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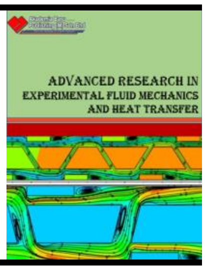


# The Influence of Upstream Installation of D-53° Type Cylinder on the Performance of Savonius Turbine

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**ABSTRACT**

The purpose of this research is to study experimentally and numerically the effect of installing a D-53° type cylinder in front of the returning turbine blade on the performance of the Savonius wind turbine. The D-53° type cylinder is a circular cylinder cut only on the front side of the cylinder, at a cutting angle of 53°. The cylinder having a diameter relative to the rotor diameter (d/D) = 0.5 is installed at a distance relative to the rotor diameter (S/D), which remains constant at 1.6. The configuration was tested at Reynolds number 95,000 experimentally in the open air and numerically using Commercial CFD software, Ansys Fluent. The results showed that the presence of a D-53° type cylinder in front of the returning turbine blade effectively increased the power coefficient of the turbine. The increase in the maximum power coefficient is about 24.56%. It also obtained good agreement between the numerical and experimental results.

***Keywords:***
Savonius turbine; D-53° type cylinder; returning turbine blade; power coefficient; moment coefficient

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## 1. Introduction

Realising the depletion of fossil energy reserves on earth, humans and all policymakers in countries in the world must be more aware of the importance of cultivating renewable energy sources. One of them is the wind which is very abundant and relatively cheap. One way to process wind energy is by using wind turbines. The difference in the wind speed range in each region makes not all types of turbines suitable for operation in every region. The turbine that is suitable for operation at low wind speeds and from all wind directions is the Savonius type turbine. This type of turbine has two blades: the advancing blade in the form of a concave and the returning blade in a convex shape. The difference in drag acting on the two blades makes the Savonius turbine produce energy. Simple and inexpensive construction is the advantage of this type of turbine. Unfortunately, the performance of this Savonius turbine is the lowest compared to other turbine types, so a lot of research is being done to improve its performance.

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In the last decade, there have been many researchers such as Mao *et al.,* [1], Kumar *et al.,* [2], Chan *et al.,* [3], Tian *et al.,* [4], Kerikous *et al.,* [5, 6], Masdari *et al.,* [7] or Saeed *et al.,* [8] who attempted to conduct research to improve the performance of the Savonius turbine by modifying the shape and/or thickness of the turbine blades. Meanwhile, other researchers such as Sheldahl *et al.,* [9], Shaheen *et al.,* [10], Basumatary *et al.,* [11], and Talukdar *et al.,* [12] are more interested in overlapping two blades or varying the number of blades, and there are even researchers such as Mahmoud *et al.,* [13] who studied the role of the endplate, the number of turbine stages and also the aspect ratio and overlap ratio of the turbine. In addition, some researchers are more interested in the role of the barrier or obstacle in the turbine to improve turbine performance than in modifying the design or construction of the turbine itself. As done by Altan *et al.,* [14], they mounted a flat plate in front of the returning blade to reduce drag acting on it, thereby increasing the positive torque of the turbine. Mohamed et al. [15] have successfully augmented the power of the Savonius turbine by 27.3% by mounting the retaining plate in front of the returning turbine blade. Regarding the installation of a flat plate in front of the returning blade, Triyogi *et al.,* [16,17] have proven that the flat plate width and Reynolds number play an important role in improving turbine performance. Researchers from the Mechanical Engineering Department of the Sepuluh Nopember Institute of Technology have done something different from other general researchers to improve the performance of the Savonius turbine. They positioned both circular and sliced cylinders as passive control either in front of the returning blade or beside the advancing blade. As done by Setiawan *et al.,* [18 & 19] for circular cylinders capable of increasing the turbine power coefficient by 41% and 17.3%, respectively, or Gunawan *et al.,* [20], who have succeeded in increasing the turbine power coefficient up to 23.6% by placing a cylinder type I-65° in front of the returning blade. Again, their compatriots, Triyogi *et al.,* [21], have succeeded in increasing the turbine power coefficient by 12.2% by installing a circular cylinder with a ratio of cylinder diameter to turbine rotor diameter d/D = 0.54 and is operated at Reynolds number = 99,000.

