

Design and performance of very low head water turbines using a surface vorticity model algorithm

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ABSTRACT

This study explores the numerical optimization of water turbine runner profile performance using a surface vorticity model algorithm. The turbine is designed on a laboratory scale and operates at a net head of 0.09 m, 400 rpm, and a water flow rate of 0.003 m³/s. The initial design of the turbine runner was optimized to minimize losses in the hydrofoil. The optimization algorithm is coded in MATLAB software to obtain the optimal stagger angle that will be used in the water turbine design. Furthermore, design validation was performed using computational fluid dynamics analysis ANSYS CFX to determine the water turbine performance. The settings used in ANSYS CFX include the reference pressure of 1 atm, turbulence model shear stress transport, and inlet boundary conditions using total pressure and static pressure outlet boundary conditions. The computational fluid dynamics analysis reveals that by optimizing the design, the efficiency of the water turbine increases by approximately 2.6%. The surface vorticity model algorithm can be applied to optimize the design of the water turbine runner.

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1. INTRODUCTION

The development of small and medium scale hydroelectric power plants in Indonesia is currently growing rapidly. This situation arises because the potential is quite large and is in line with the government's program to develop renewable energy resources. Geographically, Indonesia is an archipelagic country. The unequal distribution of electricity load centers as well as the low level of electricity demand in several regions are factors that hinder the supply of electrical energy on a national scale. Facilities in disadvantaged, frontier, and outermost areas are particularly disadvantaged. The decreasing availability of fossil energy sources and increasing awareness to preserve the environment will encourage the increased use of alternative energy sources. Pico and micro hydro power plants are also widely developed in developing countries [1]–[4].

The water turbine is one of the main components of a hydropower system. Thus, good turbine design is necessary to endow the generator with high efficiency. One method to optimize the design of the water turbine runner is to use surface vorticity model analysis which is a boundary integral method for evaluating fluid flow. The surface vorticity model has been developed and applied as a predictive tool for various engineering problems, such as for handling potential flows for any situation, including lifting bodies. Surface vorticity modelling offers the advantage of being the most natural of all boundary integral techniques [5].

In this study, an analysis of the runner profile of a two-dimensional water turbine was performed because of the advantage of computing speed relative to a more complex three-dimensional analysis. Potential flow analysis using a surface vorticity model is applied to minimize losses in the runner profile. Several other optimization methods related to potential flow analysis of the runner turbomachinery profile were conducted using a viscous vortex lattice method analysis [6], multiphase large eddy simulations [7], [8], computational fluid dynamics analysis [9]–[14] and experimental analysis [15]–[17]. In this work, design validation was implemented to determine performance using computational fluid dynamics analysis ANSYS CFX.

2. MATERIALS AND METHODS

2.1. Turbine design

In this study, the water turbine runner design employed is a propeller turbine type because the turbine can operate from very low to medium head [18]–[20]. The turbine is designed on a laboratory scale with runner specifications as shown in Table 1. The design of the main turbine dimensions and the basic shape of the runner profile uses a speed triangle analysis approach on a two-dimensional profile as shown in Figure 1. Figure 1 presents a sectional profile of a water turbine runner that is periodic in the x-axis direction. Such an approach is necessary because the nth element in all aerofoils will have the same vortex strength. Figure 1 shows a velocity triangle that occurs in the profile of a runner with a stagger angle (ξ), water flow angle (β), absolute velocity (C), relative velocity (W), tangential velocity (U), pitch (t) and chord (l). Subscript 1 is on the inlet side, and subscript 2 is on the outlet side.

