Rotational sampling of waves by tidal turbine blades

DOI:

[10.1016/j.renene.2020.10.037](https://doi.org/10.1016/j.renene.2020.10.037)

Document Version

Accepted author manuscript

[Link to publication record in Manchester Research Explorer](#)

Citation for published version (APA):

Draycott, S., Steynor, J., Nambiar, A., Sellar, B., & Venugopal, V. (2020). Rotational sampling of waves by tidal turbine blades. *Renewable Energy*, 162, 2197-2209. <https://doi.org/10.1016/j.renene.2020.10.037>

Published in:

Renewable Energy

Citing this paper

Please note that where the full-text provided on Manchester Research Explorer is the Author Accepted Manuscript or Proof version this may differ from the final Published version. If citing, it is advised that you check and use the publisher's definitive version.

General rights

Copyright and moral rights for the publications made accessible in the Research Explorer are retained by the authors and/or other copyright owners and it is a condition of accessing publications that users recognise and abide by the legal requirements associated with these rights.

Takedown policy

If you believe that this document breaches copyright please refer to the University of Manchester's Takedown Procedures [<http://man.ac.uk/04Y6Bo>] or contact uml.scholarlycommunications@manchester.ac.uk providing relevant details, so we can investigate your claim.

Rotational Sampling of Waves by Tidal Turbine Blades

S.Draycott^{a,b,*}, J.Steynor^b, A. Nambiar^b, B. Sellar^b, V. Venugopal^b

^a*School of Mechanical, Aerospace and Civil Engineering, University of Manchester, Manchester M60 1QD, UK*

^b*School of Engineering, Institute for Energy Systems, The University of Edinburgh, Edinburgh, UK, EH9 3DW*

Abstract

The presence of waves exposes tidal stream turbines to large and cyclic hydrodynamic loads which significantly influence the design requirements for tidal turbine blades. Here we describe a loading phenomenon not previously considered in literature caused as blades rotationally sample an oscillating and vertically decaying wave-induced velocity field. Although implicitly incorporated into numerical models, the dominant causes and relative influence have not previously been considered.

In this article this effect is described through theoretical analysis and validated through scaled experiments; including irregular waves at angles to the rotor and current field. The associated loads are found to be strongly correlated to the wavenumber. The nature of the rotational-sampling-effect is confirmed through analysis of the experimental results, where characteristic sidebands are effectively predicted in the blade root bending moment spectra. It is estimated to account for between 8%–16% of the fatigue damage and between 7%–13% of the peak root bending moment for the conditions tested. A key finding is that two bilaterally-symmetrical oblique wave conditions do not produce equivalent loading patterns: one produces higher frequency oscillations. Additionally, it is found that the frequency of these loads reduces linearly with rotational speed; highlighting another consideration for tidal stream turbine operation.

Keywords: tidal stream turbine, irregular wave loads, blade fatigue, rotational sampling, oblique waves, sidebands

1. Introduction

Tidal stream turbines (TSTs) must be designed to survive the harsh ocean environment, and as such they must withstand the large unsteady hydrodynamic loads induced by the presence of waves, turbulence and shear. The combination of large waves and energetic currents in particular has the potential to cause catastrophic damage on TSTs whilst significantly contributing to fatigue. It is therefore important that the true nature of the loading resulting from waves is understood. This article addresses a previously under-explained phenomenon: rotational sampling of wave-induced velocities by tidal turbine blades. This effect occurs as the blade *samples* the vertically decaying wave-induced velocity field at its rotational frequency f_{rot} , where the wave-velocities themselves

*Corresponding author

Email address: Samuel.Draycott@manchester.ac.uk (S.Draycott)

are oscillating at the wave frequency, f_{wave} . The combined effect results in a high frequency loading pattern on the blades with dominant frequencies $Nf_{rot} \pm f_{wave}$ in a similar manner to amplitude modulation (where N is any integer and magnitudes decrease with N). The magnitude of these ‘sideband’ frequencies indicate their relative contribution to the loads and can be identified in experimental data.

There has been a significant amount of experimental work assessing TST loads and performance in the presence of waves, with a large proportion focusing on the influence of monochromatic regular waves. The first published work documenting this was that of Barltrop et al. (2006). This study, as well as Faudot and Dahlhaug (2012), Galloway et al. (2014), Fulton et al. (2015) and Guo et al. (2018) were used to validate Blade Element Momentum Theory (BEMT) codes including the presence of regular waves. Wave-induced variations in power and thrust have been further reported in Gaurier et al. (2013); De Jesus Henriques et al. (2014); Martinez et al. (2018); Draycott et al. (2019c), with additional studies discussing the influence of wave phase and turbine depth (Lust et al., 2013; Luznik et al., 2013).

To experimentally confirm the nature of the aforementioned rotational sampling effects it is required to have measurements on the blades of a TST model. A number of studies have included such measurements yet either the effect is non-existent or has not been adequately discussed or explained. Milne et al. (2013) specifically assessed blade loads in oscillatory flow but as there was no wave shear the rotational sampling effects were not present. Lust et al. (2013) looked at the significance of wave phase, but did not analyse individual blade loads as a result of this relative phase. This effect can be seen in experimental data in McCann (2007); Faudot and Dahlhaug (2012), where root bending moment and blade thrust values respectively show the characteristic high frequency oscillations. Additionally, Galloway et al. (2014, 2011) shows high frequency velocity oscillations as observed in the reference frame of a blade element in BEM code which further indicate this phenomenon.

As will become apparent, the presence of rotational sampling induced loads is more simple to identify in the frequency domain, and indeed, were noted in the spectral analysis presented in Gaurier et al. (2013). In that study the strain on blades in the presence of waves was reported and the characteristic high frequency peaks in the spectra were commented on briefly as “resonance phenomena”.