Some researchers such as Bouak *et al.,* [22], Tsutsui *et al.,* [23], Lee *et al.,* [24] for different purposes have also used a small circular cylinder to influence the drag force acting on the larger circular cylinder behind it. They have all proven the role of the small cylinder as a passive controller in reducing the drag acting on the large circular cylinder behind it. For example, Tsutsui *et al.,* [23] have installed a small circular cylinder 0.025 to 0.25 times the diameter of the large circular cylinder behind it and succeeded in reducing the drag of the large cylinder by 63%. Likewise, Lee et al. [24] have succeeded in reducing the drag force acting on large cylinders by up to 25%. It is obtained for the ratio of rod diameter and large cylinder diameter d/D = 0.233, which is installed at a distance ratio of S/D = 1.833, and the flow is operated at Reynolds number (Re) = 20,000. Meanwhile, other researchers such as Triyogi *et al.,* [25], Imron *et al.,* [26, 27, 28] and Hakam *et al.,* [29] have used an I-type cylinder to reduce the drag force acting on the large cylinder behind it. They all have claimed that the I-type cylinder has succeeded to diminish the drag force of the large cylinder installed behind it. Triyogi *et al.,* [25] proved that for Re = 53,000, the I-type cylinder has a cutting angle of 65° and installed at a distance ratio of S/D = 1.375 has reduced the drag of the large cylinder behind it by about 52%.

The decision to use of I-65˚ type of cylinder as passive control installed in front of the returning turbine blade in the research of Gunawan *et al.,* [20] was inspired by the researches of Aiba *et al.,* [30]. In fact, that Aiba *et al.,* [30] have suggested two types of bluff bodies cut from a circular cylinder; i.e. D-type cylinder and I-type cylinder, where D-type is a small circular cylinder cut only on the front side of the small cylinder, so it resembles the letter "D" and I-type cylinder is cut on both sides that is like the letter "I". They have claimed that each type of bluff bodies with a cutting angle of 53˚ have the minimum drag force, which is about 50% of the circular one. They have proved that the bluff bodies with a cutting angle of more than 60° have a drag coefficient higher than the circular cylinder.

Inspired by the various studies above, this research aims to study numerically and experimentally the effect of the upstream installation of a D-53° type cylinder on the performance of the Savonius wind turbine, where the D-53° type cylinder is installed in front of the returning blade with the cutting surface of the cylinder facing perpendicular to the direction of the coming wind. It is to determine the role of the D-53° type cylinder as a passive control to reduce the drag force on the returning turbine blade. As far as it is known, the use of the D-53° type cylinder, which is placed in front of the returning blade to improve the performance of the Savonius turbine, has never been done by other researchers. Some other researchers have focused more on using a curtain plate or flat plate, which only block the flow to the returning blade, but there is no flow interaction between the plate and the turbine blades. It is, of course, different from the use of a D-53° type cylinder installed in the front of the returning blade as proposed in this study, which allows a flow interaction between the cylinder and the turbine blade. The shear layer detached from the cylinder will attach to the returning blade surface and affect the flow characteristics at the front and on the returning blade surface. The wake formed behind the cylinder will reduce the pressure on the front side of the returning blade, and meanwhile, the pressure behind it will increase, according to Tsutsui *et al.,* [23] and Triyogi *et al.,* [25]. As a result, pressure drag acting on the returning blade will decrease, and the positive torque of the turbine will increase the finally will increase the power of the turbine. On the other side, the shear layer detached from the cylinder will attach to the surface of the returning blade, it will disturb the pressure distribution and boundary layer on the returning blade surface; therefore, the separation point will be delayed, and the wake which is formed behind the returning blade will be narrower. It will reduce the drag force acting on the returning blade and increase the positive torque, which in turn will increase the performance of the turbine. It is the uniqueness of this research. This phenomenon will be studied in this paper experimentally and numerically using commercial computational fluid dynamics software, i.e. Fluent Ansys 18.1. The D-53° type cylinder used in this study has a diameter of about 0.5 times the rotor diameter (d/D = 0.5), is placed at a distance relative to the rotor diameter set at S/D = 1.6 and the flow operated at Reynolds number (Re) = 95,000. The expected result in this study is an increase in turbine performance due to the installation of the D-53° type cylinder in front of the returning turbine blade, which is determined based on the power coefficient ($C_p$) and moment coefficient ($C_m$) of the tested turbine.

