pressure-Linked Equations (SIMPLE);(2) the spatial discretization for pressure with second order, for momentum, turbulent kinetic energy, and specific dissipation rate with the second order upwind schemeto achieve accurate results;and (3) the transient formulation with second order implicit. For calculation, the number of time step 0.001 with 20 iterations per time step is used. The simulation was considered to have converged when the residuals of all conserved variables are below 1 x 10^{-5}.

3. RESULT AND DISCUSSION

3.1 Validation

Validation of simulation was performed using experimental data obtained from SANDIA Laboratories [35]. To verify the simulation parameters used in this simulation, turbine geometry was simulated under the same experimental test [35]. The semi-circular blade contained a full 180 arc with a radius, r, of 0.25 m, the dimensionless gap width, s/d, of 0.1 and the radius of the rotor, R, of 0.4762 m (configuration No.11), with the test Reynolds number of 4.32 × 10^5. Two-dimensional transient simulations using the SST κ-ω turbulence model have performed at various TSR. The averaged value of moment coefficient (C_M) generated from simulations. Figure 4 shows the comparison of moment Coefficients as a function of Tip Speed Ratio (TSR) between the experimental data and numerical result. The numerical results indicate the conformity of the experimental tests, at low to medium rotational speeds, using these simulation parameters. However, it can be observed that the difference between numerical simulations and the experimental data grows with the increase of the TSR. At very high rotational speeds and wind velocities, the flow separation occurs, which is a limiting characteristic of k − ω model [36].

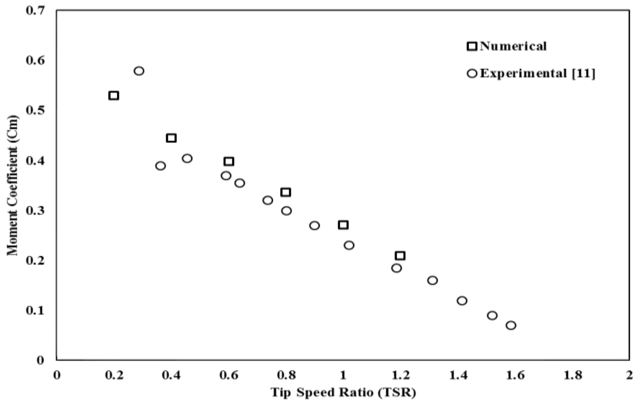
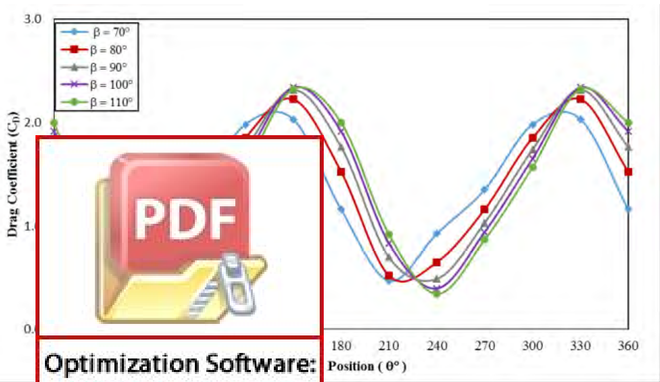


Figure 4: Moment Coefficient as a function of Tip Speed Ratio for numerical validation

3.2 Static Simulations

The study was conducted to determine the effect of modification straight blade angle by bending it inward and outward. Modified blade models with bend angle changes are shown in Figure 1. The diameter and distance between the blades of the five models are set equal to 40% of the chord length, and the bend angle varies from β = 70° to β = 110° with 10° intervals. Static and dynamic simulations are performed to observe turbine performance. Static simulations are performed to see the aerodynamic performance of the modified blades at various angular positions (0° - 150°, with interval 30°) to show some of the characteristic features of the flow around the turbine. For 180° to 360° positions are repeated (equal to 0°position) due to symmetrical blade shape, but there is a change of position between the advancing blade and the returning blade. Two parts of vortices produced at the advancing blade, one part is provided at the top of the tip of the advancing blade and moves downstream. The other part is built on the bottom of the tip of the advancing blade, which is divided into two ways: the first way runs along the concave side of the returning blade and the second way moves downstream. Thus there are two separation points. The vorticity influenced the average value of the drag coefficient and lift coefficient at different angle rotor positions.



at different angular position

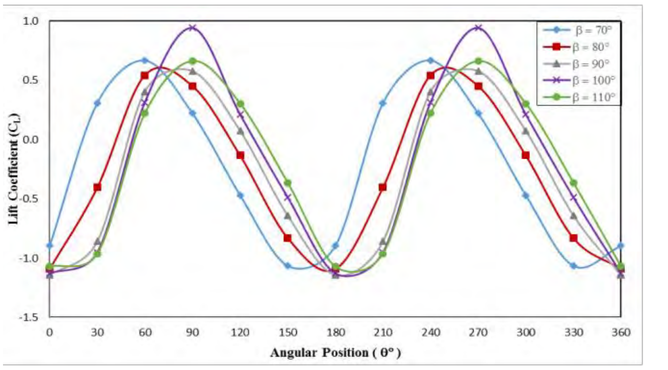


Figure 6: Lift coefficients at different angular position

Figure 5 shows that the drag coefficient (C_D) decreases from 0° to 60° angular positions and then gradually increases from 60° to 150° and then returns the same between the 180° position with the 0° position due to the symmetrical blade. A relatively high C_D value was obtained for a modified turbine at an angle bend position outward (β = 100° and 110°), compared to the other models. While Figure 6 shows the lift coefficient (C_L) value of the five-blade models. It is seen that the lift coefficient shows an increase from an angular position of 0° to 90°, and then decrease from 90° to 150°. This indicates that there are lift forces on the blade at the 60° and 90° positions. The negative lift coefficient occurs at 150° and 180° (same as 0° positions) as the acceleration of the returning blade moving into the flow creates a force in the clockwise direction. Thus the turbine does not operate purely with drag as the only contributing force, but in this case, lift force also contribute to increase power. The highest value of lift coefficient is obtained on bend angle β = 100° at a 90°and 270°positions.

3.3 Dynamic Simulations

Dynamic simulation is done by giving a specific rotational speed to the rotating zone. This study was made in the range of TSR = 0.2 to 1.4. The value of the moment coefficient (C_M) and the power coefficient (C_P) is averaged over the time interval to see the performance ratios among the various proposed models.

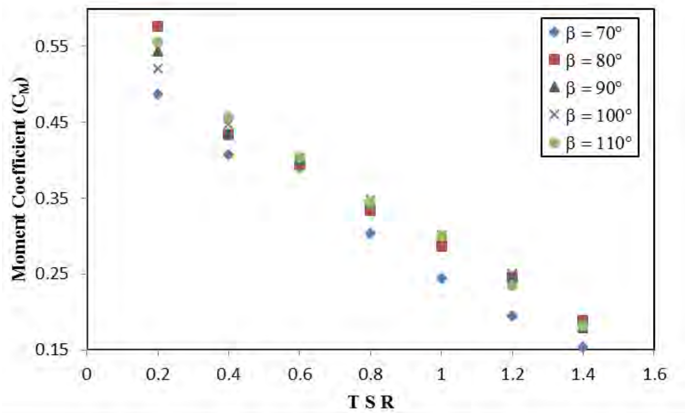


Figure 7: Moment coefficients at different TSRs

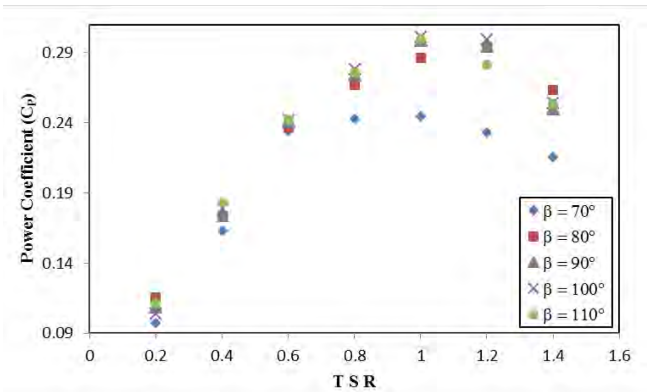


Figure 8: Power coefficients at different TSRs

Figure 7 and Figure 8 show the variation of moment coefficient and power coefficient for all models examined in this study. The average moment coefficient as a function of the tip speed ratio is presented in Figure 7. As it can be observed, the average value of the moment coefficient decreases approximately linearly as the tip speed ratio increases. The geometry design with β = 80° has the highest torque coefficient at lower tip speed ratios (TSR = 0.2), while at the higher tip speed ratios (TSR > 0.8) rotor with β = 100°, the torque coefficientis the highest. Figure 8 presents a comparison of C_P characteristics for all the geometries. It can be seen that there are maximum value for each curve and the power coefficient increases with TSR up to a certain point after which it drops down as TSR further increases. All geometries attain maximum C_P at TSR = 1. In that TSR, the model with β=100° geometry produces the highest power output.

The Savonius turbine is a device that utilizes the drag force to gain power, but its performance is low due to the negative moment on the returning blade. Savonius blade with Bach type, the negative effect is significantly reduced due to backflow through the overlap region and the flow acceleration leading to the returning blade side. When the straight blade is bent inward (β = 70° and β = 80°), the blade overlap distance becomes shorter than the baseline blade. On the other hand, the modification with the straight blade is bent outward (β = 100° and β = 110°), the blade overlap distance becomes longer than the baseline blade. With the overlap distance too short, the return flow through this area is shorter than the overlap distance on the baseline blade, resulting in reduced aerodynamic performance. However, too long overlap distance can also decrease its performance, since the backflow time becomes longer and does not produce a positive effect on the returning blade. So in this case, the blade model with β = 100°generates the optimal distance, capable of maximizing the return flow over the overlap area, and giving the highest average power coefficient, CP = 0.302. To explain the flow physics around the Savonius wind rotor, the velocity contours between advancing blade and returning blade are plotted in Figure 9.

