Renewable Energy 74 (2015) 318–325

Contents lists available at ScienceDirect

Renewable Energy

journal homepage: www.elsevier.com/locate/renene

Performance measurements of cylindrical- and spherical-helical cross-flow marine hydrokinetic turbines, with estimates of exergy efficiency

Peter Bachant*, Martin Wosnik

Center for Ocean Renewable Energy, University of New Hampshire, Durham, NH 03824, USA

ARTICLE INFO

Article history:
Received 23 October 2013
Accepted 26 July 2014
Available online 29 August 2014

Keywords:
Marine
Hydrokinetic
Wind
Helical
Vertical-axis

ABSTRACT

Power and drag (or thrust) measurements were performed in a towing tank for two different helical cross-flow marine hydrokinetic energy conversion devices—a cylindrical Gorlov Helical Turbine (GHT) and a Lucid Spherical Turbine (LST). The turbines are compared with respect to their various design parameters, with the GHT overall operating at higher power and drag coefficients. An estimate for the exergy efficiency of a turbine in free flow is formulated using momentum theory, and this quantity is computed for both devices. The GHT's exergy efficiency advantage over the LST was higher than that based on the power coefficient. Momentum theory-based blockage corrections were applied to the measurements and compared with the non-corrected data. The results presented here will help increase the amount of experimental data for helical devices in the literature, which is necessary for the development of more accurate engineering tools that take into account the unique three-dimensional nature of these devices.

1. Introduction

Marine hydrokinetic (MHK) turbines extract mechanical power from moving water without the need for a dam. At low Froude numbers they operate on essentially identical principles as wind turbines and are of interest for harnessing renewable energy sources such as rivers, drainage flows, tidal flows, and ocean currents.

Cross-flow turbines consist of a number of blades that rotate around an axis that is perpendicular to the flow. These turbines can receive flow from any direction, as long as it is approximately perpendicular to the axis of rotation, and unlike axial-flow turbines do not require yaw control. Cross-flow turbines, such as straight-bladed or Darrieus turbines have been explored for wind power applications since the 1970s, producing some large scale research projects, e.g., the 625 kW Sandia 34 m test bed turbine and the 3.6 MW Lavalin Eole 64 m research turbine [1]. Despite these advances, Darrieus turbines ultimately suffered from poor fatigue-life predictions, e.g., the Eole turbine had to be down-rated to 2 MW, and the now omnipresent three-bladed axial-flow propeller-style concept became the dominant design for utility-scale wind turbines. Paraschivoiu states that this may be more a result of relatively larger investment in research and development for axial-flow turbines rather than inherent technical superiority [1].

More recently, however, the cross-flow turbine concept has re-emerged as an attractive design for smaller-scale wind, offshore wind [2], and MHK applications [3] due to its ability to operate without a yawing mechanism, to place heavy drive train components lower in the assembly, and to be designed with virtually any frontal area shape. This attribute allows cross-flow turbines to be more closely packed in arrays, a fact that—along with its wake characteristics—recently led to the claim that cross-flow turbines may outperform axial-flow by an order-of-magnitude with respect to power generation per land area when arranged in arrays or farms [4]. Their arbitrary frontal area also gives an advantage when applied to tidal channels or rivers, where blocked area must be optimized to retain shipping channels, etc.

Helical cross-flow turbines, invented by Gorlov in 1995 [5], have blades that are swept azimuthally along their height. While adding manufacturing complexity, helical turbines have the technical advantage of smoothing out torque variation—or “ripple”—at the shaft. Furthermore, the overall streamwise and cross-stream forces on the turbine are less unsteady, decreasing vibration and fatigue loading of mounting structures.

* Corresponding author.
E-mail addresses: pxL3@unh.edu (P. Bachant), martin.wosnik@unh.edu (M. Wosnik).

http://dx.doi.org/10.1016/j.renene.2014.07.049
0960-1481/

Gorlov's final technical report included testing of comparable helical vs. straight-bladed turbines, showing an improvement in overall power output for the helical version [6]. Shiono et al. performed experimental studies that contradict Gorlov's results, observing a decrease in power output for a turbine with helical blades compared with a straight-bladed version [7]. The design parameter space for these turbines is quite large—blade shape, number of blades, blade chord/radius—i.e., solidity, helical sweep angle, blade end configuration, shaft configuration—and these conflicting answers make it difficult for developers to assess the economic and technical benefits of the added complexity of the helical turbines.

Despite a lack of published research on helical cross-flow turbines, they are present in industry. In 2012 the Ocean Renewable Power Company (ORPC) installed the first grid-connected helical cross-flow tidal turbine in Cobscook Bay, ME [8]. Urban Green Energy (UGE) has developed and has been installing helical cross-flow wind turbines for smaller-scale wind applications in urban or built environments, e.g., for electric vehicle charging stations [9].