## 2. Methodology

### *2.1 Experimental Arrangement*

Figure 1 shows the schematic diagram of the constructed polyvinyl chloride (PVC) Savonius turbine with the diameter ($D_s$) of 303.4 mm and 294.4 tall with D-53° passive control cylinder, which is a 76.2 mm diameter circular cylinder cut at 53° on the surface facing the flow. This 500 mm tall cylinder is placed upstream of the returning blade at the distance of 1.6 times the diameter of the blade, calculated from the centre of the turbine to the centre of the cylinder (S/D). The fluid velocity (U) generated by the axial fan is set to 5 m/s, with a reference length of the turbine diameter of 303.4 mm, which corresponds to the Reynolds number of approximately $9.5 \times 10^4$. The axial fan is positioned 3200 mm apart from the turbine, with a honeycomb pattern panel in between to ensure the uniformity of the flow, as shown in Figure 2 based on research by Gunawan et. al [20]. The fan in this experiment is an axial fan CKE SF-46, and the rotation of the turbine is measured using tachometer Omega HHT13.

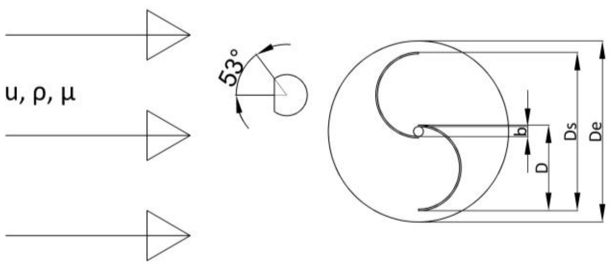



**Fig. 1.** Schematic diagram of the Savonius turbine and the passive control cylinder

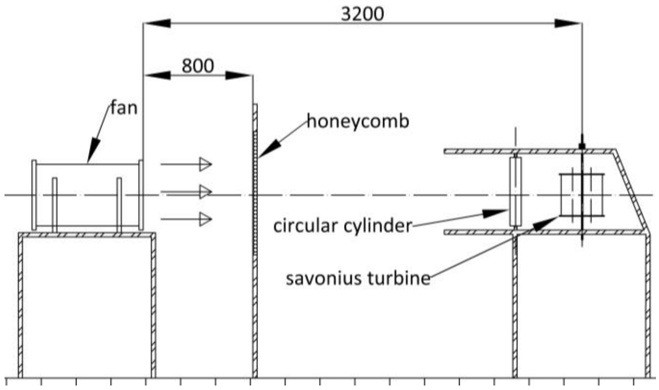



**Fig. 2.** Arrangement diagram of the experiment

Figure 3 shows the rope brake dynamometer system to measure the torque of the turbine, based on research by Mahmoud *et al.*, [13], as reference. It consists of HTC TAL 220 load cell and balancing mass connected by a nylon string. The torque generated is measured according to gradual mass addition to balancing weight. By adding 15.5 gram for each step, from no load until the turbine stops, the weight acting on the load cell changes from minimum at tip speed ratio $\lambda \approx 1$ to maximum at $\lambda = 0$. The difference between the weight of the balancing weight and the weight acting on the load cell is calculated to obtain the net torque generated by the turbine. The torque generated by the turbine and the tip speed ratio at which the turbine is rotating is then considered to calculate the dynamic moment.

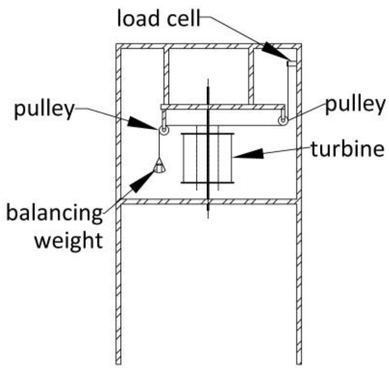



**Fig. 3.** Rope brake dynamometer system

*2.2 Data Calculation*

Based on the data obtained from the balancing mass and the mass on the load cell, the dynamic torque is then calculated based on equation

$$T = F.r \tag{1}$$

Where F is the force on the rotor shaft and r is the radius of the shaft. Which in this case:

