Experimental and Numerical Study on Performance of Shrouded Hydrokinetic Turbines

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Abstract— Hydrokinetic turbines, also known as marine current turbines, have the potential to be a major component of the world’s renewable energy portfolio. Improving the efficiency of turbines is critical to making this technology more widespread and cost-effective. This work focused on raising power output for a given turbine blade design and flow speed through the addition of a straight-diffusing shroud. A shroud works by lowering pressure at the turbine outlet and thereby accelerating the flow through the turbine. The performance (measured by the power coefficient, $C_p$) of a shrouded horizontal axis hydrokinetic turbine was studied experimentally in a tow tank at the United States Naval Academy. Two different shroud designs were built and tested, and performance was compared to the bare turbine. Both shrouds featured the same diffuser angle ($20^\circ$) but different area ratios, cross-sectional shapes, and tip gaps. The second shroud design was optimized using numerical flow simulation data, and resulted in a 21% increase in power output compared to the bare turbine. Both shroud designs lowered the stall speed of the turbine and allowed operation at lower tip speed ratios.

Keywords—hydrokinetic turbine; marine current turbine; shroud; diffuser; efficiency; performance

I. INTRODUCTION

Hydrokinetic turbines, also known as marine current turbines, function by converting a portion of the kinetic energy of moving water into rotational torque, which is then used to turn an electrical generator. Hydrokinetic turbines have the potential to be a major component of the world’s renewable energy portfolio, with applications in both river (unidirectional) and tidal (bidirectional) situations. However, the technology has yet to achieve widespread use for a number of reasons, including environmental policy restrictions and typically low efficiencies. Therefore, improving the efficiency of turbines is critical to the widespread and cost-effective implementation of this technology.

There are a number of ways to approach the problem of improving hydrokinetic turbine efficiency. The first is the optimization of the turbine blades. The optimization of turbine blades is well-understood, largely because of the similarity to propeller design and the extensive experience in that field. This work assumes an optimized turbine blade as a starting point, and focuses on raising power output for a given turbine design and flow speed through the addition of a shroud.

This study began as an undergraduate student design project at the United States Naval Academy during the spring of 2015. The goal of the initial project was to design and build a hydrokinetic power system small enough to be transported in backpacks by one or two people. The system would be deployed in small, shallow streams and would produce on the order of 100W, useful for powering or recharging small electronic devices. In order to maximize the power output from the necessarily small turbine, a shroud was implemented [1]. The design concept proved feasible; however, the shroud design was not optimized nor exhaustively tested. The goal of this work was to optimize the shroud design through numerical flow simulation, and to demonstrate the resulting efficiency improvement.

II. HYDROKINETIC TURBINE FUNDAMENTALS

The power output of a hydrokinetic turbine is given by the following equation:

$$P = \frac{1}{2}C_p \rho AV^3$$  (1)

where $C_p$ is the power coefficient, $\rho$ is the fluid density, $A$ is the turbine swept area, and $V$ is the fluid velocity. The power coefficient, $C_p$, is a measure of turbine efficiency and is the ratio of actual turbine power output to the maximum power available in the free-stream tube of cross-sectional area $A$:

$$C_p = \frac{\frac{P}{\rho AV^3}}$$  (2)

The turbine power can also be calculated as:

$$P = \tau \omega$$  (3)

where $\tau$ is torque and $\omega$ is the rotational velocity. The tip speed ratio (TSR) is defined as:

$$TSR = \frac{R\omega}{V}$$  (4)

where $R$ is the turbine radius. Turbine performance is typically presented as a plot of $C_p$ vs. TSR.

The theoretical maximum value of $C_p$ is 0.59, and is termed the Betz limit [2]. The Betz limit is based on the fact that a turbine requires fluid moving through it to generate power. If too much kinetic energy were extracted from the flowing fluid, it would come to a stop and the turbine would cease to operate.