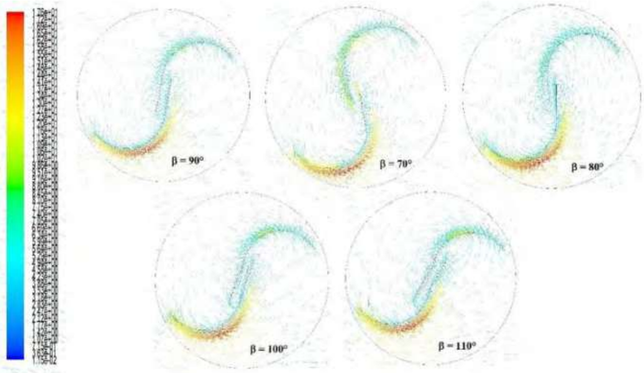


Figure 9: Velocity vector plots at various bend angle.

Figure 9 shows the difference of the velocity vector flow around the rotor of the five models tested: baseline (without bending or β = 90 °), bend inward (β = 70 ° and β = 80 °) and bend outward (β = 100 ° and β = 110 °). There are five flow conditions on the Savonius rotor: the free flow areas at the inlet, accelerating flow area at the rear edge of the returning blade, the overlap flow area between the blades, lifting flow on the convex of the advanced blade, and returning flow. The overlap flow area is the flow which provides a thrust effect on the advancing blades and the returning flow, where the stream is moved toward the concave side of the returning blade, helping to increase the torque of the rotor. With the same distance between blades and different at the overlap distance due to the effect of bending on the straight blade, the flow around the rotor causesthe differences in the overlap area between the blades. This is due to the flow acceleration effect of continuing the thrust on the advancing blades. Beside that, the flow which provides the thrust on the advancing blade, is lower on the bending blades and the baseline blades related to short overlap spacing. However, with higher overlap spacing due to larger bend angle (β = 110 °), it can contribute to cause a blockage effect so that the return flow towa ... ed, which causes its performance to decl ... design is obtained at the outward benc

4.

This ... estigation to simulate and verify the effec ... the performance of themodified bach-type ... re are five turbine models including base ... t the point of the curve and straight blad ... nwards (β=70° and β=80°), and two models bent outwards (β=100° and β=110°). Both the static and dynamic simulations were performed using 2D CFD Fluent. In the static simulation, the drag and lift coefficient on the Savonius turbine were directly calculated at every angular position. From static simulations, it has been observed that a relatively high CD value was obtained for a modified turbine at an angle bend position outward (β = 100° and 110°) and the highest value of lift coefficient is obtained on bend angle β = 100° at a 90° and 270° positions, compared to the other models. Furthermore, the dynamic two-dimensional simulation data have been collected at different tip speed ratio values, and the results show that the rotor with a straight blade angle of β = 100° has the highest coefficient of power, 0.302,for the tip speed ratio 1, which is 0.84% higher than a baseline model, and the rotor with a straight blade angle of β = 70° has the lowest coefficient of power, 0.244,for the tip speed ratio 1.

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The 8th Joint Conference on Renewable Energy and Nanotechnology (JCREN 2019)

Department of Mechanical Engineering
Engineering Faculty of Hasanuddin University
Secretariat: Mechanical Building, Kampus Unhas Gowa, Makassar Indonesia 92171

Makassar, Indonesia, October 28th, 2019

To:
Khairil Anwar, Syukri Himran, Luther Sule and Nasruddin Azis

Dear Authors,
On behalf of JCREN 2019 Committee, Department of Mechanical Engineering, Engineering Faculty of Hasanuddin University, we would like to inform you that we received your **FULL PAPER** to present as **Oral Presentation** at The 8th Joint Conference on Renewable Energy and Nanotechnology (JCREN) 2019 on 5-6 November 2019 at Faculty of Engineering Hasanuddin University, Makassar, Indonesia.

The title of your oral presentation is

Performance of Bach-type Savonius Wind Turbine with Modification of Straight Blade Angle

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If you need any help do not hesitate to contact us. Thank you very much.

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PERFORMANCE OF BACH-TYPE SAVONIUS WIND TURBINE WITH MODIFICATION OF STRAIGHT BLADE ANGLE

KEMENTERIAN RISET, TEKNOLOGI DAN PENDIDIKAN TINGGI UNIVERSITAS HASANUDDIN FAKULTAS TEKNIK DEPARTEMEN TEKNIK MESIN

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Performance of Bach-type Savonius Wind Turbine with Modification of Straight Blade Angle

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Abstract

This study aims to investigate the modification of Bach-type Savonius wind turbine on the straight blade section, by bending on the straight blade inward and outward, to obtain optimal performance. The Savonius wind turbine model was tested experimentally to predict the performance of the Savonius wind turbine in terms of the moment coefficient (CM) and power coefficient (CP) parameters concerning the tip speed ratio (TSR). The experiments are carried out at a certain wind speed by placing the savonius turbine models at the wind tunnel exit. The results obtained show that the blade model with the straight blade bent outwards 10° (ß=100°) has a better power coefficient than other tested models.

1. Introduction

The use of renewable energy has received close attention along with the depletion of energy resources, mainly from fossil energy. Among the various types of renewable energy available, wind energy is one of the best choices for producing energy that is economical and environmentally friendly. As one of the renewable energy sources, wind power is a sustainable form of energy, with considerable potential but not yet fully utilized. The utilization of wind energy is more accessible than other conventional power plants so that it is possible to make it on a small scale for the energy needs of rural and urban communities.

The wind turbine converts the wind energy into mechanical energy, which can be categorized into two types based on its rotational axis: horizontal axis wind turbine (HAWT) and vertical axis wind turbine (VAWT). One kind of VAWTs is the savonius wind turbine, proposed by Sigurd Johannes Savonius in 1922 that has a relatively simple construction [1]. This turbine is a drag type device, which consists of two or more semi-circular blades. In a drag-type turbine, the wind pushes against the blade, forcing the rotor to spin on its axis. Although the savonius wind turbine has a lower efficiency than other types of wind turbines, this turbine has advantages such as low manufacturing and operational costs, being able to receive wind from any direction with relatively low wind speeds, excellent start characteristics and produced relatively large torque. The savonius wind turbine is generally applied for electric power generation [2–3], public lighting system [4], power drive such as the irrigation pumps [5, 6], an element of mechanical or electromechanical systems [7] and micro-scale energy harvesters [8]. With these advantages, this turbine is widely studied and modified by many researchers to improve its performances.

...t studies on the savonius wind turbines developed with many ... provide notable performances. Published results show that Savonius ...haracteristics are influenced by shape and model of the blade [9,10], ...ters such as aspect ratio [11], overlap size [12,13], and gap size [14]. Some ...igating savonius rotor have performed studies on the number of the blades [15–18] and rotor stages [19, 20]. Several studies also have presented that wind guide and augmentation contribute to reducing negative torque in the returning blade section. [21–25].

Over the years, many different models and shapes of the Savonius rotor were investigated. The model and shape change of the rotor results in a difference in the performance curve with considering different parameters of the rotor and airflow. Dris et al. [26] have examined numerical simulation and experimental validation of the turbulent flow around a small incurved blade, that the incurved Savonius wind rotor affects the local characteristics. Also, some researchers have worked to enhance the Savonius turbine performance by changing the structure of the turbine that studied the aerodynamic characteristics of a modified Savonius turbine with helical blades [27–29].

Numerical investigation of modified and conventional Savonius wind turbines, with three configurations of Savonius wind turbines, namely Classical, Elliptical designs and Bach-type has been reported by Kacprzak et al. [30]. For the performance parameter CP, the Bach-type turbine is more excellent compared to the other tested models, and at the same time, the Elliptical type shows better performance characteristics than the Classical one. Modi and Fernando [31] has already drawn attention to the design parameters of the Bach-type, which are gap size, blade overlap, blade aspect ratio, and blade arc angle to increase the performances. In another study, Roy et al. [32] have made experiments of newly developed two-bladed Savonius-style wind turbine meant explicitly for a small-scale energy conversion, which demonstrates a gain of 34.8% in maximum CP with the newly developed two-bladed turbine.

The aforementioned findings demonstrate that the developments in the blade shape configuration of savonius wind turbines could improve the overall performance. Many innovative models improvements have been made on Bach-type savonius blade, but the simplest change is to modify the geometry of the turbine blade. The Bach-type blade has a blade structure that is not fully curved, but there are the straight shaped blades at the inside end in the overlap area [33]. Several previous studies conducted experiments on the side of the curved blade to examine the effect of the curved blade arc angle on the performance characteristics [34, 35]. While on the straight side of the blade, there has not been much investigation by the researchers. The straight blade part is also significant because it affects the movement of fluid flow along the surface of the curved blade towards the overlap area, which potentially decreases the negative torque effect of the returning blade side. The previous study by authors [36] has investigated the straight blade section, by bending the straight blade inward and outward at the joint of the curved blade and the straight blade which shows the influence of increasing turbine performance. However, this study was still done numerically to predict the bending effect at straight blade sections on the aerodynamic performances.

