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MODELLING TECHNIQUES FOR UNDERWATER NOISE GENERATED BY TIDAL TURBINES IN SHALLOW WATERS

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ABSTRACT

The modelling of underwater noise sources and their potential impact on the marine environment is considered, focusing on tidal turbines in shallow water. The requirement for device noise prediction as part of environmental impact assessment is outlined and the limited amount of measurement data and modelling research identified. Following the identification of potential noise sources, the dominant flow-generated sources are modelled using empirical techniques. The predicted sound pressure level due to inflow turbulence for a typical turbine is estimated to give third-octave-bandwidth pressure levels of 119 dB re 1 μPa at 20 metres from the turbine at individual frequencies. This preliminary estimate reveals that this noise source alone is not expected to cause permanent or temporary threshold shift in the marine animals studied.

INTRODUCTION

The development and installation of renewable energy devices, including tidal turbines, requires extensive environmental impact assessment studies. Part of this procedure involves predicting the noise from devices and its likely effect on marine animals [1, 2]. However, no attempts to model turbine noise at the design stage appear to exist.

Tidal turbine noise will be influenced by many factors, such as blade design, water depth, tidal channel velocity profile and local bathymetry. Thus prediction of noise and its impact on marine life will be site and device specific and a limited set of measured data will not be sufficient for all designs. Thus further studies into the effect of tidal turbines on marine life are necessary [3]. Fig 1. shows a typical horizontal axis tidal turbine (HATT) similar to that studied in this investigation.

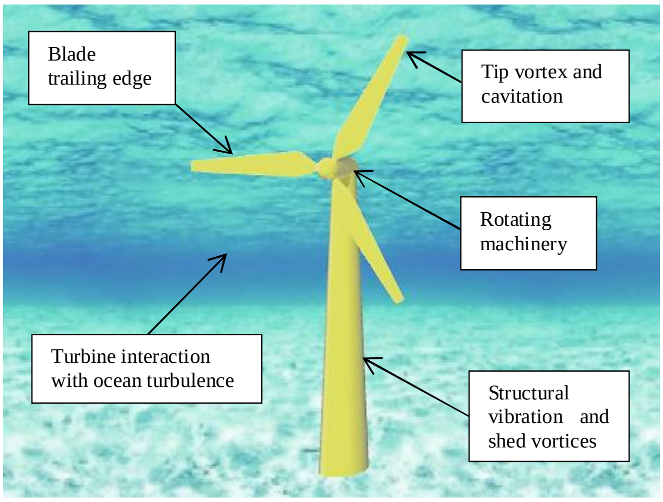

Figure 1. Potential Sources of Noise for a Typical Horizontal Axis Tidal Turbine (HATT).

This paper concerns the modelling of noise sources for use in the assessment of noise influence on marine life. Firstly, the problem is clearly defined, along with an identification of possible turbine noise sources. Secondly, modelling approaches are outlined and a case study presented using an empirical analysis of two possible noise sources. Finally, further work is outlined along with the conclusions from this investigation.

UNDERWATER NOISE

In the assessment of an underwater noise source and its impact on the marine environment, the existing background noise can be significant. The world's oceans can be considered noisy [4, p.11], and thus the sound produced by a turbine must be assessed relative to the 'ambient' noise of the site. Acoustic noise sources in the North Sea are outlined by Ainslie *et al.* [4, p.11] under the headings of natural sources and anthropogenic (both intentional and unintentional) sources. An operation tidal turbine lies in the third class of noise source (i.e. unintentional anthropogenic), which also includes activities such as shipping and offshore operations. Factors such as weather, which contributes to surface noise, and variety of marine species inhabiting the area will have an effect on the ambient noise of a site, with higher ambient noise generally at lower frequencies (< 3 kHz) [1]. Thus measurement of site specific ambient noise is important to give baseline data in an impact assessment.

Since tidal turbines will generally operate in shallow water, acoustic transmission losses and other shallow water propagation effects must be considered when predicting far field sound pressure level (SPL).

Effect on Marine Environment

When assessing the effect of anthropogenic noise on marine life, a number of contributing factors must be taken into account. These include the frequency range, acoustic pressure and exposure duration. The possible effects include: masking, behavioural change, hearing threshold shift (both temporary and permanent), and even death [3].

The SPL alone does not provide enough information to assess acoustic impact on marine species. The duration of exposure to the noise is also important. This can be hard to assess, as the behavioural patterns and movements of an animal within the region of a renewable energy device are hard to predict. Richards *et al.* [1] apply dosage criteria intended for humans in the assessment of acoustic impact of the 'SeaGen' marine current turbine, as well as formulations for estimating temporary threshold shift (TTS) and permanent threshold shift (PTS). This is due to the fact that no official criteria exist for application to marine animals in the context of noise impact assessment.

Another applicable model is that of Richardson *et al.* [5] termed 'Theoretical zones of noise influence', based on omnidirectional spreading and categorises the effect of noise on marine life based on distance from the source. An audiogram can also be used to compare SPL to the hearing threshold of marine animals over a range of frequencies [4, p.16]. This data is important when identifying which noise sources are likely to affect certain species. A review of techniques relating to the effects of noise exposure on marine mammals is provided by Southall *et al.* [6].