Shrouds have been shown to improve performance of hydrokinetic turbines in some situations, bringing $C_p$ closer to the Betz limit. In theory, a shroud will increase the fluid
velocity through the turbine and reduce blade tip losses [3]. Shrouds also have the secondary benefit of providing some degree of physical protection to the turbine blades. Lawn [4] used one-dimensional fluid mechanics theory to explore different shroud inlet/outlet area ratios, and demonstrated up to 30% improvement in power output. Gadén and Bibeau [5] conducted numerical studies modeling the turbine as a momentum source, and showed significant improvement in power output through use of a diffusing shroud. Shahsavaranifar, Bibeau, and Birjand [3] conducted an experimental study of two different shroud profiles: one convergent-divergent, and one purely divergent. Tests were conducted in a variable speed water tunnel at the University of Manitoba and provided a solid set of experimental data useful for validating future theoretical or numerical models.

III. TURBINE AND SHROUD DESIGNS

A. Turbine

This project used a 3-bladed turbine design from a previous project at the United States Naval Academy. Overall turbine diameter was 26.54cm (10.45in), with a 5.08cm (2in) hub. The blades utilized a NACA 4412 foil shape, with chord length of 5.08cm (2in) at the root and 2.54cm (1in) at the tip. Pitch angle varied from 21° at the root to 0° at the tip. The blades were designed using SolidWorks CAD software, 3D printed, then nickel-plated to improve the stiffness. Fig. 1 shows the complete CAD model of the turbine, and Fig. 2 shows the actual turbine blades before plating.

Fig. 1. CAD model of turbine  
Fig. 2. 3D printed blades

B. Original Shroud

While the original shroud design was not fully optimized, it did adhere to some design parameters recommended in the literature. The chosen shroud profile consisted of a straight cylindrical section around the turbine followed by a diffuser section (see Fig. 3a). The other key parameters of the shroud design are the cross-sectional shape, diffuser angle, and outlet/inlet area ratio. Following results from Gadén and Bibeau [5], the diffuser angle was set at 20°. The desired area ratio was 1.56; however, the outlet area size was limited by the constraints of the available 3D printer used to manufacture the parts. The final dimensions were an inlet diameter of 28.83cm (11.35in) and outlet diameter of 31.78cm (12.51in), giving an area ratio of 1.22. For simplicity, the cross-sectional shape was uniform thickness (1cm) with rounded leading and trailing edges, not a true foil shape.

A key concern with the original shroud was the tip gap between turbine blades and shroud. Because of uncertainty in the rigidity of the shroud mounting assembly (see Fig. 3b) and resulting vibration during use, the tip gap was made relatively large (0.64cm). This tip gap and the turbine diameter drove the shroud inlet diameter.

Fig. 3. Original shroud design

(a) CAD model  
(b) Attached to test rig with 4 mounting brackets

C. Optimized Shroud

The shroud optimization process consisted of two parts: (1) reduction of the tip gap, and (2) incorporation of a foil cross-sectional shape. The tip gap reduction was accomplished by first improving the mounting assembly. The original assembly consisted of four aluminum mounting tabs connected to four radial threaded brass rods. The brass rods threaded into a PVC collar mounted to the non-rotating horizontal shaft. During testing, the rods experienced significant vortex induced vibration (VIV), and the PVC collar flexed and deformed under load. The assembly was re-designed with a brass collar replacing the PVC. Further, small segmented foil sections were placed over the rods to reduce VIV. The improved mounting system is shown in Fig. 4. Testing of the improved mount assembly with the original shroud showed significant reduction in vibration and deflection. As a result, the tip gap for the new shroud could be reduced from 0.64cm to 0.32cm. Reducing the tip gap had the secondary benefit of shrinking the inlet area, which subsequently allowed achievement of the optimal area ratio of 1.56, while still fitting within the 3D printer constraints.

Fig. 4. Improved shroud mounting assembly
To investigate the effect of the cross-sectional shape, four different shrouds were designed in SolidWorks. Two different foil shapes (NACA 0006 and NACA 4412) were used, with two different diffuser angles (20° and 30°) for each. Each design was numerically analyzed using SolidWorks FlowSim software. Specifically, the velocity and pressure distributions and the streamlines were analyzed for each design. The best design was the one with the largest flow velocity at the turbine location and the smoothest streamlines. Parameters of the final shroud design, along with the velocity profile and streamline plot, are shown in Table I and Fig. 5.

