Experimental and Numerical Investigation of I-65° Type Cylinder Effect on the Savonius Wind Turbine Performance

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Abstract— In this study, a circular cylinder cut by 65 degrees on both sides was placed aligned with the axis returning turbine vane of a Savonius rotor turbine. This type of cylinder is called an I-65° type cylinder and is designed to bring down the drag forces on the returning blade, aerodynamically. Wind turbine performance was investigated experimentally and numerically under two conditions, with and without the installation of a type I-65° cylinder in line with a horizontal axis returning turbine blade. The study was conducted with Reynolds number Re = 9.9 x 10⁴ based on free stream velocity (U) 5 m/s and a characteristic length L = 2D-b, where D is the outer diameter of the vane and b is the diameter of the rod. The I-65° type cylinder, which has a diameter of 0.5D, is placed at a distance of 1.4D in front of the returning turbine blade. In numerical studies, a 3D simulation was conducted to analyze the Savonius turbine flow using Commercial CFD software, Ansys Fluent version 19.1. The experimental output shows that installing an I-65° type cylinder in front of the returning blade can increase the Cp of the Savonius turbine. Compared to the conventional turbine, the maximum power coefficient of the Savonius turbines increased to around 23.6% due to the installation of an I-65° type cylinder, and this was achieved at a tip-speed ratio (TSR) of 0.8. This result is justified by numerical results carried out in this study.

Index Term— Savonius turbine; I-65° type cylinder; returning blade; performance; transient; sliding mesh.

I. INTRODUCTION

In order to participate in efforts to increase new and renewable energy sources, many studies have conducted that utilize environmentally friendly sources of energy such as wind energy, ocean energy, biomass, solar energy, and geothermal energy. Wind energy is one of the abundant and easily found energy sources, many studies have conducted that utilize wind energy such as solar energy, biomass, and geothermal energy. Wind energy is one of the popular wind turbines Morshed K.N. [1].

![Fig. 1. Power coefficient function Tip-Speed Ratio (λ) for various wind turbines Morshed K.N. [1].](image-url)
The experimental method for reducing the main cylinder drag by installed I-type bluff body cut performed by [2]. The I-type bluff body cut off in several scission angles between $0^\circ \leq \theta_s \leq 65^\circ$ were installed in upstream and aligned at the line axis of the main cylinder at a distance $S/d = 1.375$. The result presented that optimum drag reduction is obtained in the bluff body within cutting angle $65^\circ$, Reynolds number $Re = 5.3 \times 10^4$, which is the primary cylinder drag decreased up to 52%. Investigation of the Installation D-shape and I-shape cylinder in front of the primary cylinder circular conducted by [3]. The ranges of cutting angles for both shape cylinders between $50^\circ-53^\circ$ and the Reynolds number $Re > 2.3 \times 10^4$. The output shows that the wake behind the main cylinder becomes narrowed, and vortex arises downstream. The experimental investigation of the bluff body cut related to the flow characteristics from the circular cylinder reported by [4]. The bluff body volume reduction is identical to $d/2(1 - \cos \theta_s)$, where $d$ and $\theta_s$ defined as the diameter and the position angles, respectively. Where $\theta_s$ are stretched in the interval $0^\circ \leq \theta_s \leq 72.5^\circ$, and Reynolds Number $Re = 2.0 \times 10^4$ and $3.5 \times 10^4$. The result shows the optimum decrease in the drag coefficient $C_D$ is obtained at $\theta_s = 53^\circ$ when $Re > 2.5 \times 10^4$. Whereas for $Re=3.1\times10^4$ and $\theta_s > 60^\circ$, the coefficient drag $C_D$ is higher than the circular one for the D-type. The other reports by [5], experimentally put down the rod upstream circular cylinder with diameter $D = 40$ mm. The rod diameter varies from $1 \leq d \leq 10$ mm, the spaces from center to center between main cylinder and