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RUBENS CAVALCANTE DA SILVA

**Numerical Evaluation on the Effect of Boss Cap
Geometries on AUV Propeller Noise**

São Paulo–SP

2021

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Revised Version

Dissertation submitted to the Polytechnic
School at the University of São Paulo in
partial fulfillment of the requirements for the
degree of Master of Science

Field of Study: Naval Architecture and
Ocean Engineering

Supervisor: Prof. Dr. Gustavo R. S. Assi

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*To Tito, Darin and Gustavo Felipe,
for showing me a new face of love.*

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*“How lucky am I,
to have something that makes saying goodbye so hard.”*
(Alexander A. Milne)

Resumo

DA SILVA, Rubens Cavalcante. **Avaliação Numérica do Efeito de Geometrias de Caps no Ruído do Propulsor de um AUV**. 2021. Dissertação (Mestrado em Engenharia Naval e Oceânica) – Escola Politécnica, Universidade de São Paulo, São Paulo, 2021.

Este trabalho realizou a análise numérica do ruído emitido por um propulsor de alto skew desenvolvido para operar em um veículo submarino autônomo (AUV), considerando-se 5 diferentes geometrias de bosso (elíptico e divergentes) e utilizando um código comercial de CFD. Foi utilizado o modelo acústico de Ffowcs Williams-Hawkings (FW-H), baseado na formulação de Dunn-Farassat-Padula para domínios rotativos subsônicos, acoplado ao modelo de turbulência DES (*Detached Eddy Simulation*). A validação do modelo foi feita através da comparação com dados disponíveis em literatura do propulsor INSEAN E779A. Os resultados numéricos de eficiência para cada geometria foram comparados com os obtidos experimentalmente em túnel de cavitação, onde se observou uma ligeira queda para os bossos divergentes se comparado com a geometria paradigma (bosso elíptico). Os resultados numéricos hidroacústicos mostraram que, para determinadas bandas de frequências, houve uma diminuição sensível do ruído para a geometria mais divergente de bosso.

Palavras-chave: Hidroacústica. AUV. propulsor. bosso. modelo acústico de Ffowcs Williams-Hawkings. dinâmica dos fluidos computacional. DES.

Abstract

DA SILVA, Rubens Cavalcante. **Numerical Evaluation on the Effect of Boss Cap Geometries on AUV Propeller Noise**. 2021. Master's Dissertation (Naval Architecture and Ocean engineering) – Polytechnic School, University of São Paulo, São Paulo, 2021.

This study numerically analyzed the underwater noise emitted from an autonomous underwater vehicle (AUV) high-skewed propeller (Mod5), considering five different boss cap geometries (elliptical and divergent) and using a commercial CFD code. The numerical model was developed by means of the Ffowcs Williams-Hawkings (FW-H) acoustic model, as well as the Dunn-Farassat-Padula formulation to subsonic rotating blades, coupled with a Detached Eddy Simulation (DES) turbulence model. The model was validated by comparing the numerical results with acoustics data from INSEAN E779A propeller. Numerical and experimental results showed that divergent boss caps presented lower values of propeller efficiency than the elliptical geometry. On the other hand, the most divergent geometry presented a slightly reducing of noise to certain frequency bands.

Keywords: Hydroacoustics. AUV. Propeller. boss cap. Ffowcs Williams and Hawkings acoustic model. Computational fluid dynamics. DES.

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List of abbreviations and acronyms

AUV	Autonomous Underwater Vehicle
FW-H	Ffowcs Williams-Hawkings
IMO	International Maritime Organization
CFD	Computational Fluid Dynamics
EFD	Experimental Fluid Dynamics
DES	Detached Eddy Simulation
DDES	Delayed Detached Eddy Simulation
PBCF	Propeller Boss Cap Fins
LES	Large Eddy Simulation
RANS	Reynolds-Averaged Navier–Stokes Equations
LabHidro	<i>Laboratório de Hidrodinâmica</i> (In Portuguese)
INSEAN	<i>Istituto Nazionale per Studi ed Esperienze di Architettura Navale</i> (In Italian)
IPT	<i>Instituto de Pesquisas Tecnológicas</i> (In Portuguese)