$$T = (|m - s|g)r \tag{2}$$

Where m is the balancing mass (gram), s is the mass sensed at the load cell (gram), g is the gravity acceleration (m/s$^2$), and r is the shaft diameter (m). Tip speed ratio ($\lambda$) can be calculated using:

$$\lambda = \frac{\frac{2\pi n}{60} R}{U} \tag{3}$$

Where n is the RPM measured, R is the radius of the turbine (m), and U is the freestream velocity (m/s). The coefficient of moment ($C_m$) is calculated using:

$$C_m = \frac{T}{\frac{1}{2} . \rho . A . U^2 . R} \tag{4}$$

Where ρ is the fluid density (kg/m$^3$), A is the swept area of the turbine (m$^2$) calculated by multiplying the height (H) by the diameter of the turbine ($D_s$), U is the freestream velocity (m/s), and R is the radius of the turbine (m). The coefficient of power can be calculated based on the coefficient of moment based on the equation:

$$C_p = C_m . \lambda \tag{5}$$

or based on the equation:

$$C_p = \frac{\omega . T}{\frac{1}{2} . \rho . A . U^3} \tag{6}$$

*2.3 Numerical Simulation*

The numerical simulation of the flow near the turbine is a complex and time-consuming process; therefore, a careful approach considering the accuracy of the result and the duration of the process should be implemented. Figure 4 shows the boundary dimension of the numerical simulation based on research by Setiawan *et al.*, [19]. The total width of the boundary is set 6 times the diameter of the turbine to prevent the blockage effect. Upstream reaches 10 times the diameter of the turbine, and downstream reaches 14 times. The rotating domain and wake domain are separated by interface and consist of finer mesh than the fixed domain. As shown in Table 1, the meshing parameter used is based on the research by Larin *et al.,* [32] as a reference with $y^+ = 30$. The 2-dimensional Savonius turbine geometry has difficult surface contours; thus, the rotating and wake domain unstructured meshing would be more suitable.

Commercial ANSYS 18.1 program is used to run the simulation. Unsteady Reynolds-Average Navier Stokes (URANS) and Semi-Implicit Method for Pressure-Linked Equations (SIMPLE) algorithm is applied in the numerical simulation. All variables are discretised by finite-volume formulation and second-order upwind scheme. SST k-ω is the most suitable turbulence modelling based on Nasef *et al.*, [31]; therefore, it is used in numerical simulation. The flow is considered unsteady because of the rotation of the turbine and the unsteadiness of the flow; in this case, sliding mesh motion used. The turbine's rotation speed is set to correspond to each tip speed ratio of 0.4, 0.5, 0.6, 0.7, and 0.8 based on freestream velocity of 5 m/s to understand better the flow in tip speed ratio in which the power reaches a maximum value. The time step is set to correspond to 1° per time step based on the research by Setiawan et al. [19]. The convergence criterion reaches if the continuity and other residuals pass below $10^{-5}$. Table 2 indicates the turbine rotation speed and time step corresponds to each tip speed ratio.

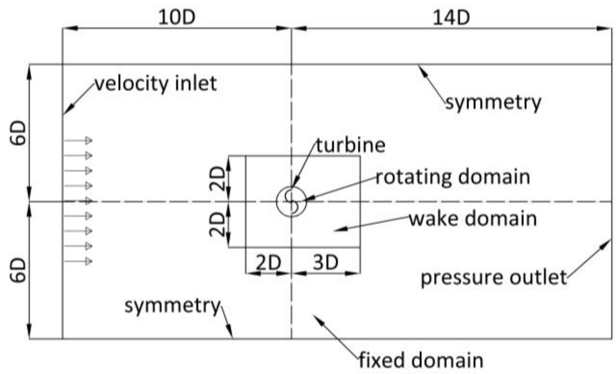


**Fig. 4.** Numerical simulation boundary dimension

**Table 1**
Meshing parameters

| | |
|---|---|
| Interface cell size | $1.21 \times 10^{-2}$ D |
| Rotating domain cell size | $1.21 \times 10^{-2}$ D |
| Wake domain cell size | $1.21 \times 10^{-2}$ D |
| Static domain cell size | 0.606 D |
| $y^+$ | 30 |
| Growth rate | 1.1 |
| Sub-iteration convergence criterion | $10^{-5}$ |