rod $50 < L < 120$ mm and Reynolds Number $Re = 1.5 \times 10^4 - 6.2 \times 10^4$. The result show the peak conditions of the pressure drag reductions are $d/D = 0.25$, $L/D = 1.75-2.0$. In these ways, the drag reduced up to 63% contrast with a normal cylinder. The experimental investigation corresponds to the installation of a small control rod in the direction of airflow or upstream the central cylinder executed by [6]. The control rod diameter $(d)$ varies between $0.133 \leq d/D \leq 0.267$, where $D$ is defined as the diameter of the main cylinder (30 mm). The coefficient of drag of the main cylinder dropped about 29% when a regulator rod with the diameter $d = 7$ mm $(d/D = 0.233)$ is mounted at a ratio of the pitch near to the significant value of $Lc/D = 2.081$. Concerning the study about the Savonius turbine, the scientific reports [7] investigated the installation of a curtain upstream of the Savonius rotor turbine in order to disappear returning blade negative moments within both numerical and experimental approaches. Performance investigation carried out with the use and unused curtain installation. The result shows that significantly increased in Savonius rotor turbine performance has been achieved by the installation of the curtain. The curtain varies in dimension involved of both side parts length $(l_1$ and $l_2$) and angles $(\alpha$ and $\beta$). The Power Coefficient $(C_p)$ increase up to 38.5% with optimal curtain dimension $l_1 = 45$ cm and $l_2 = 52$ cm and $a = 45^\circ$ and $\beta = 15^\circ$. The research reports [8] investigated experimentally in order to improve the final power parameters of the Savonius rotor turbine include the static moments, which calculated the turbine's self-starting capability. Both objectives achieved by the geometry skeleton line that is maximalized in the existence of the disturbance plate. The result shows that the turbine's coefficient of power climb up to 40% at the tip-speed ratio $TSR (\lambda) = 0.7$. The static torque as a benchmark of self-starting ability strengthened in the positive area at any angle, it means the turbine need external devices to rotate. The investigation in diverse Savonius geometries is conducted experimentally with the intention to decide the most successful operation parameters involved in power coefficients and static torque carry out by [9]. The results show that two blades rotor, rotor with endplates, second stages rotor, and rotor without overlap is more efficient than the others. The investigation prove that the aspect ratio directly proportional to the power coefficient. Static torque measurement of all types of rotors at different speeds verify the results of the investigation. The corresponding reports by [10] have studied numerically about the impact of the breadth of a unique curtain on the capability of the Savonious rotor turbine. They proved that the installing of the curtain in front of the returning blade of the turbine for improving the performance of the turbine is dependent on the width of the curtain and the Reynolds number. For the width of the large curtain of $S/D = 2.0$ at $Re = 9 \times 10^4$, the performance of the turbine is estimated lower than when the turbine without the curtain. In the same case, the scientific reports [11] have also investigated the problem experimentally. They also proved that the placement of the curtain plate in front of the convex blade for improving the performance of the turbine depends on the breadth of the curtain plate and the $Re$. Where for the wider curtain $(S/D>1.4)$ at $Re = 9 \times 10^4$, the coefficient of power turbine is less than when the turbine without the curtain plate. In this way, the $C_p$ of the turbine with the curtain plate may fall to 60.8% of the turbine without the curtain plate, for $S/D = 1.83$ at $Re = 9 \times 10^4$. The numerical investigation related to the Savonius wind turbine with Novel blade shapes developed from Myring Equation performed by [12]. The results show that the effect of blade fullness increased peak power coefficient 10.98% higher than conventional Savonius wind turbine obtained at tips speed ratio 0.8.