List of symbols

ρ	Fluid density [Kg/m ³]
p	Fluid pressure [N/m ²]
p'	Acoustic pressure [N/m ²]
u_i, u_j, u_k	Velocity components [m/s]
t	Time [s]
c	Speed of sound [m/s]
c_p	Specific heat at constant pressure of the fluid
k	Thermal conductivity of the fluid [W/m.K]
k_e	Turbulent kinetic energy [m ² /s ²]
$\tau_{i,j}$	Viscous stress tensor
ω	Specific dissipation rate [1/s]
μ	Dynamic viscosity [Kg/m.s]
μ_t	Turbulent eddy viscosity [m ² /s]
x_i, x_j, x_k	Spatial coordinates
δ_{ij}	Dirac delta function
T	Fluid temperature [K]
$T_{i,j}$	Lighthill stress tensor
τ^R	Reynolds stress tensor
r	Radial coordinate [m]
θ	Angular coordinate [rad]
$\mathbb{D}^2$	D'Alembert operator
M_r	Mach number of the source in the radiation direction

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Introduction

1.1 Relevance and Motivation

Ship underwater noise has become a great concern of several international regulatory institutions and governments during the past years, as well as the focus of substantial discussion around its interaction with marine environment. According to the International Maritime Organization (IMO), underwater-radiated noise from commercial ships may have negative consequences on sea life both in the long and short terms, especially physiological damages on marine mammals. Since ships routinely cross international boundaries, the correct management of such noise requires a well-coordinated international response.

In 2008, the IMO Marine Environment Protection Committee (MEPC) agreed to develop non-mandatory technical guidelines to provide general advice about reduction of underwater noise to designers, shipbuilders and ship operators, as well as consider common technologies and measures that may be relevant for most sectors of the commercial shipping industry.

The pace in which the anthropogenic noise sources have increased in the last decades is certainly faster than the evolution of knowledge of underwater radiated noise. As stated by Rolland *et al.* (2012), maritime activities, such as seismic exploration by the oil and gas industries, military and commercial use of sonar, recreational boating and shipping traffic, are the main anthropogenic sources of noise in maritime environments. Therefore, further research is needed to better understand and measure the contribution to underwater emissions of each source of noise on the ship and also the overall impact on marine fauna.

Due to the broad sound spectrum (Figure 1), several anthropogenic sources of noise on oceans can significantly affect fish and other marine organisms. According to the National Research Council of the National Academies - NRCNA (2003), such

interaction can lead the marine animals from behavioral disturbances such as subtle changes in surfacing and breathing patterns, to cessation of vocalizations and active avoidance or escape from the region of the highest sound levels.

Figure 1 – Hearing ranges of fishes and mammal species and anthropogenic noise sources

Source: Walker *et al.* (2019)

Furthermore, underwater noise presents some particular properties that other pollutants do not. For instance, sound waves propagate much faster and reach longer distances in deep water than conventional pollutants, becoming a powerful and almost unknown contaminant. Hence, researches about the properties of generation and propagation of underwater noise have become more necessary to understand its impacts on marine life. The control of noise in submarine vehicles, such as military submarines and Autonomous Underwater Vehicles (AUV) employed in plethora of activities in the ocean environment has ever been an issue due to their strict operational requirements. Wynn *et al.* (2014) explain that AUV's have a variety of applications and are increasingly being used in different tasks in the scientific, military, commercial, and policy sectors.

According to Ianniello *et al.* (2012), the propeller is one of the main sources of

noise radiated from marine vessels (in both non-cavitating and cavitating conditions), and acts not only as a direct noise source (being a body moving in the fluid), but also as a sort of indirect source, since it excites the stern vault of the hull by an unsteady (periodic) hydrodynamic load. In addition, Carlton (2007) stated that the noise produced by a propeller, in terms of both its intensity and its spectral content, has been of considerable importance to several vessels, such as warships and merchant ships.

Hence, given the important role that the marine propeller plays in the acoustic emission of a vessel, the propeller design has taken into account many geometric adaptations to diminish the acoustic emissions. At this moment, one important question that arises is: what geometric features of the propeller would be changed to impact the radiated noise?

To answer such question, we must know the geometry of a marine propeller and the turbulent wake generated during its regular operation. As we can observe in figure 2b, the propeller wake is compounded by turbulent structures, such as the tip vortex and the hub vortex, which is commonly a concentrated vortex emitted by the propeller hub. As the boss cap is located at the end of the propeller hub (Figure 2a), a change in its geometric feature would lead to a change in the hub vortex behavior. This hypothesis is the core of the present study, as we will see in the foregoing chapters.