In this article we provide a description of the nature of this phenomenon along with accompanying theory and experimental validation of the theory presented. The influence of wave angle on the rotational-sampling induced loads are explored, and critical angles are identified which serve to significantly alter the dominant frequencies of the loading patterns by means of creating asymmetry in the sidebands. Irregular waves at several angles are used for the experiments and the loading on a 1:15 scale tidal turbine is analysed and reported. In addition to exploring the rotational sampling phenomena, experiments are carried out in conditions which have not previously been reported. Indeed the only known study to the authors assessing the effect of wave angle on scale turbine loads does so in regular wave conditions (Martinez et al., 2018). Only a small number of studies assess TST loads in irregular wave conditions (Stallard et al., 2013; Sos et al., 2017; Ordonez-Sanchez et al., 2019); where in all cases waves are collinear to the current direction, in many cases due to the nature of the test-facilities utilised.

The remainder of the article is laid out as follows. A theoretical analysis is presented in Sec-

tion 2, including the incorporation of wave angle on the velocities experienced along a rotating tidal turbine blade and the resulting root bending moments. Section 3 details the specifics of the test set-up: instrumentation, equipment & specified environmental conditions. Results are presented in Section 4, including key theoretical and experimental results and a comparison between theory and experiment. Further discussion and concluding remarks are offered in Section 5 and Section 6 respectively.

2. Theoretical analysis

This section lays out a theoretical analysis in order to understand the loads resulting from the rotational sampling of wave-induced velocities. The focus is on the wave-induced fluid velocities as seen in the reference frame of the blade (Section 2.1), including wave angles (Section 2.1.1). Simple approximations for the resulting normalised loads on the blade are detailed in Section 2.2.

2.1. Velocities experienced by a rotating tidal turbine blade

For a simple channel with no transverse shear, and with waves and currents collinear to each other, the velocity measured at a given position in a tidal channel can be approximated by the velocity contribution from the mean vertical shear, $\bar{U}(z)$, turbulence, $u'(x, y, z, t)$, and waves, $u_w(x, z, t)$:

$$U(x, y, z, t) = \bar{U}(z) + u'(x, y, z, t) + u_w(x, z, t) \quad (1)$$

where x is defined as positive in the streamwise direction, and y and z are the transverse and vertical co-ordinates respectively.

For the purpose of the analysis it is useful to ignore the turbulence as its effect in the time-domain cannot be deterministically described. The vertical shear and wave-induced velocities, however, can be. To first order, the wave-induced velocity is (Baddour and Song, 1990):

$$u_w(x, z, t) = \frac{gaK}{\sigma_r} \frac{\cosh(K(z+h))}{\cosh(Kh)} \sin(Kx - \sigma_{wave}t + \Phi_{wave}) \quad (2)$$

where a is the wave amplitude, g is acceleration due to gravity, σ is the angular frequency and σ_r is the relative angular frequency i.e. the frequency observed in a reference frame moving at the same velocity as the mean current. Φ_{wave} is the phase of the wave referenced at $t = 0$ and $x = 0$. Wavenumber, K , and σ_r are defined below:

$$\frac{\sigma}{K} = U \cos(\beta) + \sqrt{\frac{g}{K} \tanh(Kh)} \quad ; \quad \sigma_r = \sqrt{Kg \tanh(Kh)} \quad (3)$$

where β is the angle between the wave propagation direction and the current (used in Section 2.1.1).

The vertical shear is often well approximated as a power law (e.g. Dalrymple (1977)):

$$\bar{U}(z) = U_s \left(\frac{h-z}{h} \right)^{1/n} \quad (4)$$

where U_s is the velocity at the surface, and $1/n$ is the order of the power-law fit, often assumed to be $1/7$ in tidal channels. In FloWave this is closer to $1/15$.

To understand the wave and shear induced velocity variations at radial position, l , on the blade, one way to approach this is to obtain the instantaneous z position at a location l on the blade. For all of the presented analysis it is assumed that the rotor-plane-normal is aligned with the mean current direction. This z position then depends on the angle of the blade, θ , which for a turbine operating in speed control (as in the present case) can be expressed as a function of the rotational angular frequency, σ_{rot} , and initial phase of the blade being studied, Φ_{rot} . This is depicted in Fig. 1. The instantaneous z position can be described as:

$$z(l, t) = z_0 + l \cos(\theta(t)) = z_0 + l \cos(\sigma_{rot}t + \Phi_{rot}) \quad (5)$$

Figure 1: Diagram of turbine

The time-varying flow velocity, normal to the rotor plane, as a function of l can therefore be expressed as:

$$U(l, t) = U_s \left[\frac{h - (z_0 + l \cos(\sigma_{rot}t + \Phi_{rot}))}{h} \right]^{1/n} + \frac{gaK}{\sigma_r \cosh(Kh)} \cosh(K(z_0 + l \cos(\sigma_{rot}t + \Phi_{rot}) + h)) \times \sin(Kx - \sigma_{wave}t + \Phi_{wave}), \quad (6)$$

which provides the time-varying velocity at a point on the blade as a function of the wave parameters and turbine state. It accounts for both the turbine rotational frequency and the wave frequency, but ignores the reduction in velocity at the rotor as a result of induction. The reason why this can be neglected is detailed in Section 2.2 below.

2.1.1. Extension to include waves at angles

When waves are at an angle, β , to the current, one way to formulate this is to treat this as a time-varying effective position relative to the wave propagation direction, x_w . This is depicted in Fig. 2.

Figure 2: Top-down diagram of turbine, where wave is at an angle β to the current and rotor plane normal.

Under this approach the value of x_w can be determined by:

$$x_w(l, t, \beta) = l \sin(\sigma_{rot}t + \Phi_{rot}) \sin(\beta) \quad (7)$$

The formulation for the streamwise velocity at a point l on the blade becomes:

$$U(l, t) = U_s \left[\frac{h - (z_0 + l \cos(\sigma_{rot}t + \Phi_{rot}))}{h} \right]^{1/n} + \frac{ga(\beta)K \cos(\beta)}{\sigma_r \cosh(Kh)} \cosh(K(z_0 + l \cos(\sigma_{rot}t + \Phi_{rot}) + h)) \times \sin(-\sigma_{wave}t + \Phi_{wave} + Kl \sin(\sigma_{rot}t + \Phi_{rot}) \sin(\beta)) \quad (8)$$

Noting that in addition to the streamwise component of velocity (normal to the rotor-plane) reducing with $\cos \beta$ the amplitude, $a(\beta)$, of the wave is also a function of the wave-current angle. Formulations for the theoretical change in wave amplitude with current velocity can be found in Jonsson et al. (1970); Jonsson (1990).

