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Experimental measurements of the hydrodynamic performance and structural loading of the transverse horizontal axis water turbine: Part 2

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A B S T R A C T

This paper is the second of three, which outline the procedures and results for a set of experiments on various configurations of the Transverse Horizontal Axis Water Turbine (THAWT), which is a horizontally orientated variant of the Darrieus cross-flow turbine. Tests were conducted in the combined wind, wave and current tank at Newcastle University on a 0.5 m diameter rotor, while the flow depth and velocity were varied over a range of realistic Froude numbers for tidal streams. Various configurations of the device were tested to assess the merits of varied blade pitch, rotor solidity, blockage ratio and truss oriented blades. Experiments were carried out using a speed controlled motor/generator, allowing quasi-steady results to be taken over a range of tip speed ratios. Measurements of power, thrust, blade loading and free surface deformation provide extensive data for future validation of numerical codes and demonstrate the ability of the device to exceed the Lanchester–Betz limit for kinetic efficiency, by exploiting high blockage. This second paper covers the instrumentation and analysis for the structural loading for the parallel bladed variant of the THAWT device. The first paper covers the experimental setup and hydrodynamic performance of the parallel bladed rotor, and the third paper covers both performance and loading of the truss configured THAWT device.

1. Introduction

The transverse horizontal axis water turbine (THAWT) has previously been proposed as an alternative design to the common axial-flow tidal stream device, and is intended to allow the scaling of long stiff multi-bay rotors by using a truss configuration of blades [1], as shown in Fig. 1.

This paper is the second of three, which together describe the procedures and results for a set of experiments, in the combined wind, wave and current tank at Newcastle University, on various configurations of the transverse horizontal axis water turbine (THAWT). This paper covers the instrumentation and analysis of the blade loading for the parallel bladed variant of the THAWT device shown in Fig. 2. The first paper covers the experimental setup and hydrodynamic performance of the parallel bladed rotor [2] and the third paper covers both performance and loading of the truss configured THAWT device [3].

Unlike axial-flow devices, the blades of a cross-flow rotor experience a reversal of blade loading during each turbine rotation. With over 1×10^7 cycles of loading during the lifetime of a cross-flow turbine, cyclic fatigue governs structural failure, and the accurate prediction of blade loading is vital in the design of a device [4].

While there have been experimental measurements of the blade loadings on cross-flow wind turbines [5,6], as well as numerical predictions of tidal stream blade forces [7,8], there have been no experimental measurements of blade loading on tidal stream cross-flow devices, especially in high blockage and solidity environments for which numerical simulations are difficult. The experimental measurements obtained in this study can be used as validation for future numerical codes, as well as indicating how the structural loading of the parallel bladed device is affected by variations in fixed offset blade pitch, rotor solidity, channel blockage and flow direction.

2. Experimental setup

2.1. Basic flume setup and instrumentation of hydrodynamic performance

The experiments on the cross-flow rotor were carried out in the combined wind, wave and current tank at Newcastle University, with plan dimensions and the turbine located as shown in Fig. 3.

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Nomenclature

A	Cross sectional area of turbine, m^2
b	Test section width, m
$B = A/hb$	Turbine blockage ratio
c	Blade chord length, m
$C_{EL} = w/\frac{1}{2}\rho[(\lambda + \cos\theta)u_r]^2 c$	Equivalent lift coefficient
$C_N = w/\frac{1}{2}\rho c u_r^2$	Blade normal force coefficient
$C_P = P/\frac{1}{2}\rho A u_r^3$	Power coefficient
$C_T = T/\frac{1}{2}\rho A u_r^2$	Thrust coefficient
d	Diameter of turbine, m
$Fr = u_r/\sqrt{gh_r}$	Dimensionless Froude number
h	Depth of flow, m
L	Lift force, N
n	Number of turbine blades
F	Resultant blade force, N
P	Measured power produced by turbine, W
T	Measured streamwise rotor thrust, N
r	Turbine blade radius, m
$Re = vc\rho/\mu$	Dimensionless blade Reynolds number
$s = nc/\pi d$	Turbine solidity
x	Chordwise coordinate of blade profile, m
y	Coordinate of blade profile normal to chord, m
u	Volume averaged flow velocity, m/s
u'	Flow velocity at a blade, m/s
v	Blade resultant flow velocity, m/s
w	Distributed loading normal to blade chord, N/m
μ	Dynamic viscosity of water, mPa s
$\lambda = r\omega/u_r$	Tip speed ratio
ρ	Density of water, kg/m^3
θ	Blade rotation angle
φ	Foil fixed offset pitch angle

