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Feasibility study into a computational approach for marine propeller noise and cavitation modelling

Artur K. Lidtke [a,*], Victor F. Humphrey [b], Stephen R. Turnock [a]

[a] Fluid Structure Interactions Group, University of Southampton, SO16 7QF, UK
[b] Institute of Sound and Vibration Research, University of Southampton, SO17 1BJ, UK

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ABSTRACT

There is increased interest in the ability to predict the noise associated with commercial ship propellers. Key components of the computational analysis process are considered for two test cases and the future direction in resolving the associated challenges is presented. Firstly, the Potsdam Propeller Test Case is used to compute tonal blade passage noise using the Ffowcs Williams–Hawkings acoustic analogy. Cavitation extents predicted using the Sauer and Schnerr mass transfer model agree well with the experiment but show little unsteadiness due to URANS being used. A complementary study of initial results from the study of cavitation noise modelling attempt is presented for a NACA0009 section, used as a simplified representation of a propeller blade. Large Eddy Simulation and FW-H acoustic analogy are used in order to estimate the cavitation-induced noise. Results indicate that the discussed approach provides the means for identifying low-frequency noise generation mechanisms in the flow, but does not allow for the fine-scale bubble dynamics or shockwave formation to be resolved. It is concluded that the discussed approach is a viable option to predict large parts of the marine propeller noise spectra but still further work is needed in order to account for the broadband components.

1. Introduction

Concerns about limiting the input of noise into the Oceans have been increasingly more pronounced in recent years. One may associate the anthropogenic noise with multiple mechanisms, shipping being one of the larger contributors (Hildebrand, 2009; Urick, 1984). The significance of this is even greater given that a large part of the energy of the ship-related noise falls within the 10–1000 Hz regime and thus has a high potential to effect marine wildlife (Lloyd et al., 2014).

Hence, several initiatives have been established in order to investigate how to mitigate the impact of shipping on the marine environment (Van der Graaf and Ainslie, 2012; Tasker and Amundin, 2010). These have contributed to the debate as to whether regulation should be introduced and updated where necessary in order to limit the noise induced by commercial vessels (Kellet et al., 2014; Bertschneider et al., 2014). According to the review by the ITTC Specialist Committee on Hydrodynamic Noise (Bertschneider et al., 2014) the noise due to the vibration of the hull structure induced by its interaction with the propeller and ship board machinery will be of smaller interest as far as marine wildlife is concerned, although it is of key importance for the crew and passenger comfort on board a commercial ship. It may, however, play a role if the dominant machinery frequencies, such as engine rpm, will coincide with frequency range of particular importance to a given species. This may be overcome by increasing vibration impedance of the structure, for example by avoiding rigidly mounted engines.

The tonal noise sources associated with the propeller, cavitating and non-cavitating, are typically considered to be dominant when assessing the environmental impact of shipborne noise. This is because of their high sound power and low attenuation resulting in the potential to affect the largest area most severely (Bertschneider et al., 2014). It is likely, however, that other noise sources, such as those due to machinery or broadband cavitation, will become of greater importance in off-design conditions, such as when operating in shallow coastal waters, during manoeuvring or while at port.

The tonal, non-cavitating sound component is caused primarily with the loading noise caused by the blade passing through non-uniform wake of the hull as it rotates (Lloyd et al., 2015; Ianniello et al., 2013). Due to the induced change to the pressure distribution on the blades this phenomenon also has an effect on periodic cavitation. This fluctuation of cavitation volume will act as a strong monopole noise source (Park et al., 2009; Seol et al., 2005; Salvatore and Ianniello, 2002). This may also be expected to be

* Corresponding author.
E-mail address: akl1g09@soton.ac.uk (A.K. Lidtke).

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accompanied by contributions from higher order acoustic sources, particularly for smaller source–receiver distances (Seol, 2013).

The unsteadiness of the flow will play a crucial role in determining the noise signature of a lifting surface such as a propeller or hydrofoil. Thus, while some useful insights may be gained into the cavitation phenomena using approaches such as unsteady RANS or boundary element methods, it is likely that Large Eddy Simulation (LES) will be required to develop a deeper understanding of the underlying flow.

The work reported contributes to a wider study into the assessment of the environmental impact of a ship on marine ecosystems which requires computation of propeller-induced noise levels. Specifically the current focus is to assesses the potential benefits and disadvantages of turbulence and cavitation modelling techniques from the numerical propeller noise modelling perspective.

In order to allow more detailed analysis to be undertaken a basic understanding of the limitations of the modelling methods constituting the current state of the art must be developed. This is done on the example of the Potsdam Propeller Test Case (PPTC). This has seen a significant amount of both experimental and theoretical attention (Abdel-Maksoud, 2011), thus becoming one of the more established validation problems. The presented results were obtained using the Schnerr–Sauer mass transfer cavitation model for the flow being solved using unsteady RANS with the k–ω SST turbulence model (Sauer and Schnerr, 2001).