2.2. Loads Resulting from Rotational Sampling Effects

For comparison to experimental measurements, root bending moments, *RBM*, values are utilised. This can be defined as a function of the local thrust force, w , at a point l on the blade as:

$$RBM(t) = \int_0^L w(l, t) l dl \quad (9)$$

where L is the length of the blade.

In uniform flow conditions it is often observed that the variation of force on a rotating blade is a linear function of l (Bir et al., 2011). The thrust force will also be a linear function of $\rho U(l, t)^2$ as both the lift and drag forces that provide the resultant thrust are a function of this term. The angle

of attack, and hence relative size of the lift and drag forces, and corresponding resulting thrust will also some function of the local instantaneous velocity, say $C(U(l, t))$. Hence:

$$w(l, t) = C(U(l, t))\rho U(l, t)^2 l \quad (10)$$

However, it is useful to assume that $C(U(l, t)) \approx C(\bar{U})$. This is valid under the assumption that variations in $U(l, t)$ are relatively small relative to the mean velocity, and the twist of the blade creates the same angle of attack along the blade length. Under these circumstances:

$$RBM(t) = C(\bar{U})\rho \int_0^L U(l, t)^2 l^2 dl \quad (11)$$

A normalised bending moment, RBM_{norm} , can be effectively used for comparison to measurements:

$$RBM_{norm}(t) = \int_0^L U(l, t)^2 l^2 dl \quad (12)$$

3. Experimental set-up and test plan

This section details the test set-up, including the TST model (Section 3.1.1), the FloWave basin (Section 3.1.2), and the instrumentation (Section 3.1.3). The wave-current conditions used in the experiments are described in Section 3.2.

3.1. Test set-up

3.1.1. The Model Turbine

The design and manufacture of the model scale TST is well described in Payne et al. (2017), and has been used in previous studies (Draycott et al., 2019a,c). Only a very brief description is provided in this article.

The TST is designed to be a 1:15 scale representation of a generic three-bladed bed-mounted horizontal axis device. The rotor diameter, D , is 1.2 m and the rotor axis is 1 m from the base. Integrated load sensors capture streamwise root bending moment, RBM , for each blade, along with thrust, T , and torque, Q , on the entire rotor. The generator is simulated using a direct drive servo motor, and angular position, θ , is obtained from the motor's encoder. For these tests the turbine was operated in speed control with a constant rotational velocity, ω , of 90 rpm: corresponding to a nominal tip-speed-ratio, λ , of 7.

3.1.2. The test facility

All experiments presented in this article were carried out at the FloWave Ocean Energy Research Facility (FloWave, www.flowave.eng.ed.ac.uk), Edinburgh, UK. The facility is a large circular combined wave-current tank with a 25 m diameter and a water depth of 2 m. The circular nature of the tank removes limitation on the direction of either the wave or current fields, which enables the generation of waves at arbitrary angles to the current. This capability enables the conditions defined in Section 3.2 to be produced. More information on the facility can be found in

Figure 3: Wave gauge array layout. Turbine rotor position is indicated by the light blue box.

Draycott et al. (2019c); Sutherland et al. (2017); Noble et al. (2015) where diagrammatic representations of the facility can be found alongside baseline measurements of flow variability and turbulence.

For the flow velocities used in these experiments the turbulence intensity is around 7%. The blockage ratio with the turbine installed is 2.3%. Due to the large horizontal dimension of the tank, and representative vertical dimension, there is not expected to be any unrealistic blockage effects during the tests.

3.1.3. Instrumentation and configuration

The instrumentation layout is described in Draycott et al. (2019a,b,c); where tables and to-scale drawings can be found. In addition to the turbine-integrated sensors discussed in Section 3.1.1, resistance-type wave gauges to measure surface elevation, η , an Acoustic Doppler Velocimeter (ADV) to measure current velocity, U , V , W , (in x , y , and z directions respectively), and a bottom-mounted six-axes load cell to measure the total forces, F , and moments, M , on the entire TST structure were used. A large wave gauge array has additionally been included, mounted over the turbine rotor, to infer wave directionality. This array layout is depicted in Fig. 3, with x and y positions given relative to the rotor centre.

3.2. Wave-current conditions

For these tests a single input wave spectrum was used in a mean current velocity of 0.8 m/s. The input wave condition was defined using a JONSWAP spectrum (Hasselmann et al., 1973) with a peak period (T_p) of 2.5 s and a significant wave height (H_{m0}) of 0.15 m. The angle of the input wave spectrum was varied to assess the influence of angle on the observed rotational sampling effects. The wave angles relative to the current chosen were 150° , 165° , 180° , 195° & 210° , with corresponding test identification (ID) values detailed in Table 1. Values of β_{in} are given relative to the mean current direction. Due to the opposing or near-opposing currents chosen, the measured H_{m0} values are larger than the input values due to the wave-current interaction. Near-opposing angles were chosen due to the higher wavenumbers and larger expected rotational sampling effects.

Table 1: Wave-current angles and corresponding test names

Input wave angle, β_{in} [°]	150	165	180	195	210
Test ID	A	B	C	D	E

4. Results

This section presents both the theoretical and experimental results. Examples of theoretical time and frequency domain *RBM* values are given in Section 4.1.1, before the effect of frequency and angle on the sideband amplitudes are assessed in Section 4.1.2 and Section 4.1.3 respectively. Experimental results are presented in Section 4.2: first the incident wave angles are calculated (Section 4.2.1) and used to estimate theoretical *RBM* spectra which are compared to measurements in Section 4.2.2.

4.1. Theoretical analysis of turbine blade response

4.1.1. Example load time-series and spectra

To understand the nature of the loads resulting from the rotational sampling of wave-induced velocities, it is useful to look at the theoretical results; where any effects of turbulence and rotational speed variation can be removed and the rotational sampling effects isolated. A simulated regular wave example is utilised in this section to explore this phenomenon (using Eq. (12) with Eq. (8)); with parameters chosen to match those of the turbine deployed in FloWave.

