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Multi-mode evaluation of power-maximizing cross-flow turbine controllers

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ABSTRACT

A general method for predicting and evaluating the performance of three candidate cross-flow turbine power-maximizing controllers is presented using low-order dynamic simulation, scaled laboratory experiments, and full-scale field testing. For each testing mode and candidate controller, performance metrics quantifying energy capture (ability of a controller to maximize power), variation in torque and rotation rate (related to drive train fatigue), and variation in thrust loads (related to structural fatigue) are quantified for two purposes. First, for metrics that could be evaluated across all testing modes, we considered the accuracy with which simulation or laboratory experiments could predict performance at full scale. Second, we explored the utility of these metrics to contrast candidate controller performance. For these turbines and set of candidate controllers, energy capture was found to only differentiate controller performance in simulation, while the other explored metrics were able to predict performance of the full-scale turbine in the field with various degrees of success. Effects of scale between laboratory and full-scale testing are considered, along with recommendations for future improvements to dynamic simulations and controller evaluation.

1. Introduction

Hydrokinetic turbines convert the kinetic energy in fast-moving water to electricity. Such flows can be found in rivers, tidal channels, or the western boundary currents of oceans [1–3]. Energy extraction is possible without impoundment, potentially making current energy an environmentally viable option in areas where a dam or barrage would be prohibitive or impossible [4]. Current turbines can be generally subdivided into two types: axial-flow, for which the axis of turbine rotation is aligned with the mean inflow direction, and cross-flow, where the axis of rotation is perpendicular to the mean inflow direction. In addition, there are novel designs, such as oscillating foils [5,6]. The present work focuses on cross-flow turbines. While axial-flow turbines are dominant for utility-scale wind energy, cross-flow turbines have a number of benefits in current energy applications. In a vertical orientation, cross-flow turbines do not require yaw control (e.g., a yaw-drive to align the projected area of the rotor with the flow direction) or, in a horizontal orientation, can operate in bidirectional tidal flows without yawing and have a lower profile in the water column, allowing operation in shallower channels. The rectangular projected area of cross-flow turbines is more amenable to high-blockage arrays, which can augment individual turbine power conversion efficiency of kinetic energy [7]. Cross-flow turbines also generally operate at lower blade speeds than

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axial-flow turbines, reducing the potential for cavitation, decreasing noise production [8,9], and reducing the risk of collision for marine animals [9].

As for wind turbines, current turbines can incorporate control mechanisms to maximize power capture and maintain power quality [10]. A poorly designed or implemented control algorithm can reduce energy capture, cause the turbine to stall [11], or induce excessive loads or large amplitude oscillations in the turbine drive-train components or support structure that decrease the device service life. It is clearly desirable to test new control schemes through scaled experiments and simulation prior to implementation on a full-scale system, but is not intuitive how the dynamics of a scaled laboratory experiment (and the implied controller effectiveness) will translate to a full-scale device in a field setting. In general, experimental scaling is complicated by physical limitations of experimental facilities and inability to maintain multiple relevant geometric and hydrodynamic parameters across scales, as recently summarized in [12].

Here, we evaluate three candidate control algorithms intended to maximize power capture. In the wind and hydrokinetic turbine literature, this is often referred to as "Region 2" control, denoting a region of flow speeds between turbine cut-in and rated power, in which the control objective is to maximize power. These controllers were tested in simulation, scale-model experiments, and on a full-scale cross-flow turbine deployed at a field site. The usefulness of metrics to differentiate between control performance at each mode of testing are considered and compared to results of full-scale testing. The ability of the proposed methods to anticipate the best-performing controller from among several candidates is examined, and the necessary caveats for extrapolating laboratory results to full-scale devices are explored. This study is unique in its exploration of cross-flow turbine control across multiple development modes and device scales.

2. Background

2.1. Turbine dynamics and performance

The available kinetic power in a moving flow is generally given as

$$P(t) = \frac{1}{2}\rho A U(t)^3 \tag{1}$$

where ρ is fluid density (kg/m^3, assumed constant), A is turbine projected area (m^2), and U is the free-stream velocity (m/s), a function of time t (s).[1] A turbine rotor converts a portion of the incident kinetic resource to mechanical energy, and then the power train (i.e. gearbox, generator) converts a portion of this to electrical energy. The fraction of power extracted is described by a turbine-specific performance curve, a non-dimensional functional relationship expressing the primary conversion efficiency from kinetic to mechanical energy in terms of the tip-speed ratio

$$\lambda(t) = \frac{R\omega(t)}{U(t)} \tag{2}$$

where R is the turbine radius (m), and ω is turbine angular velocity (rad/s). Instantaneous primary conversion efficiency ($C_P(t)$) is given as

$$C_P(t) = \frac{\tau_h(t)\omega(t)}{P(t)} \tag{3}$$

where τ_h is the hydrodynamic torque produced by the turbine. Overall conversion efficiency ($\eta(t)$) is calculated as

$$\eta(t) = \frac{I(t)V(t)}{P(t)} \tag{4}$$

where I and V are the generator output current (amps) and voltage (volts), respectively. Either formulation may be used to gauge the efficiency of a turbine system and the choice depends on several factors. For example, in the case of laboratory testing, a generator may be impractical and unnecessary for evaluating rotor performance, while the objective of a full-scale system is to generate electricity. Consequently, the choice of performance metric is often intuitive. Note that for a given turbine-generator system, C_P and η are related by a balance of system efficiency η_0 (the combined efficiency of generator, gearbox, and power electronics) by $C_P\eta_0 = \eta$. Balance of system components have been shown to play a significant role in overall system performance and may not be well-described by a constant factor [13].

A performance curve typically has one global maximum, designated C_P^* (or η^*) at an optimal λ^*. Quantities denoted with a* superscript herein signify this global maximum. A Region 2 controller can maximize power capture by holding $\lambda = \lambda^*$. Because $U(t)$ is uncontrolled and changing in time, holding λ constant requires active control of turbine angular velocity. Turbines that can adjust their angular velocity are referred to as variable-speed, in contrast to fixed-speed variants that operate at a constant angular velocity [10].

[1] As originally noted by Garrett and Cummins (2007), the "available" energy is, in truth, the sum of the kinetic and potential energy in a flow [7]. When turbines are deployed in a manner such that their projected area is an appreciable fraction of the channel cross-section, the extracted energy can exceed the kinetic energy reference value given by Eq. (1) because the array also harnesses a portion of the potential energy in the flow.

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