The present work, the experimental method is conducted to investigate the effect of the blade straight blade angles leading to find the optimal angle corresponding to the maximum moment coefficient (CM) and power coefficient (CP).

2. Method

The research was conducted experimentally to verify the effect of modifying straight blade angles by bending inward and outward. All blade rotor models were tested, where the blade model with overlap distance variation was model I: baseline blade (β = 90°), Model II or straight blade bent inward of 10° (β = 70°), Model III or [illegible] bent inward 20° (β = 80°), Model IV or straight blade bent outward of 10° [illegible] finally model V or straight blade is bent outward of 20° (β = 110°).

2.1 Blade model

The baseline model of the Bach-type savonius wind turbine refers to Roy and Saha [34], was built with the blade curve angle ϕ = 135°; the blade overlap, e = 40% of the blade chord distance; the gap between the blades, a = 10% of the blade chord distance; the straight blade length (S1) is the same with the blade radius (S2) of 42% of the blade chord distance. The baseline model is modified that the straight blade section is bent outward and inward to set an angle (ß) between of 70°-110° by 10° (the baseline model without being bent or ß = 90°). The fixed parameters are the turbine diameter (Di), turbine height (H), the blade curve angle (ϕ), the straight blade length (S1), the blade radius (S2) and the gap between the blades (a). (**Fig.1**).

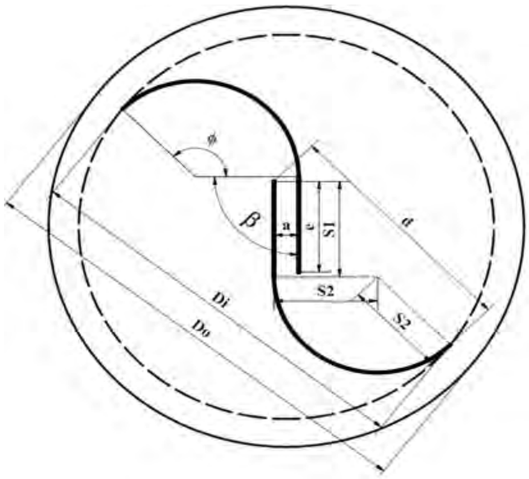


Fig. 1 Dimension parameters of the Bach-type savonius wind turbine.

Table 1 Dimensions of turbine models used in experiments.

Models	ß	ϕ	Di (mm)	H (mm)	a (mm)	e (mm)	a/e
I	90°	135°	195	195	11.4	44.7	0.26
II	70°	135°	195	195	11.4	6.6	1.73
III	80°	135°	195	195	11.4	28.7	0.40
IV	100°	135°	195	195	11.4	55.4	0.21
V	110°	135°	195	195	11.4	62.8	0.18

The specific dimensions of the modified design under the experiment are shown in the **Table. 1**. The turbine blades and endplates were made of acrylic materials, with the turbine diameter (Di) and height (H) of 0.195 m (aspect ratio 1 or D/H = 1) and thickness of the blades and end plates (t) of 3 mm. The endplates diameter (Do) is greater than the diameter of the rotor of 10% to obtain desirable performance as previously suggested by Blackwell et al. [16]. All turbine prototypes used in this test have the same rotor swept area so that all models get the same amount of wind energy. el modified by changing the bend angle is shown in **Fig. 2**.

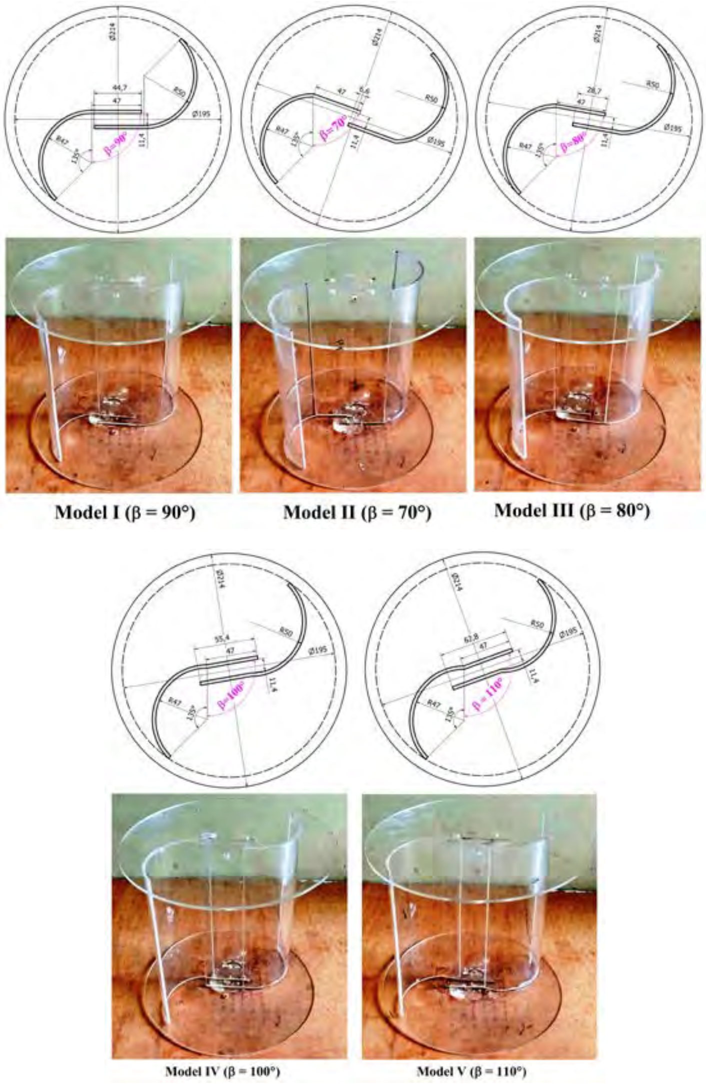


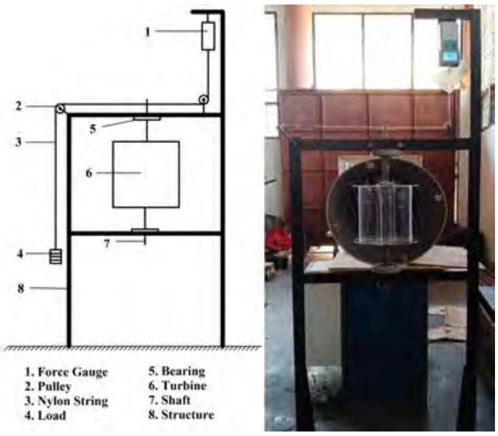
Fig. 2 Designs and models of modified savonius wind turbine.

2.2 Experimental Setup

Experimental set-up to conduct experiments on rotating savonius wind rotor is shown in **Fig. 3a**. This facility consisted of a wind tunnel and structural test rig housing the savonius model with measurement devices. The rotor system tested was placed 500 mm from the outside of the wind tunnel so that the center of the turbine was in line with the center of the wind tunnel's exit area [37]. The Savonius rotor was placed in the appropriate position using a steel frame structure. Two bearings mounted on the frame to support the Savonius rotor. The use of studs, nuts, and bolts in making frames facilitate the replacement of various tested geometries from the Savonious rotor and also help in determining the exact position of the rotor axis in the midline of the wind tunnel. ...ne then was connected with a mechanical torque measurement system, as ...b.

...periments were performed in the Open jet sub-sonic wind tunnel (Delta ...der open type test section [38]. Rotor savonius was supported by a shaft ...) of 5 mm, and a rope for measuring torque with a radius (r_n) of 0.5 mm. ... and load indicators used the ELK-50 digital force gauge instrument, with accuracy ±0.5%. The rotational speed of the rotor was measured by digital photo contact tachometer (Reed Instruments K4010) with an accuracy of ± (0.5%+1 digit) and operating range from 5 to 99,999 r/m. The wind velocity was measured by a hot wire anemometer (Lutron AM-4204) with a measurement range from 0.2–20 m/s at a resolution of 0.1 m/s. Bearings on the rotor shaft at the bottom and top were given grease for each initial data collection to avoid friction. Measurement of force gauge loads (M_2) was carried out on each addition of load mass (M_1) until the rotor shows a specific rotation (N).

(a)



(b)

Fig. 3 Experimental set up (a) Wind tunnel and structural test rig (b) Savonius model with mechanical torque measurement system.

Experiments carried out at an average wind speed (v) to match the expected Reynolds number [39]. The parameters calculated on each rotation are given by [40]:

$$T = (M_1 – M_2) (r_s + r_n) g / 1000 \quad (1)$$

where T is torque (N), M_1 is the load (g), M_2 is the gauge load (g), r_s is shaft radius (mm), r_n is string radius (mm) and g is acceleration of gravity (m/s^2).

$$C_M = 4T / (\rho \cdot v^2 \cdot D^2 \cdot H) \quad (2)$$

where C_M is the moment coefficient, ρ is the fluid density (kg/m^3), v is free stream velocity (m/s), D is the turbine diameter (m) and H is the turbine height (m).

$$\omega = 2 \pi \cdot N / 60 \quad (3)$$

ular velocity (rad/s) and N is the turbine rotation (rpm)

$$TSR = \omega \cdot D / 2 \cdot v \quad (4)$$

he Tip Speed Ratio

$$C_P = TSR \cdot C_M \quad (5)$$

moment coefficient

3. Results and Discussion

A graph displays the experimental results the torque with respect to the rotational speed of the rotor, the power coefficient (CP) and the moment coefficient (CM) against the tip speed ratio (TSR) to show the performance of savonius wind turbine.

