Simulation Study on Cross Flow Turbine Performance with an Angle of 20 ° to the Variation of the Number of Blades

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***Abstract*—Water turbine is one of the important components in the development of renewable energy in Indonesia. Indonesia has the potential of 94 GW to be sourced from hydro energy. The development of the water turbine will positively impact the development of renewable energy in Indonesia. On the other side would help achieve the government's target on the composition of new renewable energy was increased to 23% - 31% in 2050. This study has used a water turbine type of cross-flow and simulation using the ANSYS CFX Solver Application. The purpose of the research that has been carried out is to determine the best number of blades in the cross-flow turbine to produce maximum performance in the water turbine. The runner design used has 80 mm in diameter, 130 mm in length, and 20° blade angle. Variations in the number of blades were carried out on blades 32, 36, 40, 44, and 48. This simulation has been carried out under steady-state conditions and the turbulent Shear Stress Transport type. The cross-flow turbine operates at a water velocity of 3m / s. The results show that the Coefficient of Power Maximum generated for blades 32, 36, 40, 44, and 48 is 20%, 21.85%, 18.96%, and 13.47%. These results show that the cross-flow type hydro-turbine generates the maximum performance with a runner with a number of blades 40.**

***Index Terms*—Renewable energy, hydro-turbine, coefficient of power, computational fluid dynamics**

I. Introduction

Indonesia is one of the countries that have enormous new and renewable energy potential. Table I shows the data potential of Indonesia's renewable energy sources obtained from Indonesia's energy outlook [1]. However, to supply the national electricity demand, renewable energy sources are only 14% [2]. There have been many renewable energy developments, such as analyzing hydro energy sources, solar thermal energy, wind energy, and bioenergy [3]. Table I shows that hydro energy has tremendous potential among other energy sources. This shows an opportunity for research in hydropower to develop new and renewable energy in Indonesia. It will help achieve the program. The government decides to target renewable energy use by 23% - 31% in 2050 [4].

TABLE I. Potential of Renewable Energy Source [1]

Types of Renewable Source	Potential
Hydro	94,3 GW
Thermal	28,5 GW
Bio Energy	32,6 GW
Solar	207,8 Gwp
Wind	60,6 GW
Marine energy	17,9 GW

Water turbines are the primary means of developing hydroelectric power. Water turbines are divided into axial flow water turbines and cross-flow water turbines shown in Fig. 1. The axial flow turbine consists of a horizontal axis and an inclined axis turbine. In contrast, the cross-flow turbine type consists of a vertical axis and an in-plane axis turbine[5]. Several types of turbines included in the cross-flow turbine are savonius turbines, darrieus turbines, and banki turbines.

Hydro-turbine research is significantly developed in recent years. The research was carried out on the Darrieus turbine [6], Kaplan turbine [7], Savonius turbine [5], and

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cross-flow water turbine [8]. The variations included blade shape [9], diameter ratio [10], number of stages [5], and thickness ratio [11], either experimentally or numerically. This research was conducted to get the best turbine performance, as seen from the coefficient of power generated.

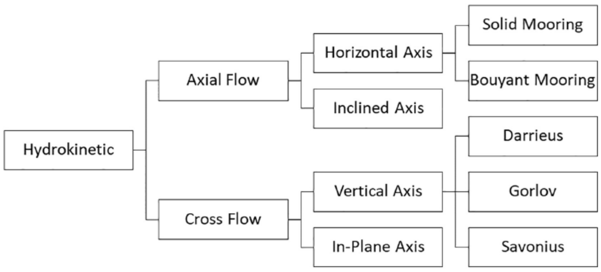

Figure 1. Classification of turbine [5].

The cross-flow type turbine is one of the hydro-turbines and can be used as a micro-hydropower plant [12-13]. Some of the factors that affect the cross-flow turbine's performance are the number of blades, runner diameter, thickness, nozzle, and nozzle entry [14]. This research is to analyzes the effect of geometry on the performance of water turbines.