The flow over a propeller may be regarded as complex and is thus not well suited for preliminary simulations aimed at assessing the mechanism of cavitation noise. Hence, a simpler test case of a NACA0009 hydrofoil is considered, where LES is used instead of RANS to solve the equations of motion of the flow. The far-field sound pressure level is computed using a porous Ffowcs Williams–Hawkings acoustic analogy implemented in OpenFOAM. The presented analysis focuses on correlating the relationships between the predicted flow features and the corresponding noise signals, allowing for preliminary conclusions to be drawn with respect to the aptness of the presented approach to the modelling of noise of a complete propeller.

2. Numerical modelling

2.1. Cavitation

Cavitation may be described as the transition of liquid into vapour in regions of low pressure. This is caused by the presence of small gas nuclei in the liquid (Plesset and Prosperetti, 1977). When subject to tensile stress, these nuclei expand and lead to different types of cavitation, such as sheet or bubble cavitation, depending on the flow conditions (Vallier, 2013).

It is possible to simulate the behaviour of individual cavitation bubbles, as described, for instance, by Jamaluddin et al. (2011) and Hsiao and Chahine (2004). However, because of the small size of the cavitation nuclei, ranging between 2 and 50 μm for standard sea water (Woo Shin, 2010), it would not be feasible to compute the behaviour of every individual bubble in full detail for a flow over a full-scale propeller or a hydrofoil.

Alternatives involve, for instance, the use of volume-of-fluid or level-set multi-phase flow solvers in order to describe the physics governing the motion of large cavities. Schnerr–Sauer cavitation model has been used here in order to account for the pressure-induced phase change of liquid into vapour and vice versa (Sauer and Schnerr, 2001). This is done based on solving the transport equation for a volume fraction, α, with an additional source term introduced on the right-hand side to account for the evaporation and condensation:

$$\frac{\partial \alpha}{\partial t} + \nabla \cdot (\alpha \mathbf{U}) = -\frac{\dot{m}}{\rho}, \tag{1}$$

where $\dot{m}$ denotes the rate of change of mass of the liquid–vapour mixture, ρ is the density of the mixture and $\mathbf{U}$ is the fluid velocity. The presence of the additional source term also modifies the continuity equation which now becomes

$$\nabla \cdot \overline{\mathbf{U}} = \left(\frac{1}{\rho_v} - \frac{1}{\rho_l}\right)\dot{m}, \tag{2}$$

where subscripts v and l refer to vapour and liquid phases, respectively. One may also define the density and viscosity of the liquid–vapour mixture as

$$\rho = \alpha\rho_v + (1-\alpha)\rho_l,$$
$$\mu = \alpha\mu_v + (1-\alpha)\mu_l, \tag{3}$$

respectively.

In order to close the system of equations, an expression for the rate of mass transfer between the liquid and the vapour has to be introduced. In the approach proposed by Sauer and Schnerr this is done by considering the equation of motion of a single bubble and rearranging it as

$$\dot{m} = \frac{\rho_l \rho_v}{\rho}(1-\alpha)\alpha\frac{3}{R}\sqrt{\frac{\frac{2}{3}(p-p_v)}{\rho_l}}, \tag{4}$$

where R is modelled based on the specified characteristic nuclei radius, R_0, and their volumetric density, n_0.

2.2. Large eddy simulation

In the discussed hydrofoil study Large Eddy Simulation (LES) was used in order to model the fluid flow. Use was made of the implicit PISO solver on a collocated finite-volume grid, as implemented in OpenFOAM 2.2.2. The LES approach is based on resolving the most prominent turbulent structures and modelling the remainder of the turbulent kinetic energy spectrum. This is achieved by filtering the momentum equation yielding

$$\frac{\partial \overline{\mathbf{U}}}{\partial t} + \nabla \cdot (\overline{\mathbf{U}} \otimes \overline{\mathbf{U}}) = -\frac{1}{\rho}\nabla \overline{p} + \nu \nabla^2 \overline{\mathbf{U}} - \nabla \cdot \boldsymbol{\tau}, \tag{5}$$

where the overbar denotes the filtering operation, p is the fluid pressure and ν is the kinematic viscosity. Similarly, the continuity equation becomes

$$\nabla \cdot \overline{\mathbf{U}} = 0. \tag{6}$$

the non-linear subgrid stress tensor, $\boldsymbol{\tau}$, used to describe the effect of the filtered eddies on the flow in Eq. (5), may be expressed as

$$\boldsymbol{\tau} = \overline{\mathbf{U} \otimes \mathbf{U}} - \overline{\mathbf{U}} \otimes \overline{\mathbf{U}}. \tag{7}$$

In order to model this quantity one may consider the Boussinesq hypothesis, whereby the stress tensor is assumed proportional to the fluid strain-rate and an assumed subgrid viscosity, ν_{SGS}, yielding

$$\boldsymbol{\tau} - \tfrac{1}{3}\boldsymbol{\tau} \cdot \mathbf{I} = 2\nu_{SGS}\mathbf{S} \tag{8}$$

In the above $\mathbf{I}$ is the identity matrix, and the strain rate may be computed as

$$\mathbf{S} = \tfrac{1}{2}\left(\nabla \mathbf{U} + \nabla \mathbf{U}^T\right). \tag{9}$$

An expression provided by the Smagorinsky model assumes the subgrid scale viscosity to be dependent on a constant coefficient, C_S, and the filter width, Δ, dictated by the mesh density. These