Whilst the impact of sound on the marine environment is not fully understood, the techniques discussed here represent current practice in the field [1, 7]. The focus of this work is on improving noise source prediction rather than the development of impact assessment techniques.

Existing Tidal Turbine Noise Measurements

Only two sources of measured data are known to the authors, that of Parvin *et al.* [8] on a full-scale device of 0.3MW rated power, and the experimental work of Wang *et al.* [9] on a 0.4 metre diameter model scale HATT in a cavitation tunnel. Direct comparison of the data reported with the present study is not appropriate, due to differences in the turbines being studied. However, these results provide some indication of the likely SPLs that may be expected from a HATT.

Parvin *et al.* [8] report an overall effective source level of 166 dB re 1µPa at one metre for the 'SeaFlow' turbine, rated at 0.3MW. This value is predicted from measurements made at distances ranging between 100 metres and one kilometre from the turbine, accounting for ambient noise. Richards *et al.* [1] scale these measurements to a 1MW turbine assuming that sound power scales linearly with generator power. The maximum tidal velocity is expected to be approximately three metres per second. Wang *et al.* [9] scale experimental measurements, accounting for background noise, for a turbine of 12 metre diameter operating in a tidal velocity of 2.57 metres per second. They report a maximum SPL of ~131 dB re 1µPa at one metre (at 20 Hz), reducing to ~112 dB re 1µPa at one metre for higher frequencies (~500Hz), presented for 1 Hz bandwidths. The equivalent SPLs for one-third-octave bandwidths at these frequencies are ~ 138 dB re 1µPa at one metre and ~133 dB re 1µPa at one metre for 20 Hz and 500 Hz respectively.

NOISE MODELLING APPROACHES

'Low Level' Approaches

These approaches are characterised by simplified modelling of the fluid flow and low computational cost, allowing fast analysis of multiple candidate designs. Tidal turbine performance can be investigated using blade element momentum (BEM) codes, such as *CCAV*, a version of *CWIND* [10]. As well as predicting turbine power, *CCAV* provides prediction of likely cavitation type and location for a turbine operating in a defined water depth and an assumed water column velocity profile. Since the interaction of ocean turbulence with a structure is a noise source, the characterisation of turbulence is important. Gant and Stallard [11] note there is a limited amount of measured data for turbulent ocean boundary layer profiles but nevertheless provide recommendations for modelling turbulence in shallow water.

Sound pressure levels due to noise sources generated by cavitation and turbulence can be modelled empirically (e.g. see [12, 13]). Such approaches consist of formulae derived from experiment or full scale data. However, their applicability to specific problems should be carefully considered, as the relationships may be derived from limited data sets.

‘High Level’ Approaches

Computational fluid dynamics (CFD) approaches can be used to model fluid flows more accurately than simpler techniques and allow detailed visualisation of the flow. This technique is computationally expensive, reducing the potential for high-number ‘design selection’ type studies but is more suited to device optimisation at later design stages. For analysing turbine power, the Reynolds Averaged Navier Stokes (RANS) equations are used [14], where all scales of turbulence are modelled. However, for problems involving estimation of noise, direct numerical simulation (DNS) or large-eddy simulation (LES) may be used. These methods resolve turbulence scales up to a specified cutoff wave number, allowing more accurate noise source prediction. The far field sound is then predicted using a propagation model such as the Ffowcs-Williams Hawkings (FW-H) or Kirchoff formulations (see [15] for a summary of hybrid approaches to aeroacoustic problems). Whilst such approaches have been extensively validated for cases such as single aerofoils [16], their successful application for noise prediction of full propeller geometries at high Reynolds numbers is still in its infancy. One attempt to compute sound pressure level of a ducted fan using LES data [17] reports poor accuracy below 2.5 kHz, the range of interest for tidal turbines according to [1].

Shallow Water Acoustic Models

A water column can be considered ‘acoustically shallow’ if the sound propagates to a receiver by repeated reflections from the seabed and surface [18, Chap. 6]. Thus assessment of received sound levels close to the device, where the most harm may be expected, may not require shallow water effects to be taken into account. However, at large distances, sound wave reflections as well as transmission losses should be taken into account, such as the effects of water column velocity and temperature profiles, as well as bathymetry and the seabed medium. Two common theories for predicting sound propagation in shallow water are *ray theory* (for short ranges) and *normal-mode theory* (for longer ranges) with the cross-over range defined as h^2/λ [18, Chap. 6]. These models are readily available, for example in the implementation by Duncan and Maggi [19], but normally apply for point sources only.

A range of more sophiscated models have also been developed for modeling acoustic propagation in acoustic channels (see for example review by Etter [20]). The model of Marsh and Schulkin [21] includes the near-field anomaly, attenuation factor and seawater absorption coefficient, and is implemented by Carter [7] in the study of noise propagation from renewable energy devices in shallow water.

Acoustic Impact Assessment

Richards *et al.* [1] apply the dosage criteria previously mentioned to a generic hearing threshold envelope which encompasses a number of species, using the scaled measured noise data for the ‘SeaGen’ turbine. They note that TTS and PTS are possible if an animal is exposed to SPLs above the of 75 dB and 95 dB or greater respectively above the hearing threshold, for longer than 8 hours in a 24 hour period.

