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Assessing the influence of inflow turbulence on noise and performance of a tidal turbine using large eddy simulations

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ABSTRACT

Large eddy simulations of a model scale tidal turbine encountering inflow turbulence have been performed. This has allowed both unsteady blade loading and hydrodynamic noise radiation to be predicted. The study is motivated by the need to assess environmental impact of tidal devices, in terms of their acoustic impact on marine species.

Inflow turbulence was accounted for using a synthetic turbulence generator, with statistics chosen to represent the gross features of a typical tidal flow. The turbine is resolved in a fully unsteady manner using a sliding interface technique within the *OpenFOAM®* libraries. Acoustic radiation is estimated using a compact source approximation of the Ffowcs Williams–Hawkings equation.

It is observed that the long streamwise length scale of the inflow turbulence results in characteristic 'humps' in the turbine thrust and torque spectra. This effect is also evident in the far-field noise spectra. The acoustic sources on the blades are visualised in terms of sound pressure level and "Powell's source term". These measures show that the dominant sources are concentrated at the blade leading edges towards the tip. This results from the high loading of the turbine blades, and causes the sound to radiate more akin to a monopole than a dipole.

The full scale source level, obtained from scaling of the simulation results, is found to be lower than comparable measured data reported in the literature; this is attributed to additional sources not included in the present study. Based on the predicted source level, no physical impact on fish is expected.

1. Introduction

The level of acoustic emission from tidal turbines is part of environmental impact assessment [1]. It is possible that tidal turbine noise will have some impact on marine life, but attempts to quantify this are limited [2]. We direct the reader to work involving full-scale turbine noise measurement [3,4], as well as noise estimation for smaller devices [5,6]. Turbine noise sources are commonly defined in terms of a sound pressure level (SPL) measured at some far-field distance. These are often corrected back to a source level at 1 m from the rotor. A typical level defined in this way is of the order of 166 dB re 1 μPa^2 at 1 m [3]. Richards et al. [4] expect the dominant noise from horizontal axis tidal turbines (HATTs) to be due to rotating machinery in a frequency range $\approx$1–100 Hz.

Wang et al. [5] measured the noise of a 0.4 m diameter device, using a scaling procedure recommended by ITTC [7]. The reported maximum third-octave bandwidth SPLs (for a freestream velocity of 2.57 ms^{-1}) were approximately 115 dB and 125 dB for model and scaled results respectively.

Numerical studies of tidal turbine noise are less commonly reported. The noise of a vertical axis tidal turbine was estimated using a discrete vortex method by Li and Çalişal [6]. These authors found the peak SPL occurs at 4 Hz, and related their findings to the hearing sensitivity of fish, without making direct environmental impact assessments. No studies of HATT noise have been located; by contrast, noise simulations of horizontal axis wind turbines are more commonplace [8–10].

It is important to study the dynamic forces experienced by a turbine, since they contribute to fluid structure interaction effects, such as blade fatigue [11] or potential improvements in power capture [12]. Thus studying this behaviour in a dynamic environment would seem appropriate. The effect of inflow turbulence on

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turbine wakes will also influence the layout of arrays in order to optimise total power capture [13,14].

The effect of inflow turbulence on the length of the turbine wake has been studied using computational fluid dynamics [15,14]. Various approaches to simulating unsteady loading have been used. Churchfield et al. [16] used a library of instantaneous turbulence realisations as an inlet condition to replicate a turbulent boundary layer. Alternatively, 'synthetic' turbulence can be generated using a numerical method to represent the inflow turbulence. In Ref. [15], this approach was used to show that the inflow turbulence reduces the length of the turbine wake. This effect was also reported by McNaughton et al. [14], who included the turbine geometry in the simulation, as opposed to the porous disk representation in Gant and Stallard [15]. Afgan et al. [17] showed that large eddy simulation (LES) is more capable than an unsteady Reynolds-averaged Navier–Stokes equations solution at predicting the complex flow features and mean performance of a model scale turbine. LES also provides better resolution of the inflow turbulence and thrust spectra.

We have previously presented predictions of tidal turbine noise [18] as well as impact assessment [19], using empirical modelling. The approach was based on modelling the turbine unsteady thrust spectrum due to inflow turbulence, and predicting the noise assuming free field radiation [20]. Inflow turbulence is known to be the dominant noise source due to turbulence in this case [18,21]. The estimated spectral source level (SSL) was ≈ 145 dB re $1 \mu\text{Pa}^2\ \text{Hz}^{-1}$ at 1 m across a frequency range of ≈ 10–100 Hz. This paper presents a development in terms of noise simulation of HATTs, and is based on the methodology in Lloyd et al. [22].