Figure 2 – (a) A typical marine propeller and (b) The turbulent wake of a marine propeller

Source: Author

Particularly, divergent boss cap profiles have been the focus of few studies about vortex and propeller efficiency, as an earlier stage design of propeller boss cap fins (PBCF) (Figure 3), as reported by Katayama *et al.* (2015). According to Carlton (2007), a slightly divergent hub form is desirable to avoiding certain issues such as blade root erosion, as well as reducing the root vortices in some type of vessels (warships and patrol vessels), preventing further negative impacts in other ship components like rudders.

Figure 3 – Detail of a propeller boss cap fins (PBCF)

Source: Katayama *et al.* (2015)

Nonetheless, due to the lack of references about the effect on underwater noise of divergent boss caps, further studies are necessary to fulfill the need of understanding about the influence of such device on propeller acoustic emissions, this kind of investigation should be a quite important benchmark to develop a feasible apparatus to improve the design of efficient and acoustically optimized propellers. Besides, as the boss cap presents a simple and regular geometry with a relative ease of construction and installation, such element is a feasible candidate to be modified and studied in order to define the impacts on propeller features, such as the radiated noise.

1.2 Objective and Methodology

The main objective of this study is to evaluate the effect of variation of boss cap geometry on the noise of a marine propeller (Mod5 Propeller) (NETO *et al.*, 2018) developed to operate in an DARPA Suboff (GROVES *et al.*, 1989) shaped AUV and designed in accordance with the methodology proposed by Sbragio (1995), which uses the Lifting Line and Lifting Surface Theories. Four divergent geometries of boss caps and an elliptical boss cap based on the DARPA Suboff stern shape were analyzed in order to check the main differences on the features of the broadband noise of each geometry. To correctly measure the underwater acoustic emissions, the numerical simulations were developed in the commercial CFD code StarCCM+, by means of the Ffowcs Williams-Hawkings acoustic model and the Detached Eddy Simulation (DES) as turbulence model.

According to Kaltenbacher *et al.* (2008) this type of methodology is a hybrid approach widely used in aeroacoustics for solving the flow and providing acoustic sources in order to yield a precise prediction of the flow induced noise. It is important to notice that this approach is also used in hydroacoustic problems with some alternative considerations and hypothesis took into account to modelling the pressure fluctuations of the flow in incompressible fluids. Figure 4 shows the typical numerical methods which are employed when using this kind of methodology to acoustics problems.

Figure 4 – Some of the possible strategies when using an aeroacoustic hybrid approach

Source: Kaltenbacher *et al.* (2008)

The CFD simulations of the Mod5 propeller with each boss cap geometry were validated with experimental tests in cavitation tunnel. Furthermore, due to the available data in several references, the INSEAN E779A propeller was adopted as benchmark test to assess the performance of the numerical hydroacoustic model developed in this work.

1.3 Thesis Structure

This dissertation is divided into 5 chapters, including the chapter 1 – Introduction: Chapter 2 presents a brief literature review about some aspects of divergent boss caps geometries, as well as the development of the Lighthill acoustic analogy, culminating in the Ffowcs Williams and Hawkings acoustic analogy and the formulations that account the linear and non-linear noise terms. Chapter 3 is concerned about the description of the mathematical problem that governs the hydroacoustic phenomena of interest. A reasonable explanation is provided about turbulence modeling and the numerical scheme itself, embracing the final form of FW-H porous formulation and some numerical requirements followed by the computational model. Chapter 4 firstly presents the validation of the numerical model by means of the benchmark E779A propeller. Subsequently, the Mod5 propeller will be analyzed with all boss cap geometries considered. The validation of the numerical results was done by comparing to several experimental results Chapter 5 presents the conclusions and suggests further developments to improve the results and the methodology developed hereby.

Literature Review

This chapter presents information about divergent boss caps and its features as well as the main studies about the numerical application of Hydroacoustic theories on the prediction of propeller non-cavitating noise, in particular the Ffowcs Williams-Hawkings acoustic model and its peculiarities related to the subsonic underwater flow.