According to the above summarize, this study is conducted in objectives to improve the performance of the Savonious rotor turbine experimentally and numerically by mounting an I-65° type cylinder upstream convex of the turbine. The experimental is performed by decide the center to center distance between axes of returning blade and I-65° cylinder $S/D = 1.4$ and at Reynolds Number $Re = 9.9 \times 10^4$. The numerical computation is conducted by use commercial solver ANSYS-Fluent 19.1 in order to validate the experimental data and recognized the flow behavior at an adjacent turbine wall. According to Triyogi et al. [2], that an I-65° type cylinder gives the highest reduction in drag of the main cylinder behind it, due to the highest drag coefficient $(C_D)$ of the I-65° type cylinder. As it is known that the highest $C_D$ is generally due to the broader wake region behind the cylinder, so the shear layer of the I-65° type cylinder is most effective in affecting a bluff body in the downstream. Inspired by Triyogi et al. [2], the idea of using a type I-65° cylinder, as a passive control, mounted upstream of the convex blade is intended to reduce the drag force of the convex blade so that positive torque increases and ultimately increases Savonius turbine power.
II. METHODS

A. Experimental Arrangement

Figure 2a describes the Savonius wind turbine configuration with the installation I-65° type cylinder cut upstream returning blade, where this I-65° type cylinder cut is a circular cylinder with \( d = 88.5 \, \text{mm} \) in diameter cut at both sides in parallel with y-axis. The turbine constructed from polyvinyl chloride plastic material with blade diameter \( D = 165.2 \, \text{mm} \), it gives the \( d/D = 0.54 \) (this diameter size is chosen without special consideration, but a similar research is currently being carried out for variations in circular cylinder diameters sizes by other groups at the same laboratory) and height \( H = 294.4 \, \text{mm} \), with the shaft diameter \( b = 19 \, \text{mm} \). The turbine rotor is equipped with an end-plate that has a diameter of 333.7 mm. The I-65° type cylinder is also assembled by polyvinyl chloride plastic material whose height is 500 mm installed in the upstream returning blade. The I-65° type cylinder is also set within distance center to center relative to the turbine diameter \( S/D = 1.4 \). This is because, according to Triyogi et al. [2], small-type I cylinder with cutting angles \( \theta_s = 65^\circ \) and mounted at the distance \( S/D = 1.4 \) in front of the main cylinder give the highest drag reduction among the small cylinders tested in their investigation. Also, based on the results theses of Sakti [13], that placing an I-65° type cylinder upstream the convex blade at a distance relative to the blade diameter \( S/D = 1.4 \) provides the highest increase in Savonius turbine power, among other distances tested in his study. The experiments are conducted with freestream speed \( (U) \) of 5 m/s coincide to \( Re = 9.9 \times 10^4 \). The Reynolds Number is obtained from the length of characteristic \( L = (2D - b) \) and free stream speed \( (U) \) from the blower. Figure 2b describes the turbine dynamics moment that is measured by a rope-brake type dynamometer, based on research Mahmoud et al., Kadam & Patil [9, 14] as references. The rope-brake dynamometer assembly contains pulley, weighing pan, spring balanced, and nylon string. The weighing pan is linked by spring balance with nylon string 1.0 mm in diameter, were the wire string is wrapped one loop over the pulley.

![Fig. 2. (a). The Savonius wind turbine configuration with an I-65° type cylinder is installed in front of Returning Blade. (b). The Schematic diagram of rope-brake dynamometer measurement](image)

Figure 3 describes the schematic diagram of the experimental equipment assembly, where the turbine is set at 3200 mm from the blower fan. The specification of the fan is CKE SPF-45, diameter = 450 mm, speed = 1800 rpm, airflow \( = 12 \, \text{m}^3/\text{min}, \text{power} = 1700 \, \text{Watt} \). The honeycomb plate 1x1 m is positioned 800 mm in front of the blower fan to make convinced that the flow is uniform as possible. The freestream velocity is sensed by anemometer Omega type HHF141. The measuring range of this anemometer as wide as 1.5 to 35 \( \text{m/s} \) with an accuracy of \( \pm \, 1\% \). Non-contactable tachometer OMEGA series HHT12 is used to censor the rotational speed of the turbine. This optical tachometer has to measure range 5 to 99,999 rpm and the accuracy up to 0.01%. The static torque meter is censored by LUTRON model TQ-8800 that the measuring ranged from 0 to 147.1 Ncm. The accuracy of this sensor up to 0.01 N.cm measures the torque while the turbine is in static condition. The turbine shaft equipped with R12ZZ unsealed type ball bearing. In order to minimize the bearing friction, the grease inside the roller is removed and periodically sprayed with WD40. Nagata C-5 spring balance type that has a measurement range of 0 to 10,000 grams and accuracy up to \( \pm \, 50 \) grams is assembled as part of the rope-brake dynamometer. The rope-brake dynamometer mass balances are calibrated by Shimadzu ELB300 with measuring a range of 0-300 grams with accuracy up to 0.01 grams. The turbine torque is detected by loaded the rotor gradually (adding 20 grams of mass each time) from unloading conditions to maximal loaded that characterized by the cessation of rotation of the rotor [13]. This condition gives Tip-Speed Ratio in a range of \( 0 < \lambda < 1.3 \). The data information from spring balance and turbine rpm for every condition is recorded, up to the 99,999 rpm and the accuracy up to 0.01%. The static torque is measured by KISTLER model 9257B dynamometer with measuring range of this anemometer as wide as 1.5 to 35 m/s velocity is sensed by anemometer Omega type HHF141. The dynamic moment of this anemometer as wide as 1.5 to 35 m/s.

![Fig. 3. Schematic diagram of the experimental set-up](image)

B. Data reduction

In this study, the Savonius wind turbine mechanical power such as shaft revolution and dynamics moment investigated at \( Re = 9.9 \times 10^4 \). The Reynolds number calculated based on the length characteristic and freestream from the blower fan, as shown in the following equation:

\[
Re = \left( \frac{\rho U L}{\mu} \right) = \left( \frac{\rho (2D - b)}{\mu} \right)
\]

Where \( L \) defines as the length characteristic of the turbine \( (L = 2D - b) \), \( D \) refers to the diameter of the turbine blade (m), and \( b \) is the overlap diameter (m), \( \rho \) is air density \( (\text{kg/m}^3) \), \( \mu \) is dynamic viscosity \( (\text{kg/m.s}) \) \( U \) is freestream velocity \( (\text{m/s}) \). The shaft angle speed and dynamics torque based on the
experiment result is used to determine the mechanical power $P_m$ with the following equation:

$$P_m = T\omega$$  \hfill (2)