The dimensions of the turbine and facility have been fixed to the experimental equivalents. The mean current velocity is chosen to be 0.8 m/s, with no shear ($n=\infty$), $a = 0.05$ m, and $\beta = 180^\circ$. To ensure no spectral leakage, f_{rot} and f_{wave} have been chosen based on the simulation repeat time, T_r , of 64 s. These frequencies have also been chosen to match the experiments, and hence $f_{rot} = \frac{96}{64} = 1.5$ Hz, and $f_{wave} = \frac{26}{64} = 0.4063$ Hz. Random wave phases and initial blade rotational positions have been added to the time-domain simulation but make no difference to the resulting load spectra.

The results of the simulation in the time-domain and frequency domain are shown in Figs. 4 and 5 respectively. It is apparent from Fig. 4 that the wave-induced loads are complex, featuring oscillations at much higher frequencies than the wave frequency. This is also evident in Fig. 5 whereby there are a large number of visible high frequency peaks. It is noted that these peaks fall into 4 categories, defined in Table 2, which are colour coded in the figure. Away from the dominant peaks at lower frequencies, many of the small high frequency peaks would be impossible to separate from turbulence and noise in real measurements, but indicate the complex nature of the effect.

Assessing the load spectra it is apparent that the dominant peak is at the f_{wave} , with other significant peaks at $f_{rot} \pm f_{wave}$ and $2f_{rot} \pm f_{wave}$. For collinear wave-current conditions the magnitude of peaks at $Nf_{rot} + f_{wave}$ and $Nf_{rot} - f_{wave}$ are equal. The importance of these largest peaks is demonstrated in Fig. 4, where a reconstructed time-series using only these five peaks (by means of an Inverse Fast Fourier Transform (IFFT)) matches the full time-series well. This highlights that the category 1 and 3 peak types are of most importance: type 1 could be classified as the dominant

Figure 4: Example theoretical normalised RBM values from wave-induced velocities. Shown for a regular wave with $a=0.05$ m, $\beta = 180^\circ$, $f_{wave} = 0.4063$ Hz, and with $f_{rot}=1.5$ Hz.

Figure 5: Example theoretical normalised RBM spectrum from wave-induced velocities. Shown for a regular wave with $a=0.05$ m, $\beta = 180^\circ$, $f_{wave} = 0.4063$ Hz, and with $f_{rot}=1.5$ Hz.

Table 2: Categories of *RBM* peaks induced by rotational sampling of wave-induced velocities

Category	Notation	Description
1	Nf_{wave}	Multiples of the wave frequency
2	Nf_{rot}	Multiples of the rotational frequency
3	$Nf_{rot} \pm f_{wave}$	Multiples of rotational frequency $\pm$ wave frequency
4	$Nf_{rot} \pm 2f_{wave}$	Multiples of rotational frequency $\pm 2 \times$ wave frequency

wave effect whilst type 3 could be seen as the dominant rotational sampling effect. This rotational sampling effect is that which is described in Sections 1 and 2.

4.1.2. Relative size of sidebands

From the example shown in Section 4.1.1 it is evident that the rotational sampling of waves creates a complex high frequency loading pattern on the blades. This effect is largely explained by the type 3 peaks defined in Table 2, and are analogous to sidebands for amplitude modulated signals (e.g. Noguchi et al. (1986)). The amplitude and frequency of these sidebands are of critical importance as these will determine the number and magnitude of the associated loading cycles; the key drivers for the fatigue attributed to this effect.

The frequency and amplitude of the sidebands are a function of different phenomena. For a given wave frequency, the frequency of the important sidebands ($f_{rot} \pm f_{wave}$ and $2f_{rot} \pm f_{wave}$) is decided by the turbine rotational speed alone. Changing the rotational speed does not change the amplitude of the sidebands, but only the frequency they occur at. This suggests that the rotational speed can be lowered to reduce the number of fatigue cycles associated with the rotational sampling effects.

The amplitude of the sidebands is a result of a more complex phenomenon. Effectively, the larger the velocity variation observed in the reference frame of the blade through a rotation, the larger the amplitude of the rotational sampling induced loading would be. Hence, the amplitude of the sidebands are dominated by the wavenumber and the amplitude of the wave, along with the rotor position within the water column (and to a lesser extent blade length, profile etc.). Higher wavenumbers give rise to a greater vertical attenuation of wave-induced velocities and hence increase the velocity variation observed by a blade rotating within.

To quantitatively assess this effect a range of wave frequencies were simulated (1/32 Hz to 47/32 Hz in steps of 1/32 Hz) both following and opposing the current direction and the magnitude of the sidebands of the *RBM* spectrum have been extracted. This is presented in Fig. 6, showing type 3 peaks with N from 1 to 4 for both following and opposing waves. The magnitudes have been presented relative to the dominant peak at f_{wave} to remove the influence of wave amplitude. Apparent from Fig. 6 is that, as expected, the magnitude of the sidebands increases with wave frequency due to the increase in wavenumber. Also evident is that the sideband amplitudes for opposing waves are much larger than the following wave equivalents. Assessing Eq. (3), it is clear that this is because opposing currents increase the wavenumber for a given wave frequency whilst following currents cause a reduction. Wave frequencies above around 0.5 Hz are blocked in 0.8 m/s of opposing current and hence no results are presented for this region. It is evident

Figure 6: Value of peaks at $N\sigma_{rot} \pm \sigma_{wave}$ relative to peaks at σ_{wave} . Results in opposing waves are shown in dashed lines, while results in following waves are shown in solid lines. Up to $N=4$ shown.

from Fig. 6 that the magnitude of the sidebands can in some cases be comparable to the loads at the fundamental wave frequency yet are occurring at a much higher frequency and hence can be significant contributors to fatigue under these conditions.