Factors that affect turbine performance include geometry, fluid properties, and turbine material [15]. The material factor used in the turbine influences the performance, structural strength, and cost-efficiency of the turbine manufacturing process. The availability of raw materials also has an important influence on the design [16].

This research was conducted on a cross-flow turbine design with an angle of 20 ° with a variation of blades 32, 36, 40.44, and 48. This study was conducted to determine the number of blades most effective in producing the hydro turbine's best performance. This research was carried out at a fluid velocity of 3 m / s.

II. Runner Design and Data Reduction

The turbine used in this study is a type of water turbine cross-flow with blades 32, 36, 40, 44, and 48. The study was conducted by simulating the three-dimensional object. Fig. 2 shows the three-dimensional runner design for each of the variations in the number of blades. Fig. 2 Shows the runner with a) 32 blades, b) 36 blades, c) 40 blades, d) 44 blades, and e) 48 blades.

Fig. 3 shows the dimensional position for the runner on top projection. Description of Runner dimension is the outer diameter (D1) 80 mm, the inner diameter (D2) 60 mm, the runner length 130 mm, the blade radius (rr) 10 mm with a slope (ϴ) 20 ° and the thickness of the runner 1, 5 mm.

The equation used in calculating the performance of the hydro-turbine cross-flow is as follows:

$$\omega = \frac{2\pi n}{60} \quad (1)$$

$$P_{Rotor} = T \times \omega \quad (2)$$

$$P_{Available} = 0{,}5 \times \rho \times A \times V^3 \quad (3)$$

$$Cp = \frac{P_{Rotor}}{P_{Available}} \quad (4)$$

Where ω is the Angular velocity, n is the rotational speed, T is the torque, A is the runner projection area, ρ for the density of water, V is the velocity of Fluid Flow, and Cp is the Coefficient of Power. [17].

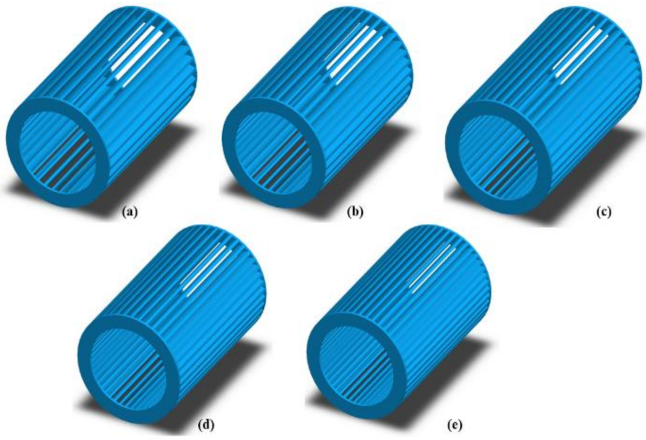

Figure 2. 3D runner design with a) 32 blades, b) 36 blades, c) 40 blades, d) 44 blades and e) 48 blades.

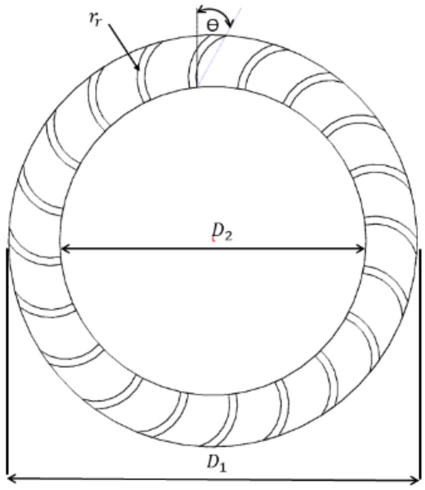

Figure 3. a description of the dimensions in the runner design.

III. Numerical Modeling

This cross-flow hydro turbine research was conducted using the CFD method, using Ansys Student Version 2021R software with CFX Solver. The hydro-turbine was simulated using three-dimensional modeling. The runner axis is horizontal, and the inlet fluid flows from the top of the runner.