**Table 2**
Turbine rotational speed and time step for each tip speed ratio

| TSR (λ) | ω (rad/s) | time step (s) |
|---|---|---|
| 0.4 | 13.18392 | 0.001324 |
| 0.5 | 16.47989 | 0.001059 |
| 0.6 | 19.77587 | 0.000883 |
| 0.7 | 23.07185 | 0.000756 |
| 0.8 | 26.36783 | 0.000662 |

## 3. Results

### *3.1 Coefficient of power and coefficient of moment*

Figure 5 shows the comparison of the power coefficient between Savonius turbine with passive control D-53° type cylinder, hereinafter referred to as cylinder only, at the distance of S/D = 1.6 and turbine without passive control cylinder, or just called a conventional turbine. The maximum Cp value obtained by the conventional turbine is 6.26%; this number increases up to 7.93% with the installation of a passive control cylinder upstream of the returning blade. This data indicates that the presence of a passive control cylinder upstream of the turbine effectively increases the maximum coefficient of power by 24.56% at TSR, close to $\lambda = 0.6$ based on experimental study. As for tip speed ratio $\lambda > 0.5$, visually, there is an observable improvement with installing a passive control cylinder.

Moreover, as the result of numerical simulation, Cp reached a good agreement with the $C_p$ due to the experiment at a tip speed ratio of $0.5 < \lambda < 0.8$; the difference between simulation and numerical simulation is less than 10%. However, at tip speed ratio lower and higher than those values, the difference reaches up to 19.58%. It is expected because the 2-dimensional nature of simulation makes it difficult to portray the value compared to 3-dimensional turbine accurately.

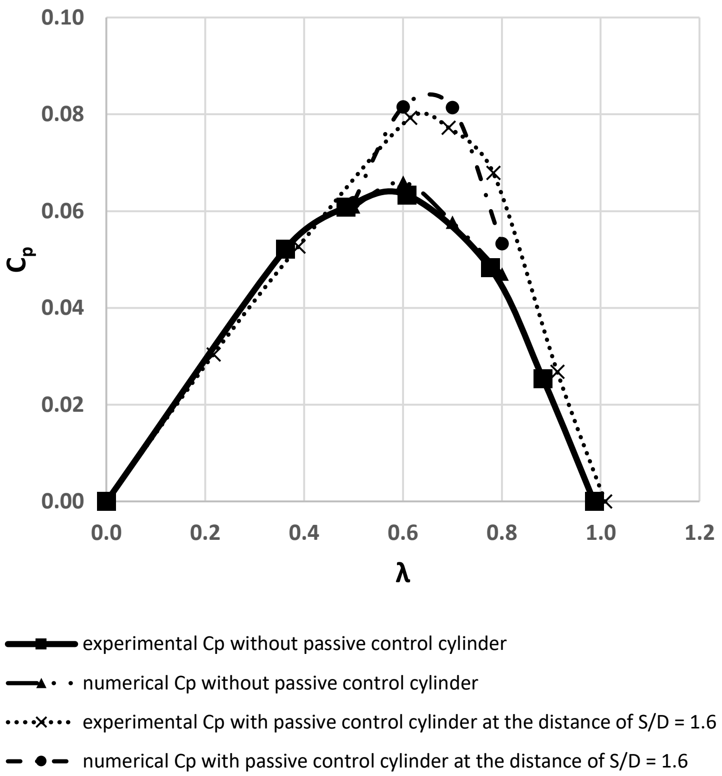



**Fig. 5.** Comparison of $C_p$ between turbine with and without passive control cylinder

The experimental and numerical coefficient of the moment is shown in Figure 6. The coefficient of moment value is inversely proportional to the tip speed ratio of the turbine. At tip speed ratio λ > 0.41 $C_m$ of the turbine with passive control, cylinder tends to be higher than the conventional turbine, however, at tip speed ratio λ < 0.41, this trend changes, as the moment of the conventional turbine tends to be higher than one with the passive control cylinder placed upstream. At λ ≈ 0.6, the difference between the two study methods reaches up to 23%, with the conventional turbine generating 10.42% of $C_m$ and turbine with a passive control cylinder generated 12.9% of $C_m$.