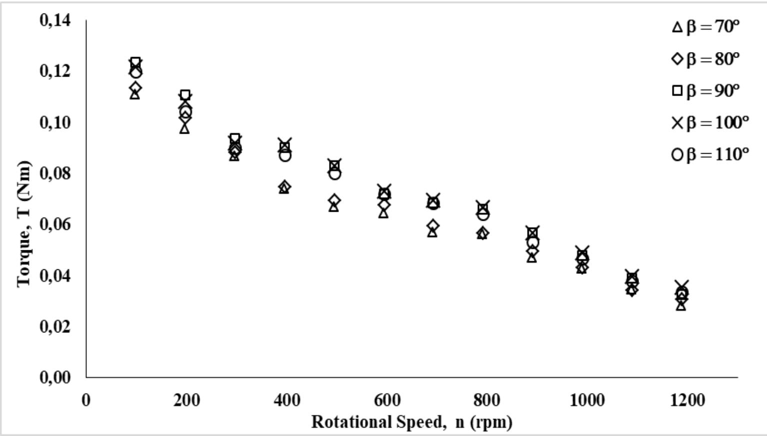
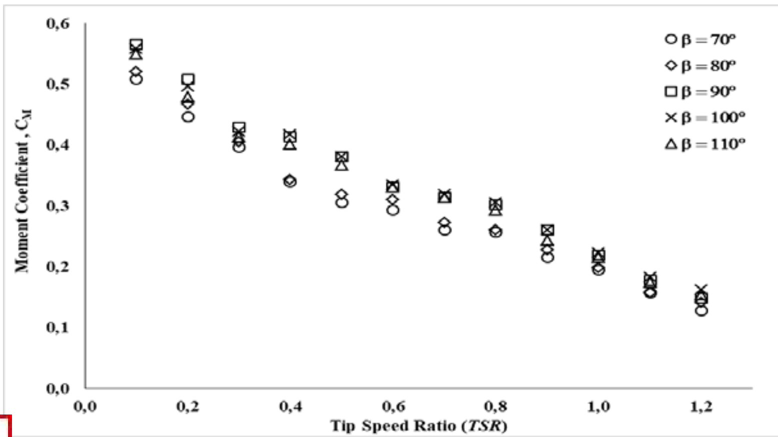


Fig. 4 The effects of straight blade angles on the torque with respect to the rotational speed.

Fig. 4 shows the effects of straight blade angles on the torque concerning the rotational speed where it is noticed that for all turbine models, the torque value decreases with increasing turbine rotation. Changes in torque value are obtained due to gradual loading where the rotational speed of the rotor increases when the load decreases at constant wind speeds. So, reducing the load causes the produced torque also decreases. From the five-turbine models tested, it can be observed that the Savonius wind rotor with straight blade angle β of 100 shows better torques than the other tested Savonius rotor models. Turbine model IV has the optimum distance at overlap area for generating the highest torques.



ffects of straight blade angles on the moment coefficient with respect to the tip speed ratio..

The effects of straight blade angles on the moment coefficient for the tip speed ratio are shown in **Fig. 5**. For all the turbine models, it may be seen that the value of the moment coefficient decrease with the increase of TSR due to the load applied to the rotor shaft leading to the rotational speed of the rotor decreases. From the five-turbine models, it can be identified that the higher moment coefficient (CM) curve is obtained on the turbine model ß = 100° of 0.305 at TSR = 0.8, while the lowest moment coefficient values are found in turbine model ß = 70° of 0.256 also at TSR = 0.8.

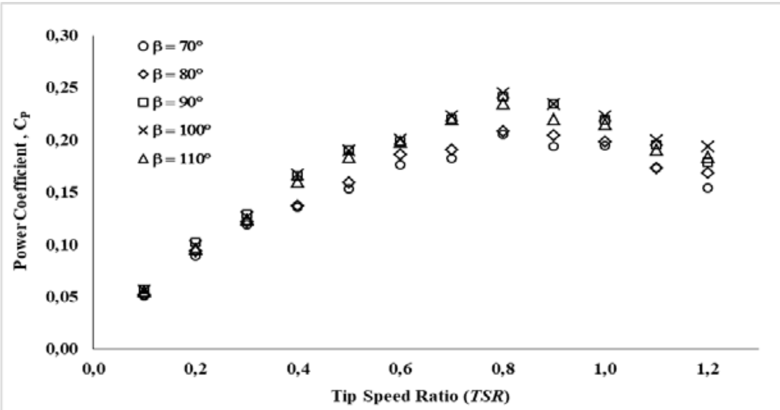


Fig. 6 The effects of straight blade angles on the power coefficient with respect to the tip speed ratio.

Fig. 6 shows the effects of straight blade angles on the power coefficient concerning the tip speed ratio. In all the cases, the coefficient of power increases with increasing TSR to a specific maximum value (0,8), then decreases after 0,8 TSR. From the tested five-turbine models, the turbine model IV (ß=100°) has the highest power coefficient as compared to the other tested models. The turbine model II (ß=70°) displayed the lowest power coefficient. The maximum power coefficient of 0.244 is obtained for the turbine model IV at TSR = 0.8. Whereas, the maximum power coefficient for model I (ß=90°), model V (ß=110°), model III (ß=80°) and model II (ß=70°) are found to be 0.241, 0.234, 0.208 and 0.205, respectively at TSR = 0.8.

The savonius wind rotor is a mechanism that applies the drag force to get power, but its work is low due to the negative effect on the returning blade. With the Bach-type blade, this negative moment is considerably reduced because of the flow acceleration through the overlap area, which allowed an extra force to the returning blade at the concave side. The five models have the same distance between blades, but they have a difference in the distance of overlap due to the bending effect on straight blade parts. It causes a flow difference around the rotor, especially in the area of advancing the blade and the area between the blades leading to the acceleration of the flow, which provides the additional force to the returning blade [36]. From the five tested turbine models, it can be analyzed that with a shorter overlap distance (in models ß=70°, ß=80°, and ß=90°), the backflow time through this area is shorter time and smaller acceleration which cause the reduced aerodynamic performance. However, if the overlap distance is too long (model ß=110°), it can also reduce its aerodynamic performance, because the backflow onger and does not produce a positive effect on the returning blade. With ficant overlap distance owing to the greater bending angle of straight the blockage effect so that the return flow towards the concave side is s case leads to a decrease in aerodynamic performances. Therefore, the ith a bend angle ß=100° produces the optimal distance in order to maximize backflow through the overlap area, which provides the best power coefficient compared to the other tested models.

4. Conclusion

This study experimentally examined the effect of modification of straight blade angles on the performance of the Bach-type savonius wind turbine. There are five tested models of rotors including the baseline model (without bending at the joint of the curved blades and straight blades or β = 90°), two models with straight blades were bent inward (β=70° and β=80°), and two models with straight blade bent outward (β=100° and β=110°). Experimental studies are carried out by placing the Savonius turbine model at the wind tunnel exit. The results obtained from five-turbine models show the turbine model IV (β=100°) presents the performance improvement as compared to other tested models, with Cpmax of 0.244 at TSR = 0.8. Whereas, the lowest is in the turbine II model (β=70°) with Cpmax of 0.205, also at TSR = 0.8.

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Static performance of bach-type savonius wind rotor with different straight blade angle

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Abstract. The purpose of this study is to examine the modification of the geometric shape of the Savonius Bach-type wind turbine blade to get optimal static performance. Static performance contributes to Savonius rotors to be able to start rotation. Modifications are made to the straight blade by bending the blade inward and outward, at the curved blade's connection and the straight blade. There are five rotor models tested including the baseline model (without bending or β = 90°), two models are bent inward (β = 70° and β = 80°), and two models are bent out (β = 100° and β = 110°). The study was conducted experimentally and static CFD simulations to obtain aerodynamic characteristics at several rotational angles of the rotor. For static simulations, the results showed a higher value of the drag coefficient (C_D) obtained in the modification of the turbine with outward bending angles (β = 100° and 110°) at the rotor positions 150° and 330° compared to other models. The same results were obtained in static experimental testing for the value of the drag coefficient.

Keywords: Savonius Wind Turbine, Bach Type, Straight Blade Angle, Drag Coefficient, Flow Visualization.

1. Introduction

Savonius wind rotors are drag axis based vertical devices and are used as small scale renewable energy production. The simplicity of construction, low investment cost, well-starting capability, and independent of wind direction with low operating velocity are the superiority of this turbine. However, the efficiency is lower than other types of wind rotors because of the high negative torque produced by one of the blades. Due to the low efficiency of rotor Savonius, various studies have been carried out to increase it over the past several years. A large amount of study has been carried out to raise Savonius rotor performance by optimizing various aspects, but most studies concentrate on dynamic [illegible] the parameters of power and torque coefficients. These studies generally focus on [illegible]ade profile [1], design parameters such as overlap ratio [2,3], aspect ratio [4], gap [illegible] curvature [6]. Some researchers have examined the effects of rotor stages [7,8], the [illegible]9–11], and end plates [12,13]. Several studies have also shown that augmentation [illegible] influence the reduction of negative torque on the returning blade [14–18].

Savonius rotor, as an energy extractor with better static performance, will be easy to start its rotation at relatively lower wind velocity. Rotation starts when the force produced by the airflow exceeds the friction force between the shaft and the rotor, where the principal aerodynamic forces acting on the turbine are the drag force (F_D) and the lift force (F_L). Drag and lift are relative wind velocity functions on the area of the turbine blades. For Savonius rotors, the drag force is the primary moving force, but the lift force as well contributes to producing power at certain position [19].

