UMERC+METS
2024 Conference

7-9 August | Duluth, MN, USA

Effects of foil and trailing edge thickness on the performance of cross-flow turbines

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Abstract

Ocean Renewable Power Company (ORPC) developed a set of hydrofoils with a blunt trailing edge geometry. There are two base foil geometries, NACA 0018 and NACA 0024, which each have been adapted to have 0.4% and 4.0% trailing edge thickness as a percentage of chord length. The blades will be tested in the ORPC Model Reference cross-flow turbine, which has three blades and a 1 meter diameter.

The ORPC Model Reference turbine with the new blades will then be installed in the UNH cross-flow turbine test bed. This allows turbine models to be of a size where performance becomes independent of Reynolds number while maintaining reasonable blockage.

The objective of these experiments is to explore the effects of increased mean and trailing edge thickness on turbine power performance. Tow tank testing was repeated with each blade set at varying tow speeds to determine the Reynolds number independent regime. Turbines were then tested in this regime at varying tip speed ratios to determine full power curves and the peak power coefficient.

Keywords: Tidal energy; Laboratory testing; Blunt trailing edge blades

1. Introduction

Cross-flow turbines are an attractive option for tidal energy conversion because of their ability to operate regardless of the water inflow direction, as long as the flow is perpendicular relative to the turbine axis. One of the downfalls of these devices, however, is that their blades operate under dynamic stall, which causes separation and reattachment of the flow [1]. There are many potential solutions to this problem, including optimizing performance otherwise. In these experiments, the effects of foil thickness and trailing edge thickness on overall performance is explored.

Blades with different trailing edge thickness and different base blade geometries will be tested in the UNH tow tank testing facility, where the blades will be attached to the ORPC Model Reference turbine. They will be tested for varying Reynolds numbers and tip speed ratios to compare the power coefficient for different trailing edges. A blunt trailing edge was chosen because of the potential for a delay in dynamic stall, which would increase the lift and potentially increase the power coefficient [2]. This contribution will report on the experiment design and preliminary results.

Nomenclature

Blockage	Ratio of turbine area to the facility test section cross-sectional area [3]
TSR	Tip speed ratio, $\lambda = \frac{\omega R}{U_\infty}$
Re_D	Reynolds number based on free stream speed and turbine diameter, $Re_D = \frac{U_\infty D}{\nu}$
Re_c	Reynolds number based on relative velocity and blade chord, $Re_c = \frac{U_{rel} c}{\nu}$
c	Blade chord length (m)
C_p	Power coefficient = $\frac{T\omega}{0.5\rho A U_\infty^3}$
C_L	Lift coefficient = $\frac{L}{0.5\rho A U_\infty^3}$
D	Turbine diameter (m)
R	Turbine radius (m) = $\frac{D}{2}$
ν	Water viscosity (m^2/s)
ω	Angular velocity (s^{-1}) = $\frac{2\pi * RPM}{60}$

2. Methods

2.1. Blunt trailing edge blades

Cross-flow turbines typically use blades with thin, sharp trailing edges. These blades with sharp trailing edges are often costly and difficult to fabricate [4]. One way to reduce cost in fabrication would be by having a blunt trailing edge. A potential fluid dynamic advantage to a blunt trailing edge is that the stall may be delayed, which can allow for an increase in the coefficient of lift, C_L [2].

In these experiments, there were four sets of blades tested with two different base geometries, NACA 0018 and NACA 0024. Each geometry had two different trailing edge thickness, 0.4% of chord and 4.0% of chord. Table 1 shows all the blade geometries. The blade profile was altered from the maximum thickness point backwards to increase the thickness of the trailing edge. The chord length for both foils is 0.095 m, and the overall height of the foil is 0.9 m tall. These blades are made of a 3D printed ABS plastic over a rectangular 9.5 mm thick and 38 mm wide steel core.

Table 1. Foil geometries

Foil Name	Base Geometry	TE Thickness
N18_T04_C33	NACA 0018	0.4% chord
N18_T40_C33	NACA 0018	4.0% chord
N24_T04_C33	NACA 0024	0.4% chord
N24_T40_C33	NACA 0024	4.0% chord

2.2. Towing Tank

The UNH Wave and Tow Tank was the facility used for testing, which is 12 ft wide, 8 ft deep, and 120 ft long. This tank allows for experiments where test bodies can be towed, subjected to wave action, or both. Experiments performed in a tow tank are provided with a clean, uniform inflow, which is useful for turbine performance evaluation. This facility also provides the option for a turbulent inflow from a turbulence generating structure. The turbine test bed is installed beneath the carriage that rides on case-hardened Thomson rails, driven by a 30 hp servo motor via an endless toothed belt along the length of the tow tank.