The results presented in Richards *et al.* [1] indicate that for a source strength of 157 dB, representing a 1MW tidal turbine, and accounting for shallow water effects, PTS can be expected up to 16 metres from the device for a 30 minute exposure, and TTS at up to 934 metres, assuming an 8 hour exposure. Tollit [22] also notes that continuous noise from tidal turbines is likely to cause disturbance to marine animals, but has been shown to be below injury threshold levels.

TIDAL TURBINE NOISE SOURCES

Noise can generally be thought of as unwanted sound and can range from ‘annoying’ to ‘harmful’. Potential noise sources due to the operation of a tidal turbine are categorised in Fig. 1 and Fig. 2. Table 1 characterises the flow-generated noise sources. This survey of potential noise sources allows the expected dominant sources to be identified and modelled appropriately. Table 1 suggests that the main flow-generated noise sources could be:

- unsteady loading;
- trailing edge;
- vortex shedding;
- cavitation.

Based on the parameters outlined in Table 1, an estimate of the likelihood of cavitation has been made using the *CCAV* code. This reveals that cavitation is unlikely in this case, and thus noise sources due to cavitation will not be studied further here. However, it is expected that a more thorough investigation of turbine cavitation behaviour will be required using site and device specific parameters before cavitation noise can be fully neglected for a particular turbine design study.

ESTIMATION OF TIDAL TURBINE NOISE

The remaining three dominant sources can be analysed using empirical relationships presented by Blake [13]. In this study, unsteady loading (inflow turbulence) and trailing edge noise have been considered. According to Blake [13], vortex shedding noise may also be important at frequencies below 150 Hz in this case. This noise source will be the subject of further work.

Noise due to Inflow Turbulence

The formulae implemented in this study for the computation of inflow turbulence noise are taken from Blake [13, Chap. 10] and are reproduced in the Appendix. These are empirical formulae derived for propellers, where the noise frequency spectrum can be predicted based on rotational speed, rotor dimensions and inflow turbulence characteristics. A frequency spectrum of acoustic pressures is derived from an unsteady thrust loading spectrum (see Eq. (4) in the Appendix) based on the turbine parameters in Table 2 as well as the inflow turbulence parameters.

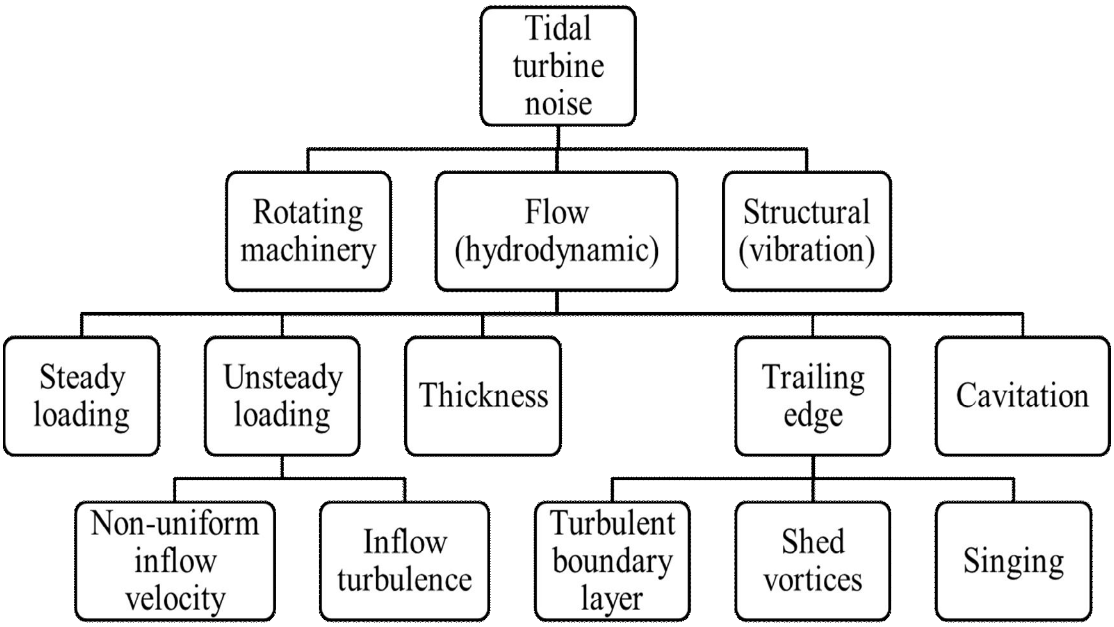

Figure 2. Categorisation of Hydrodynamic Tidal Turbine Noise Sources [1, 13].

Table 1. Characterisation of Potential Tidal Turbine Flow-Generated Noise Sources.