Some studies have been carried out relating to static performance with drag and lift characteristic rotor parameters. A study of the drag and lift characteristics can result in improved efficiency. Chauvin and Benghrib [20] examined the effects of drag and lift coefficient on Semicircular Savonius rotor based on the pressure difference between the lower and upper surface of the rotor. Gavalda et al. [21] obtained the value of the maximum drag coefficient, $C_D = 1.6$ for $\theta = 0°$ and the minimum value, $C_D = 0.65$ at the relative position $75 < 0 < 90$. Irabu and Roy [22] experimentally obtained the highest drag coefficient for the conventional rotors to be 1.56 with $\alpha = 90°$ and 270° at the overlap ratio of 0.0. It was also obtained that with the raise of overlap ratio, the drag coefficient was diminished. The highest and lowest lift coefficient is about 0.6 and 0.2 at the overlap ratio of 0.0 and 1.0, respectively. Jaohindy et al. [23] examined the effects of drag and lift coefficient numerically on the conventional blade, where the highest drag coefficient was obtained about 2.2 at $\alpha = 60°$ -70° and at the tip speed ratio of 0.6. The highest lift coefficient is about 1.72 at $\alpha = 30^{o}$ and 210° at the same tip speed ratio. Roy and Ducoin [24] conducted a two-dimensional numerical approach to predict the drag and lift coefficient for the new blade geometry. It was presented that the maximum drag coefficient was 2.24 at $\alpha = 80°$. The average and maximum lift coefficient were obtained about 1.19 and 2.07 at the tip speed ratio of 0.6. From some of the studies mentioned above, it is still limited mainly to the conventional blade. Therefore, Alom [25] examines the drag coefficient and lift coefficient for the conventional, Benesh, and modified Bach-type blades. The study indicates that the drag coefficient for the modified Bach geometry is greater than the Benesh geometry. The results are compared with the conventional blade under the same conditions.

Bach-type Savonius turbine has a blade structure that is not fully curved, but there are straight-shaped blades on the inside in the overlapping area. Several previous studies conducted experiments on the curved blade's sides to study the influence of the blade's angle on the performance characteristics [19,26]. While on the straight side of the blade, there have not been many investigations by researchers. This straight blade section can affect the movement of fluid flow along the curved blade surface towards the overlap area, potentially reducing the negative torque effect from the returning blade side. Anwar et al. [27] have investigated the straight blades by bending inward and outward straight blades at the connection of curved and straight blades, which shows the effect of increasing turbine performance. However, that is only done numerically to predict aerodynamic performance against the effects of bending on the straight blade. There is no detailed explanation of the instantaneous flow field around the blade due to the rotational angle of the rotor. Further flow visualization analysis is needed to investigate the power-producing mechanism because it is closely associated with the power and torque performance of the Savonius rotor.

This study investigates numerically and experimentally the static performances of the straight blade angle modification of Bach-type Savonius wind turbine, to find the optimal angle based on the drag coefficient (CD) and obtain visualization of the flow over the blade at several rotational angles of the rotor.

2. Method

sted for static performance is the modification of a straight blade's bend angle with between the blades (variation of the overlap distance). The diameters and overlap e models are determined to be the same, that is, 40% of the length of the chord and ies from $\beta = 70°$ to $\beta = 110°$ at intervals of 10°. CFD simulations and static were carried out to obtain the aerodynamic performance of Bach-type blade ous angular positions (0°, 30°, 60°, 90°, 120°, and 150°), with the drag coefficient parameter (CD). For 180° to 360° angular positions will be repeated or equal to angular positions 0 to 150° because they are related to the symmetrical blade shape, but only switch positions between the returning and advancing blade. The rotational angles of the rotor are arranged such that the curved blades to the five-blade models remain parallel. The rotational angle position of the rotors of the five Savonius turbine models tested is shown in figure 1.

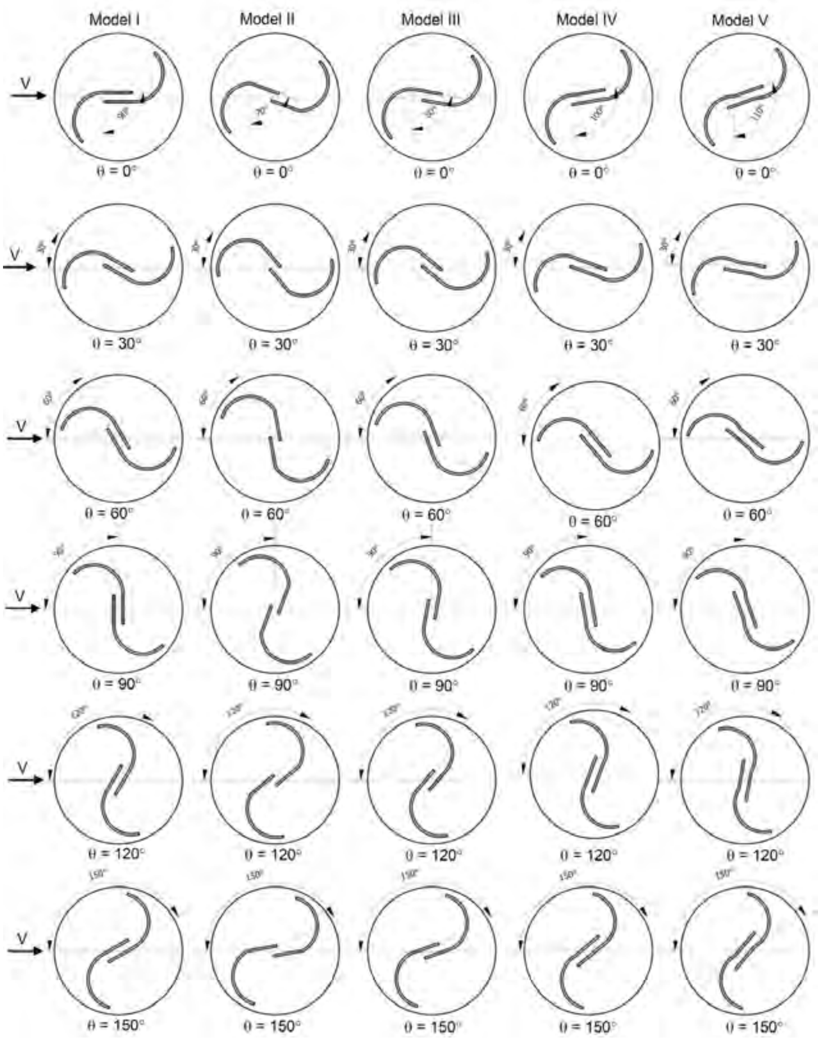


Figure 1. Rotational angle position (θ) of five blade models.

2.1. CFD Simulation

The computational domain is discretized using quadrilateral mesh (figure 2). The computational domain must be large enough that its boundaries are not too close to the Savonius rotor. To reach the optimal domain, it must meet the accuracy requirements and to avoid increasing cell numbers and computational time [28]. The computational domain size is taken as 24D × 12D [19] (figure 3). The size function is given with the start size of 0.08 mm, a growth rate of 1.1, and a size limit of 3 mm to ional results and take the boundary layer created near the blade surface. Airflow in e blade model is regarded to be turbulent, and the effect of molecular viscosity is le.

nditions on the upper and lower walls are symmetry, the inlet is determined as the he outlet opens with pressure outlets at atmospheric pressure. The rotor blades are shear conditions and the boundary condition as wall. The level of turbulence intensity is set at 5% for the inlet and outlet boundary. The standard k-ε turbulence model at FLUENT is used to simulate computational flow around the Bach-type Savonius rotor model with changes in the straight blade angle. Pressure-velocity coupling is achieved using the SIMPLE (Semi-Implicit Method for Pressure-Linked Equations) method. Discretization for the momentum equation's solution uses the second-order scheme for pressure, while for the momentum, turbulent kinetic energy (k) and turbulent dissipation rate (ε), second-order upwind is chosen. The standard k-ε turbulence model is a semi-empirical model based on the transport model equation for the kinetic energy of turbulence (k) and its dissipation rate (ε).

The iterative convergence solutions is attained if the quantity of the absolute differences of the solution variables between two consecutive iterations is below the pre-determined small amount, which is chosen at 1×10^{-5} in this simulation. All blade models using the k-ε turbulence model convergence principle are set and tested for continuity velocity, x velocity, y velocity, kinetic energy (k), and turbulent dissipation rate (ε). The drag coefficient is determined from the simulation-based on drag force, the surface area of the rotor, fluid velocity, and density.

Figure 2. Mesh generation around the Savonius rotor.

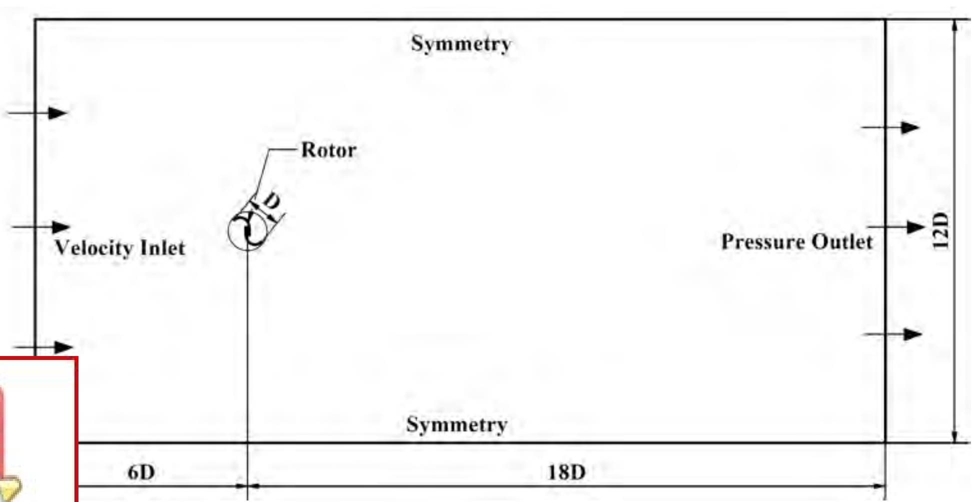


Figure 3. Computational domain and boundary conditions.