2.1 Divergent Boss Caps

According to Carlton (2007), the design process of a propeller is a complex and creative task that aims to resolve several constraints to produce an optimal solution. The author enumerated few characteristics that would be considered in a multivariable optimization process in the propeller design, such as the blade number, propeller diameter, skew, pitch-diameter ration and the hub form. About the latter feature, Carlton (2007) also explained that the hub form has a considerable importance in the propeller design, and an adequate divergence of hub would avoid cavitation and erosion problems on root blades of vessels such as fast patrols (Figure 5) and warships.

Figure 5 – Divergent boss cap of: (a) composite propeller for submarine class 212A; (b) high-speed patrol vessel

Source: (a) Thyssenkrupp (2011); (b) Carlton (2007)

Katayama *et al.* (2015) analyzed the behavior of few boss cap geometries by CFD in order to account their influences on propeller efficiency and optimize the use of propeller boss cap fins (PBCF) in such devices. The authors investigated contraction, straight and diffusion (divergent) boss caps geometries and concluded that the diffusion type presented the highest pressure in the hub vortex region (Figure 6).

Figure 6 – Pressure distributions in the hub vortex region

Source: Katayama *et al.* (2015)

The main idea on the use of divergent boss cap boss in propeller design is that, due to the increasing of the hub vortex diameter by the divergent geometry, the radial velocity of the vortex tends to decrease, as the distribution of circulation along the propeller blades is assumed to be the same. Hence, is expected that the vortex sound

diminishes as the intensity of the vortex is lower.

It is important to highlight that cavitation phenomena will not be considered in this study, as the propeller is designed to operate in an AUV, i.e., to operate at deep water regions, where the pressure is higher than in the water surface region and the cavitation is harder to occur. In the following sections, some numerical and experimental results will be presented in order to assess the influence of the boss cap divergence on the propeller efficiency and on the broadband noise. Particularly, the Lighthill acoustic analogy is the starting point of the theoretical background needed to develop the Ffowcs Williams-Hawkings acoustic analogy.

2.2 The Lighthill Acoustic Analogy

One of the most meaningful theories on the acoustic field is the so-called Lighthill Acoustic Analogy, in which M. J. Lighthill (LIGHTHILL, 1952; LIGHTHILL, 1954) developed a methodology that focused on the understanding of the generation of noise by jet engines. The Lighthill's approach considers a limited region β of fluctuating fluid in arbitrary motion immersed in an extensive volume of fluid at rest Θ (Figure 7).

Figure 7 – A generic fluid region in arbitrary motion

Source: Author

Lighthill (1952) developed a density-based acoustic analogy by rearranging the continuity (2.1) and momentum (2.2) equations considering the sound propagation in a uniform medium, with no sources or external forces. The following algebraic development is described according the study of Orselli (2012):

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u_i)}{\partial x_i} = 0 \quad (2.1)$$

$$\frac{\partial(\rho u_i)}{\partial t} + \frac{\partial(\rho u_i u_j)}{\partial x_i} = \frac{\partial \tau_{ij}}{\partial x_j} - \frac{\partial p}{\partial x_i} \quad (2.2)$$

Here, ρ is the fluid density, u_i and u_j are the velocities in the x_i and x_j directions, p is the fluid pressure and τ_{ij} is the viscous stress tensor, given by:

$$\tau_{ij} = \mu \left\{ \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3} \left(\frac{\partial u_k}{\partial x_k} \right) \delta_{ij} \right\} \quad (2.3)$$

By time deriving equation 2.1 and applying the divergent operator to equation 2.2, we led to, the equations 2.4 and 2.5:

$$\frac{\partial^2 \rho}{\partial t^2} + \frac{\partial^2(\rho u_i)}{\partial t \partial x_i} = 0 \quad (2.4)$$

$$\frac{\partial^2(\rho u_i)}{\partial x_i \partial t} + \frac{\partial^2(\rho u_i u_j)}{\partial x_j \partial x_i} = \frac{\partial^2 \tau_{ij}}{\partial x_i \partial x_j} - \frac{\partial^2 p}{\partial x_i^2} \quad (2.5)$$

Afterwards, combining the equations 2.4 and 2.5, we have:

$$\frac{\partial^2 \rho}{\partial t^2} = \frac{\partial^2(\rho u_i u_j)}{\partial x_j \partial x_i} - \frac{\partial^2 \tau_{ij}}{\partial x_i \partial x_j} + \frac{\partial^2 p}{\partial x_i^2} \quad (2.6)$$