4.1.3. Effect of wave angle

For the examples shown in Sections 4.1.1 and 4.1.2 the waves were aligned with the current direction, and it is observed that the magnitude of peaks at $Nf_{rot} + f_{wave}$ (positive sideband) and $Nf_{rot} - f_{wave}$ (negative sideband) are equal. This is no longer the case when waves are at an angle to the rotor. Wave heights and wavenumbers will be modified if they are interacting with a current at an angle. In addition, as described in Eq. (8), a rotating tidal turbine blade will ‘sample’ the x variation in wave velocity as well as the z variation. As the z and x wave-induced velocity variations are being sampled at the same frequency, f_{rot} , and the wave velocity itself is also oscillating at a fixed frequency, f_{wave} , these two rotational sampling effects (x and z) are effectively phase locked. This phase is determined by the relative wave angle and the wavenumber. This phase locking means that the apparent frequencies observed by the blade is biased either positively or negatively, and as such either the positive ($N\sigma_{rot} + \sigma_{wave}$) or negative ($N\sigma_{rot} - \sigma_{wave}$) sideband will be larger than the other. The larger sideband will be determined by the direction of rotation of the turbine relative to the wave direction.

To visualise the effect of wave angle an example is presented in Fig. 7. In this figure, the relative magnitude of the positive and negative sidebands (at $N = 1$) are presented when the incident wave direction is at angles ranging from 0 to 2π relative to the current (and rotor plane normal). This is shown for the same physical conditions and set-up used in Section 4.1.1, with $f_{wave} = 0.4036$ Hz and $f_{rot} = 1.5$ Hz. From Fig. 7 it is evident that there are certain angles when one of the sidebands essentially disappears. These angles are a function of the wave frequency and current velocity. For

Figure 7: Effect of wave angle on relative theoretical size of positive and negative sidebands for $f_{wave} = 0.4063$ Hz and $f_{rot} = 1.5$ Hz.

this example, the critical angle for the near-opposing waves is at around $\pi \pm \pi/4$; where at $\pi + \pi/4$ there is only the positive sideband and at $\pi - \pi/4$ there is only the negative sideband. To further visualise this effect simulated *RBM* time-series for these angles are shown in Fig. 8. The observed difference matches the frequency domain observation, in that the wave incident at $\pi + \pi/4$ induces higher frequency load oscillations than at $\pi - \pi/4$. This has further implications for the fatigue associated with rotational sampling effects.

Fig. 7 is extended to include a range of wave frequencies in Fig. 9. The black lines on this figure indicate the angle at which the maximum relative value of the sideband occurs for every frequency. This figure highlights the wavenumber dependency of these critical angles; indicating that the wave angle which produces the phase-locking associated with this critical angle occurs at lower angles for higher frequencies/wavenumbers. The larger wavelengths (lower frequencies/wavenumbers) require a larger angle to achieve the same phase relationship between x and z rotational sampling effects.

4.2. Experimental

This section presents the results of the experiments detailed in Section 3, and compares the results to the theory developed in Section 2.

Figure 8: Example time-series for two incident wave angles with $f_{wave} = 0.4063$ Hz and $f_{rot} = 1.5$ Hz.

Figure 9: Effect of wave angle on relative theoretical size of positive and negative sidebands for frequencies ranging from 0.125 Hz to 0.5 Hz. Black lines indicate the angle at which the maximum relative value of the sideband occurs for each frequency.

Figure 10: Calculation of frequency-dependent wave angles using the PTPD method. Input angles are given as dashed lines, with calculated values in solid lines of the same colour.

4.2.1. Calculation of wave angle at turbine location

Prior to comparing measurements to theory it is required to know the angle of the waves at the turbine location. Although the input wave direction was specified as detailed in Section 3.2 the wave angles at the model location will differ due to refraction (Jonsson and Skovgaard, 1978). This is particularly complex in FloWave due to the circular nature of the wave basin. These refraction effects are made more complex by the turbine wake, resulting in a highly spatially variable flow field on which waves must propagate. To estimate the wave angles at the turbine location a Phase-Time-Path-Difference (PTPD) approach was used (see Draycott et al. (2016); Fernandes et al. (2000) for details) in combination with the wave gauge measurements taken with the array detailed in Fig. 3. The results of this PTPD analysis is shown in Fig. 10. Only results up to 0.4 Hz were found to be consistent (gauge triads providing similar angle estimates) and hence higher frequencies are omitted: providing values up to the peak frequency of the spectrum. For simplicity (and somewhat out of necessity) the mean value computed for this reliable range is used as the input angle of the whole spectrum for the theoretical comparisons.

Assessing Fig. 10 a number of observations are apparent. The first is that the measured apparent angles differ from the input angles, yet the observed refraction effects are not symmetrical about π . Second, the deviation from the input angles show no regular trend with frequency. One would expect the level of refraction to increase with frequency, and hence would consistently expect the deviation from the input angles to increase with frequency, for all input angles. The fact that the observed discrepancy is not symmetrical and does not increase with frequency highlights the complex nature of the spatially variable current field in the wake of the turbine.

4.2.2. Spectral analysis and comparison to theory

Spectra of the blade root bending moments *RBM*, thrust *T*, horizontal velocity components *U* & *V*, and surface elevation η , are presented in Fig. 11 for the five wave angles. Additionally, Fig. 12 shows the *RBM* spectra with theoretical values overlaid. These theoretical values have been calculated using the simulated data used to plot Fig. 9: the normalised amplitudes of the sideband *RBM* values have been multiplied by the measured *RBM* values at f_{wave} .

Assessing Fig. 11, it is evident that the *RBM* spectra for all of the blades are near identical, and all show energy content at the aforementioned sideband frequencies. The second set of sidebands (at $2f_{rot} \pm f_{wave}$) are also just visible. It is also apparent that peaks in *RBM* from rotational sampling effects are negligible in the thrust spectra, yet shear/tower shadow effects are present at 3p (4.5 Hz). This is because the phase relationship between the blades for the rotational sampling effects cause the net associated thrust to approximately cancel, which is not the case for the tower shadow/shear. The integrals of the sidebands, representing the total load variance from the rotational sampling effects, are significantly larger than the tower shadow/vertical shear effects in all cases.

Also apparent in Fig. 11 is that there is a double peak in the *RBM* spectra for case A (as defined in Table 1). This, however, is not apparent in the sidebands (when assessing Fig. 12). This double peak is difficult to fully explain. Initially, it appears to be related to wave reflection as it is also apparent in the surface elevation spectrum. However, if the reduced amplitude part of the spectrum was due to reflection (partial standing wave), then this would be associated with a decrease in vertical velocity and increase in horizontal velocity. Hence, if reflection-based one would expect the trough in the surface elevation spectrum to correspond to a peak in the *RBM* spectrum, which is not observed. Additionally a reduction in velocity is noted in the horizontal *V* component rather than an increase, but is not apparent in *U*.

