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Horizontal-Axis Tidal Turbine Blade Loading for Multi-Frequency Oscillatory Motion

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Abstract

This paper presents results from an experimental study which analysed the hydrodynamic response of the out-of-plane blade root bending moment for a horizontal-axis turbine exposed to multi-frequency oscillatory motion. Estimates of the amplitude and phase agree well with those for single frequency oscillatory motion, which suggests that a model based on the principles of linear superposition is applicable. When minor flow separation is experienced, linear superposition is likely to offer conservative estimates. The findings are likely to be of interest to designers of turbines deployed in tidal streams, rivers or canals, and who are seeking low computational approaches for assessing the dynamic blade loads.

Introduction

If tidal stream turbines are to offer a viable means of renewable energy generation they must be reliable and economical to manufacture and operate. This is by no means trivial, as key components such as the blades are subject to harsh operating conditions, where turbulence and wave action induce a spectrum of loads. Adding to the challenge is a lack of experimental studies from which models may be validated or industry guidelines developed for predicting the unsteady hydrodynamic blade loads and fatigue life. Consequentially, designers of early generation turbines have resorted to employing considerable degrees of conservativeness, in an attempt to mitigate failures, but resulting in increased cost.

Previous experimental studies by Bahaj and co-workers [1, 2] have demonstrated that for steady flow, traditional quasi-steady based models, such as that derived from blade element-momentum theory, can lead to reliable predictions. However, in unsteady flow, complex interactions in the loading exist due to circulation contained in the trailed wake, coupled with both circulatory and non-circulatory effects at the blade section level. Barltrop et al. [3] have presented experimental results for a tidal turbine exposed to surface waves. These studies showed that the magnitude of blade loading can be significantly greater than that for steady flow.

Using surface waves, typically limits the range of unsteady conditions which are able to be studied however, due to the size of the facility. Whelan [4] has previously performed a series of experiments to investigate the sensitivity of the rotor thrust using planar oscillatory motion superimposed on a mean current. The resulting phase difference between the velocity and thrust, or effective added mass, was found to be small compared to the loads for steady flow, but it was considered to be significant for controller design and fatigue estimates. However, relatively high blockage levels and the presence of a seiche wave again restricted the parameters which could be studied and made determining underlying relationships between forcing parameters difficult. Literature dealing with the effects of separated flow, where dynamic stall can result in significant load ranges, appears to be restricted to the wind turbine and helicopter sectors.

Objectives

The response of multi-frequency forcing is investigated in this present study, and is a further step towards providing the tidal turbine industry with valuable insight into the nature of the loads which will be imparted by the unsteady flow. Of particular interest is the degree of linearity exhibited in the loading, and whether low computationally demanding stochastic models may be viable. Subsequently, one would also like to understand the conditions in which the assumptions of linearity lose validity, and necessitate more complex modelling techniques. The findings of this paper are likely to be of interest to not only designers of tidal turbines, but also of axial turbines deployed in rivers and canals.

Methodology

In the present study a horizontal-axis turbine was subjected to planar oscillatory motion, whilst being towed at constant speed in a still water towing tank. This approach was deemed to be the most suitable for meeting the objectives of the study, in that it permitted a relatively wide parameter envelope, whilst ensuring that the onset velocity was as controllable and repeatable as possible. To further reduce external effects of structural inertia from components such as the rotor shaft, it was the blade root bending moment which was measured and analysed, rather than the rotor thrust.

The coherent loading experienced by the blades, whilst not necessarily depicting realistic small scale turbulence, is an attempt to simplify an already complex situation, and is considered to still offer highly valuable insight into the underlying hydrodynamics as well as provide data from which to validate models. Rotational sampling of turbulence, in which frequencies equivalent to the rotational frequency are of interest, is likely to also result in coherent loading over a significant portion of the blade.

Experimental Set-up and Turbine Design

Specifications of the Towing Tank and Carriages

The experiments were conducted at the University of Strathclyde's 55m long, 4.5m wide still water towing tank. For all tests the water depth was maintained at 2.20m. The effect of reflective waves is mitigated using a 9m long beach installed at one end of the tank, with reflection coefficient typically less than 5 percent. The main carriage velocity is computer controlled, and was ramped up at constant acceleration at the start of each test, and then maintained constant for the duration of the test period, before being ramped down again.