Where the letter $T$ as dynamic torque (N.m), $\omega$ is shaft angle speed or angular velocity (rad/s). Individually the active torque is calculated with the following equation:

$$T = Fr$$  \hfill (3)

Where $r$ expressed as the radius of the pulley and the letter $F$ is the force step on the rotor shaft that could be determined with the following equation:

$$F = (M - s)g$$  \hfill (4)

The angular velocity $\omega$ could be calculated by the equation below:

$$\omega = \frac{2\pi N}{60}$$  \hfill (5)

Where the letter $N$ expressed as the number revolution per minute (rpm). The coefficient of a moment for each turbine blade determined for a sequence with calculated the swept area before by using the following equation:

$$A = HD$$ \hfill (6)

Where $A$ defines as the swept area (m²), and the letter $H$ described as the height of the rotor (m). Then the following equation uses to calculate the coefficient moment.

$$c_m = \frac{4T}{\rho AU^2}$$ \hfill (7)

After getting the angular speed $\omega$, the tip speed ratio $TSR$ or $\lambda$ determined with the following equation below:

$$TSR = \lambda = \frac{\omega L}{2V}$$ \hfill (8)

The coefficient power is the power efficiency of the turbine or ratio between turbine power input and turbine power output that could be determined by the following equation:

$$C_p = \frac{P_m}{P_w}$$  \hfill (9)

Where $P_w$ defines as wind power act as input, and $P_m$ defined as mechanical power act as an output. The wind power $P_w$ could be determined by the equation below:

$$P_w = \frac{1}{2}\rho AU^3$$  \hfill (10)

In the fullness of time, refers to the (2), (3), (4), (5), (9), and (10) the power coefficient determined by the equations below:

$$C_p = \frac{grN(M-s)}{15\rho AU^3}$$ \hfill (11)

Based on the multiplication between of coefficient moment and tips speed ratio found the alternatives equation in order to calculated coefficient power as explained below:

$$C_p = \frac{P_m}{0.5\rho AU^3}$$ \hfill (12)

Based on Young D. Hugh (1962), the uncertainty value is a critical consideration in an experimental study [18]. Table 1 presented the uncertainty of coefficient moment and coefficient power that computed at the maximum amount of 5.2% and 5.3%, respectively.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Uncertainty (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reynolds Number</td>
<td>2.7</td>
</tr>
<tr>
<td>Coefficient Moment</td>
<td>5.2</td>
</tr>
<tr>
<td>Coefficient Power</td>
<td>5.3</td>
</tr>
<tr>
<td>Tips Speed Ratio</td>
<td>2.0</td>
</tr>
</tbody>
</table>

C. Numerical analysis

Based on the literature, it can be understood that a precision CFD simulation of the flow behavior near Savonius turbine is very complex and crucial work, primarily due to its time consuming and the system efficiency strongly influences by flow separation. Therefore it is essential to observe the simulation procedures carefully with full attention. Following the series of that procedures, the results of the numerical simulation approach must be validated.

Initially, the three-dimensional 3D geometry of the Savonius wind turbine constructed with commercial software and has been saved in .IGES format file. The simulation boundary condition is built by importing .IGES format file on Design Modeller. The commercial software ANSYS Fluent 19.1 takes the lead role for further numerical analysis in this study. Figure 4 illustrated the specific dimension surrounding the turbine. For each side of the turbine, that is determined as five times of rotor diameter called a stationary zone and designed in such a way to avoid blockage effect. The outflow kept away from the turbine as far as 15 times width in order to prevent backflow effect during computation, and it is determined as a pressure outlet. The internal interface called a wake zone stretched along as ten times diameter in aft turbine and five times diameter in the fore turbine. The rotating zone radius is 0.6 times the diameter of the turbine. All of these three-zones are separated by their interfaces. Overall of the three-dimensional (3D) of the simulation domain confirm with another research by [13, 15, 16, 21, 22, 23].
The mesh generation is performed based on three domain zones consist of the rotating zone, wake zone, and stationary zone, detail illustrated in Fig 5. The Three-dimensional (3D) turbine Savonius geometry has complex surface contours that would be appropriately meshed with the unstructured mesh type. The rotating zone and wake-zone use local control sizing mesh with the number mesh element denser than the fixed-zones. The fixed region uses structural mesh inserted with the tetrahedron hex-dominant method in order to reduced time consuming for the CFD calculation.