Subscripts

∞	far field, upstream flow
r	at streamwise rotor plane

Measurements were taken during the experiments of volume averaged flow velocity, upstream and downstream flow depths, rotor torque, rotor angular velocity, rotor angular position and hydrofoil loading normal to the blade chord. Details of the rotor support geometry, drive train components, depth measurement instrumentation, flow characterisation and parallel bladed hydrodynamic performance are given in Ref. [2].

2.2. Rotor dimensions

As shown in Fig. 2 the rotor consists of two aluminium endplates, each 10 mm thick and 540 mm diameter, with 5 mm $\times$ 45° chamfers at the outer edge of each face. The 1.53 m long blades have a chord of $c = 65.45$ mm and are evenly distributed at the pitch circle radius of $r = 0.25$ m, resulting in a solidity of $s = 0.25$ for a six-bladed rotor (see nomenclature for definition). The rotor axis is mounted 0.425 m above the flume base, leaving a clearance of 0.175 m below and 0.325 m above to the free surface in a 1.0 m deep flow.

The hydrofoils use the NACA0018 section, but the profile has been conformally mapped onto the pitch circle of the blades using the following transformations:

$$x' = (r + y)\sin\left(\frac{x}{r}\right) \quad (1)$$

$$y' = (r + y)\cos\left(\frac{x}{r}\right) \quad (2)$$

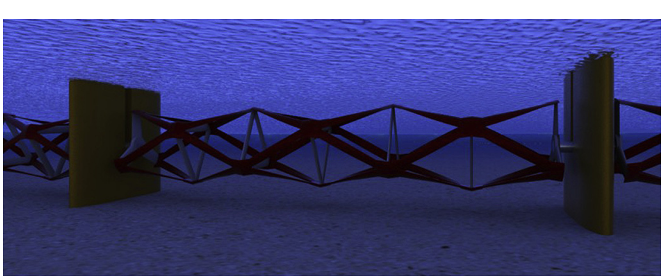

Fig. 1. Rendered image of a string of THAWT devices, provided by Kepler Energy Ltd. and Mojo Maritime.

Threaded aluminium inserts in the blade ends allow the blades to be bolted to the rotor plates. Slots in the rotor plates allow the rotor blades to be pitched prior to a test. A hydrofoil is defined as neutrally pitched when the blade chord at the quarter chord point is tangential to the rotor pitch circle. A positive fixed offset pitch angle φ is defined in degrees as a nose-in pitch about the quarter chord point.

2.3. Blade load instrumentation

When considering the structural design of a cross-flow turbine hydrofoil, the most significant load is the lift generated approximately perpendicular to the blade chord. This flapwise bending causes the greatest stresses, due to the small second moment of area in bending about the blade chord, and the fact that the lift force generated by a hydrofoil at peak lift to drag is between one and two orders of magnitude greater than the drag force. The rotor blades were therefore instrumented to measure the blade loading normal to the blade chord.

During the moulding of the instrumented carbon fibre blades, plates were placed inside the moulds to create recesses in the blade profile on the top and bottom of the foil, at four spanwise positions and located approximately on the neutral axis of chordwise bending. Strain gauges in these recesses are connected in a full bridge bending circuit, and potted with epoxy to recreate a smooth blade profile, as shown in Fig. 4. The strain gauge signals are carried by wire leads out of the blade and through the hollow shaft to a waterproof enclosure, mounted on the shaft inside the streamlined fairing. The waterproof enclosure houses Mantracourt T24-SAf strain gauge instrumentation modules which amplify, condition, digitise and wirelessly transmit the measured strains at 2 kHz to T24-AO1 receivers outside the flume tank, as shown in Fig. 2. The measured strains were calibrated by applying known bending moments in four point bending tests prior to the experiments. Prior to each test, the rotor is turned at a speed of approximately 1 revolution per minute in a stationary flow and the averaged measured strain for each gauge during a rotation is calculated to provide the zero calibration measurement.

It is approximated that the maximum and minimum blade loads given in the subsequent analysis, are prone to a maximum error of 2.0%, due to instrumentation and calibration errors.