DESIGN MODELING OF SAVONIUS-DARRIEUS TURBINE FOR SEA CURRENT ELECTRIC POWER PLANT

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Abstract
Turbines convert the kinetic energy of ocean currents into electrical energy produced by the sea current electric power plant. This study aims to design a power generator turbine modeling that is carried out using the Computational Fluid Dynamic (CFD) approach by comparing the geometric performance based on the angle of attack and the Tip Speed Ratio (TSR) value of the Savonius-Darrieus Turbine. Having done several trials and errors during collecting the data, the value of the TSR 1.427; 2.853; 4.28; 5; and 5.7 is proposed. Here, the NACA 0018 series has been adopted on the current design of Savonius-Darrieus Turbine. The turbine has three blades, length of the span 357 mm, the diameter of turbine 428 mm, and length of the hydrofoil chord 40 mm. Effect of various angle of attacks from 0°up to 10° has been taken into account in the computational to obtain the coefficient power for each variation. The results revealed that the turbine with an angle of attack of 5°and TSR value of 5.0 has higher power coefficient value by 0.469 as compared with its angle of attack of 10°. It should be noted here that the increase of the angle of attack up to 10° resulted in a significant reduction of the power coefficient value of 0.206 as the value of TSR about 4.28. The addition of the Savonius Rotor results in increasing efficiency of the turbine for sea current applications.

INTRODUCTION
Indonesia is a maritime-based nation which is two-third of the area are high seas. Seas are one of the biggest resources than Indonesia has from its seafood to its energy potential. According to Paris Convention to reduce greenhouse gas emissions then Indonesia plans to achieve 23% of renewable energy use by 2025 and reach 31% by 2050. One of the sources is to use the sea as a source of targeted current energy reach 3.1 GW in 2025 [1, 2, 3, 4].

In 2017, BPPT Hydrodynamics Technology Center (BTH-BPPT) conducted a double shaft turbine performance test at Suramadu Bridge, which observed the amount of rotor rotation and the electric power generated as a function of the speed and direction of ocean currents. The results of the study are an initial reference for further research about sea current electric power plant in Indonesia [5]. Research by Kanyako & Janajreh shows the comparisons of NACA blades profile and the number of the profile power coefficient. By this research, NACA 0018 was chosen because of its best capability to provide the best starting torque compared to NACA 0015 and NACA 0021. Three blades configuration was used because of its better power coefficient compared to four blades, even though two blades are better for its power coefficient. Meanwhile, two blades provide least starting torque compared the other two [6, 7, 8].

In this research, an investigation to analysis Savonius-Darrieus Turbine design using Computational Fluid Dynamics (CFD) approach was presented. The CFD Analyze 2D Model of the Savonius-Darrieus Turbine with variations of velocity for the environment and the angle of attack of the hydrofoil. The Savonius-Darrieus Turbine was chosen due to overcome negative
start off torque problem in the Darrieus Turbine [9] [10]. Basically, this research by changing the Darrieus Turbine to the Savonius-Darrieus Turbine aim for seeing the effect of Savonius Rotor addition with an angle of attack variation [11] [12].

METHOD

Geometry Modeling and Meshing

The geometry of the 2D model was created using Autodesk Inventor software and then exported to ANSYS Fluent to do the simulation. The geometry 2D model was used because of the unchanged surface towards height. The simplicity solution had done to minimize the use of the computer's RAM when meshing activity, so the high accuracy of the mesh and data collecting can be obtained.

The Geometry consists of two parts, as depicted in Figure 1, which is an enclosure and rotating part. Separation of the two parts had done to turn one of the parts and keep the other part to not turning. In this case, the turning part is rotating, which is representing the turbine itself. The enclosure part represents the free stream fluid or sea current in terminology.

After doing trials and errors when collecting the data, variations of the geometry were determined to find the most optimum geometry. This research simulates the turbine with variations of Tip Speed Ratio (TSR) and the angle of attack of the hydrofoil. Details of the variations of the geometry that determined to find the most optimum geometry [13] [14], as explained in Table 1.

![Figure 1. Turbine Model Geometry](image)

Table 1. Variations of Velocity of Turbines for Simulation

| v (m/s) | 1.5 | 0.75 | 0.5 | 0.428 | 0.375 |
|---------|-----|------|-----|-------|-------|
| ω (rad/s) | 10  |      |     |       |       |
| r (m)   | 0.214 |     |     |       |       |
| TSR     | 1.42 | 2.85 | 4.28 | 5     | 5.71  |

Where

v : Speed of the free stream flow (m/s)
ω : Rotational speed of the turbine (rad/s)
r : Radius of the turbine (m)
TSR : Tip Speed Ratio, calculated by v, ω, and r

Figure 2 shows the visualization of the meshing result. The meshing process settings use multi-zone (quad) to create the computation domain dominated by hexahedral or square mesh. This shape is ideal by its resolution and the efficiency when creating a detail mesh without using much element in total, so the use of the RAM will be saved [14]. While the dimension of the mesh, we choose maximum 3mm for the entire computation domain.