The simulation on the present study uses commercial software ANSYS-Fluent 19.1. The equation of Unsteady Reynolds-Average Navier Stokes (URANS) solved with SIMPLE (Semi-Implicit Method for Pressure-Linked Equations). All of the variables and turbulence intensity discretized within finite-volume formulation in the second-order upwind scheme. The turbulence model uses k-epsilon realizal with enhanced wall treatment and the unsteady flow solved by SMM (Sliding Mesh Model), overall confirm with reports by [21, 23]. The time step is counted for every degree increment angle of the turbine rotation. The maximum number of iteration for every time step up to 80, it indicated that the convergence obtained within 80 iterations for 1° turbine rotation. Convergence criterion reaches when the continuity and other residual parameters value pass under 10^{-5} [17].

The time needed for 1° turbine rotation defines as the Time Step Size (TSS) counted from the time period for once turbine revolution (T_{rot}) multiply the number of degree increment angle (\theta) divide by 360. The following equation use to determine the time period for once the turbine rotation (T_{rot}) and further use to calculate the required time for \theta = 1° increment angle or time step size (TSS).

\[ TSS = \frac{T_{rot} \theta}{360} = \frac{60}{N} \times \frac{\theta}{360} = \frac{\theta}{6(N)} \]  

(13)

Table 2 shows the time step and time step size for a various number of tips speed ratio, conducted in velocity upstream 5 m/s, \(Re = 9.9 \times 10^5\) at the TSR 0.5 \(\leq \lambda \leq 1.1\). Refer to \(\lambda = 0.8\) the \(\omega\) obtained 28.089 rad/s, the time period spent for one turbine revolution \(T_{rot}\) reach 0.224 s, and the time needed for \(\theta = 1°\) increment angle determined as 0.000621 s. Hereinafter, the simulation result aligned with current experimental data for specified number 0.5 \(\leq \lambda \leq 1.1\).

The numerical verification should be performed in order to reduce errors in the numerical solution, and the achievement of approval between the numerical and exact solutions would get better. Several dissimilar three-dimensional unstructured mesh with an advance in density and quality, consist of mesh element about 950,000, 1,100,000, 1,200,000, and 1,400,000 respectively, have been simulated for Savonius turbine with I-65° type cylinder. Figure 6. described the coefficient moment as a function of the turbine blade increment angle in four different mesh configurations. The grid independence result shows that mesh within more than 1,200,000 elements provides the moment coefficient with the identical trend results. The coefficient moment trends for various mesh configurations, especially on the enlargement portion as a sample, give clear evidence that 1,200,000 mesh elements in line with 1,400,000 mesh elements. Considered time-consuming, which is directly proportional to the number of mesh elements, the common 1,200,000 mesh element retained for the further numerical solution in this study [16, 17].

FIG. 6. The coefficient moment distribution as a function of the angle of the turbine blade aligned between four different mesh elements.
The numerical validation has been performed by aligned the simulation results and experimental data from [19]. The comparison presented in Fig. 7, correspond to the $C_p$ between the experimental result and present numerical data in a specified tips-speed ratio of $0.3 \leq \lambda \leq 0.1$. Refers to the $\lambda = 0.8$, the coefficient power between simulation results for the conventional turbine and the data from [19] reach 0.1735 and 0.1734, respectively. The gap between the two counted on 0.07% indicated that both experimental and numerical works quite similar. Furthermore, the most significant deviation seen at both ends is $\lambda = 1.0$ and 0.3, but for another TSR point, the gap has no more than 4.4% in the maximum and still under the uncertainty value of $C_p$, which is 5.3% as shown in Table 1. Based on these considerations, it can be concluded that the numerical simulation and all of the configurations valid for use in further analysis.

![Fig. 7. The Validation of the coefficient power $C_p$ for specified $0.3 \leq \lambda \leq 0.1$.](image)

### III. RESULTS AND DISCUSSION

#### A. Coefficient Moment and Coefficient power

Figure 8 shows the result of the experimental and numerical of the turbine $C_p$ as a function of TSR, for $S/D = 1.4$ and $Re = 9.9 \times 10^4$.

![Fig. 8. The outgrowth of the power coefficient of the turbine as a function of Tip Speed Ratio, for $S/D = 1.4$ and Reynolds number $Re = 9.9 \times 10^4$.](image)

If compared to the conventional turbine, the placement of an I-65° type cylinder upstream convex blade of the Savonius wind turbine is useful for increasing turbine power. The experimental maximum TSR between a conventional turbine and turbine with an I-65° cylinder reaches 1.1, and 1.25, respectively. This different maximum TSR for both turbines configuration gives an initial indication that the installation of an I-65° cylinder provides performance enhancement. Furthermore, the experimental $C_{p_{\text{max}}}$ for both turbines configuration reaches a value of 0.177 and 0.218, respectively. Where in this condition, the turbine $C_{p_{\text{max}}}$ with I-65° cylinder at TSR of 0.8 has increased up to 23.61% higher than the conventional turbine.