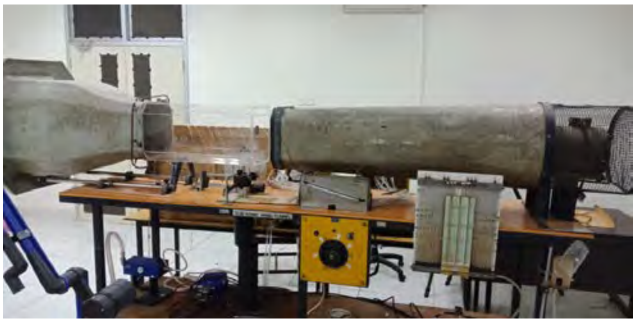
2.2. Experimental setup

Experimental testing of static Savonius wind rotor performance to get the value of the drag coefficient (C_D) is done by placing a Savonius wind rotor model (figure 4) in the ARMFIELD model C2 subsonic wind tunnel test section with a motor rating of 1SkW (figure 5). Savonius turbine models are tested statically at various rotational angles of the rotor θ: 0°, 30°, 60°, 90°, 120°, and 150°. The drag force measurement instrument uses the load cell that calibrated by applying a digital balance, with the statistical uncertainty of the predicted force measurement around ± 2%. The drag coefficient due to drag force working on the rotor models is formulated as:

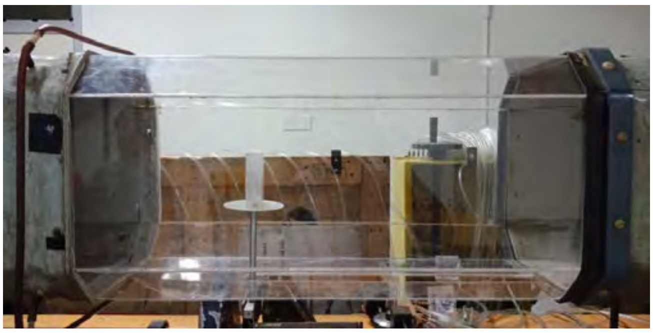
$$C_D = \frac{F_D}{\frac{1}{2} \cdot \rho \cdot A \cdot V^2} \tag{1}$$

Where, F_D is the total drag force working on Savonius rotor blade as measured by the load cell (N), ρ is the density of the air (kg/m^3), A is the frontal area (the area projected on the rotor normal to the direction of flow), and V is the free-stream wind velocity (m/s).

Figure 4. Savonius turbine model for drag coefficient testing.



(a)



(b)

Figure 5. a) ARMFIELD Wind tunnel; b) Test Section.

around the Savonius turbine blade is visualized using vertical suction-type wind & PARTNER LTD ENGINEERS, with the smoke-wire device (figure 6). The model of the Savonius model is fixed at the wind tunnel center. The dimensions of the rotor model used are the same as the drag coefficient test by considering the capacity of the flow visualization test apparatus. The blade of the rotor test model is painted with black color while the upper endplate is made of transparent acrylic material in order that the flow pattern can be seen clearly along the blade surface (figure 7). The visualization of the flow around the blade is tested in a stationary or static state at various rotational angles: 0°, 30°, 60°, 90°, 120°, and 150°.

Figure 6. Flow visualization test equipment.

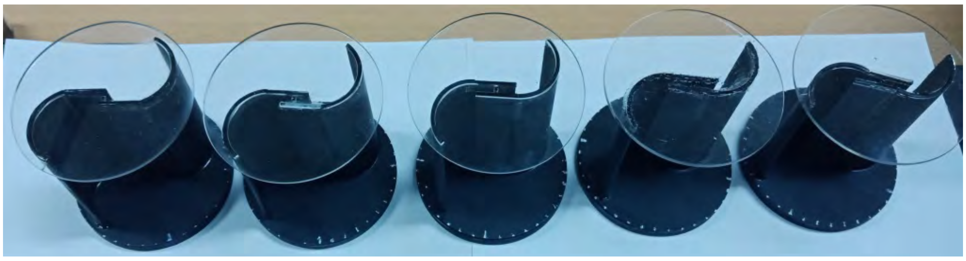


Figure 7. Savonius turbine model for flow visualization testing.

3. Result and Discussion

3.1. Drag Coefficient

Drag force is usually described as the force that is parallel to the direction of the fluid flow. It is caused by two viscous friction forces on the surface of the rotor and unequal pressure on the surface of the rotor facing and away from the approaching flow. For the Savonius rotors, the drag force is a function of the relative wind velocity on the surface area of the rotor. The drag coefficient (CD) indicates the size of the fluid resistance received by Savonius rotors. The drag coefficient produces nius rotor model. The static performance testing of Savonius wind turbines was entally by placing a Savonius wind turbine prototype in the test section of wind he results obtained in the CFD simulation, in this case, the drag coefficient value. on the straight blade angle produce different aerodynamic characteristics respect to of the rotor. Fig. 8 and 9 shows the drag coefficient at the rotational angle position ent from the five rotor models tested.

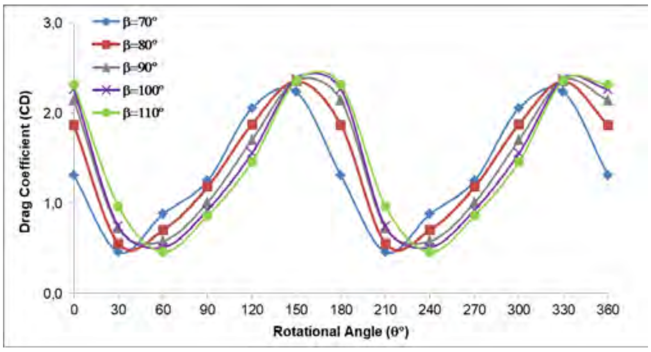


Figure 8. CFD simulation result of drag coefficient to the rotor angle position.

Figure 8 presents CFD simulation result of the drag coefficient of the static Savonius rotor as a function of the relative position of the rotor. It can be observed that the drag coefficient value varies depending on the rotor position, where the rotor is at different projection positions in the reception force of the fluid. For model β=90°, β=100° and β=110, drag coefficient decreases when the rotor is rotated from position 0° to 60°. For model β=70° and β=80°, it decreases from position 0° to 30° and then increases gradually from position 60° to 150° and then returns to the same position between 180° and 360° due to the symmetrical blade shape. For all blades tested, the highest drag coefficient value is obtained at the rotational angle of 150°. Whereas, compared to the five rotor models, a relatively high drag coefficient value was found for the turbine that was modified in the outward bending position or β = 100°.

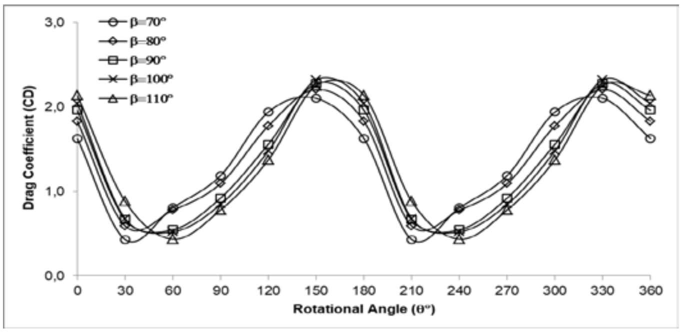
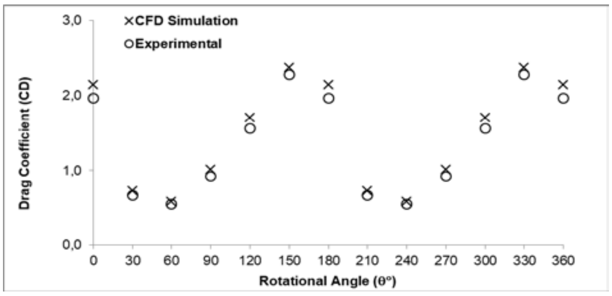
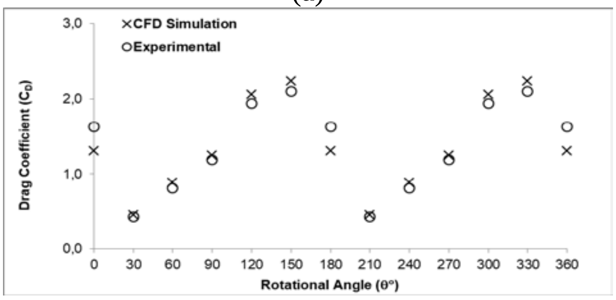


Figure 9. The experimental result of drag coefficient to the rotor angle position.