The main carriage is equipped with a digitally-controlled auxiliary carriage, which is driven by a linear actuator which can obtain accelerations in the order of up $2g$. A simple axial harmonic displacement time history of frequency f was input into

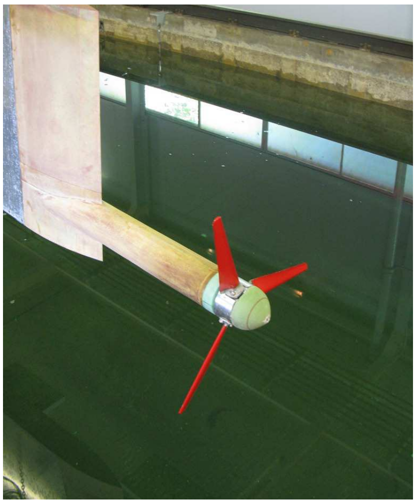

Figure 1: Photo showing the rotor and blades prior to installation on the carriage

the computer controlling the sub-carriage, such that the desired velocity upstream of the turbine $u(t)$ was of the form

$$u(t) = U + \sum_{i=1}^{n} \mu U \sin(2\pi f_i \cdot t) \tag{1}$$

where U is the velocity of the main carriage (assumed constant), and μU is the maximum velocity of auxiliary carriage for the i^{th} frequency component.

All data signals were acquired with a sampling rate of 400Hz. A low pass digital filter with a cut-off frequency of 300Hz was applied in real time on all signals. For each oscillatory test, the auxiliary carriage motion commenced whilst the main carriage was accelerating. This facilitated the decay of transients in the oscillatory kinematics and blade loads sample histories, therefore allowing for longer test runs. Spectral analyses of the carriage velocity and acceleration showed that structural vibrations were minimal.

Specifications of the Rotor and Blades

The nominally 20^{th} scale tri-bladed rotor used in the experiments had a rotor diameter of 780mm, and is shown in figure 1. The profile of the R = 367mm long blades have a non-uniform spanwise chord and twist distribution, and blade sections which conform to the 24 percent thick, NREL S814 profile. The three blades were instrumented with water-tight strain gauges on the cylindrical blade root, at a radius of r_{sg} = 36mm from the rotor axis, and which were enclosed within the rotor hub. The out-of-plane bending moment from one of these blades is analysed in this study.

The rotor motor, slip ring, and thrust and torque transducer were enclosed in a submerged rectangular box, at a distance of approximately 2 diameters downstream of the rotor. The box was mounted rigidly to the sub-carriage from above, and the rotor axis was 0.70m below the mean free water surface. The tests were performed at constant rotational speed, using a closed-loop digital controller.

Rotor Performance and Blade Loads for Steady Flow

Prior to performing oscillatory tests the power and loading characteristics of the rotor were established at a range of steady flow speeds with the auxiliary carriage disabled. The rotor power, thrust, and in-plane and out-of-plane root bending moment coefficients, defined by equations (2 - 4), are presented in figure 2 as a function of tip-speed ratio $\lambda (= \Omega R/U)$.

$$C_P = \frac{\Omega R}{\frac{1}{2}\rho U^3 A} \tag{2}$$

$$C_T = \frac{F_x}{\frac{1}{2}\rho U^2 A} \tag{3}$$

$$C_{M_{x,y}} = \frac{M_{x,y}}{\frac{1}{2}\rho U^2 A R} \tag{4}$$

where ρ is the density of water, and Ω is the rotational speed, F_x is the thrust, and A is the swept area of the rotor.