Table 1. Specification and dimensions of the turbine runner

Symbol	Value	Description
H_{net}	0.09 m	Head netto
Q	0.003 m ³ /s	Debit
n	400 min ⁻¹	Rotational speed
n_q	133.33 min ⁻¹	Specific speed
C_m	0.565 m/s	Meridional speed
D_1	0.0949 m	Outer diameter of turbine runner
D_N	0.0475 m	Diameter of hub
z	5	Number of propeller blades

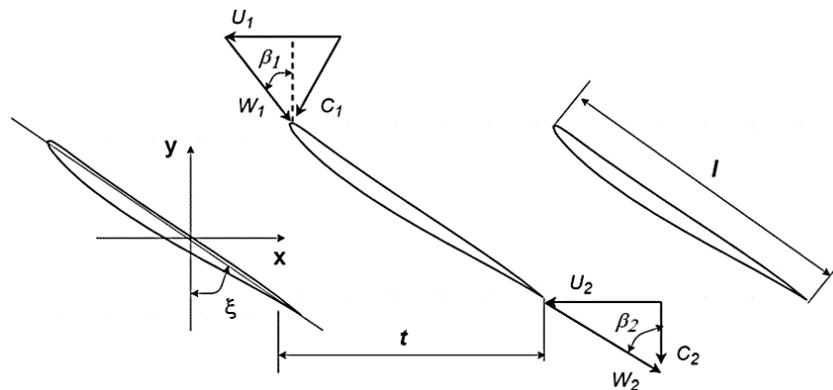


Figure 1. Velocity triangles of cascade geometry for a runner turbine

2.2. Potential flow analysis

Surface vorticity modelling offers advantages over turbine runner profile panels which are actually direct simulations of ideal fluid flow. This method is the most natural of all boundary integral techniques. Fluid flow passing through a two-dimensional water turbine runner profile in a plane (x,y) with a uniform flow velocity (W_∞) and tilt angle (β_∞) were analysed with this surface vorticity model. Figure 2 shows a flow diagram for runner potential flow analysis. The outlet flow angle (β_2), which is the output of the surface vorticity model algorithm, will be used to calculate the new stagger angle (ξ).

Some of the equations for calculating the potential flow analysis on the water turbine runner profile are as in (1) to (12):

$$\Delta S_n = \sqrt{[(X_{n+1} - X_n)^2 + (Y_{n+1} - Y_n)^2]} \quad (1)$$

$$\beta_n = \text{arc tan} \left(\frac{y_{n+1} - y_n}{x_{n+1} - x_n} \right) \tag{2}$$

$$x_n = \frac{1}{2} (X_{n+1} + X_n) \tag{3}$$

$$y_n = \frac{1}{2} (Y_{n+1} + Y_n) \tag{4}$$

$$K(s_m, s_n) = \frac{\Delta s_n}{2t} \left\{ \frac{\sin \frac{2\pi}{t} (y_m - y_n) \cos \beta_m - \sinh \frac{2\pi}{t} (x_m - x_n) \sin \beta_m}{\cosh \frac{2\pi}{t} (x_m - x_n) - \cos \frac{2\pi}{t} (y_m - y_n)} \right\} \tag{5}$$

$$K(s_m, s_m) \approx -\frac{1}{2} - \frac{1}{8\pi} (\beta_{m+1} - \beta_{m-1}) \tag{6}$$

$$K(s_{opp}, s_m) = -\frac{1}{\Delta s_{opp}} \sum_{\substack{n=1 \\ n \neq opp}}^M K(s_n, s_m) \Delta s_n \tag{7}$$

$$\sum_{n=1}^M K(s_m, s_n) \gamma(s_n) = K(n, 1) \gamma(s_1) + \dots + (K(n, te) - K(n, te + 1)) \gamma(te) + \dots + K(n, M) \gamma(M) \tag{8}$$

$$rsh_m = -U_\infty \cos \beta_m - V_\infty \sin \beta_m \tag{9}$$

$$C_p = 1 - \left\{ \frac{\gamma(s)}{W_\infty} \right\}^2 \tag{10}$$

$$\beta_2 = \text{arc tan} \left\{ \left(\frac{1 - \frac{\Gamma_v}{2t}}{1 + \frac{\Gamma_v}{2t}} \right) \tan \beta_1 - \left(\frac{2}{1 + \frac{\Gamma_v}{2t}} \right) \frac{\Gamma_u}{2t} \right\} \tag{11}$$