Type	Source	Origin	Importance	Freq. Type	Directivity	Ref.
Self noise	Steady loading	Blade loading distribution	Larger than thickness but generally insignificant when addressing unsteady loading scenarios	Broadband	Maximum aft of rotor between axis of rotation and plane of rotor	[23-25]
	Thickness	Blade geometry	Negligible for subsonic rotors	Broadband	Maximum in axis of rotation	[13, 24, 25]
	Trailing edge	Eddy convection past blade trailing edge	Generally dominant at high frequency (> 1 kHz), but smaller than unsteady loading	Broadband	Maximum in axis of rotation	[13]
	Vortex shedding	Wake instabilities	Can be highest noise but appears as narrow band tone related to vortex shedding frequency. Affected by trailing edge shape	Tonal	Maximum aft of rotor between axis of rotation and plane of rotor	[13]
Interaction noise	Unsteady loading	Non-uniform inflow velocity; inflow turbulence	Generally largest noise source at low frequency (< 1kHz)	Broadband	Maximum in direction of turbine axis of rotation	[13]
	Cavitation	Cavitation bubble collapse; blade-bubble interaction	Depends on propeller design and maintenance. Can be largest noise source if cavitation prevalent	Broadband	Maximum in plane of rotor	[12, 26, Chap. 10]
Hydroelastic noise	Singing	Not fully understood. Vibration, related to vortex shedding	Tone similar to vortex shedding. Magnitude associated with trailing edge shape	Tonal	Maximum aft of rotor between axis of rotation and plane of rotor	[13]

Table 2. Main Turbine Parameters used in Study.

Parameter	Symbol	Value	Unit
Number of blades	B	3	-
Diameter	D	20	m
Tip chord	C_T	0.62	m
Blade shape	-	NACA4415	-
Blade twist angle	γ	16 (hub) – 1 (tip)	deg.
Tidal velocity	U_∞	2.5	ms^{-1}
Tip speed ratio	λ	7	-
Tip Reynolds no.	Re_T	9.2×10^6	-
Rated power	-	1.22	MW
Water depth	h	40	m
Turbine depth	-	20	m

Two formulations for the unsteady thrust loading spectrum exist (Eqs. (5) and (9) in the Appendix). Use of the appropriate equation depends on the relative sizes of the turbulence length scale (Λ_1) and rotor pitch (P). For the case that $\Lambda_1 > P$, Eq. (5) is applied, whilst for $\Lambda_1 < P/B$, Eq.(9) is used. In doing so the effect of the turbulence being 'cut up' by the rotor blades is accounted for in the acoustic frequency spectrum.

The turbulence length scale is a characteristic length that describes the turbulence spectrum expected in the ocean boundary layer. The ocean boundary layer turbulence characteristics used in this study are taken from Gant and Stallard [11] who base their assumptions on the recommendations of the IEC [27] for wind turbines and measurements of shallow water ocean boundary layers. These are summarized in Table 3. Note that this approach does not account for non-uniform inflow velocities across the turbine rotor, such as those seen by a turbine operating in an ocean boundary layer velocity profile.

Table 3. Ocean Turbulence Parameters used in Study.

Parameter	Symbol	Value	Unit
Axial length scale	Λ_1=0.7D	14	m
Length scale factor	μ	6	-
Radial, circumferential length scales	$\Lambda_R, \Lambda_\theta = \Lambda_1/\mu$	2.33	m
Turbulence intensity	I	0.1	-
Mean square fluctuating velocity	$\overline{u^2} = (I\, U_\infty)^2$	0.0625	m^2s^{-2}

The rotor pitch is calculated from

$$\pi\tan(\gamma) = P/D \qquad (1)$$

using a mean value for the blade angle, γ, of 5.7 degrees. This results in a pitch of approximately 6.3 metres which is much less than the turbulence axial length scale given in Table 3. The unsteady thrust loading spectrum can thus be calculated using Eq. (5) presented in the Appendix.

The SPL in decibels (dB) is calculated as

$$\mathrm{SPL}=10\log_{10}\left(\frac{\overline{p^2}}{p_{ref}^2}\right) \qquad (2)$$

using the appropriate mean square pressure (MSP) values and a reference pressure of 1μPa, which is the standard reference unit for underwater acoustics [2]. Fig. 3 shows the low frequency SPL calculated based on two axial length scales, 14 metres (dotted line, corresponding to Eq. (5)) and 2.1 metres (dashed line, corresponding to Eq. (9)). 2.1 metres represents the largest axial length scale for which Eq. (9) is valid, i.e. P/B. SPL is presented at a distance of 20 metres from the source (turbine hub), assuming that this is the start of the 'far field' region, where the summation of incoherent sources is valid. Data is presented for 1 Hz bandwidths for a small frequency range in order to show clearly the blade passing tones and allow comparison to the data of Wang *et al.* [9]. The maximum value of SPL is 115 dB re 1μPa for the case $\Lambda_1 < P/B$. Note that there are many tones due to the low rotational speed of a tidal turbine compared to a ship propeller.