The numerical $C_{p_{\text{max}}}$ for conventional turbine and turbine with I-65° cylinder reaches a value of 0.173 and 0.217, respectively. This means the I-65° cylinder upstream returning blade increases the $C_{p_{\text{max}}} 25.32\%$ higher than the conventional one. Moreover, the comparison of the $C_p$ between the numerical and experimental results for the traditional turbines and turbine with I-65° cylinder type individually obtained a good agreement. This agreement approved in which for the conventional turbines, the maximum gap of the $C_p$ appears on the $0.4 < \lambda \leq 1.0$, reaches below 2.3%. Whereas for the turbine with an I-65° cylinder configuration, the maximum difference of the $C_p$ appears on the tips-speed ratio of $0.6 < \lambda \leq 1.0$, reaches below 3.26%. This means that the gaps no more than the uncertainty value, i.e., 5.3% for the coefficient power, which presented in Table 1.

![Fig. 9. The outgrowth of the coefficient moment as a function of TSR for $Re = 9.9 \times 10^4$ and $S/D = 1.4$.](image)

The experimental and numerical proceeds for the coefficient moment $C_m$ at specified Reynolds number $Re = 9.9 \times 10^4$ and $S/D = 1.4$, shows in Fig. 9. Visually, the coefficient moment for the conventional turbine and turbine with I-65° cylinder inversely proportional to the TSR and the numerical simulation result validated with each other with experimental data. The significant improvement also presented in this figure, where the placement of the I-65° cylinder upstream to the convex turbine blade increased the coefficient moment, convincingly. The maximum gap between numerical and experimental results is obtained at TSR $\lambda = 0.8$ for turbine conventional and TSR $\lambda = 0.9$ for a turbine with an I-65°
cylinder type. This particular condition of the maximum gap calculated reaches 2.65% and 4.25%, respectively. The whole coefficient moment \( Cm \) specific for TSR number ranges to 0.3<\( \leq 1.0 \) considered still below the uncertainty value, i.e., 5.2% from table 1. This means that the deviation of the \( Cm \) value between both turbines configuration still in an acceptable area. Consider the facts founded, the coefficient moment at all TSR points confirms the improvement of the turbine performance.

### B. Self Starting Capability

Another principal consideration affiliated with turbine performance is the turbine capability for starting its self. In order to resolve this issue, the experimental static torque measurement conducted and plotted only in a given angle \( 0^\circ \leq \theta \leq 180^\circ \) due to the periodicity of the turbine rotation [8, 16]. Figure 10 shows the achieved static torque (Ncm) for the conventional turbine and turbine with an I-65\(^{\circ}\) cylinder specified at \( Re = 9.9 \times 10^4 \). Compared to the conventional turbine, the experiment result leading to significant improvement of static torque as an effect of placement I-65\(^{\circ}\) cylinder upstream returning blade. The static torque of the conventional turbine is partly in the negative domain, e.g., 127\(^{\circ}\)≤\( \theta \leq 163^{\circ} \). This negative domain indicates that the conventional turbine does not have the self-starting capability, or this means that when the turbine starts to received wind energy at these angles, they need other external forces to begin in its rotation. For the turbine with an I-65\(^{\circ}\) cylinder mounted upstream from the returning turbine blade, the negative static torque zone completely disappears with a specific static torque value of more than 0.13 N.cm. The peak static torque obtained at the blade position angle \( \theta = 30^\circ \) for both the conventional turbine and turbine with I-65\(^{\circ}\) cylinder installation and its reaches values of 3.75 N.cm and 4.29 N.cm, respectively. The maximum static torque obtained at \( \theta = 30^\circ \) confirms with other experimental results by [13] and numerical results by [8]. Based on this reason, the blade position angle \( \theta = 30^\circ \), 90\(^{\circ}\), and also 150\(^{\circ}\) considered as the critical angle for the subsequent further numerical investigation in this report. The overall analysis, according to the static torque measurement, the installation of the I-65\(^{\circ}\) type cylinder upstream of the returning blade, has a substantial positive effect on the turbine performance, where the self-starting capability is obtained at every angle position in the turbine configuration.

### C. Pressure contour and Velocity contour

The Savonius turbine works based on the difference in pressure drag between the returning and advancing blade [23]. In this section, the consideration of turbine performance based on the pressure drag changes and the associated of static torque as an effect of placement the I-65\(^{\circ}\) cylinder. The value of TSR \( \lambda = 0.8 \) and the position angle (\( \theta \)) within 30\(^{\circ}\), 90\(^{\circ}\), and 150\(^{\circ}\) (represent the maximum coefficient and the static torque critical angles) used as reference points in this consideration.