The first step in this study was to conduct a mesh independence study. A Mesh independence study was performed by making changes to the number of elements [5]. The present study has been conducted in six mesh settings. The results obtained are shown in Fig. 4. From this graph, the fifth mesh adjustment is used, with the number of elements 990,991. Because starting at the fifth set, the results of the simulation have not changed.

This research begins with the process of validating the boundary conditions. This process is the process of duplicating a study, then comparing the results [5]. The

Validation results are shown in Fig. 5. Validation is done by comparing experimental research used as benchmarking in Makarim et al.'s [13]. The average difference in the benchmarking value is 6%, where this value is acceptable because it is less than 10%.

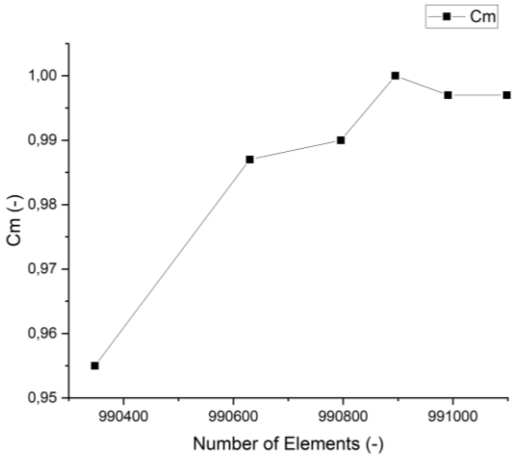

Figure 4. Graph of the relationship between the number of mesh and the Moment Coefficient.

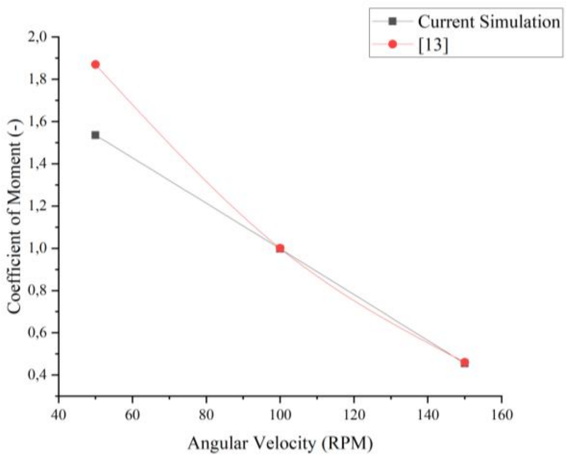

Figure 5. Validation of relationship between Rotation Speed with Moment Coefficient.

The simulation process is carried out in the Ansys workbench. Meshing on the domain uses the tetrahedron method [18] and uses inflation on the runner walls and stationary domain walls. Inflation is used on solid parts that are in contact with the fluid [19].

The meshing results in the rotary domain are shown in Fig. 6. The stationary domain are shown in Fig. 7. The average skewness in the meshing process is 0.3. This value falls into the category of good mesh quality. Good mesh quality is when the skewness value is less than 1 [20].

Fig. 10 shows the schematic and boundary conditions of the modeling. The inlet boundary condition is that the fluid used is water, with a density of 1000 kg / m3 and a speed of 3 m / s. Water flow is a subsonic flow. The wall boundary uses the wall boundary condition with the assumption of a no-slip condition. At the outlet, the opening limit condition is used with a pressure of 1 atm. The stationary domain interfaces and the rotary domain link the two domains with fluid to fluid conditions [21]. The simulation is carried out at a steady and uses the turbulent Shear Stress Transport type. The use of turbulent type Shear Stress Transport has better accuracy than other turbulent types [22]. The simulation is carried out at a rotation speed of 50 to 350 RPM

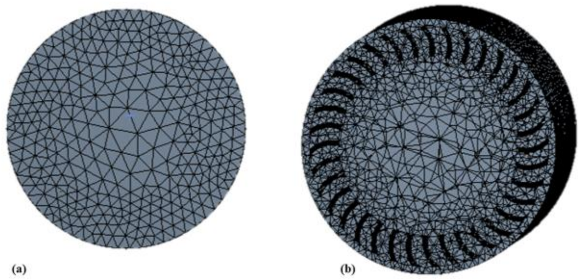

Figure 6. The meshing of Rotary Domain.