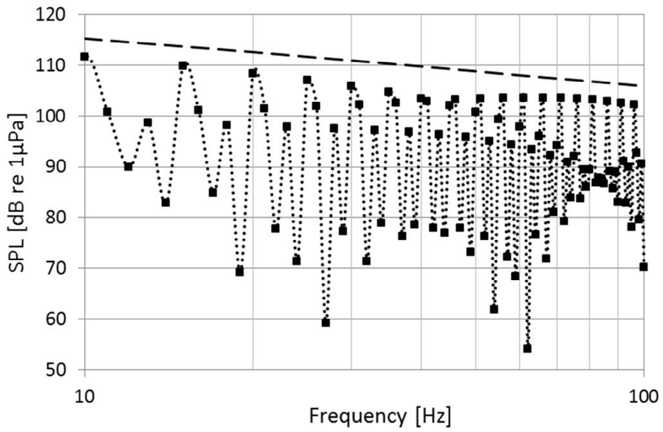

Figure 3. Predicted Sound Pressure Level in 1 Hz Bandwidths for Inflow Turbulence Noise, at 20 Metres from the Source in the Direction of the Turbine Axis of Rotation: Dotted Line ($\Lambda_1 > P$); Dashed Line ($\Lambda_1 < P/B$).

Noise Due to Blade Boundary Layer Turbulence

Higher frequency noise is also generated by the blade trailing edge. In this case, the noise is a function of rotor rotational speed and dimensions, as well as number of blades and blade boundary layer characteristics (see Eq. (10) in the Appendix, taken from Blake [13, Chap. 10]). The predicted SPL at 20 metres for this noise source is plotted in Fig. 4, alongside the inflow turbulence noise (for $\Lambda_1 < P/B$), using one-third-octave bandwidths. This allows comparison of SPLs due to different noise sources across a large frequency range.

The maximum SPL due to blade boundary layer turbulence is estimated to be approximately 44 dB re 1μPa, occurring between 10 and 100 Hz. Note that this SPL is below the hearing threshold of all the marine animals considered across the entire frequency range.

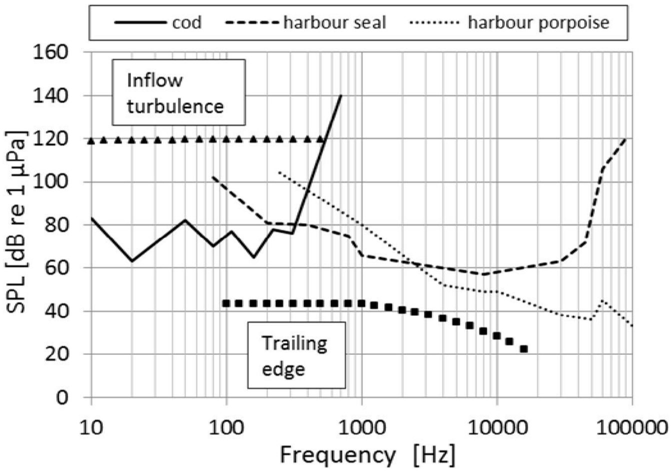

Figure 4. Predicted Sound Pressure Level for Turbulence Noise Sources at 20 Metres from the Source, on the Turbine Axis of Rotation for One-Third-Octave Bandwidths. Comparison Made to Hearing Threshold Data for Three Marine Animals [4, p.16].

The noise due to the blade trailing edge is seen to be insignificant compared to the inflow turbulence source. Assuming the noise sources to be incoherent, the MSPs can be summed allowing the overall SPL to be calculated. For example, at a frequency of 100 Hz, there is a 9 orders of magnitude difference between the MSP values for the two noise sources computed here, thus noise due to inflow turbulence dominates. This agrees qualitatively with Richards *et al.* [1] who state that the dominant noise sources for HATTs will be in the range 0-500 Hz, as well as the prediction of Blake [13, Chap. 9] for a translating hydrofoil.

Comparison to Measurement Data

Considering the noise due to inflow turbulence only, the corresponding maximum source level at one metre is approximately 141 dB re 1μPa assuming 1 Hz bandwidths (Fig. 3), allowing comparison to available experimental results. This result lies between the maximum source levels for the two measured data sets already discussed. The predicted SPL at 20 Hz for 1 Hz bandwidths is approximately 10 dB higher than the equivalent value presented by Wang *et al.* [9]. However, the turbine studied here has a diameter of 20 metres compared to 12 metres in [9] and thus the results are not directly comparable.

Since not all potential noise sources have been modelled in this study, the total SPL of the turbine investigated here may be higher, bringing the maximum SPL value closer to that measured in [8]. It should also be noted however that the effective source level of 166 dB quoted by Parvin *et al.* [8] is inferred from measurements at distances of more than 100 metres from the turbine and thus its accuracy is unknown. Richards *et al.* [1] also note of the data that there is a "large degree of variability" [p.38] and that shipping noise is included in the recordings.

The measurements of Wang *et al.* [9] have also been made at a distance from the device and extrapolated to full scale, implying similar uncertainties. As one metre is within the near field the prediction of a SPL of 141 dB re 1μPa at one metre does not mean that this pressure will be generated at this location. It should also be noted that a distance of one metre from the source is likely to lie within the turbine nacelle, and thus does not provide a SPL for environmental impact assessment but is presented to allow comparison to measurement data.