2.3. Test Bed

The ORPC Model Reference turbine was tested using the UNH cross-flow turbine test bed. This is a low-drag hydrokinetic turbine test bed, with a submerged frame made from extruded aluminum NACA 0020 foils and can be used to test cross-flow turbines up to 1 meter diameter. This allows testing to occur at a scale where performance becomes independent of Reynolds number while still maintaining reasonable blockage, in this case a blockage of 10%. It has been shown that turbine performance is affected by blockage when it exceeds 10% [5]. The UNH turbine test bed allows for testing to be done according to international marine energy specifications: IEC TS 62600-202, *Marine energy - Wave, tidal and other water current converters - Part 202: Early stage development of tidal energy converters - Best practices and recommended procedures for the testing of pre-prototype scale device* (published 2022) [3].

2.3.1. Torque, angular velocity, and thrust measurements

The tow tank carriage and cross-flow turbine test bed are equipped with several instruments to measure torque, angular velocity, and thrust during testing, as shown in Figure 1. Torque is measured using an Interface T8-200 torque transducer with a nominal accuracy of ±0.5 Nm, as well as a Sentran ZB3-200 torque arm for additional torque measurement with a nominal accuracy of ±0.2 Nm. The turbine rotor is controlled using a Kollmorgen AKM62Q servo motor with 10^5 pulse/revolution resolution. The turbine shaft angle is measured using the encoder output from the servo motor, which can be converted to turbine angular velocity. The test bed is supported by linear bearings connected to two Sentran ZB3-500 load cells with accuracy ±0.6 N, with one on each side of the carriage to measure thrust.

In addition to performance testing, tare drag and tare torque testing is conducted. Tare drag tests are done with the turbine test bed frame with no turbine installed towed over the same range of tow speeds run during performance testing. This allows for the mean thrust value of the test bed to be subtracted during data processing to estimate the thrust of the rotor itself. Similarly, tare torque tests are performed by rotating the turbine shaft in air without turbine blades or struts over the range of angular velocities used in testing. These torque values are then fit with a linear regression and added to the turbine torque during data processing.

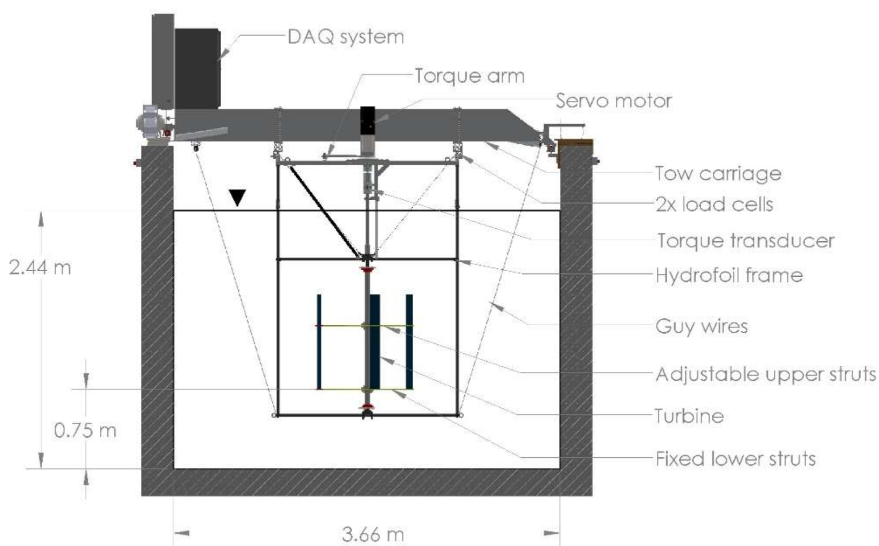

Figure 1. Schematic of UNH test bed with current sensors [6].

2.4. Turbine to be tested

Ocean Renewable Power Company (ORPC) developed a set of hydrofoils with a blunt trailing edge geometry. The blades were tested in the ORPC Model Reference cross-flow turbine, seen in Figure 2. The turbine has a diameter of 1 meter and has two support struts, which were fixed for this testing at a location of z/H = 1, where z is the upper strut location in reference to the bottom strut, and H is the overall blade length.

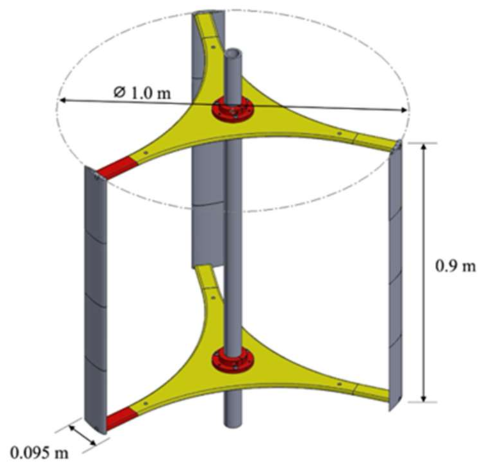

Figure 2. Model of ORPC Model Reference Turbine with struts at z/H = 1.