Motivated by the aforementioned lack of experimental data for helical cross-flow turbines in the archival literature, and the premise that progress in the development of more accurate engineering tools depends on this data, we set out in the present study to experimentally measure and compare the power and drag (a.k.a. thrust) of two different helical cross-flow turbines—one cylindrical and one spherical.

1.1. Exergy efficiency

Studies on wind or MHK turbines typically use the power coefficient as the main performance metric for a device. The power coefficient C_P is defined as the ratio of shaft power output to fluid kinetic energy flux with free stream velocity, U_∞, and pressure through an area the size of the turbine frontal area, A_f,

$$C_P \equiv \frac{P}{\frac{1}{2}\rho A_f U_\infty^3}. \tag{1}$$

Considering the 1-D momentum theory developed by Rankine [10] and Froude [11], later used by Betz to calculate the Betz limit [12], the power flux through the turbine itself is lower than what is used to compute C_P, i.e., the denominator of Eq. (1). This means that the power coefficient, although a good indicator of device effectiveness in capturing fluid power, is not necessarily a measure of efficiency.

We can draw an analogy to a heat engine, where the thermal (first law) efficiency is defined as the (rate of) net mechanical work out divided by the (rate of) heat energy input. This relation does not take into account the exergy of the heat rejected, or the entropy generation (irreversibility) that occurs in the device. The second law, or exergy efficiency takes these effects into account, providing another metric for system performance.

When evaluating wind and MHK turbines, exergy efficiency may be a more relevant metric for array performance. For instance, an upstream turbine with low exergy efficiency, regardless of power coefficient, may severely decrease the performance of one downstream, decreasing overall array performance. It may be the case that operating individual turbines at a C_P slightly below an individual turbine's maximum but with higher exergy efficiency will optimize overall array performance. Furthermore, low device exergy efficiency increases environmental effects per unit energy generated, which is especially important in tidal channel flows [13,14].

There are a few studies known to the authors that discuss exergy efficiency of turbines that convert kinetic energy, i.e., wind and MHK turbines. Corten [15] considered the heat generation by a wind turbine, concluding that even an ideal turbine operating at the Betz limit would convert 1/3 of the wind's kinetic energy to heat. Pope et al. [16] compared energy and exergy efficiency of various wind energy systems using computational fluid dynamics (CFD) simulations. Ozgener and Ozgener [17] investigated the reliability and exergy efficiency of a small horizontal axis wind turbine (HAWT). Bachant [18] and Bachant and Wosnik [19] estimated exergy efficiency based on measured power and drag combined with actuator disk theory, a concept which is refined in the present study.

2. Exergy efficiency formulation

We seek to define a simple enough expression for exergy efficiency such that it can be estimated from power and drag coefficients alone, since these are the measurements at our disposal. We will derive a relationship using 1-D momentum theory, for which the actuator disk and bounding streamtube are sketched in Fig. 1.

We define the second law, or exergy efficiency, η_{II}, as the fraction of flow exergy removed from the mean flow that is converted to shaft work. This can be written as

$$\eta_{II} = \frac{\text{Useful work output}}{\text{Change in flow exergy}}, \tag{2}$$

or

$$\eta_{II} = \frac{P}{\frac{D}{Dt} m\psi}, \tag{3}$$

where ψ is the specific flow exergy of the fluid passing through the turbine control volume.

The Reynolds Transport Theorem,

$$\frac{DN}{Dt} = \frac{\mathrm{d}}{\mathrm{d}t}\int_{CV} \rho\eta \mathrm{d}V + \int_{CS} \rho\eta(\vec{u}\cdot\vec{n})\mathrm{d}A, \tag{4}$$

defines the rate of change of a quantity N for a fluid particle as it passes through a control volume, where η is the quantity of interest per unit mass [20].

Since we are assuming a steamtube-bounded control volume with steady, uniform flow, Eq. (4) simplifies to

$$\frac{DN}{Dt} = \rho(\eta_1 A_1 U_1 - \eta_4 A_4 U_4), \tag{5}$$

which is further simplified using conservation of mass

$$A_1U_1 = A_2U_2 = A_4U_4, \tag{6}$$

giving

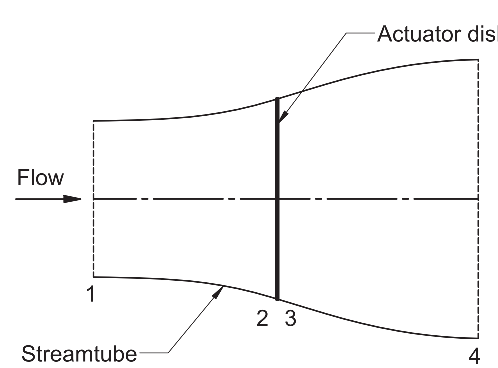

Fig. 1. Schematic of a turbine modeled as an actuator disk for 1-D momentum theory analysis.