Discussion of Predicted Sound Pressure Levels

Also depicted in Fig. 4 is audiogram data for typical marine animals found in the North Sea [4, p.16]. This indicates that inflow turbulence alone does not constitute a difference large enough to lead to threshold shift, with a maximum excess of approximately 56 dB. However, high frequency or other broadband noise sources (such as cavitation noise) could lead to a higher chance of threshold shift where animals' hearing threshold is lower. It should be noted that the findings of Richards *et al.* [1] in terms of the highest PTS and TTS refer to high frequencies (~15 kHz) and the hearing threshold of a toothed whale (not considered here).

Figure 5 illustrates the directivity of the sound due to rotor-turbulence interaction based in Eq. (9), for the case of $\Lambda_1 < P/B$. The SPL value at 100 Hz has been chosen as a typical value from Fig. 4. It can be seen that there will be no sound emitted directly in the plane of the rotor. This provides a potential detection issue for animals in this region of the turbine. Whilst other noise sources, such as steady loading, can be heard in the plane of the rotor [23, Chap. 3], the magnitude of these is not known and was initially considered insignificant. Thus further investigation may be required into the potential collision risks tidal turbines pose to marine animals [7].

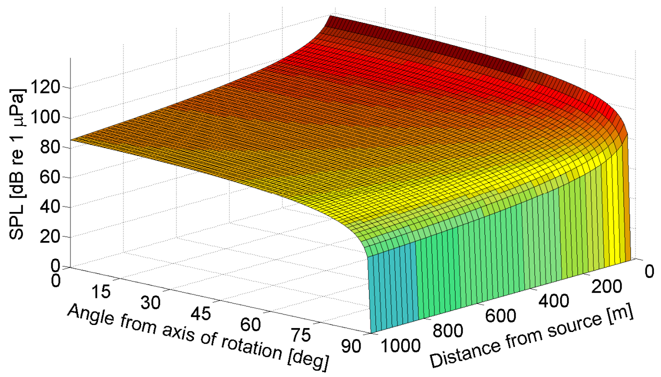

Figure 5. Predicted Far Field Sound Pressure Level for Inflow Turbulence Noise at a Frequency of 100 Hz ($\Lambda_1 < P/B$).

The approximate nature of this modelling approach must be taken into account when considering the SPL values obtained. As well as being derived empirically, the equations used only allow for the specification of a few single-value parameters to define the turbulence and do not fully include the effects of blade chord and pitch distribution, or airfoil shape. For example the airfoil section used to derive Eq. (10) is a NACA0012, i.e. symmetric airfoil, which has a quite different form to the NACA4415 of the turbine studied here.

CONCLUSIONS

The initial stages of a methodology for predicting tidal turbine noise are presented, which reveal that inflow turbulence is expected to be the dominant flow-generated noise source. The predicted maximum SPL however is not seen to be high enough to cause threshold shift in marine animals based on standard measures. The empirical nature of the method used was discussed, although it was noted that it provides a cost effective assessment of the effect design parameters have on turbine noise. The limited amount of measured turbine noise data available means full validation of the results is not possible, although the predicted noise agrees qualitatively with existing data [13, Chap. 9].

This work represents the first step in modelling tidal turbine noise, which has the potential to be developed into a comprehensive model. The use of such techniques in tidal turbine design will help with environmental impact assessment and certification, as well as allowing early consideration of noise in the design process.

FURTHER WORK

The immediate extension of this work will be to model all noise sources, including machinery noise in order to estimate overall SPL for a tidal turbine. Studies modelling generator noise are scarce and adaptation of research on shipping noise has been used previously [1]. A methodology for predicting cavitation noise may also be required depending on the specific turbine being studied. Again, equations derived for ship propeller cavitation may be the basis for such an analysis [12].

Included in accurate noise prediction is the need to account for the shallow water effects discussed in this paper, as well as the use of site specific ambient noise and ocean turbulence information. Once a full noise prediction methodology has been developed, this can be applied to single devices, or used to investigate the effect of tidal array size and spacing on overall SPL. Since tidal turbines are fixed structures that are required to operate without regular maintenance, as is possible with ships, the effect of surface degradation should also be investigated. This would include assessing the effect of biofouling as a potential noise source.

If complex interaction effects between turbulence and the turbine blades are expected to cause significant noise, a CFD approach may be the next step in better understanding the nature of the flow-generated noise sources. This will allow for their reduction during the design stage, where there may be a payoff between noise and other design drivers such as cost and efficiency.

NOMENCLATURE

Upper Case Roman Symbols

B	Number of Blades	
C_T	Blade Chord at Tip	m
D	Rotor Diameter	m
P	Rotor Pitch	m
R_T	Rotor Radius at Tip	m
Re_T	Reynolds Number at Tip = $C_T U_T / \nu$	
U_T	Blade Tip Velocity = ΩR_T	ms^{-1}

Lower Case Roman Symbols

c_o	Speed of sound	ms^{-1}
h	Water Depth	m
k_o	Acoustic Wave Number = ω / c_o	m^{-1}
p_{ref}	Reference Pressure	μPa
$\overline{p^2}$	Mean Square Pressure	$kg^2m^{-2}s^{-4}$
q_T	Dynamic Pressure at Tip = $1/2\ \rho(U_T^2 + U_\infty^2)$	$kgm^{-1}s^{-2}$
r	Observer Distance from Source	m