<table>
<thead>
<tr>
<th>Cross-section: NACA 0006 foil</th>
<th>Diffuser angle: 20°</th>
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<tbody>
<tr>
<td>Area ratio (Aout/Ain): 1.56</td>
<td>Velocity ratio (Vturbine/Vinlet): 1.42</td>
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</table>

TABLE I. OPTIMIZED SHROUD DESIGN

IV. EXPERIMENTAL TESTING AND RESULTS

Testing was conducted for three configurations: (1) bare, unshrouded turbine, (2) turbine with original shroud, and (3) turbine with optimized shroud. Testing was conducted in a 36.5m (120ft) towing tank at the United States Naval Academy, at flow speeds ranging from 0.61m/s to 1.52m/s.

A. Experimental Setup

The mounting assembly shown in Fig. 4 was suspended from a movable carriage such that the turbine axis was 0.76m below the surface (i.e., mid-depth of the tank). Care was taken to align the turbine axis exactly with the direction of travel. The rotating turbine shaft was connected via a 90° gear assembly to a vertical shaft that extended inside a hydrodynamic fairing above the surface to the carriage. Instrumentation at the upper end of the vertical shaft consisted of a torque meter, RPM meter, and a voltage brake, as shown in Fig. 6.

The voltage brake was used to create a variable resistance load on the rotating shaft. Increasing the load increased the torque and decreased the RPM, thus creating a range of TSRs for a given flow speed. For each flow speed, the brake voltage was started low (yielding a high TSR) and slowly increased until the turbine stalled. In most cases, approximately ten TSRs were tested per flow speed.

For each test run, the average torque and RPM were measured, and power was calculated using Eq. (3). The power coefficient, \( C_P \), was then calculated from Eq. (2) and plotted vs. TSR.

B. Experimental Results

The three configurations were tested at four different flow speeds, and the resulting \( C_P \) vs. TSR plots are shown in Fig. 7. The results reveal the performance of the three configurations relative to each other, as well as the optimal TSR (i.e., highest \( C_P \)) for each case.

Comparison of the bare turbine and original shroud configurations does not show a significant difference in performance. In fact, for the lower two flow speeds the original shroud actually lowered the \( C_P \) over most of the operating range, due to the increased drag and flow disruption caused by the shroud. For the upper two flow speeds, the \( C_P \) values were nearly identical for the two configurations. However, the original shroud had the positive effect of lowering the stall speed of the turbine, indicated by the data points at lower TSRs. Addition of the shroud allowed operation at higher torque/lower RPM conditions, so that the maximum \( C_P \) and optimal TSR could be determined. For all flow speeds, the optimal TSR was between 3 and 4. This is roughly consistent with previously published results [3, 6].

Comparison of the optimized shroud to the other configurations shows significant improvement. At all flow speeds, the maximum \( C_P \) with the optimized shroud is higher than the bare turbine or original shroud. The optimal TSR appears unaffected by the shroud, still falling between 3 and 4. The performance of the optimized shroud relative to the original is summarized in Table II.
The results clearly indicate that this turbine is more efficient at the higher flow speeds. The upper end of the flow speed range was limited by the capabilities of the test rig and structural limits of the turbine blades. It is expected that the maximum $C_p$ would continue to increase if flow speed were increased, until either blade failure occurred or the efficiency reached its peak and began to decrease. It is unclear, without further analysis, which would occur first.

V. CONCLUSIONS

The performance of a shrouded horizontal axis hydrokinetic turbine was studied experimentally in a tow tank at the United States Naval Academy. A previously-optimized 3-bladed turbine design was used, and two different shrouds were designed, built, and tested. Testing was conducted at four different flow speeds, ranging from 0.61 m/s to 1.52 m/s, and a range of TSRs for each flow speed.

The first shroud design consisted of a uniform, non-foil shaped cross-section and a 0.64 cm tip gap. This shroud showed no significant efficiency improvement compared to the bare turbine, but did serve to lower the stall speed of the turbine at all tested flow speeds.

The second shroud design was optimized by evaluating several candidates with SolidWorks FlowSim and choosing the design with the greatest flow velocity at the turbine location and the smoothest streamlines. Testing of the optimized shroud showed significant improvement in $C_p$ compared to the original shroud design (increased from 0.37 to 0.45). The results demonstrate that a carefully designed straight-diiffusing shroud...
can bring turbine performance closer to the Betz limit ($C_p=0.59$). Testing results also indicated the optimal TSR for maximum efficiency at each flow speed. The shrouds tended to lower the optimal TSR compared to the bare turbine, to a TSR value between 3 and 4.

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REFERENCES