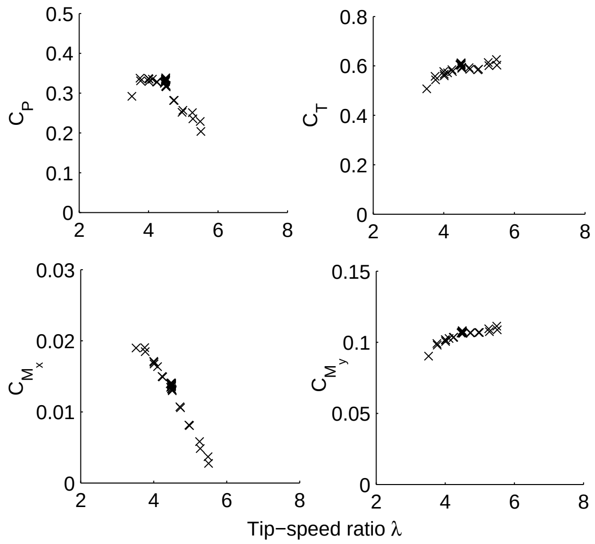

Figure 2: Clockwise from top left: rotor power, thrust, out-of-plane and in-plane blade root bending moment coefficients, as a function of tip-speed ratio, for steady flow velocities at a constant rotational speed of Ω = 96rpm.

The optimal power of the rotor is obtained at tip-speed ratios of approximately λ = 3 - 3.5, which is similar to the rotor used in the study by Whelan [4], though slightly lower than that of Bahaj and co-workers [1, 2]. At lower-tip speed ratios the power is limited by stall, and at higher tip-speed ratios the rotor enters the turbulent wake state, which limits power whilst increasing loading. Zero power is achieved at a tip-speed ratio of approximately λ = 7, which is expected to be lower than that for a full-scale tidal turbine. This is predominately a result of the drag on the blade sections being higher due to the model scale Reynolds numbers of around 100,000, being an order of magnitude lower.

Blade Loading for Planar Oscillatory Flow

Parameter Selection

The steady flow results were used to inform the mean operating conditions at which the oscillatory tests were to be conducted. A set of experiments were performed at a mean tip-speed ratio of λ = 4.5, which is near maximum power and where the flow is still primarily attached over the blades. These operating points were deemed to be of most interest to tidal turbine designers. The rotor speed and carriage speed at this operating point were Ω = 96rpm and U = 0.82m/s respectively.

The axial oscillatory motion was aimed to replicate the velocity perturbations imposed by surface waves and turbulence. Velocity amplitudes of up to 25% of the mean flow and periods ranging from 9 - 1s are likely to form the typical parameter envelope of interest [4]. Froude number based scaling was employed to determine the model scale parameter values. The parameter scope encompassed current numbers ranging from $0.075 \leq \mu \leq 0.25$ and oscillatory frequencies $0.4 \leq f \leq 1$Hz. Considering a representative blade section at $0.75R$, these frequencies represent reduced frequencies $k(= \pi cf/\Omega r)$, of $0.05 \leq k \leq 0.02$. Physcial limitations with the control system and thrust and torque dynamometer restricted the upper ranges of parameters to be studied.

Data Processing

All measured signals were post-processed using a band-pass IFR digital filter in both the forward and reverse directions to preserve the phase relationships. The low and high pass frequency cut-offs were set at $10f$ and $4f$ respectively. The relatively stringent high frequency cut off was implemented to reduce the contamination caused by non-perfect oscillatory motion. Such an approach therefore inherently removed the presence of hydrodynamic non-linearities above these frequencies.

Estimates of the relative amplitude and phase of the bending moment response were computed by fitting a linear sinusoidal model to both the velocity and bending moment time histories, using a minimisation of least squares technique. The fit was applied to two cycles at a time, and all estimates then averaged. This enabled the presence of any underlying surface waves caused by the carriage motion to be identified and excluded. The effect of such waves was found to be the greatest for oscillations at f = 0.5Hz.

Sensitivity to Oscillatory Frequency and Velocity Amplitude

The bending moment amplitude and phases, relative to the oscillatory velocity are first considered for single frequency oscillations. These are presented in figure 3, as a function of oscillatory frequency and current number. An underlying trend of an increase in the relative amplitude ratio and a decrease in the phase with an increase in frequency can be observed. The effect also appears to be greater for an increase in frequency than current number, for the parameter ranges investigated. The magnitude of the phase is small, which suggests that the loads attributed to flow acceleration are relatively minor compared to that associated with velocity. The positive phase is also indicative of an 'added mass' which acts in phase with acceleration. This has been attributed to the combined effects of dynamic inflow and the hydrodynamic added mass [4].