Assessing Fig. 12 it is evident that good agreement is found with theory in general. For all but case E the sideband asymmetry direction is predicted correctly and the peak magnitudes are comparable; particularly for cases B and C. For case D the negative sideband almost completely disappears, which is expected from theory at around this angle (see Figs. 7 and 9) for this frequency range. The opposite is true for case A, where the positive sideband is much smaller. In general, the smaller peaks are not particularly well predicted due to the larger influence of turbulence on the loads at these frequencies.

Case E is not predicted well and seems to be an outlier in the data. It appears that the angle inferred from the wave gauges does not align with that observed by the turbine. There are a number of indicators for this. One is the amplitude of the surface elevation spectra. It is actually larger than cases A and D in size ($\pm 30^\circ$ from opposing), yet one would expect the amplitude to be further reduced at a smaller relative angle to the current. From the relative H_{m0} value, a much smaller angle value of 199.8° would be suggested. The ratio of *U* to *V* wave-induced velocity components, however, does agree well with the wave gauges, and suggests an angle of 225.5° . In addition, the asymmetry in the sideband peaks in the *RBM* is not as expected, and is the only case not well predicted by theory. Lastly, the fact that case D appears to have refracted by 15° (when case B had negligible inferred refraction) suggests that waves arriving from one side of the tank are going through a very different flow to the other side. The authors suggest that, due to these contradictory indicators, there are likely multiple paths for waves to arrive at the turbine rotor for waves input at this angle. This is perhaps due to the complex spatial variation in flow due to the

Figure 11: Amplitude spectra of measured environmental and turbine parameters for five input angles as defined in Table 1

Figure 12: Spectra of *RBM* compared to theory. Top plots show close up of first sideband frequencies in region enclosed by grey box in bottom plots. A-F correspond to the wave angles defined in Table 1

turbine wake and current generation system in FloWave. The fact this is only observed on case E and not case A (same relative angle input) is suspected to be a combination of the turbine being located away from the tank centreline (0.6 m), and the asymmetrical flow field resulting from the wake.

5. Discussion

Section 4 confirms the nature of the rotational sampling effects and the applicability of the theory presented. This section aims to assess the significance of the rotational sampling effects in terms of its contribution to blade fatigue and peak loads (Section 5.1), before assessing potential mitigation strategies in Section 5.2.

5.1. Implications of rotational sampling effects on blade fatigue & peak loads

In order to effectively assess the relative contribution from rotational sampling induced loads, actual experimental data is utilised; accounting for the effect of turbulence and shear on the total fatigue loading. A rainflow analysis was performed on the experimental data. To assess the proportion of fatigue associated with rotational sampling, the energy content associated with the first set of sidebands at $(f_{rot} \pm f_{wave})$ was removed from the *RBM* spectra. *RBM* time series were generated from these modified spectra using an IFFT and were compared with the non-filtered experimental results. A notch filter was applied to $f_{rot} \pm f_{wave}$, where the range of f_{wave} values

Figure 13: Example of filtering process for removal of sidebands for case C (defined in Table 1). FFT and time-series shown with and without sidebands removed. Also shown is the equivalent with just the wave frequencies considered. Mean values have been added to the time-series to aid visibility.

has been taken to be 0.3 Hz to 0.6 Hz. Additionally data is low pass filtered to 4 Hz to remove the unequal contribution of sensor noise on the apparent *RBM* oscillations. It is to be noted that this does not enable assessment of the complete rotational sampling induced loads (as described in Section 4.1), however, does account for the dominant effect as observed in the experimental results (Section 4.2). Removing all frequency components associated with rotational sampling (see Fig. 5) would additionally remove a significant proportion of loading due to turbulence.

An example of the implemented filtering process is shown for Case C in Fig. 13. The FFT and resulting time-series is depicted with and without the sidebands removed. Also shown for interest is the result if only the wave frequencies are considered (0.3 Hz to 0.6 Hz). Resulting from rainflow analysis, the amplitudes of the load cycles for the filtered (No Side Bands) and unfiltered (Total) data is presented in Fig. 14 for all three *RBM* measurements and for each of the five input angles. For completeness the equivalent result with only the wave frequencies considered is shown. It is evident that in all cases the removal of the sidebands serves to decrease the number of high amplitude oscillations and slightly increase the number of low amplitude oscillations.

This reduction in large amplitude load oscillations and increase in smaller amplitude oscillations is observed equally across all three blades indicating the phase-independence of the result. As the amplitude of the load fluctuations is increased when the sidebands are present, this would indicate an increase in expected fatigue damage.

To further quantify the fatigue damage, Miner's rule can be utilised which is defined as:

$$D = \sum_{i=1}^k \frac{n_i S_i}{N_i S_i} = \sum_{i=1}^k \frac{n_i}{N_i} \quad (13)$$

where D is the accumulated damage, the number of cycles to failure at stress S_i (and mean M_i) is N_i , and the number of cycles at S_i (and mean M_i) is n_i .

As the scaled blade will not have analogous $S - N$ relationships as a full-scale blade, and the $S - N$ curves for the blade in question is unknown, it is useful to normalise the effective damage.

Figure 14: Histograms of cycle amplitudes for *RBM* time-series with (black) and without (red) first set of sidebands removed for cases A-E (defined in Table 1). Also shown is the equivalent histograms with just the wave frequencies considered (blue). Note that for each test all three blades are presented using different lines, and cycle counts include half cycles (0.5). Amplitude bins are in 0.5 Nm increments.