$$C_{L\infty} = 2 \left(\frac{t}{l} \right) (\tan \beta_1 - \tan \beta_2) \cos \beta_\infty \tag{12}$$

with runner profile input data coordinates (X_n, Y_n) , element lengths (ΔS_n) , profile slopes (β_n) , pivotal points (x_n, y_n) , coupling coefficients $K(s_m, s_n)$, right hand sides (rsh_m) , the back-diagonal correction $K(s_{opp}, s_m)$, the Kutta-condition $K(s_m, s_n) \gamma(s_n)$, the surface pressure coefficient (C_p) , and the lift coefficient $(C_{L\infty})$ [5].

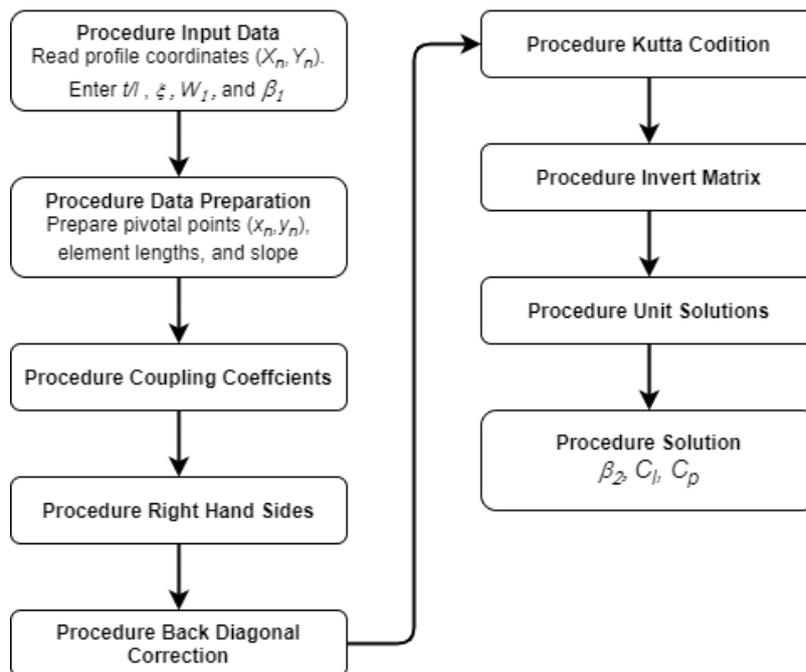


Figure 2. Flow diagram for runner potential flow analysis

2.3. Analysis of computational fluid dynamics (CFD)

CFD is a science that studies how to predict fluid flow, heat transfer, chemical reactions and other phenomena by solving mathematical equations to produce three-dimensional data. The continuity equation and the Navier-Stokes equation in cylindrical coordinates are described in the (13) to (16):

$$\frac{\partial v_r}{\partial r} + \frac{\partial v_z}{\partial z} + \frac{v_r}{r} = 0 \quad (13)$$

$$v_r \frac{\partial v_\theta}{\partial r} + v_z \frac{\partial v_\theta}{\partial z} - \frac{v_r v_\theta}{r} = \nu \left(\frac{\partial^2 v_\theta}{\partial r^2} + \frac{1}{r} \frac{\partial v_\theta}{\partial r} - \frac{v_\theta}{r^2} + \frac{\partial^2 v_\theta}{\partial z^2} \right) \quad (14)$$

$$v_r \frac{\partial v_r}{\partial r} + v_z \frac{\partial v_r}{\partial z} - \frac{v_\theta^2}{r} + \frac{\partial \rho}{\rho \partial r} = \nu \left(\frac{\partial^2 v_r}{\partial r^2} + \frac{1}{r} \frac{\partial v_r}{\partial r} - \frac{v_r}{r^2} + \frac{\partial^2 v_r}{\partial z^2} \right) \quad (15)$$

$$v_r \frac{\partial v_z}{\partial r} + v_z \frac{\partial v_z}{\partial z} + \frac{\partial \rho}{\rho \partial z} = g + \nu \left(\frac{\partial^2 v_z}{\partial r^2} + \frac{1}{r} \frac{\partial v_z}{\partial r} + \frac{\partial^2 v_z}{\partial z^2} \right) \quad (16)$$

where v_θ , v_r and v_z are the tangential, radial, and axial velocity components, ρ is the density of water, g is gravity and ν is the kinematic viscosity.