Estimates of the amplitude and phases from a select number of multi-frequency oscillation cases are compared with those from the single frequency tests in figure 4. For these cases the current number of each component was μ = 0.1, which implies that the effective current number was of the order of μ = 0.2. The estimates appear to agree relatively well, given the inherent difficulty in analysing a phase of such small magnitude. The findings suggest that for these test parameters, the principles of linear superposition are reasonably valid.

Limitations Imposed by Flow Separation

Whilst the unsteadiness exhibited by the loads for attached flow may be considered to be approximately linear, facilitating a transfer function type approach, this is however, not expected where separated flow is present. Therefore, the conditions in which flow separation is expected, and the magnitude of the associated errors expected are of interest.

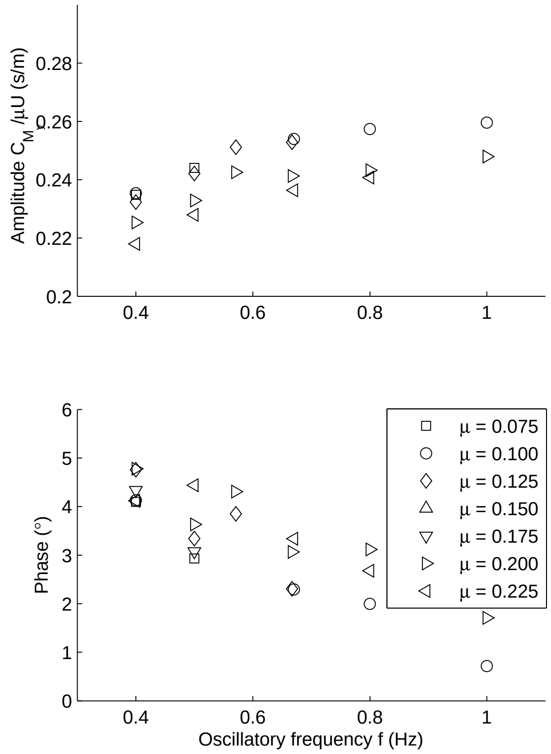

Figure 3: Effect of the oscillatory frequency and current number on the amplitude and phase response of the out-of-plane blade root bending moment, for single frequency planar oscillations about λ = 4.5.

The effect of flow separation is demonstrated for two multi-frequency oscillation cases in figure 5. The two frequency combinations of f = 0.4Hz and 0.67Hz, and f = 0.67Hz and 1Hz are considered, and each frequency component has an current number of μ = 0.15. As can be observed, a linear superposition is not able to reconstruct the sudden drop off in load associated with flow separation, which in these cases results in an over-prediction of the maximum loads. This effect is more pronounced for the lower frequency case and demonstrates the delay of flow separation and re-attachment to greater angles of attack with an increase in frequency, as is commonly observed for oscillating foils.

Discussion

Designers of tidal turbines are likely to seek computationally inexpensive methods by which to assess loads, particularly for preliminary calculations. The findings of this study suggest that a stochastic approach, in which the loading is assumed to be linear, is likely to allow for reasonably accurate predictions of the blade root bending moment in conditions where the flow is primarily attached. Stochastic techniques may therefore allow fatigue estimates to be obtained readily.