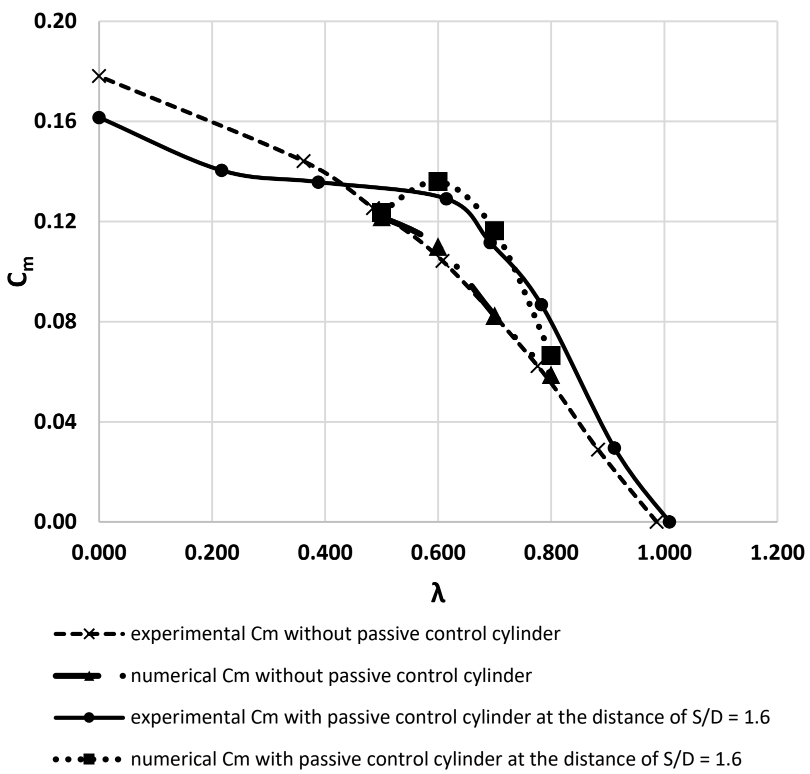



**Fig. 6.** Comparison of $C_p$ between turbine with and without passive control cylinder.

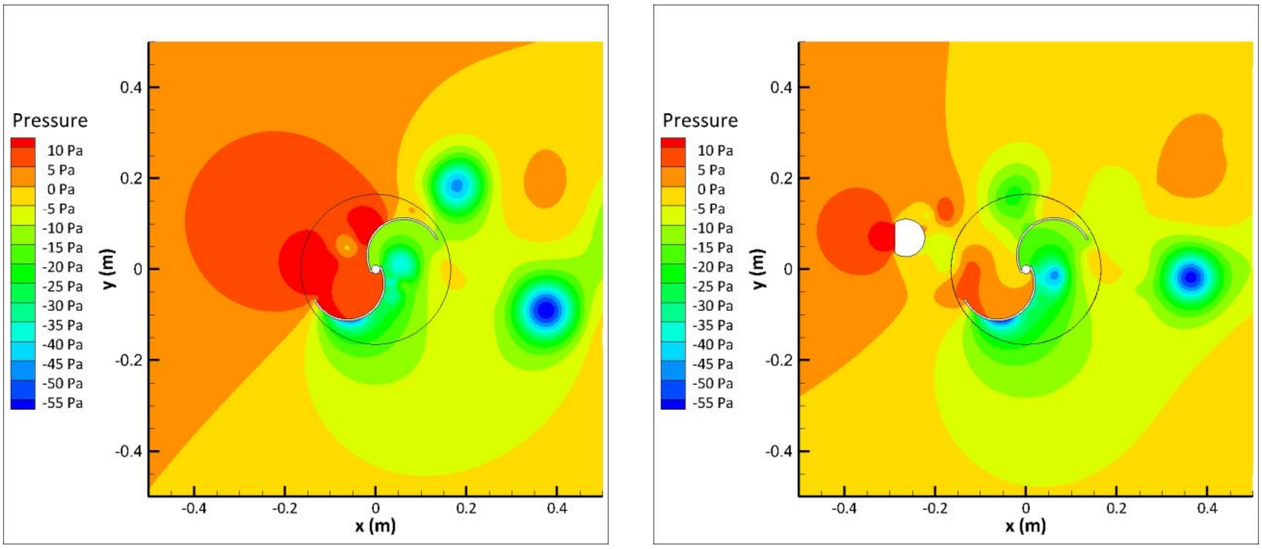
### *3.2 Pressure Analysis*

Savonius turbine, as a type of vertical axis wind turbine, utilises drag as a force to generate torque and thus power. The pressure of fluids around the turbine needs to be studied better to understand the reason behind the Savonius turbine performance improvement. Flow around the turbine with and without the passive control cylinder is compared to analyse the influence of the flow phenomenon on the drag force.