The main advantage of CFDs is their ability to quickly produce results at low cost, thereby making them especially suitable for optimization [21]. However, CFDs also require rigorous quantitative validation by physical models before they are used for design purposes because the results from CFDs can be higher than those from real experimental conditions [22]–[24].

Numerical simulations were performed on ANSYS CFX with a three-dimensional water turbine runner model and hexahedral mesh elements. The settings used include the reference pressure of 1 atm and turbulence model of shear stress transport. The inlet boundary conditions include total pressure and static pressure outlet boundary conditions. High resolution type turbulence numeric with double precision were employed. Given the axisymmetric shape of the runner, the analysis in this work utilized the turbo mode, namely, modelling with one propeller blade.

3. RESULTS AND DISCUSSION

3.1. Runner potential flow analysis

The runner is the part that directly converts the potential energy contained in the water into torsional energy on the turbine shaft. Accordingly, the initial design of the water turbine runner in this work was optimized using a surface vorticity model algorithm on the runner profile coded in MATLAB. The output of the algorithm aims to obtain the outlet flow angle (β_2) so that the optimal stagger angle (ξ) can be calculated and the losses that occur in the hydrofoil can be minimized. The water turbine runner blade is divided into five segments: segment 1 is near the side of the hub, segment 3 is in the middle of the runner and segment 5 is on the outer side of the runner. The results of the surface vorticity model algorithm are shown in Table 2. For the type 1 turbine, the initial design was achieved using the speed triangle analysis method on a two-dimensional runner profile as shown in Figure 1. The type 2 turbine is a turbine design that was optimized by employing the surface vorticity model algorithm.

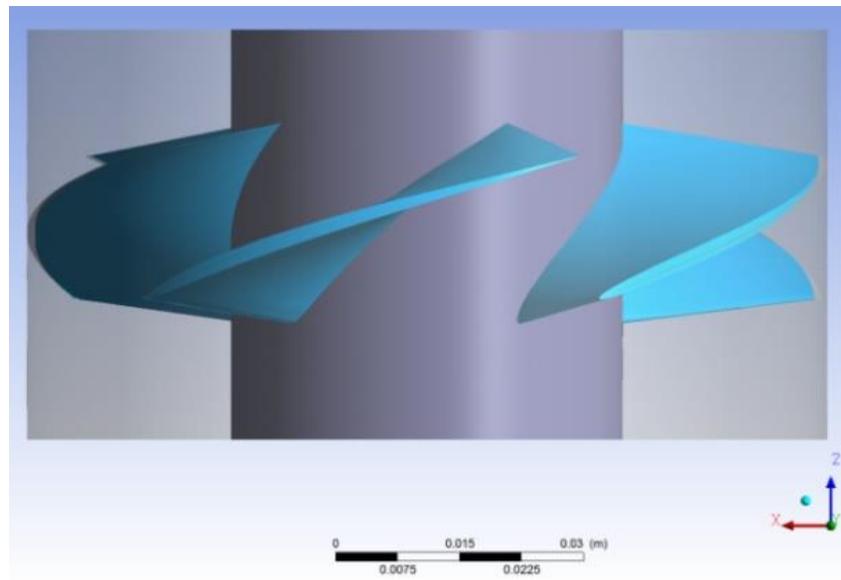
Table 2. Comparison of design stagger angles and optimization results

	Stagger angle (ξ)	Segment 1	Segment 2	Segment 3	Segment 4	Segment 5
Turbin type 1	ξ_{initial}	47.48	58.99	65.48	69.64	72.56
Turbin type 2	ξ_{optimasi}	23.71	50.91	62.17	68.33	73.01
Difference	$\Delta \xi$	23.77	8.08	3.31	1.31	-0.45