For a given M_i , it is often observed that S is inversely proportional to N and approximately linear (as observed in Sutherland and Mandell (2005) and others). Under this assumption the normalised damage can be expressed as:

$$D_{norm} = \sum_{i=1}^k n_i S_i \quad (14)$$

which, if the effect of cycle means is assumed insignificant (relatively valid for this application due to dominant mean load coming from the mean current), is equivalent to the amplitude of the load fluctuations multiplied by the corresponding count which can be computed using the first moment of the plots presented in Fig. 14. To assess the relative influence of the rotational sampling induced loads on the blades the total damage with and without the first set of side bands has been assessed. The increase in total normalised damage with the side-band influence is presented in Table 3. It can be inferred that the total fatigue damage attributed to rotational sampling effects is approximately between 8% and 16% for cases A-E assessed in this study, which is a significant proportion of total fatigue loads which includes turbulence (up to 4 Hz), the dominant wave effect (at f_{wave}), shear and tower shadow effects.

Also shown in Table 3 is the increase in maximum *RBM* values resulting from rotational sampling. The contribution to peak loads is found to be between 7% to 13%.

5.2. Potential mitigation strategies

As the rotational sampling induced loads are a function of both the environmental forcing and the turbine state it is possible to mitigate their effect through the control of the turbine. As noted in Section 4.1.2, theoretically, it can be seen that the magnitude of the loads cannot be reduced by the

Table 3: Increase in normalised damage as a result of first set of sidebands

Name	A	B	C	D	E
$1 - \frac{D_{norm,noSB}}{D_{norm,total}}$	0.0812	0.128	0.158	0.118	0.0858
$1 - \frac{RBM_{max,noSB}}{RBM_{max,total}}$	0.0652	0.114	0.133	0.0754	0.113

turbine operation. However, the rotational speed determines the frequency of the oscillations, and hence the total number of associated loading cycles. Therefore a linear decrease in the count of load cycles attributed to rotational sampling can be expected with any given reduction in rotational speed. This suggests that, for exposed tidal locations, operating at a lower tip speed ratio may be beneficial; also serving to reduce cavitation effects on blades. This conclusion, however, does not consider other consequences of operating at lower rotational speeds, such as the increased likelihood/frequency of dynamic stall and its subsequent effects on loads and performance. Hence, a full assessment of the relative influence of these effects is desired for a given rotor design.

6. Conclusions

This paper focuses on the loading patterns experienced by tidal turbine blades as they rotate through a wave-induced velocity field which has notable vertical decay. Through theoretical and experimental work the effect is parameterised and the theory validated, and found to be significant when wavenumbers are high; typical when waves are high frequency or opposing the current direction. The effect of wave angle is explored and it is concluded that waves incident at equal but opposite relative angles do not result in comparable loads. One of these angles (dependent on turbine rotation direction) will induce higher frequency oscillations of equal magnitude and hence accumulate more fatigue damage. Through analysis of the experimental data it is found that the rotational sampling induced loads account for between 8% and 16% of total fatigue damage and between 7% and 13% of observed peak root bending moment for the conditions tested. It is concluded that the effect can be reduced by lowering the rotational speed of the turbine. The results in this paper therefore offer new insight into the nature of wave loading on tidal turbine blades, and provide fresh consideration for those designing and operating tidal turbines.

Acknowledgements

The authors are grateful for financial support from the U.K. Engineering and Physical Sciences Research Council through FloWTurb: Response of Tidal Energy Converters to Combined Tidal Flow, Waves and Turbulence (EP/N021487/1) and SuperGen UK Centre for Marine Energy Research (EP/M014738/1). The authors also acknowledge support from the EPSRC for funding the FloWave Ocean Energy Research facility (EP/I02932X/1). The authors are extremely grateful to the staff at the FloWave facility, in addition to the FloWTurb project team, for making this research possible.

References

References

- Baddour, R. E., Song, S., 1990. On the interaction between waves and currents. *Ocean Engineering* 17 (1-2), 1–21.
- Barltrop, N., Varyani, K. S., Grant, A., Clelland, D., Pham, X., 2006. Wave-current interactions in marine current turbines. *Proceedings of the Institution of Mechanical Engineers, Part M: Journal of Engineering for the Maritime Environment* 220 (4), 195–203.
URL <http://journals.sagepub.com/doi/10.1243/14750902JEME45>
- Bir, G. S., Lawson, M. J., Li, Y., 2011. Structural Design of a Horizontal-Axis Tidal Current Turbine Composite Blade (October), 797–808.
- Dalrymple, R. A., 1977. A numerical model for periodic finite amplitude waves on a rotational fluid. *Journal of Computational Physics* 24 (1), 29–42.
- De Jesus Henriques, T. A., Tedds, S. C., Botsari, A., Najafian, G., Hedges, T. S., Sutcliffe, C. J., Owen, I., Poole, R. J., 2014. The effects of wave-current interaction on the performance of a model horizontal axis tidal turbine. *International Journal of Marine Energy* 8, 17–35.
URL <http://dx.doi.org/10.1016/j.ijome.2014.10.002>
- Draycott, S., Davey, T., Ingram, D. M., Day, A., Johanning, L., 2016. The SPAIR method: Isolating incident and reflected directional wave spectra in multidirectional wave basins. *Coastal Engineering* 114, 265–283.
- Draycott, S., Nambiar, A., Sellar, B., Davey, T., Venugopal, V., 2019a. Assessing extreme loads on a tidal turbine using focused wave groups in energetic currents. *Renewable Energy* 135, 1013–1024.
URL <https://doi.org/10.1016/j.renene.2018.12.075>
- Draycott, S., Payne, G., Steynor, J., Nambiar, A., Sellar, B., Davey, T., Noble, D., Venugopal, V., 2019b. Environmental & load data: 1:15 Scale tidal turbine subject to a variety of regular wave conditions. *Data in Brief* 23 (March), 103732.
URL <https://doi.org/10.1016/j.dib.2019.103732>
- Draycott, S., Payne, G., Steynor, J., Nambiar, A., Sellar, B., Venugopal, V., 2019c. An experimental investigation into non-linear wave loading on horizontal axis tidal turbines. *Journal of Fluids and Structures* 84, 199–217.
URL <https://linkinghub.elsevier.com/retrieve/pii/S0889974618305073>
- Faudot, C., Dahlhaug, O. G., 2012. Prediction of Wave Loads on Tidal Turbine Blades. *Energy Procedia* 20, 116–133.
URL <http://linkinghub.elsevier.com/retrieve/pii/S1876610212007448>
- Fernandes, A. A., Sarma, Y. V. B., Menon, H. B., 2000. Directional spectrum of ocean waves from array measurements using phase/time/path difference methods. *Ocean Engineering* 27 (4), 345–363.
- Fulton, J., Luznik, L., Flack, K., Lust, E., 2015. Effects of waves on BEM theory in a marine tidal turbine environment. *OCEANS’15 MTS/IEEE*.
URL <http://ieeexplore.ieee.org/abstract/document/7401890/>
- Galloway, P. W., Myers, L. E., Bahaj, A. S., 2011. Experimental and numerical results of rotor power and thrust of a tidal turbine operating at yaw and in waves. *World Renewable Energy Congress*, 2246–2253.
- Galloway, P. W., Myers, L. E., Bahaj, A. S., 2014. Quantifying wave and yaw effects on a scale tidal stream turbine. *Renewable Energy* 63 (2014), 297–307.
URL <http://linkinghub.elsevier.com/retrieve/pii/S0960148113004977>
- Gaurier, B., Davies, P., Deuff, A., Germain, G., 2013. Flume tank characterization of marine current turbine blade behaviour under current and wave loading. *Renewable Energy* 59, 1–12.
URL <http://dx.doi.org/10.1016/j.renene.2013.02.026>
- Guo, X., Yang, J., Gao, Z., Moan, T., Lu, H., 2018. The surface wave effects on the performance and the loading of a tidal turbine. *Ocean Engineering* 156 (May), 120–134.
URL <https://doi.org/10.1016/j.oceaneng.2018.02.033>
- Hasselmann, K., Barnett, T. P., Bouws, E., Carlson, H., Cartwright, D. E., Enke, K., Ewing, J. A., Gienapp, H., Hasselmann, D. E., Kruseman, P., Meerburg, A., Muller, P., Olbers, D. J., Richter, K., Sell, W., Walden, H., 1973. Measurements of Wind-Wave Growth and Swell Decay during the Joint North Sea Wave Project (JONSWAP). *Ergänzungsheft zur Deutschen Hydrographischen Zeitschrift Reihe A*(8) (12), p.95.