Figure 7 shows the comparison between pressure contour plot of 2 variations of passive control cylinder using numerical simulation conducted beforehand at Reynolds number = 95,000, at $\lambda$= 0.6, and rotation angle (θ) of 30°, in which the maximum static torque obtained based on research by Gunawan *et al*., [20]. Figure 7a shows the pressure contour around the turbine without a passive control cylinder, whereas Figure 7b shows the pressure contour around the turbine with a passive control cylinder place at S/D = 1.6.

It can be seen in Figure 7a that the pressure reaches up to 10 Pa at the stagnation point at the centre of the convex side of the returning blade. In contrast, the pressure at the concave side reaches a negative pressure of up to -40 Pa. The value indicates a pressure drag that counters the rotation of the turbine. Downstream, a low-pressure wake in a vortex street shape formed negative pressure.

On the other side, still in Figure 7a, high pressure of 10 Pa appears on the concave side of the advancing blade. Whereas on the other side of the blade, the negative pressure on the surface reaches up to -55 Pa at a very small area near the tip of the turbine. The dominant pressure value on the surface of the convex side of the turbine is around -40 Pa. This pressure difference induces drag force at the advancing blade. A very low-pressure vortex of -55 Pa emerges from shear stress on the surface of returning blade. This low-pressure region is expected to increase pressure drag even more.




a) b)

**Fig. 7.** Pressure contour of fluid around the turbine θ = 30°, λ = 0.6 at Reynolds number = 95,000; (a) without passive control cylinder (b) with passive control cylinder.

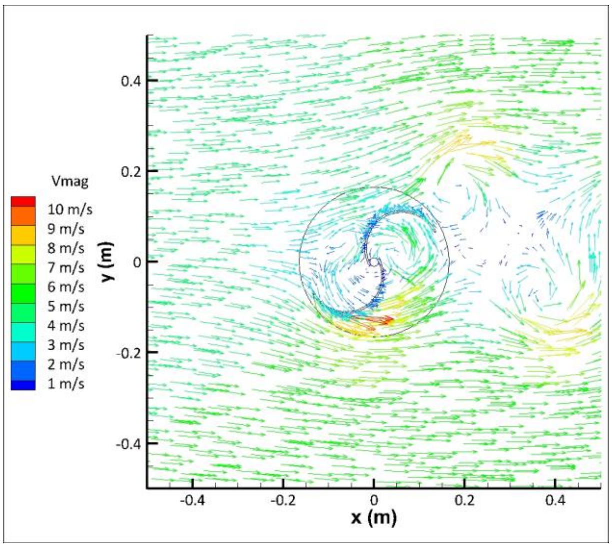
In figure 7b, it can be seen that on the surface of the convex side of the returning blade, there is no high-pressure region, especially at the area where the stagnation point should be. Exactly the opposite, a low-pressure area of -20 Pa appears on the concave side. Based on the phenomenon, the returning blade generates less drag pressure with the presence of the passive control cylinder. At the downstream side, a wake of low-pressure still appears. However, the area of this low-pressure wake becomes smaller. This phenomenon is expected to decrease the pressure drag as well.

However, on the concave side of the advancing blade in Figure 7b, a decrease in pressure occurs. The maximum pressure reaches up to more than 10 Pa on the turbine without passive control cylinder at the area mentioned. The value decreases to 5 Pa with the installation of the passive control cylinder. The difference in pressure between the concave and convex reduces; therefore, the drag force decreases.