Computational Fluid Dynamic (CFD)

The numerical data solution using CFD method with ANSYS Fluent software had done to find each torque of the geometry variations. The input parameter is very important for the credibility of the data we obtained. For this research, the SST κ-ω models were used because of the ease of convergent data can be obtained with the flows that pressure gradients tend to be unnatural like compressors, pumps, turbine and other related things and details of the parameter shown in Table 2.

![Figure 2. Meshing Visualization](image)
Table 2. Parameter of Simulations

| Time          | Transient               |
|---------------|-------------------------|
| Viscous       | SST k-ω [14]            |
| Material      | Water liquid (ρ = 1025 kg/m³) |
| Cell Zone Conditions | Mesh Motion, (ω = 10 rad/s) |
| Inlet         | Velocity inlet (V, free stream velocity) |
| Mesh Interface| Matching (Interface between the enclosure and rotating part) |
| Initialization| Hybrid                  |
| Time step size| 0.01 s                  |
| Number of time step | 50                     |
| Max iteration / time step | 20, With total 1000 (50x20) iterations had done to obtain convergent data |

RESULTS AND DISCUSSION

After the simulation process was finished, data solutions of the fluid were processed using ANSYS results. Taken parameters in this simulation are the velocity distribution plot, pressure distribution plot, and the torque obtained by the turbine every 0.04 s of time step to measure the performance of the turbine quantitatively.

From Figure 3, the Velocity distribution plot shows that fluid flows in the inner Savonius rotor area tend to be slower (±1.017 m/s) than the outer rotor. Slower speed shows stagnation area or an area with higher pressure that produces force linearly with the turbine turning direction. Higher velocity (±3.044 m/s) founded in the outer region of the rotor.

Pressure distribution from Figure 4 shows that stagnation phenomenon, as explained from the previous paragraph. In the inner rotor Savonius, the pressure tends to be higher (±560.968 Pa) that produces force to turn the turbine rotor. While at the hydrofoil, higher pressure produced at the lower area, which produces lift force of the hydrofoil to contribute extra torque for the turbine globally.

Modelling, design and simulation of the turbine geometry with variations of the hydrofoil angle of attack were analyzed using CFD software ANSYS Fluent. Because of behavior of the fluids, the torque of the turbine was obtained.

Torque data collected per 0.04s of the time step, which is represented as 22.5° of the turbine position angles. Whereas performance of the turbine was determined by average torque of the turbine itself.
Torque values from Table 3, Table 4, and Table 5 concluded that the values changing related to its angle position. 180° turn of the turbine represents the torque values of the [17] turbine globally. The differences in torque values related to TSR values influenced by the parameter of the TSR itself as explained [17].

Table 3. Torque Values against Position Angle for 0° Angle of Attack

| $\alpha$ (°) | 1,427 | 2,53 | 4,280 | 5,000 | 5,707 |
| 0 | 58,766 | 11,484 | 0,440 | -6,313 | -10,476 |
| 22,5 | 53,347 | 8,081 | 1,666 | -6,313 | -7,809 |
| 45 | 40,779 | -1,899 | -2,283 | -6,191 | -6,009 |
| 67,5 | -11,27 | -11,60 | -5,157 | -2,926 | -2,825 |
| 90 | -22,41 | -14,05 | -2,692 | 0,666 | 0,902 |
| 112,5 | 13,477 | -4,464 | -6,368 | 0,263 | 0,994 |
| 135 | 35,288 | 10,195 | 2,734 | 8,730 | 3,738 |
| 157,5 | 46,152 | 17,415 | 10,081 | 13,470 | 8,532 |
| 180 | 33,153 | 20,737 | 15,833 | 13,470 | 11,682 |
| Average | 27,476 | 3,989 | 2,221 | 1,650 | -0,140 |

Table 4. Torque Values against Position Angle for 5° Angle of Attack

| $\alpha$ (°) | 1,427 | 2,853 | 4,280 | 5,000 | 5,707 |
| 0 | 48,104 | 10,296 | 5,609 | -0,271 | -2,594 |
| 22,5 | 36,764 | 6,470 | 5,432 | -1,243 | -1,514 |
| 45 | -5,161 | -0,852 | 0,454 | -1,556 | -3,252 |
| 67,5 | -25,54 | -6,516 | -0,773 | -0,462 | -2,035 |
| 90 | -13,74 | -6,182 | 0,393 | 2,503 | 0,066 |
| 112,5 | 20,469 | 3,869 | 0,507 | 9,045 | -0,042 |
| 135 | 39,899 | 12,788 | 3,150 | 12,322 | 3,256 |
| 157,5 | 36,753 | 15,849 | 9,742 | 8,690 | 8,884 |
| 180 | 23,206 | 16,737 | 14,153 | 3,927 | 11,042 |
| Average | 17,862 | 5,829 | 4,296 | 3,661 | 1,534 |