Greek Symbols

Λ_1	Turbulence Axial Length Scale	m
Λ_R	Turbulence Radial Length Scale	m
Λ_θ	Turbulence Circumferential Length Scale	m
Ω	Rotor Angular Velocity	$rads^{-1}$
α	Airfoil Trailing Edge Shape Parameter	
β	Observer Angle From Axis of Rotation	deg
γ	Blade Twist Angle	deg
λ	Wavelength	m
ν	Kinematic Viscosity	m^2s^{-1}
ρ	Fluid Density	kgm^{-3}
ϕ_p	Frequency Spectrum of Pressures	
ω	Angular Frequency	$rads^{-1}$

Acronyms

CFD	Computational Fluid Dynamics
HATT	Horizontal Axis Tidal Turbine
MSP	Mean Square Pressure
PTS	Permanent Threshold Shift
SPL	Sound Pressure Level
TTS	Temporary Threshold Shift

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REFERENCES

[1] Richards, S.D., Harland, E.J. and Jones, S.A.S., 2007, "Underwater Noise Study Supporting Scottish Executive Strategic Environmental Assessment for Marine Renewables", Report No. 06/02215/2, QinetiQ Ltd.

[2] Faber Maunsell and Metoc, 2007, "Environmental Report Section C SEA Assessment: Chapter C17 Noise", Report for Scottish Marine Renewables SEA.

[3] Götz, T., Hastie, G., Hatch, L.T., Raustein, O., Southall, B.L., Tasker, M. and Thomsen, F., 2009, "Overview of the Impact of Anthropogenic Underwater Sound in Marine Environments", Report No. 441/2009, OSPAR Commission: Biodiversity Series, London.

[4] Ainslie, M.A., De Jong, C.A.F., Dol, H.S. Blacquiere, G. and Marasini, C., 2009, "Assessment of Natural and Anthropogenic Sound Sources and Acoustic Propagation in the North Sea", Report No. TNO-DV 2009 C085, TNO, The Hague.

[5] Richardson, W.J., Malme, C.I., Green, C.R.Jr. and Thomson, D.H., 1995, *Marine Mammals and Noise*, Academic Press, San Diego, pp. 576.

[6] Southall, B.L., Bowles, A.E., Ellison, W.T., Finneran, J.J., Gentry, R.L., Greene, C.R., Kastak, D., Ketten, D.R., Miller, J.H., Nachtigall, P.E., Richardson, W.J., Thomas, J.A. and Tyack, P.L., 2007, "Marine Mammal Noise Exposure Criteria: Initial Scientific Recommendations", *Aquatic Mammals*, **33**(4), pp.411-521.

[7] Carter, C., 2007, "Marine Renewable Energy Devices: A Collision Risk for Marine Mammals?", MSc thesis, University of Aberdeen.

[8] Parvin, S.J., Workman, R., Bourke, P. and Nedwell, J.R., 2005, "Assessment of Tidal Current Turbine Noise at the Lynmouth Site and Predicted Impact of Underwater Noise at Strangford Lough", Report No. 628 R 0102, Subacoustech Ltd.

[9] Wang, D., Atlar, M. and Sampson, R., 2006, "An Experimental Investigation on Cavitation, Noise, and Slipstream Characteristics of Ocean Stream Turbines", *Proc. IMechE, 221, Part A: J. Power and Energy*, pp.219-231.

[10] Barnsley, M.J., and Wellicome, J,F., 1993, "Dynamic Models of Wind Turbines – Aerodynamic Model Development", Final Report on Contract JOUR 0110 for the Commission of the European Communities Directorate-General XII Science, Research and Development.

[11] Gant, S. and Stallard, T., 2008, "Modelling a Tidal Turbine in Unsteady Flow", *Proc. of the Eighteenth (2008) International Offshore and Polar Engineering Conference*, J.S.Chung et al., eds., ISOPE, Vancouver. pp.473-479.

[12] Gray, L.M. and Greeley, D.S., 1980, "Source Level Model for Propeller Blade Rate Radiation for the World's Merchant Fleet", *J. Acous. Soc. Am*, **67**(2), pp.516-522.

[13] Blake, W.K., 1984, "Aero-hydroacoustics for Ships Vol. II", DTNSRDC-84/010, David W. Taylor Naval Ship Research and Development Centre, Bethesda, pp.729-1193.

[14] Münch, C., Vonlanthen, M., Gomes, J., Luquet, R., Guinard, P. and Avellan, F., 2009, "Design and Performance Assessment of a Tidal Ducted Turbine", *3rd IAHR International Meeting of the Workgroup on Cavitation and Dynamic Problems in Hydraulic Machinery and Systems*, Brno, pp.571-581.

[15] Escobar, M., 2007, "Finite Element Simulation of Flow-Induced Noise Using Lighthill's Acoustic Analogy", *Dr.-Ing. Thesis*, University of Erlangen-Nürnberg.

[16] Kim, H.-J., Lee, S. and Fujisawa, N., 2006, "Computation of Unsteady Flow and Aerodynamic Noise of NACA0018 Airfoil Using Large-Eddy Simulation", *International Journal of Heat and Fluid Flow*, **27**, pp. 229-242.