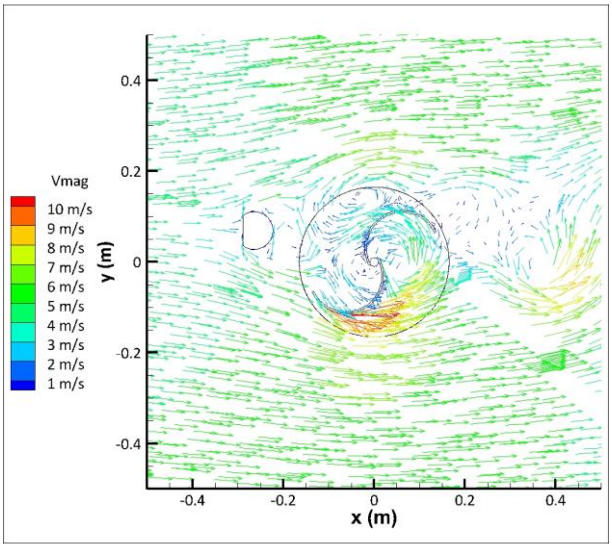
The phenomena described the effect of the overall performance of the turbine. In this case, the decrease of the pressure drag on the returning blade is expected to be more significant than the decrease of the drag force on the advancing blade. Therefore, the installation of a passive control cylinder increasing the turbine performance.

### *3.3 Velocity Vector Analysis*

Figure 8 shows the comparison of the flow velocity vector around the conventional turbine and the velocity vector around the turbine with the passive control cylinder installed. Each figure shows the turbine at rotation angle (θ) of 30° at Reynolds number = 95,000 and tip speed ratio λ = 0.6. Figure 8a shows the velocity vector around the conventional turbine, whilst Figure 8b shows the velocity vector around the turbine installing a passive control cylinder at S/D = 1.6.




a)




b)

**Fig. 8.** Velocity vector of fluid around the turbine θ = 30°, λ = 0.6 at Reynolds number = 95,000; (a) without passive control cylinder (b) with passive control cylinder.

In Figure 8a, the flow attaches to the returning blade exactly at the stagnation point. The boundary layer flow then separated near the tip. The separation induces the wake region downstream of the turbine. On the other side, the flow velocity increases significantly to up to 10 m/s right at the tip of the advancing blade.

In Figure 8b, it can be seen two vortices appear upstream of the turbine as a result of the presence of the passive control cylinder: the vortices, especially the one near the returning blade, affects the overall upstream flow. Compared to the conventional turbine, this passive control cylinder installation delays the flow attachment angle on the convex side of the returning blade. As a result, the flow attaches to the convex side of the returning blade at a steeper angle, from which the boundary layer flow shifts the separation angle steeper. In turn, the area of the wake shrinks compared to the flow around the wake of the conventional turbine, resulting in a decrease in the pressure drag on the returning blade.

## 4. Conclusions

The Savonius wind turbine is a suitable alternative to extract wind energy at places with low wind speed. However, due to its low efficiency, it is paramount to increase the efficiency of the turbine. In this study, the installation of passive control D-53° type cylinder at the distance 1.6D upstream of the returning blade has been aerodynamically expected to increase the overall performance of the turbine. Experimental results clearly indicated the improvement of the performance, especially the power generated by the turbine.

CFD simulation has been conducted to analyse the flow characteristics better. SST k-ω turbulence modelling applied to numerical consideration. Validation of the numerical result has been done to reassure the accuracy of the numerical condition and calculation. The overall result of $C_p$ and $C_m$ showed good agreement.

An improvement of maximum power by 24.56% was obtained with the installation of passive control D-53° type cylinder at the turbine rotating at tip speed ratio λ = 0.6. Simulation result with good agreement to the experimental result showed that the presence of the mentioned cylinder manipulates the flow around the turbine. A decrease of pressure upstream of returning blade as a wake of passive control cylinder and the delayed separation of boundary layer on the returning blade due to vortex formed upstream were the primary cause of the decrease of pressure drag.

However, a decrease of drag force on the advancing blade side occurred at the simulation result. The phenomenon happened as a factor of the presence of the passive control D-53° type cylinder. The mentioned cylinder caused a disturbance to the flow headed towards the concave side of the returning blade resulting in a pressure drop.

The decrease in drag pressure on the returning blade is more significant than the decrease in drag force on the advancing blade. Therefore, the Savonius turbine experienced improved power with the installation of passive control D-53° type cylinder placed 1.6D upstream of the returning blade.

**Acknowledgement**
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