2.5. Experimental procedure

Testing was done using the ORPC Model Reference Turbine with each blade set. Turbine performance curves are established for varying tip speed ratios, $\lambda = \omega R / U_\infty$, and increasing tow speeds, U_∞, to determine above which Reynolds number (Re) turbine performance becomes independent, as shown in Figure 3. Performance testing is done at a speed within the Re-independent regime, which is the region where the power coefficient no longer changes with Reynolds number.

3. Preliminary Results

Figure 3 shows the power coefficient over increasing turbine diameter-based Reynolds number. All blade profiles show that beyond a tow speed of 1.0 m/s (which corresponds to a Reynolds number of $10{*}10^5$), the power coefficient seems to converge.

Figure 4 shows the power coefficient over a range of tip speed ratios for each blade set tested at a tow speed of 1.1 m/s. The curve for both the 0024 0.4% and 0024 4.0% blade sets showed to have a higher power coefficient than the 0018 blades. This peak occurs at a lower tip speed ratio of 2.3, while the 0018 peak occurs at a TSR of 2.5-2.6. The 0018 blade sets have a bit of a difference in their performance curves, with the 4.0% trailing edge performing slightly worse than the 0.4% trailing edge.

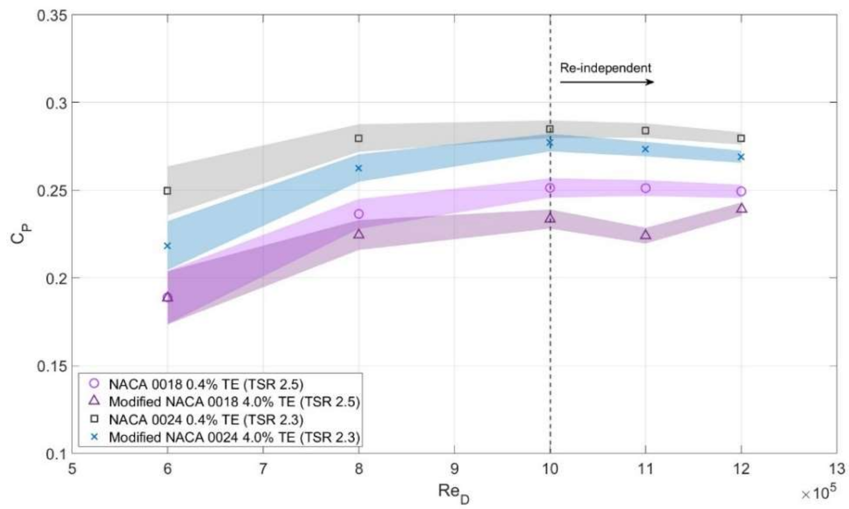

Figure 3. Turbine diameter-based Reynolds number dependence curve comparing all blade sets. Color highlight indicates range of uncertainty.

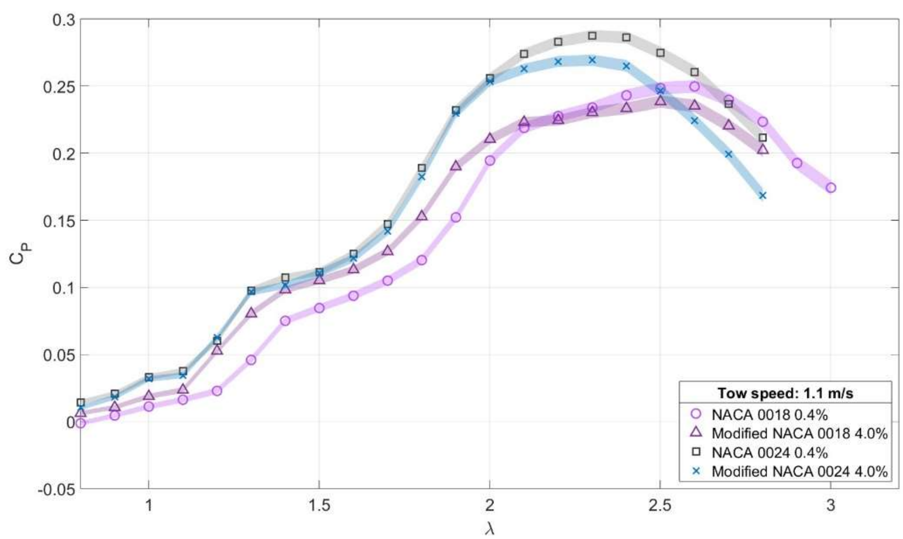

Figure 4. Performance curve comparing all blade sets tested at 1.1 m/s. Color highlight indicates range of uncertainty.

Next steps for this project include testing 0024 foils with an 8.0% trailing edge to determine if changing the trailing edge further will affect these blades.

Acknowledgements

This work is supported by the U.S. Department of Energy, ARPA-E SHARKS program.

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