Subtracting the term $c_0 \frac{\partial^2 \rho'}{\partial x_i^2}$ from both sides of the equation 2.6, one yields the following inhomogeneous wave equation:

$$\frac{\partial^2 \rho'}{\partial t^2} - c_0 \frac{\partial^2 \rho'}{\partial x_i^2} = - \frac{\partial^2 T_{ij}}{\partial x_i \partial x_j} \quad (2.7)$$

where c_0 is the speed of sound in the undisturbed flow, and T_{ij} is the Lighthill stress tensor:

$$T_{ij} = \rho u_i u_j - \tau_{ij} + (p - c_0^2 \rho') \delta_{ij} \quad (2.8)$$

Equation 2.7 is the Lighthill Acoustic Analogy, which is derived from the Navier-Stokes equation and defines a region of sound sources (right side) and another of propagation of sound waves (left side). The hydrodynamic fluctuations are considerable in the near-field (the region closer to the moving fluid), but due to viscous phenomena they tend to evanesce. The terms in the right side of equation 2.7 are significant in the near-field and tend to zero in the far-field. Nevertheless, the left side represents the propagation of acoustic perturbations at the speed of sound until the infinite, playing an important role in the far field region.

A relevant meaning that emanates from the Lighthill stress tensor is that the density fluctuations in the real flow are due to the difference between the effective stresses in the real flow ($\rho u_i u_j + p \delta_{ij}$) and the stresses in the stationary acoustic medium ($c_0^2 \rho' \delta_{ij}$). Furthermore, the convection of sound (partially due to $\rho u_i u_j$), its dissipation by conduction (partially due to $c_0^2 \rho' \delta_{ij}$) and by viscosity (in the contribution of τ_{ij}) are also incorporated to the definition of T_{ij} . It is important to remark that such theory was developed under aerodynamics assumptions and, as we will see in a foregoing chapter, despite the low Mach number (subsonic regime) of an underwater propeller operation, turbulence contribution is not negligible on propeller broadband noise.

2.3 The Ffowcs Williams and Hawkings Acoustic Analogy

Further improvement to the Lighthill analogy was done by Ffowcs-Williams & Hawkings (1969) as a way to consider the arbitrary motion of a solid surface immersed in a fluid medium, in which the governing equations of density fluctuations are arranged in an inhomogeneous wave equation similar to the Lighthill's equation (2.7), taking into account dipole and monopole inhomogeneities concentrated on the bounding surfaces and quadrupoles in the fluid volume.

It is appropriate that before starting the definition of the Ffowcs Williams and Hawkings acoustic analogy, one must ensure that the theoretical understanding of the aforementioned sound sources is clear. In this regard, Howe (2002) provides a valuable explanation about sound sources (monopoles, dipoles, quadrupoles) based on the general idea of a pulsating sphere (Figure 8) of mean radius a and normal velocity $v_n(t)$.

Figure 8 – Pulsating sphere

Source: Howe (2002) (Adapted)

The mathematical modeling developed by Howe (2002) considers that the pressure p at a given distance r from a volume point source q is given by:

$$p(t) = \rho \frac{a^2}{r} \frac{dv_n}{dt}(t) \quad (2.9)$$

Thus, for any time t , the volume flux $q(t)$ of fluid across any surface ψ surrounding a pulsating sphere may be written as:

$$q(t) = \oint_{\psi} \nabla \varphi \cdot d\psi = 4\pi a^2 v_n(t) \quad (2.10)$$

and we may also write

$$\varphi = \frac{-q(t)}{r}, \quad r > a \quad (2.11)$$

The formulation for the pulsating sphere may lead to the equations that describes the far-field sound pressure amplitude for each basic acoustic source, i.e., monopoles, dipoles and quadrupoles (lateral and longitudinal). These elements are important to modelling the sound sources in complex and turbulent flows by combining conveniently pulsating spheres at a given layout. Russel *et al.* (1999) provides the following equations that shows the spatial distribution of the sound pressure for monopoles (Eq. 2.12), dipoles (Eq. 2.13) and quadrupoles (Eq. 2.14, Eq. 2.15). Figure 9 shows the spatial distribution of pulsating spheres, whose directivities are given in figure 10.