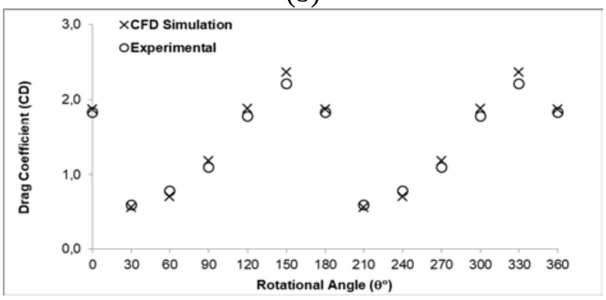
Figure 9 shows the experimental result of the drag coefficient with the change in the rotational angle of the rotor (θ) for the five Savonius wind turbine models. It can be seen that the experimental results show conformity with the CFD simulation results. For models II (β = 70°) and model III (β = 80°) the value of the drag coefficient decreases from the rotational angle of the rotor 0° to 30°, while the other three models (model β = 90° - 110°), the value of the drag coefficient decreases with increasing rotor angle from 0° to 60° and then increases with increasing rotor angle to the highest position of 150°. Furthermore, the same drag coefficient pattern repeats in the next half rotation, from 180° to 360° because of the symmetrical shape of the blade (only the advancing and returning blade nged). The maximum drag coefficient of 2.32 is obtained for the turbine model IV s, the maximum drag coefficient for model I, model V, model III, and model II are 25, 2.21, and 2.10, respectively, at θ = 150°. A comparison of the drag coefficient erimentally with CFD simulations can be seen in figures 10. The results obtained are in the same trend with numerical simulation results, where the maximum C_D otor angle position 150°.



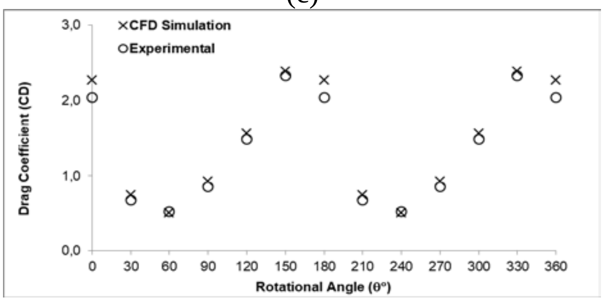
(a)



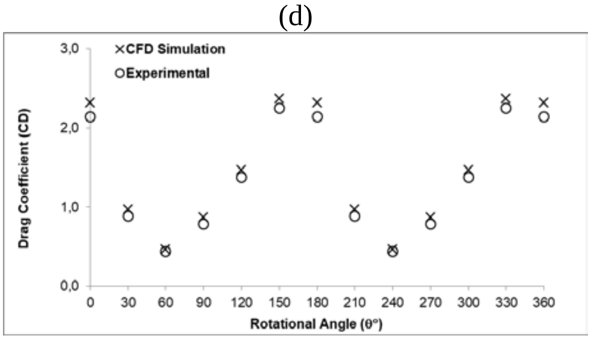
(b)



(c)



(d)



(e)

re 10. Comparison of CD between experiment and CFD simulations, model I (β = 90°), (b) model II (β = 70°), (c) model III (β = 80°), (d) model IV (β = 100°), (e) model V (β = 110°).

3.2. *Velocity and Pressure Contour*

The velocity vector and static pressure contour were obtained from CFD simulations to show a prediction of a flow field passing through the rotor blade of Savonius at several rotational angles of the rotor.

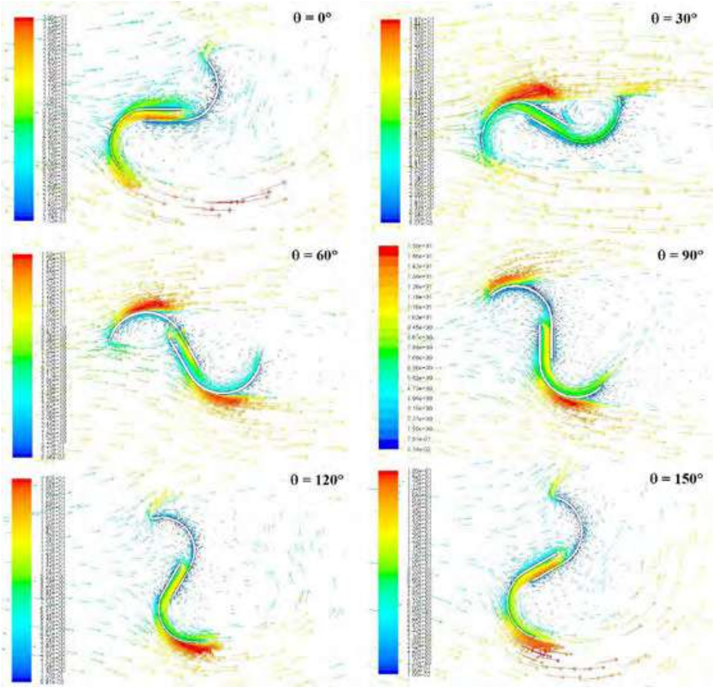


Figure 11. Velocity vector to the position of the rotor angle of the baseline model (model I, β = 90 °).

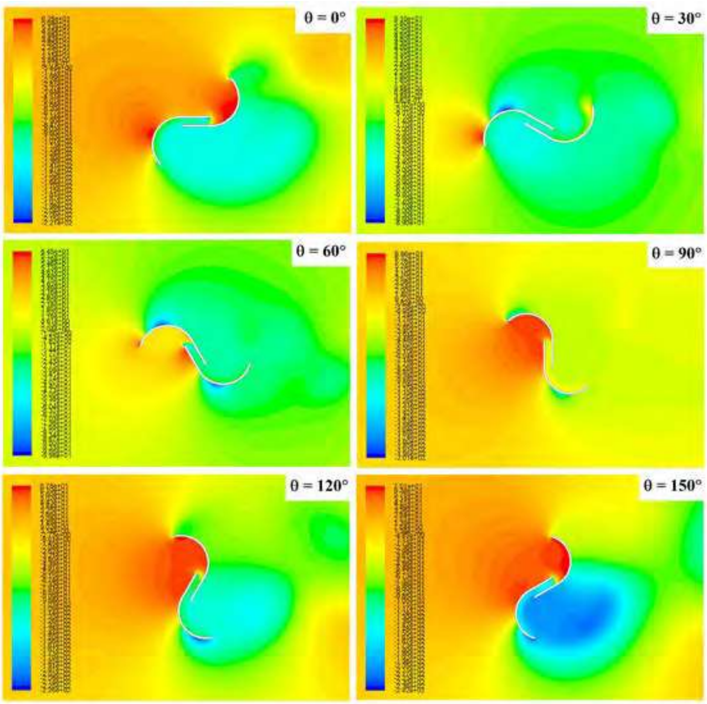


Figure 12. Static pressure at various rotational angle rotations of the baseline model (model β = 90°).

Figure 11 shows the velocity vector of the baseline blade model at the various rotational angle of the rotor. It can be seen that there are two parts of the vortex produced at the advancing blade on the convex side. A section is produced on the upper side of the tip of the advancing blade then moves towards the downstream, where the vortex rotates clockwise. This condition increases the suction pressure at the advancing blade on the convex side and causes a decrease in torque.

Figure 12 shows a static pressure contour plot for the baseline model at the rotor angle positions 0°, 30°, 60°, 90°, 120°, and 150°. For all rotational angles of the rotor, the pressure contour indicates a static pressure decrease from the upstream to the downstream side across the rotor. Pressure and velocity drop from the upstream to the downstream side of the advancing blade, which generates overall lift for the turbine. It can be seen that the maximum change in static pressure occurs from the rotor angle position 90° to 150° and decreases at the rotor angle position 0° to 60°. The advancing blade position at the 90° to 150° angle towards the direction of flow contributes to the maximum drag and lift force due to a drastic change in static pressure on both sides of the Savonius rotor blade. The same condition also occurs in other models. The velocity vector and static pressure contours for the five-blade models tested at positions 90° and 150° are shown in figure 13 and figure 14.

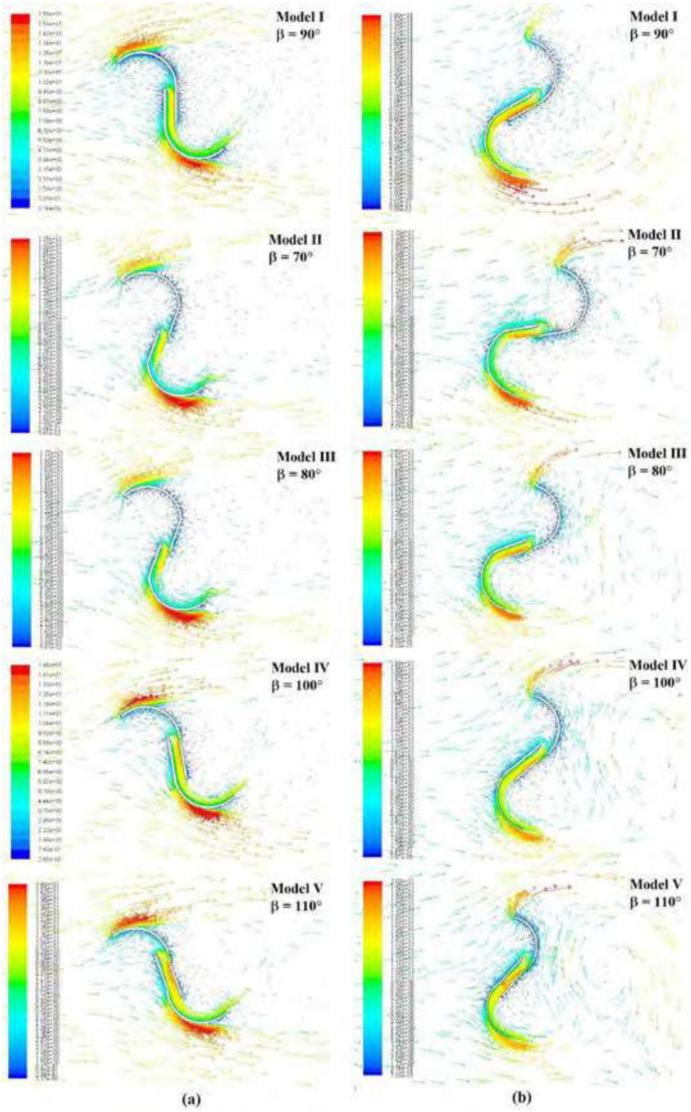


Figure 13. Velocity vector of the five models at rotational angle, θ: (a) 90°, (b) 150°.