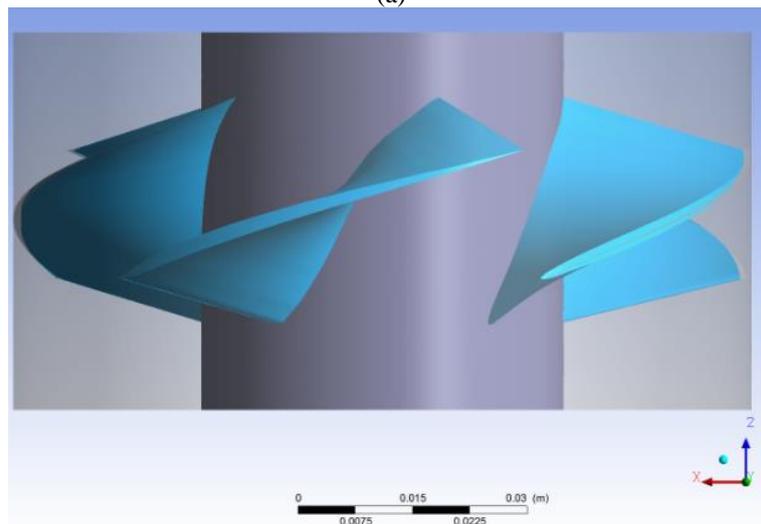
3.2. Water turbine runner modelling

The runner blade designs of types 1 and 2 turbines are then modelled using the ANSYS Design Modeler (Figure 3). As shown in Table 2, the initial stagger angle (ξ_{initial}) whose value is 47.48° , is optimized (ξ_{optimasi}) to become more upright to 23.77° for Segment 1. For Segment 2, the stagger angle difference ($\Delta \xi$) is approximately 8° . For Segment 5, the stagger angle is optimized to be slightly larger than that of the initial design. The difference in the slopes of the runner blades of the two designs are presented in Figure 3. Figure 3(a) shows isometric 3D view of the blade from the initial design. The stagger angles from segments 1 to 5 of both designs increase moderately. Figure 3(b) shows isometric 3D view of the blade from the optimization results. From Figure 3(b), which is an optimized type 2 turbine, it can be seen that the stagger angle profile is more upright when compared to the initial design type 1 turbine as shown in Figure 3(a). The

stagger angle for segment 1 to segment 4 turbine type 2 is smaller than turbine type 1 as shown in Table 2. While in segment 5, the type 2 turbine in Figure 3(b), has a slightly flatter stagger profile than the type 1 turbine in Figure 3(a).



(a)



(b)

Figure 3. Isometric 3D view of the blade from the (a) initial design and (b) optimization results

Furthermore, the design of runner blade types 1 and 2 were analyzed for their performance using the commercial ANSYS CFX CFD. ANSYS CFX works on the basis of finite volume method on object elements. The three-dimensional volume computing domain using hexahedral meshing is employed to evaluate the water turbine runner which is designed to have five blades. This analysis utilized the turbo mode so that only one runner blade is modelled. This mode is applied because the geometry of the analyzed runner blades is axis-symmetric so as to hasten the simulation process. Meshing and grid boundary conditions in the turbo mode runner blade model are shown in Figure 4. In the CFX, the pre-boundary and continuum conditions are defined to determine the inlet, outlet, runner blades, hub and shroud. Figure 4 (a) shows a one runner blade model meshed with a hexahedral structured pattern with the aim of accelerating the iteration process. Figure 4 (b) shows the inlet side face of the water inlet which is on the right in the figure and the outlet side on the left as indicated by the direction of the water flow arrow.