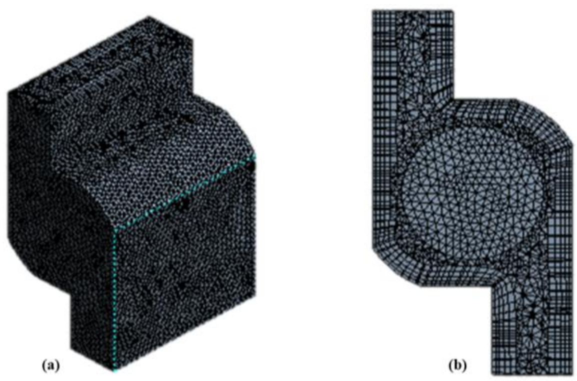

Figure 7. meshing of Stationary Domain.

The modeling schematic consists of a rotary domain and a stationary domain. Fig. 8 shows the rotary domain, consisting of up and down interfaces, interfaces, and wall runners. The stationary domain is shown in Fig. 9, consisting of inlet, wall, interface, and outlet.

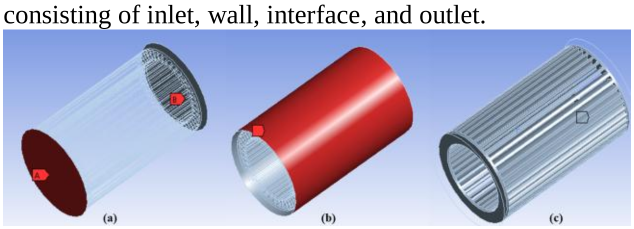

Figure 8. Rotary Domain a) interface up and down, b) interface, and c) runner.

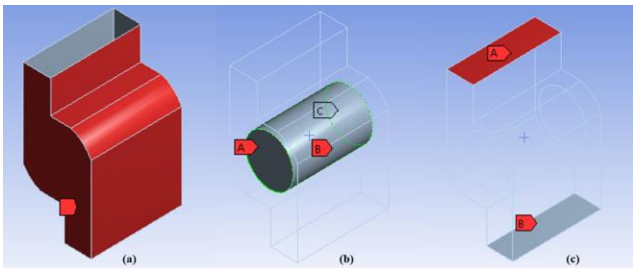

Figure 9. Stationary Domain a) Wall, b) interface, and c)inlet and outlet.

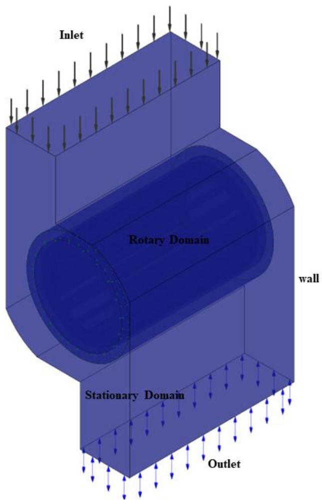

Figure 10. Schematic and boundary condition of simulation.

IV. RESULT AND DISCUSSION

Fig. 11 shows the Cp generated from the runner with all the variations in the number of blades. The runner with 32 blades shows that Cp increases up to rotational speed <150 rpm and decreases at rotational speed> 150 rpm. Cpmax generated by 20%. In the runner with 36 blades, the resulting Cpmax is 21.85% at an angular velocity of 200 RPM. With a number of blades 40, the runner increases at rotational speed <200 rpm then decreases at rotational speed> 200 rpm. The resulting Cpmax was 22.90%. For blades 44 and 48, Cp increases at rotational speed <150 rpm then decreases at rotational speed > 150 rpm. The Cpmax produced by the runner with 44 blades was 18.96%, and the runner with 48 blades was 13.47%.