The approach is less applicable when flow separation is exhibited. However, the method of superposition may introduce a degree of conservativeness into load estimates, which designers may favour. One may also expect that at higher oscillatory frequencies, such as that expected for rotational sampled turbulence, the delay in stall would be greater than that observed

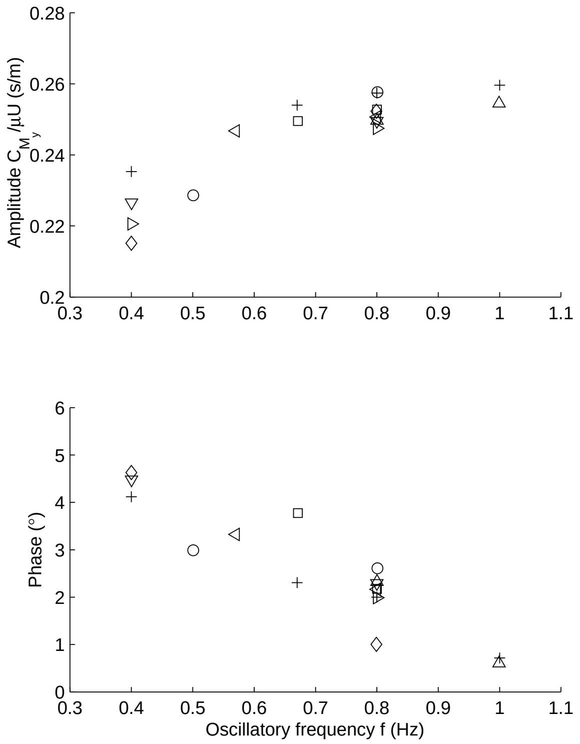

Figure 4: Amplitude and phase of the out-of-plane root bending moment for multi-frequency cases about $\lambda = 4.5$, where each frequency component has a current number of $\mu = 0.1$. Each case is depicted by a separate set of symbols, and the estimates from the single frequency oscillatory tests are shown by the (+) symbol.

here, and predictions of the maximum load from a reconstruction would be closer to that expected for a multi-frequency case.

In terms of defining the limitations of the approach, it may be concluded that the velocity amplitudes exhibited by the aforementioned cases are towards the upper ranges of what would be expected for a tidal turbine. However, if the rotor were perturbed about a lower mean tip-speed ratio, such as closer to the maximum power state, the flow separation would be expected to occur at lower current numbers. Conversely, increased severity of stall would be expected for equivalent velocity amplitudes. The cases analysed here are considered to represent relatively minor separation, and the non-linearities associated with 'deep stall' would be expected to be much more significant. In these instances, more computationally demanding numerical approaches are likely to be deemed necessary.

Conclusions

This study has presented experimental test results of the blade root out-of-plane bending moment response to multi-frequency planar oscillations. The unsteady contribution was found to be relatively linear and the component in phase with acceleration relatively small. Comparisons between the amplitude and phase of the load between single and multi-frequency oscillations suggest that linear superposition is valid when the flow remains attached across the blade. For minor flow separation, linear superposition is likely to introduce a degree of conservativeness in load predictions. The findings are expected to be of interest to tidal turbine blade designers who wish to seek low computationally demanding methods by which to assess the dynamic hydrodynamic loads imparted on turbine blades.

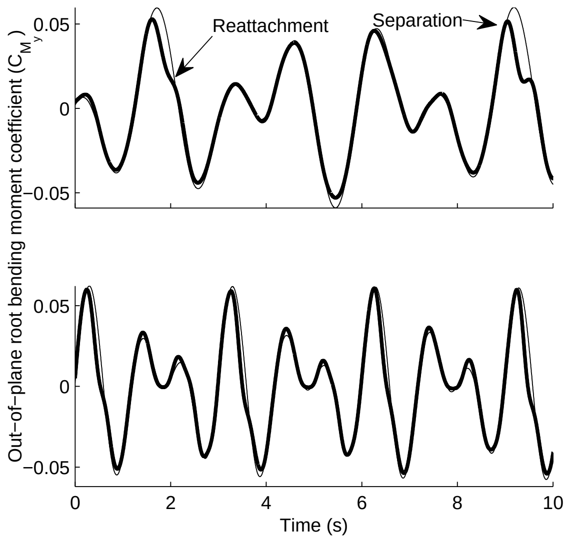

Figure 5: Time histories of the out-of-plane root bending moment for multi-frequency oscillations with frequency combinations of $f = 0.4$, 0.67Hz (upper) and $f = 0.67$, 1Hz (lower). The Current number of each component is $\mu = 0.15$, and the reconstruction using the transfer function is shown by the solid line.

Acknowledgments

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