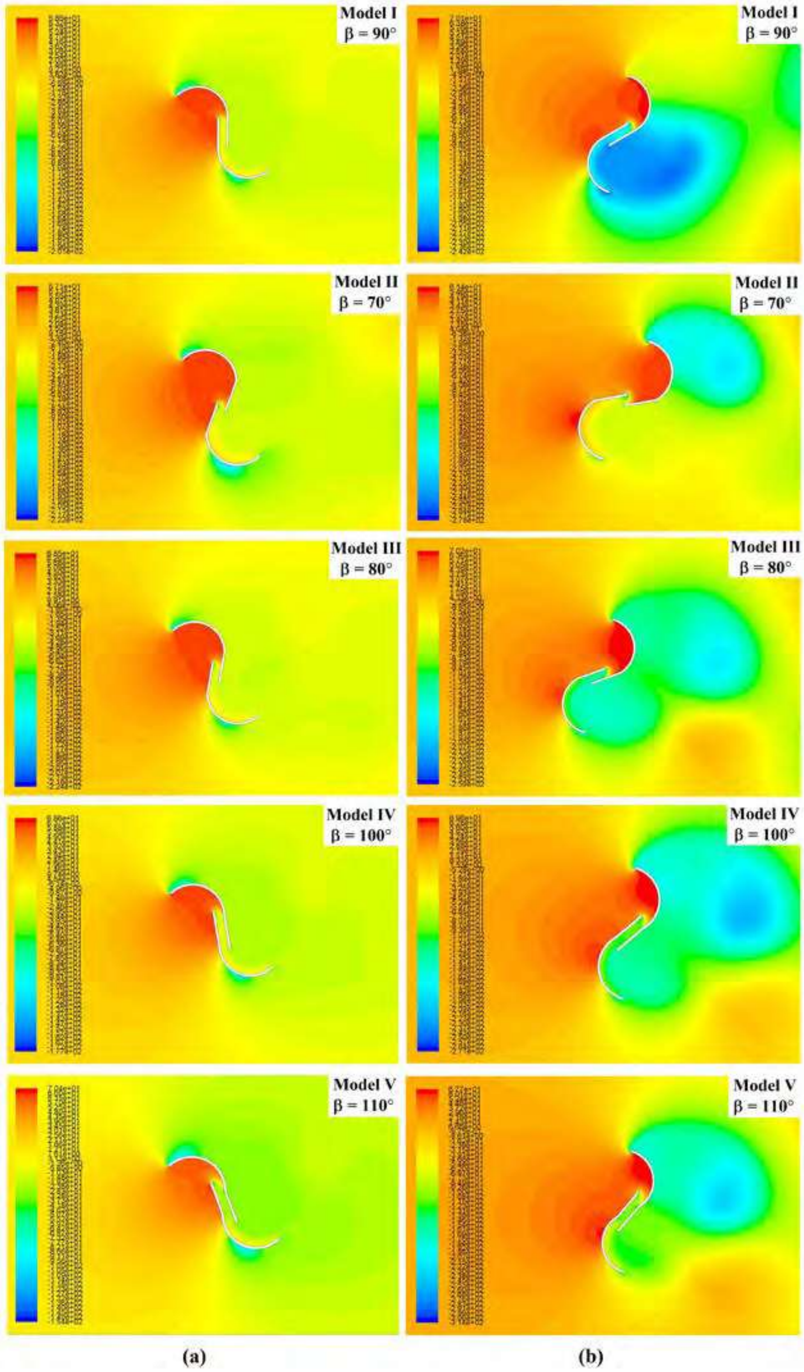


Figure 14. Pressure contours of the five models at rotational angle, θ: (a) 90°, (b) 150°.

At the rotational angle position of the rotor 90°, fluid flow flows through the convex side of the ... d the other half on the upper side of the tip of the advancing blade, which then ... back. It is seen in all blade models with almost the same pressure contours. ... seen that for model IV and model V have more significant pressure differences ... del I, model II, and model III. As seen, the advancing blade produces power, while ... reduces power output. However, in this position, it can be observed that the ... tion to the turbine power production is due to suction pressure at the advancing blade on the convex side, while only a small contribution of the work is done by increasing the pressure on the concave side. It is almost the same as the lift force produced on a standard airfoil, where the lift's dominant contribution is obtained from the airfoil at the suction side. It clearly shows that a very significant part of the turbine's power is resulted through negative pressure on the backside of the advancing blade, rather than increasing pressure on the wind. Thus the most significant contribution to power production from turbines in this position is obtained from the lift force of the advancing blade. It indicates that this type of turbine is not just a device that utilizes the drag force, as is often assumed.

At the rotational angle of the rotor 150°, the pressure contour shows the most significant decrease in static pressure from the front side to the back side across the rotor. It is especially seen in models IV and model V compared to models I, model II, and model III, where negative pressures are more significant and evenly distributed in the downstream region of the rotor. The significant pressure difference contributes much to the drag force produced compared to the position of the other rotor angles.

3.3. Flow Visualization

Flow visualization is essential for analyzing the instantaneous flow field over the Savonius blade and explaining the mechanism of the power generation. The flow pattern on the blade moved forward, which showed the contribution of lift to the rotor power mechanism [29–31]. Visualization of the flow around the Savonius blade in a static state is reviewed at various rotational angles. Characteristics of the flow fields in and along the blade are shown in figure 15.

Figure 15.a shows the flow visualization around the blade at a rotational angle of 0°. It is shown that the advancing blade position is not facing the direction of the flow, but there is a fluid flow that moves to the returning blade and partly turns into the overlap area, especially in the model β = 80°-110° model, which causes a significant drag coefficient to rotate the rotor. While at a rotation angle of 30° (figure 15b), the position of the rotor has not received significant fluid flow to provide thrust, both on the returning and the advancing blade (because the advancing blade blocked the flow). So in this position, the lowest drag coefficient is generated. However, model IV and model V are still better than the other three models in contributing the force to the rotor. At the rotational angle of the rotor of 60° (figure 15.c), the inside surface of the advancing blade has begun to open with fluid flow, but it has not yet fully contacted the blade surface so that at this rotational angle, the resulting drag coefficient is still relatively low.

At the 90° - 150° rotational angle position (figure 15.d – 15.f), significant stagnation pressures begin, especially at the advancing blade curvature (the rotor is in the broader projection position), which drives the rotor to spin in a positive direction (clockwise). On the other hand, there is also a slight pressure on the outer side of the returning blade on most surfaces that pushes the rotor in a negative direction (counterclockwise). However, the area is much smaller than the advancing blade side. Therefore the negative force on the returning blade is considered small compared to the positive force on the advancing blade at most rotor position, indicating clockwise rotation of the rotor. From figure 15.d – 15.f, it can be seen that the flow visualization of all the rotor model shows the flow passing through the advancing blade then turning towards the overlap area, wherein model II and model III, the fluid flow time is shorter than the model I, IV and V, so that the flow acceleration is much lower. While in models I, IV, and V, the flow through the overlapping region is more extended, allowing higher flow acceleration in this area. Especially in models IV and V, because of the influence of bending on the straight part, its position gives more advantage in directing the fluid towards the ver, for the 150° rotational angle position, in model V, part of the flow is blocked straight blade from the returning blade.

zation indicates that the flow above the Savonius blade is separated along the two hich is projected from the upstream direction. Therefore, the flow pattern varies e, according to the rotor projection's width. It is thought that the static rotor power y due to the different drag forces acting on the two blades.

Figure 15. Visualization of flow around the rotor at the rotational angle, θ: (a) 0°, (b) 30°, (c) 60°, (d) 90° (e) 120° , and (f) 150°.

shorter overlap distances, the time of backflow through this area will be shorter, causing aerodynamic performance to decrease. However, overlapping distances an also reduce its performance because the backflow time is longer and does not impact on the returning blade. So the rotor prototype with bend angle $\beta = 100°$ l distance, which can maximize backflow through overlap regions, providing better compared to the other models.

4. Conclusions

In this paper the drag coefficient and flow visualization of modified Bach-type Savonius wind turbine are analyzed. The result obtained show that the lowest drag coefficient is obtained at the rotational angle position 30° for model II and model III, while for model I, model IV and model V, the lowest drag coefficient occurs at the rotating angle position 60°. For all blades tested, the highest drag coefficient value is obtained at the rotational angle of 150°. A relatively high value of the drag coefficient (CD) is obtained in the modification of the turbine model IV and model V, or with an outward bending angle (β = 100° and 110°) at the rotor position 150°. The results found in the experiment are following the numerical simulations, where the maximum drag coefficient value occurs at the rotor angle position of 150°. Flow visualization provides an overview of the flow field around the Savonius wind rotor based on numerical and experimental results.

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Jabatan / Golongan : Lektor / IIIc

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