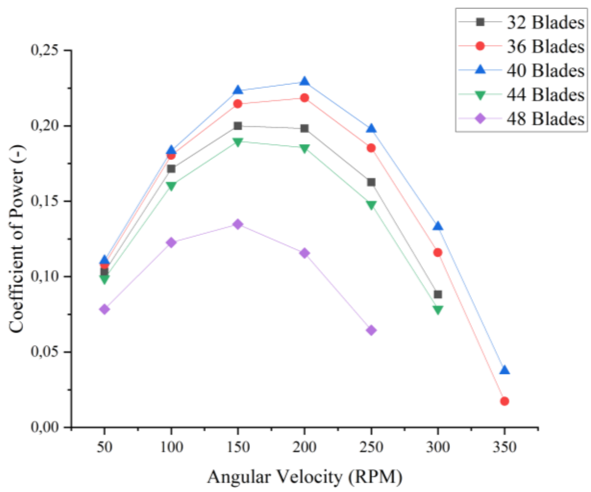

Figure 11. Graph of coefficient of power.

The coefficient of the moment is shown in Figure 12. All runner types show that C_m decreases as the angular velocity increases. Runners with a number of blades 40 have a higher C_m than other runners. Each runner produces C_{mmax} of 1.48 (32 blades), 1.55 (36 blades), 1.59 (40 blades), 1.41 (44 blades), and 1.12 (48 blades) .

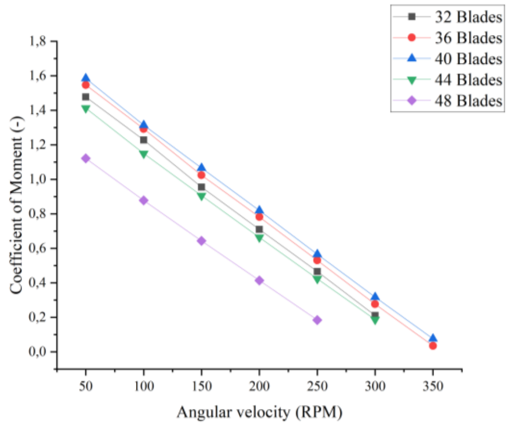

Figure 12. Graph of coefficient of moment.

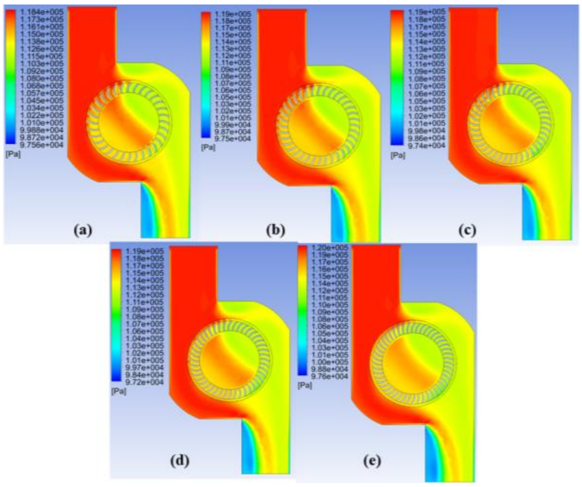

Figure 13. pressure contours of runner a) 32 blades, b) 36 blades, c) 40 blades, d) 44 blades, and e) 48 blades.

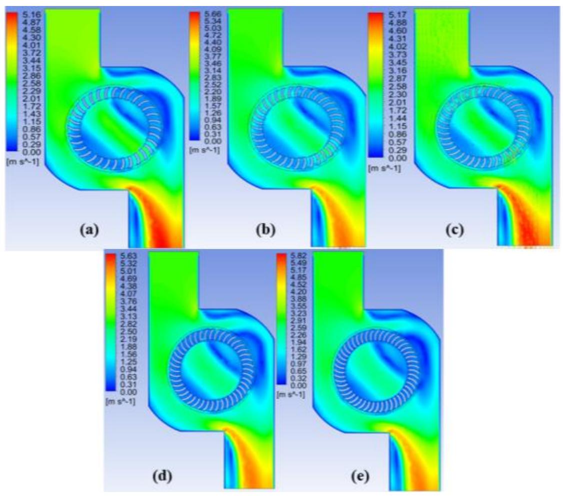

Figure 14. velocity contours of runner a) 32 blades, b) 36 blades, c) 40 blades, d) 44 blades, and e) 48 blades.