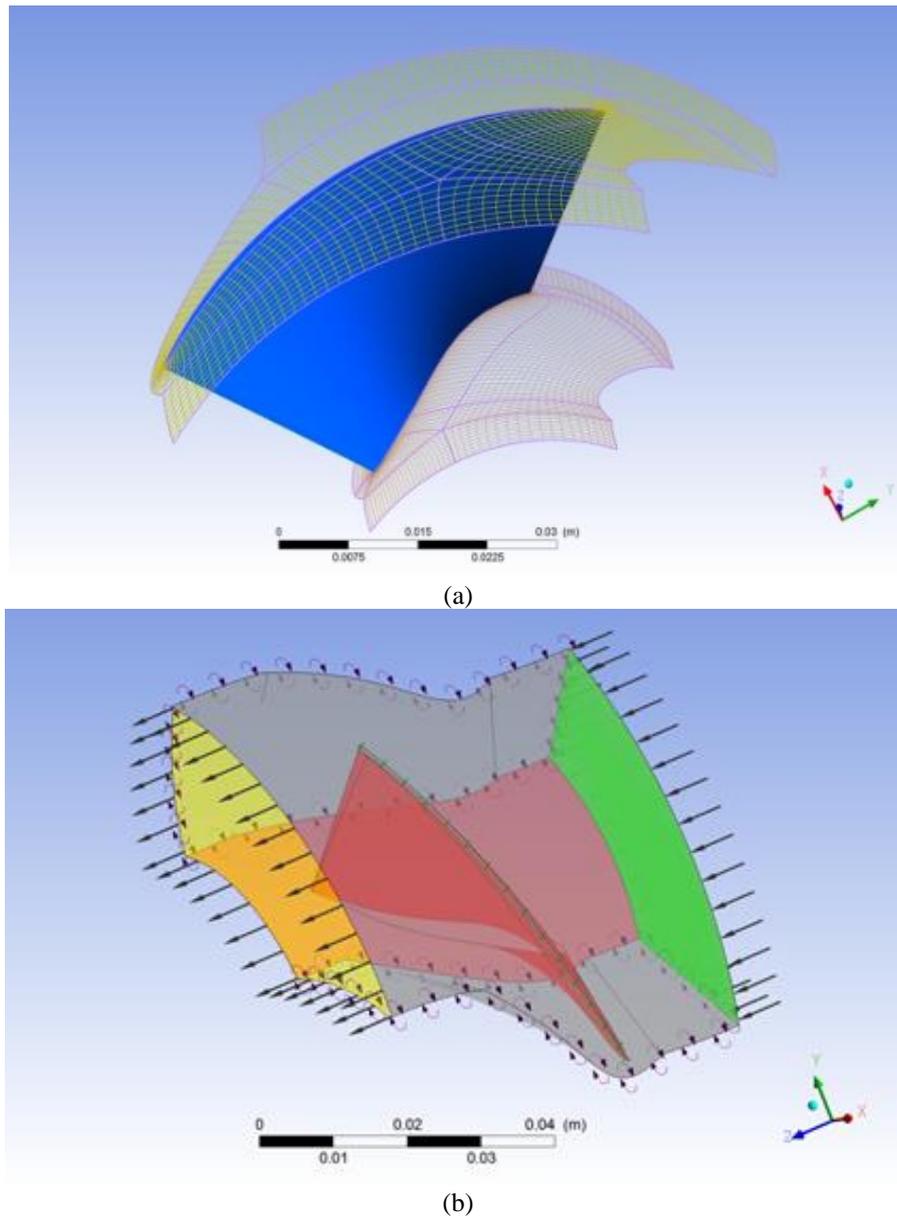


Figure 4. CFD model (a) meshing and (b) grid and boundary conditions

3.3. Turbine performance

In this study, CFD simulations were performed with various rotational speed variations ranging from 300 rpm to 500 rpm. These limits are in accordance with the specific speed range of the turbine propeller at a predetermined head and design discharge. The simulation results are as shown in Table 3, and the difference in the efficiencies from the two turbine designs is presented in graphical form in Figure 5.

As shown in Table 3, the shaft power for each type turbine design shows the same trend, namely, an increase with increasing rotational speed until maximum power is reached followed by a decrease if the rotational speed continues to be increased. Power *et al.* [25] revealed that the mechanical power output of a water turbine depends on the magnitude of the torque generated by the turbine shaft and the angular velocity (ω) of the turbine. However, every water turbine design has an optimal point according to the specifications in its design.