The paper assumes the following format. §2 outlines the numerical setup for the simulations, including the methods for generating inflow turbulence and predicting sound radiation. In §3, the test case geometry is described, along with the domain design. A model scale turbine is used since a detailed blade geometry is available, as well as mean performance data. This section also includes grid design considerations and presents an assessment of the sliding interface technique used in terms of its ability to interpolate the broadband velocity fluctuations present in the inflow. The simulation results are divided into three sections: inflow turbulence statistics (§5); turbine response (§6); and acoustic emission (§7). The model scale acoustic predictions are compared to an analytical model, since no experimental validation data is available. These are then scaled using recommended procedures in order to provide estimates of full scale turbine noise (§8). This allows discussion of possible environmental impact. Finally, conclusions are made in §9.

2. Numerical framework

2.1. Turbulence modelling

In large eddy simulation, the filtered Navier–Stokes equations are resolved in a time-dependent manner. Scales smaller than the grid are accounted for using a *subgrid* model. For a detailed description of large eddy simulation see for example Sagaut [23]. The dynamic mixed Smagorinsky subgrid model [24] is used, since it has been shown to perform well for complex flows and on coarse grids [25].

The normalised first cell height is defined as $\Delta y_w^+ = y_1 u_\tau / \nu$, where y_1 is the first cell height, u_τ the friction velocity and ν the kinematic viscosity. An average value of $\Delta y_w^+ = 40$ was achieved over the blades; due to the surface refinement technique employed by *snappyHexMesh*, the leading and trailing edges typically possessed much lower values. Since the viscous sublayer is not fully resolved (this requires $\Delta y_w^+ \approx 1$), a *wall function*, based on Spalding's "law of the wall" [26] was used.

In order to simulate stochastic loading on the turbine, an inflow turbulence generator is used. This is a numerical method for generating synthetic turbulence at the simulation inlet. Here we use the forward stepwise method; for a full description of the method, see Kim et al. [27]. Evaluations of this method for hydroacoustic predictions has previously been made.

2.2. Solution method

Simulations were performed using the *OpenFOAM*®[1] libraries. A custom solver based on the *pimpleDyMFoam* application was used. The main features of the solver are: pressure implicit splitting of operators (PISO)-type [29] correction of the velocity; outer corrector loops allowing higher time steps than PISO; grid rotation via 'dynamic meshing' and an arbitrary mesh interface (AMI); and velocity fluctuations generated by the FSM inserted during the PISO loop. All discretisation schemes are second-order, apart from convective acceleration, which uses a *hybrid* upwind-central differencing scheme, giving good accuracy in regions where a central scheme is less accurate [25,28]. Linear solution was achieved using the biconjugate gradient method for velocity, and general algebraic multigrid method for pressure. The solvers exit the iteration loop when a tolerance of 10^{-9} (velocity) and 10^{-6} (pressure) is achieved within each loop.

The *pimpleDyMFoam* solver allows the maximum Courant number $Co = |u|\Delta t/\Delta x$ to exceed unity, where $|u|$ is a local velocity magnitude, Δt the time step and Δx the cell dimension; simulations used a maximum time step $\Delta t^* = \Delta t U_0/D = 3.5 \times 10^{-5}$, based on the reference (freestream) velocity U_0, and turbine diameter D. where the maximum Courant number was also limited to four. This time step is the same as that used for other tidal turbine LES [17], and results in 40 time steps per degree of rotation. A transient phase of four turbine rotations was assumed ($T^* = TU_0/D \approx 2.3$), allowing the inflow turbulence to reach the rotor plane. Probe, force and sound pressure were then sampled at $f_{\text{sample}} = n/300$, or 100 times per blade passage, for a further $T^* \approx 6.9$, thus ensuring a complete flow-through of the domain.

2.3. Acoustic analogy

In order to evaluate the acoustic radiation from the blades, we use a formulation of the Ffowcs William-Hawkings (FW-H) acoustic analogy [30]. This has been implemented into *OpenFOAM*® using only the term relating to fluid loading, which is an acoustic dipole. This is given by

$$p'(\boldsymbol{x}, t) \approx \frac{x_i}{4\pi c_0 |\boldsymbol{r}|^2} \frac{\partial}{\partial t} \oint_S n_j p_{ij} \delta(h) \mathrm{d}S(\boldsymbol{y}). \qquad (1)$$

In Equation (1), p' is acoustic pressure, $\boldsymbol{x}$ and $\boldsymbol{y}$ denote the receiver and source locations, $|\boldsymbol{r}| = |\boldsymbol{x} - \boldsymbol{y}|$, c_0 is the speed of sound, equal to 1500 ms^{-1} in water, n_j is the normal vector to the surface S, and δ is the Dirac delta function. This form of the FW-H equation is suitable for low Mach number flows where broadband noise is of interest. It assumes the receiver to be in the acoustic *far-field* ($|\boldsymbol{r}| \gg \lambda$) and the source to be *compact* ($L \ll \lambda$, where L is the source dimension). The integration surface (corresponding to $h = 0$) is taken to be the solid boundary.

[1] www.openfoam.org/.