Runners with 40 blades have the best performance. The runner produced a Cpmax of 22.9%, which is the highest Cpmax compared to other variations. The runner with 40 blades has a wide angular velocity interval, which is 50-350 rpm.

In addition to generating C_p values, this simulation produces contour and vector output. The velocity vector for each type of runner at the C_{pmax} value is shown in Fig. 13. All pressure contours show the same pattern, where the highest pressure is at the fluid inlet side. Fig. 14 shows the velocity contour, and Fig. 15 shows the velocity vector. Figs. 14 and 15 show the velocity of water flow through stage 1 and out at stage 2. Figures show that in stage 1, not all of it is filled with water. Some are filled with air. This case shows that the runner motion is not entirely affected by stage 1. However, Fig. 13 shows the pressure contour with a high value on the inlet side. This figure shows that the runner movement is due to the pressure distribution on the inlet of the runner. It can be seen that the runner with the number of blades of 40 has an area that receives more pressure, which results in greater torque. The runner with 40 blades has the tightest speed density around the rotary domain. The outlet side has a red contour wider than the number of other blades.

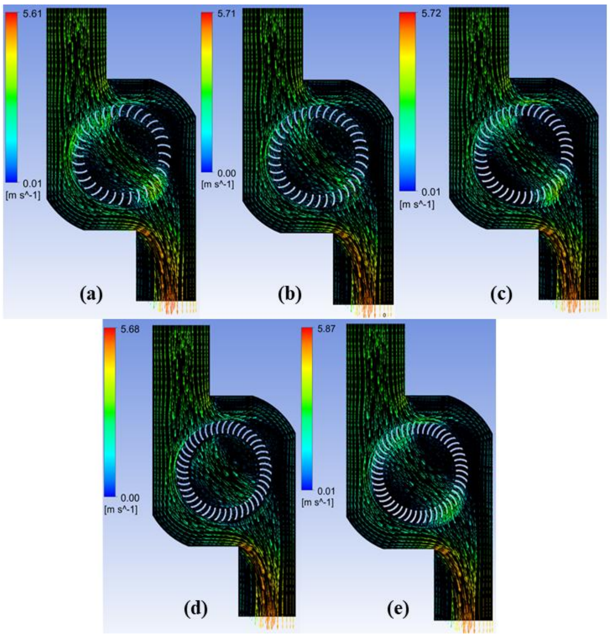

Figure 15. the velocity vector of runner a) 32 blades, b) 36 blades, c) 40 blades, d) 44 blades, and e) 48 blades.

V. CONCLUSION

The simulation results show that the number of blades on the runner affects the hydro-turbine cross-flow performance. The C_{pmax} results are shown in Fig. 10 shows that the best value is obtained in the hydro-turbine cross-flow with the number of blades 40 of 20.4%. A cross-flow hydro turbine with a number of blades of 40 can be used as an alternative design that can be used to get good performance

CONFLICT OF INTEREST

The authors declare no conflict of interest.

AUTHOR CONTRIBUTIONS

Purwanto: designing model; analysis; writing the draft manuscript; approving the final version of the manuscript. Budiono: supervising manuscript writing, checking analysis results, supporting the final version of the manuscript. Hermawan: supervising manuscript writing, checking analysis results, supporting the final version of the manuscript. Dandun Mahesa prabowoputra: conducting benchmark and analysis; writing the draft manuscript; approving the final version of the manuscript; and checking computational estimation.

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