At the design point, namely, at a rotational speed of 400 rpm, the Type 2 turbine which is the result of the optimisation design has a greater shaft power of approximately 0.72% relative to that of the initial design (the Type 1 turbine). In graphical form in Figure 5, the magnitude of the shaft power generated by the two turbines is in the form of an inverted parabola. This outcome is in accordance with the simulation and experimental results conducted by [3], [17], [26].

Table 3. Runner blade performance with variable rotational speeds

Rotational Speed n (rpm)	Type 1 Turbine		Type 2 Turbine	
	Shaft Power P (W)	Efficiency η (%)	Shaft Power P (W)	Efficiency η (%)
300	1.956	76.93	2.085	80.23
325	2.014	76.71	2.124	79.69
350	2.056	76.21	2.139	78.88
375	2.082	75.47	2.133	77.79
400	2.092	74.49	2.107	76.43
425	2.084	73.25	2.059	74.75
450	2.057	71.73	1.988	72.74
475	2.012	69.90	1.893	70.30
500	1.946	67.74	1.774	67.37

When viewed in terms of efficiency in Figure 5, turbine Type 2 almost consistently has higher efficiency than turbine Type 1 at all rotational speeds except at 500 rpm. At the design point, namely, at a rotational speed of 400 rpm, turbine Type 2 has an efficiency of 76.43%, a result which is approximately 2.6% higher relative to that of the Type 1 counterpart. The curve line in the efficiency graph in Figure 5 indicates that the efficiency decreases as the turbine rotation increases. The graph in Figure 5 is also in accordance with research conducted by [27]–[29].

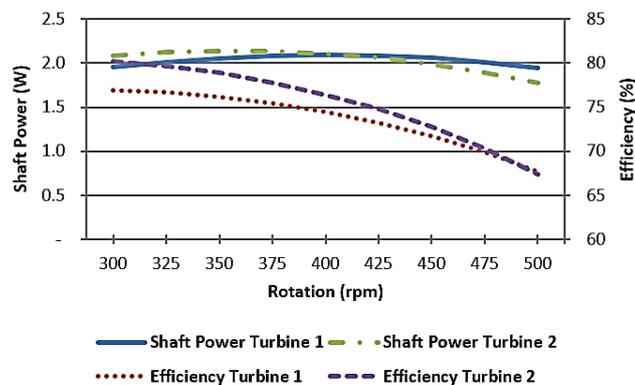


Figure 5. Turbine performance comparison

3.4. Flow behaviour inside the turbine

Contour, vectors and velocity streamlines that occur in the runner blades are shown in Figure 6. The velocity vector images at the 50% span runner blade position as shown in Figure 6 (a) indicate that the direction of flow entering the runner blades is quite good, as characterised by the absence of a reversing/turbulent flow pattern around the runner blade. The inflow of water hits the end of the leading runner blade along the runner until it exits the tealing runner uniformly. This situation is certainly very satisfactory for ensuring that the power shaft is as optimal as possible.

Figure 6 (b) shows the contour of the water velocity that occurs on the runner blade. The speed of the water increases from near the hub to the shroud at a speed of 0.2 m/s to 2.4 m/s. This result is in accordance with the mathematical concepts related to the flow motion and rotational force in a water turbine which is affected by the circulation parameter (Γ) that is related to a function of radius and viscosity [30].

Figure 6 (c) shows the contour of the flow pattern that occurs in the water turbine runner where the flow is quite satisfactory and smooth and no turbulent/turbulent flow exists. A good flow pattern on the turbine runner will certainly reduce losses in the hydrofoil so that the shaft power increases. As shown in Table 3 and Figure 5, the Type 2 turbine which is the result of optimisation has greater shaft power and efficiency than the Type 1 turbine. The Type 2 turbine produces greater torque, thereby generating greater efficiency. This outcome is in line with the assertion of Power *et al.* [25] that the mechanical power output from a water turbine depends on the amount of torque generated by the turbine shaft.