Figure 11 shows the various numerical data plotted in the pressure contour aligned between turbine conventional and turbine with an I-65\(^{\circ}\) type cylinder at the position angle of 30\(^{\circ}\), 90\(^{\circ}\), and 150\(^{\circ}\), \( S/D = 1.4 \), TSR \( \lambda = 0.7 \) and \( Re = 9.9 \times 10^4 \). Figure 11a shows the conventional turbine at an angle \( \theta = 30^\circ \), the positive pressure about +10 pa appear over the whole concave sides of the advancing blade and partial part convex sides of returning blade. Meanwhile, the negative pressure appears over the entire convex advancing blade and the entire concave returning blade. The minimum negative pressure at the downstream area detected up to -40 pa. When the I-65\(^{\circ}\) cylinder mounted in the upstream returning blade as shown in Fig.11b, the pressures decrease up to +5 pa at both the concave advancing and convex returning blade, but at the part of the convex returning blade near turbine shaft, the pressure drops to +0 pa. At the same time, the downstream area relatively steady, not affected by the I-65\(^{\circ}\) cylinder. This pressure change behavior modified the pressure difference between the upstream and downstream turbine blade causes reducing at the returning blade pressure drag. Finally, it increases the positive torque of the turbine. As to confirm with Fig.10, at this position angle \( \theta = 30^\circ \) the static torque reach the peak point for both turbines configuration.

Figure 11c and 11d identified the turbines at position angle \( \theta = 90^\circ \), given identical pressure changes behavior when the blade at a position angle of \( \theta = 90^\circ \), where the presence I-65\(^{\circ}\) type cylinder reduces pressure distribution at the convex returning blade, reducing its pressure drag and improve the turbine performance. The differentiate situation is appearance tiny positive pressure at the convex returning blade that confirmed later as a reattachment area from I-65\(^{\circ}\) cylinder separation flow. This tiny positive pressure area reduces the effect of the I-65\(^{\circ}\) cylinder and also reduces turbine performance slightly below the turbine when at position angle \( \theta = 30^\circ \) This phenomenon agreed with the evolution of static torque, as shown in Fig.10.

Figure 11e and 11f show both turbines at position angle \( \theta = 150^\circ \). The I-65\(^{\circ}\) cylinder influences the pressure difference between the upstream and downstream returning blade and also decreases its pressure drag. The role of the I-65\(^{\circ}\) cylinder is not as active as when the turbine at 0<\( \theta < 150 \) due to increasing the tiny positive pressure at the convex returning blade and the position of the blade itself related to the cylinder. This normal drag pressure in line with the lower static torque that appears when the turbine at the position angle \( \theta = 150^\circ \). The velocity contour presented to support initial supposed and confirm the previous analysis so that the aerodynamic effect of the I-65\(^{\circ}\) cylinder stated actively on the turbine Savonius performance improvement.

![Fig. 10. The outgrowth of the static torque (Ncm) as a function of the blade angle \( \theta \) for \( Re = 9.9 \times 10^4 \), comparison between the conventional Savonius wind turbine and the turbine with I-65\(^{\circ}\) cylinder for \( S/D = 1.4 \).](image-url)
Figure 11. The static pressure contour comparing between turbine conventional and turbine with I-65° type cylinder at $\lambda = 0.8$, $Re = 9.9 \times 10^4$, and $S/D = 1.4$. a) 30° the turbine conventional, b) 30° the turbine with I-65° cylinder, c) 90° the turbine conventional, d) 90° the turbine with I-65° cylinder, e) 150° the turbine conventional, f) 150° the turbine with an I-65° cylinder.

Figure 12 shows the velocity contour compared between turbine conventional and turbine with an I-65° type cylinder in several position angle. This velocity contour is presented at the Reynolds number $Re = 9.9 \times 10^4$, and the tip-speed ratio $\lambda = 0.8$. The negative value at the contour legends indicate the direction of the flow relative to the $Y$-axis; it means the backflow marks as a positive value and blue area. The red area that appears on these contours indicates the velocity reaches the peak value and also confirms with Fig. 11, where the pressure obtained at the lowest value in the same area. The following are the discussion about several types of airflow presented in the velocity contours.

Figure 12a shows the flow characteristic of the conventional turbine at $\theta = 30^\circ$, where the stagnation point occurs at a specific point on the surface of the convex returning blade. The separation point determined when the upstream flow deceleration against the friction of the surface blade and the adverse pressure gradient then start to separation with a slight increase in its velocity [2, 20]. The intensive
vortex raised up near shaft at both returning and advancing blade as an effect of turbine rotation.