Table 5. Torque Values against Position Angle for 10° Angle of Attack

| $\alpha$ (°) | 1,427 | 2,853 | 4,280 | 5,000 | 5,707 |
| 0 | 48,104 | 10,296 | 5,609 | -0,271 | -2,594 |
| 22,5 | 36,764 | 6,470 | 5,432 | -1,243 | -1,514 |
| 45 | -5,161 | -0,852 | 0,454 | -1,556 | -3,252 |
| 67,5 | -25,54 | -6,516 | -0,773 | -0,462 | -2,035 |
| 90 | -13,74 | -6,182 | 0,393 | 2,503 | 0,066 |
| 112,5 | 20,469 | 3,869 | 0,507 | 9,045 | -0,042 |
| 135 | 39,899 | 12,788 | 3,150 | 12,322 | 3,256 |
| 157,5 | 36,753 | 15,849 | 9,742 | 8,690 | 8,884 |
| 180 | 23,206 | 16,737 | 14,153 | 3,927 | 11,042 |
| Average | 17,862 | 5,829 | 4,296 | 3,661 | 1,534 |

The coefficient of power obtained by the turbine produces against the potential energy that the fluid gives. Coefficient of power equations can be seen in (1) [17]. The results were expressed in Table 6.

\[
C_{\text{power}} = T \omega / (\frac{1}{2} Av^3)
\]

Where
- $C_{\text{power}}$ : Coefficient of power
- $\omega$ : angular velocity (rad/s)
- $\rho$ : density of sea water (1025 kg/m$^3$)
- $A$ : Turbine Swept Area (m$^2$)
- $V$ : sea water flow speed (m/s$^2$)

From the coefficient power calculation results, the most optimum efficiency is from turbine geometry with a 5° angle of attack valued 0.469. Meanwhile, the turbine with the smallest efficiency was obtained by turbine geometry with a 10° angle valued 0.206. The comparisons between the results are shown in Figure 5.

Table 6. Calculation Results of the Power Coefficient

| $\alpha$ (°) | 1,427 | 2,853 | 4,280 | 5,000 | 5,707 |
| 0° | 0.082 | 0.095 | 0.178 | 0.211 | -0.027 |
| 5° | 0.053 | 0.139 | 0.345 | 0.469 | 0.292 |
| 10° | 0.028 | 0.082 | 0.206 | 0.183 | 0.034 |

Figure 5. Comparisons between $C_p$ and the TSR for the Geometries

The addition of the Savonius Rotor results in increasing efficiency of the turbine for sea current applications and wind power [11, 16, 18]. From the same geometry and variations, the Darrieus Turbine obtained 0.228 of power coefficient when the TSR valued 5.7. Meanwhile, Savonius-Darrieus Turbine can obtain as high as 0.469 of power coefficient, which is almost twice as much power can be obtained with the same geometry. The comparisons between the two turbines can be seen in Figure 6.
The character of the power of that both turbines produces is the key why Savonius-Darrieus Turbine can be more efficient than the Darrieus Turbine. Figure 7 shows that Savonius-Darrieus Turbine produces power more evenly distributed when angle position at 90°-270°. Meanwhile, the Darrieus Turbine produce power when the angle position at 90°-157.5° and then 202.5° -270°.

Figure 7. Cp Obtained from Various Angle Position

Table 7. The effects of the Flow Speed against Turbine Power

| V (m/s) | \( \omega \) (rad/s) | \( P_{_{water}} \) (Watt) | \( P_{_{turbine}} \) (Watt) |
|--------|---------------------|----------------------|----------------------|
| 1      | 23.364              | 338.137              | 158.507              |
| 1.25   | 29.206              | 660.424              | 309.584              |
| 1.5    | 35.047              | 1114.212             | 534.961              |
| 1.75   | 40.888              | 1812.203             | 849.498              |
| 2      | 46.729              | 2705.096             | 1268.055             |

Table 8. Dimensions of The Design

Dimensions of the Darrieus Rotor
- Diameter: 428 mm
- Span Length: 357 mm
- Angle of Attack: 5°
- Number of Blades: 3
- Hydrofoil Geometry Type: NACA 0018
- Chord Length: 40 mm

Dimensions of the Savonius Rotor
- Diameter: 304 mm
- Span Length: 300 mm
- Number of Blades: 2
- Bucket Spacing: 0 mm

CONCLUSION
This research concluded that the design of the Savonius-Darrieus Turbine has optimum power coefficient 0.469. Then the coefficient was achieved when the turbine TSR value is five, and the angle of attack is 5°. The drawings for this turbine are finished and ready for full-scale production.

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