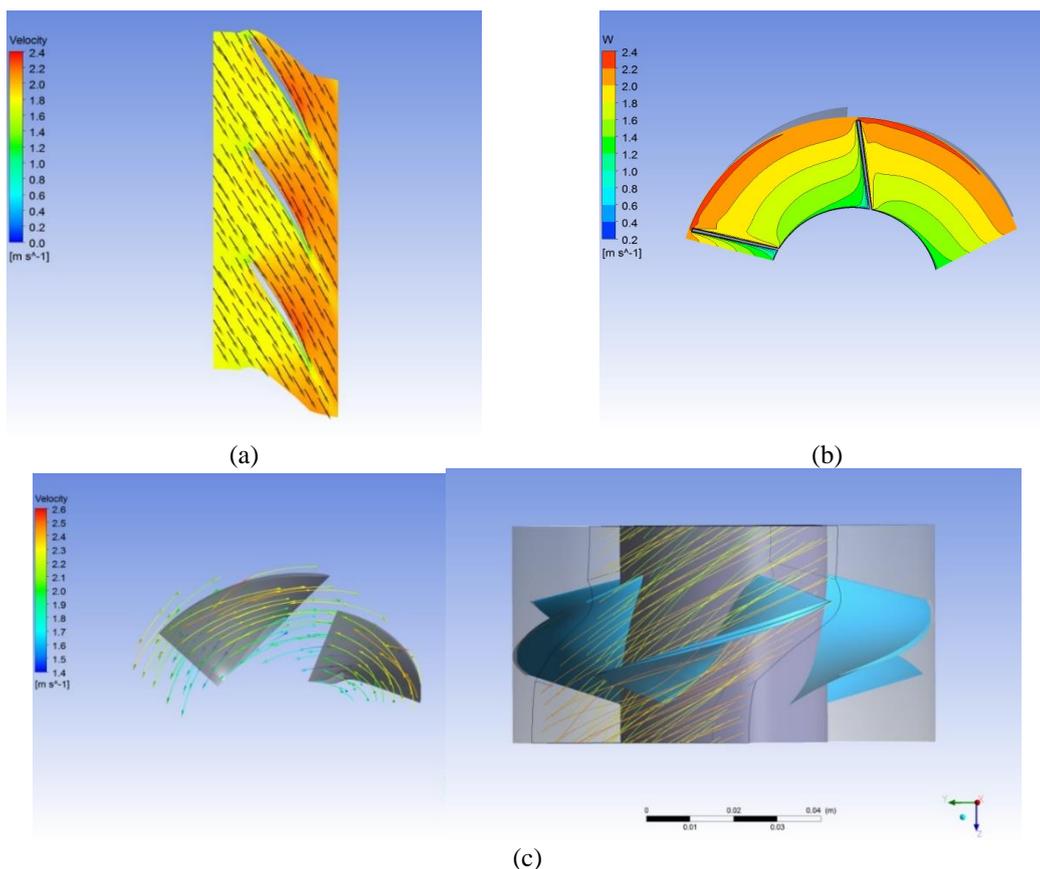


Figure 6. Velocity at blade: (a) contour of relative velocity, (b) velocity vectors at 50% span, and (c) velocity streamlines

4. CONCLUSIONS

In this research, turbine runner optimization was performed using a surface vorticity model algorithm. With this algorithm method, the efficiency and power of the water turbine is increased relative to that of the initial design. Turbine performance can be predicted by performing CFD analysis on a three-dimensional model with turbo mode to hasten the simulation process. The CFD simulation confirms that the efficiency of the optimized turbine in the design specifications has increased by approximately 2.6% relative to that of the initial design. The optimized turbine has greater efficiency at all rotational speeds from 300 to 475 rpm. Therefore, the surface vorticity model algorithm can be used as a tool to improve the performance of a water turbine runner by minimizing losses in the hydrofoil.

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