Figure 12b shows the I-65° cylinder creates both an upper and lower shear layer. The lower shear layer reattached at a specific point of the convex returning blade, and the relative shifted out than the stagnation point of the conventional turbine. This shifted attachment points also delayed the separation point at the returning blade and reduced the width of the wake downstream returning blade. The tapered wake behind the returning blade reduced the drag pressure, thus increase the turbine torque. The reduced width of the wake also indicated by the width shrinking of both vortexes near the turbine shaft. On the other hand, there are other parts of the lower shear layer also reattached at a concave advancing blade given more positive torque at the Savonius turbine.

![Fig. 12. The velocity contour comparing between the turbine conventional and turbine with I-65° cylinder at TSR = 0.8, Re = 9.9 x 10^4 and S/D = 1.4. a) 30° the turbine conventional, b) 30° the turbine with I-65° cylinder, c) 90° the turbine conventional, d) 90° the turbine with I-65° cylinder, e) 150° the turbine conventional, f) 150° the turbine with an I-65° cylinder.](image-url)
Figure 12c shows the turbine flow characteristic at $\theta = 90^\circ$, where the stagnation points identified almost at the center of the convex returning blade. Both separations point obviously fell a few degrees near the stagnation points defined as when the shear layer flow starts to leave the surface with a slight increase in velocity. Two vortexes from the returning blade appear at this turbine position angle. The reattachment of both shear layer upstream returning blade controlled by the I-65° type cylinder, as shown in Fig. 12d. The lower shear layer reattached and separated at the convex returning blade closer to the turbine shaft. The upper shear layer has a similar characteristic that the reattachment and separation points shifted closer to the tip of the returning blade. The shifted of separation points tapered the wake and reduce the pressure drag of the returning blade, thus increase turbine torque. The vortex near the tips of returning blade vanished due to the tapered wake, whereas the vortex near the turbine shaft shrink smaller indicated the wake downstream returning blade weakens. The increasing of turbine torque at position angle $\theta = 90^\circ$ not as strong as when the turbine position angle at $\theta = 30^\circ$ due to the difference in the number of separations points that fell at convex returning blade, where at this condition, the reattachment point from the lower shear layer increases the pressure drag returning turbine as described previously in Fig. 11d.

Figure 12e and 12f show the turbine position angle $\theta = 150^\circ$, the airflow behavior not too far than the previous description. Where the I-65° cylinder generated the two shear layers, the lower shear layer reattached at the convex returning blade and delayed the separation points. The delayed separation points will narrow the weak downstream returning blade and also reduced the pressure drag, thus finally improved turbine performance. The wake weakness downstream returning blade also indicated by vanishment of vortex advancing blade and the shrink vortexes returning blade. Compared to the previous position angle, the torque when the blade at $\theta = 150^\circ$ obtained at the lowest point due to the several sections of advancing blade immersed in the wake from I-65° cylinder and the pressure drag returning blade increase as an effect of attachment flow as previously described in Fig. 11f. Overall near-wall airflow investigation by the velocity contour at the blade position angles $\theta = 30^\circ$, $90^\circ$, and $150^\circ$ confirms with the previous reports by [2, 3].

IV. CONCLUSIONS

The Savonius wind turbine compatible with absorbing wind energy in relatively low wind speed, independent from wind direction and simple in design, but has low efficiency. Therefore the improvement of turbine efficiency is the main priority in this study in order to increase the coefficient power and excellent self-starting capability. For this objective, the I-65° type cylinder installed in an upstream flow of the returning turbine at a specified position and acts as a determinant factor to improve the turbine performance aerodynamically.

The flow characteristic has been well observed in numerical CFD simulation accurately. The validation against previous experimental reports shows the agreement, which is calculated with $k$-$\varepsilon$ turbulence model, medium-fine mesh, and used the vast computational domain. The I-65° cylinder modified airflow around the turbine, so that delayed the separation and reduced the width of the turbine wake and the pressure drag, then finally improving the turbine torque. The coefficient power turbine with I-65° cylinder increases up to 23.61% better than the conventional turbines at TSR $\geq 0.8$ experimentally. This improvement also in line with the numerical result with the gap to experimental data below 4.52% for coefficient power and 4.25% for the coefficient moment. The turbine Savonius with I-65° cylinder also gives a significant improvement in self-starting capability, which is the lowest value obtained 0.3 Ncm at blade position $\theta = 150^\circ$.

The pressure and velocity contour visualization also agreed with this improvement, which is the pressure contour described the pressure drag reduction, and the velocity contour presented the reduced wake area as an impact of delayed the separation flow. Finally, the overall consideration based on qualitative and quantitative data parameter agreed that the I-65° give significant enhancement on the turbine Savonius performance.

At present, this study still continues with a variety of cylinder shapes, cylinder diameter, and their position towards the returning blade of the turbine Savonius, both experimentally and numerically. The validation between experiments and numerical data also reinforced in such a way to obtain an excellent agreement.

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