Jonsson, I. G., 1990. Wave-current Interactions. In: LeMehaute, B., Hanes, D. M. (Eds.), *The Sea, Ocean Engineering Science*. Vol. 9. Wiley-Interscience publications, New York, Ch. 7, pp. 65–120.

Jonsson, I. G., Skougaard, C., Wang, J. D., 1970. Interaction between waves and currents. *Coastal Engineering Proceedings* 1 (12).

Jonsson, I. G., Skovgaard, O., 1978. Wave refraction across a shearing current. *Coastal Engineering Proceedings* 1 (16).

Lust, E. E., Luznik, L., Flack, K. A., Walker, J. M., Van Benthem, M. C., 2013. The influence of surface gravity waves on marine current turbine performance. *International Journal of Marine Energy* 3-4, 27–40.
URL <http://dx.doi.org/10.1016/j.ijome.2013.11.003>

Luznik, L., Flack, K. A., Lust, E. E., Taylor, K., 2013. The effect of surface waves on the performance characteristics of a model tidal turbine. *Renewable Energy* 58, 108–114.
URL <http://dx.doi.org/10.1016/j.renene.2013.02.022>

Martinez, R., Payne, G. S., Bruce, T., 2018. The effects of oblique waves and currents on the loadings and performance of tidal turbines. *Ocean Engineering*, 55–64.
URL <https://doi.org/10.1016/j.oceaneng.2018.05.057>

McCann, G., 2007. Tidal current turbine fatigue loading sensitivity to waves and turbulence—a parametric study. ... of the 7th European Wave and Tidal Energy ... (October).
URL http://www.gl-garradhassan.com/assets/downloads/Tidal_Current_Turbine_Fatigue_Loading_Sensitivity

Milne, I. A., Day, A. H., Sharma, R. N., Flay, R. G. J., 2013. Blade loads on tidal turbines in planar oscillatory flow. *Ocean Engineering* 60 (Supplement C), 163–174.
URL <http://www.sciencedirect.com/science/article/pii/S0029801812004465>

Noble, D. R., Davey, T., Smith, H. C. M., Kaklis, P., Robinson, A., Bruce, T., 2015. Characterisation of spatial variation in currents generated in the FloWave Ocean Energy Research Facility. In: *Proceedings of the 11th European Wave and Tidal Energy Conference*. Nantes, France, pp. 1–8.

Noguchi, T., Daido, Y., Nossek, J. A., 1986. Modulation techniques for microwave digital radio. *IEEE Communications Magazine* 24, 21–30.

Ordóñez-Sánchez, S., Allmark, M., Porter, K., Ellis, R., Lloyd, C., O'Doherty, T., Johnstone, C., 2019. Analysis of a Horizontal Axis Tidal Turbine Performance in the Presence of Regular and Irregular Waves using Two Control Strategies. *Journal of Marine Science and Engineering Under Review*.

Payne, G. S., Stallard, T., Martinez, R., 2017. Design and manufacture of a bed supported tidal turbine model for blade and shaft load measurement in turbulent flow and waves. *Renewable Energy* 107, 312–326.

Sos, M., Johnston, L., Walker, J., Rahimian, M., 2017. The Impact of Waves and Immersion Depth on Horizontal Axis Tidal Turbine Performance. *European Wave and Tidal Energy Conference*, 1–8.
URL <http://www.albayan.ae>

Stallard, T., Collings, R., Feng, T., Whelan, J., 2013. Interactions between tidal turbine wakes: Experimental study of a group of 3-bladed rotors. *Philosophical Transactions of the Royal Society of London: A* 371 (1985), 1471–2962.

Sutherland, D. R. J., Noble, D. R., Steynor, J., Davey, T. A. D., Bruce, T., 2017. Characterisation of Current and Turbulence in the FloWave Ocean Energy Research Facility. *Ocean Engineering* 139, 103–115.

Sutherland, H., Mandell, J., 2005. Optimized Goodman Diagram for the Analysis of Fiberglass Composites Used in Wind Turbine Blades. *ASME/AIAA Wind Energy Symposium (AIAA-2005-0196)*.