[17] Reboul, G., Polacsek, C., Lewy, S. and Heib, S., 2008, "Aeroacoustic Computation of Ducted-Fan Broadband Noise Using LES Data", *Acoustics 08 Paris*, Paris.

[18] Urick, R.J., 1975, *Principles of Underwater Sound for Engineers*, McGraw-Hill, New York, pp.159-168.

[19] Duncan, A.J. and Maggi, A.L., 2005, "AcTUP v2.2l Acoustic Toolbox User-Interface & Post-Processor – Installation & User Guide", Curtin University of Technology, Perth.

[20] Etter, P.C., 2001, "Recent Advances in Underwater Acoustic Modelling and Simulation", *J. Sound & Vibration*, **240**(2), pp. 351-383.

[21] Marsh, H.W. and Schulkin, M., 1962, "Shallow Water Transmission", *J. Acous. Soc. Am.*, **34**(6), pp.863-864.

[22] Tollit, D., 2010, "OREG2010", *OREG2010 Annual Conference*, Vancouver. Available online at: www.oreg.ca /web_documents/dominic_tollit.pdf [accessed 06/01/11].

[23] Goldstein, M.E., 1976, *Aeroacoustics*, McGraw-Hill, New York, pp.153-171.

[24] Salvatore, F. and Ianniello, S., 2003, "Preliminary Results on Acoustic Modelling of Cavitating Propellers", *Computational Mechanics*, **32**, pp. 291-300.

[25] Seol, H., Pyo, S., Suh, J.-C. and Lee, S., 2004, "Numerical Study of Non-Cavitating Underwater Propeller Noise", *Noise & Vibration Worldwide*, **35**(6), pp. 11-26.

[26] Carlton, J.S., 2007, *Marine Propellers and Propulsion*, 2nd ed., Elsevier, Oxford. pp.253-259.

[27] IEC, 2005, "61400-1, Wind turbines – Part 1: Design requirements", IEC, Geneva.

APPENDIX

Turbulence Noise Prediction Equations

The MSP required to calculate SPL is estimated using the approximation of Blake [13]:

$$\overline{p^2} = \frac{1}{2}\phi_p(r,\omega)\omega \tag{3}$$

where ϕ_p is an approximate pressure spectrum for the noise source being studied.

The frequency pressure spectrum due to inflow turbulence is written as

$$\phi_{p,I}(r,\omega) = \frac{k_o^2 \cos^2\beta}{16\pi^2 r^2}\phi_t(\omega) \tag{4}$$

from [13, Chap. 10] where, for the case that $\Lambda_1 > P$,

$$\phi_t(\omega) = \frac{2}{3}\pi^2\Lambda_R\rho^2 U_T^2 C_T^2 R_T \times \left|A_S\left(\frac{\omega}{\Omega}\right)\right|^2 \overline{u^2} F_\Lambda \left|S_e\left(\frac{\omega C_T}{2U_T}\right)\right|^2 \tag{5}$$

is the frequency spectrum of unsteady thrust loading, with

$$\left|A_S\left(\frac{\omega}{\Omega}\right)\right|^2 = \frac{\sin\left(\frac{\pi\omega}{\Omega}\right)e^{i(B-1)\left[\frac{\pi\omega}{B\Omega}\right]}}{\sin\left(\frac{\pi\omega}{B\Omega}\right)} \tag{6}$$

and

$$F_\Lambda = \frac{\frac{\Lambda_\theta}{U_T}}{1+\left(\frac{\omega\Lambda_\theta}{U_T}\right)^2} \tag{7}$$

The Sears function for unsteady loading approximated as

$$\left|S_e\left(\frac{\omega C_T}{2U_T}\right)\right|^2 \simeq \frac{1}{1+\frac{\pi\omega C_T}{U_T}} \tag{8}$$

This formulation predicts blade passing tones at multiples of $B\Omega$. For $\Lambda_1 < P/B$, $\phi_t(\omega)$ is re-written assuming the blades are excited independently by the turbulence as

$$\phi_t(\omega) = \frac{2}{3}\pi^2 B\Lambda_R\rho^2 U_T^2 C_T^2 R_T \times \overline{u^2} F_\Lambda \left|S_e\left(\frac{\omega C_T}{2U_T}\right)\right|^2 \tag{9}$$

The formulation for the blade trailing edge turbulent boundary layer noise is also taken from [13, Chap. 10]. The frequency spectrum of pressures is calculated as

$$\phi_{p,TE}(r,\omega) = \frac{1}{2c_o}\frac{B}{5+\alpha}q_T^2\frac{\delta^{*2}R_T}{r^2} f\left(\frac{\omega\delta^*}{U_T}\right) \tag{10}$$

where $f(\omega\delta^*/U_T)$ is a function of the boundary layer wall pressure spectrum, which is derived from experimental data [13, Chap. 7] and depends on skin friction and wall shear stress. α is a trailing edge shape parameter, where $\pi\alpha$ is the external angle of the trailing edge and $1 \le \alpha \le 2$, and the boundary layer displacement thickness at the blade trailing edge, δ^*, is approximated as

$$\delta^* \simeq 0.048 C_T \,\mathrm{Re}_T^{-0.2} \tag{11}$$