$$|p(r, \theta, t)|_m = \frac{q(t)\rho c k}{4\pi r} \quad (2.12)$$

$$|p(r, \theta, t)|_d = \frac{q(t)\rho c k}{4\pi r} k d \cos \theta \quad (2.13)$$

$$|p(r, \theta, t)|_{q1} = \frac{q(t)\rho c k^2}{4\pi r} k d_1 d_2 \cos \theta \sin \theta \quad (2.14)$$

$$|p(r, \theta, t)|_{q2} = \frac{q(t)\rho c k^2}{4\pi r} 4k d_1 d_2 \cos^2 \theta \quad (2.15)$$

Figure 9 – Sound sources scheme

Source: Author

Figure 10 – Sound sources directivities

Source: Author

The next step after defining the mathematical description of the basic sound sources is the development of Ffowcs Williams and Hawkings acoustic analogy itself. Then, let $\forall$ be a fixed volume of fluid surrounded by an arbitrary surface Σ and divided into regions 1 and 2 by a rigid moving surface S penetrating on region 2 with velocity v . Consider λ the outward normal vector from $\forall$ and n the normal to S pointing from region 1 to region 2 as indicated in Figure 11.

Figure 11 – Solid surface S in arbitrary motion

Source: Author

The derivation of FW-H equation is accomplished by rearranging the mass and momentum conservation equations to yield a solution to the Lighthill's equation (2.7), which is able to predict the noise propagation from a moving surface at the far-field.

The general form of FW-H solution may be written as a total pressure disturbance p' at the receiver position (x) divided into three contributions: Thickness term (p'_T) – monopole; loading term (p'_L) – dipole; and volume term (p'_Q) – quadrupole. Hence Equation shows how the pressure disturbance is composed by the contributions of the canonical sound sources explained in the previous section:

$$p'(\vec{x}, t) = p'_T(\vec{x}, t) + p'_L(\vec{x}, t) + p'_Q(\vec{x}, t) \quad (2.16)$$

The thickness term of surface noise is dependent only on the shape and the motion of the body and is caused by the resulting displacement of the fluid. Additionally, the loading term is an adverse effect due to the acceleration of the force distribution

on the fluid around the body due its motion. Finally, the quadrupole term is a volume distribution of sources, which accounts for nonlinearities in the flow and it is directly related to turbulence phenomena.

The final form of FW-H equation is represented by 2.17 and it is valid in the entire unbounded space. It was fully derived by Ffowcs-Williams & Hawkings (1969) and Farassat (1975), based on the Lighthill acoustic analogy.

$$\mathbb{D}^2 p' = \frac{\partial[\rho_0 U_n \delta(f)]}{\partial t} - \frac{\partial[L_i \delta(f)]}{\partial x_i} + \frac{\partial^2[H(f)T_{ij}]}{\partial x_i \partial x_j} \quad (2.17)$$

where $\mathbb{D}^2 = (\frac{\partial^2}{\partial t^2} - c_0^2 \nabla^2)$ is the D'Alembert operator, $H(f)$ is a Heaviside's unity function defined as $H(f) = 1$ if $f > 0$ (outside S) and $H(f) = 0$ if $f < 0$ (inside S). By assuming linear propagation, i.e. $p' \approx c_0^2 \rho' = c_0^2(\rho - \rho_0)$, the terms U_n and L_i may be defined as:

$$U_n = (1 - \frac{\rho}{\rho_0})v_n + \frac{\rho u_n}{\rho_0} \quad (2.18)$$

$$L_i = P_{ij}n_j + \rho u_i(u_n - v_n) \quad (2.19)$$

The solution of equation 2.17 is calculated by applying a convolution product and adopting appropriated free space Green's functions of the type $G(g) = \frac{\delta(g)}{4\pi r}$, where $g = t - \tau - \frac{r}{c}$, with τ and t as the times associated to a same acoustic wave in the source and observer respectively. Hence, by assuming a moving source in a subsonic regime, one has:

$$4\pi p'(\vec{x}, t) = \frac{\partial}{\partial t} \int_{f=0} \left[\frac{U_n}{r(1-M_r)} \right]_{ret} dS - \frac{\partial}{\partial x_i} \int_{f=0} \left[\frac{L_i}{r(1-M_r)} \right]_{ret} dS + \frac{\partial^2}{\partial x_i \partial x_j} \int_{f>0} \left[\frac{T_{ij}}{r(1-M_r)} \right]_{ret} dV \quad (2.20)$$

where the subscript *ret* stands for the retarded time, $r = |\vec{x} - \vec{y}|$ is the distance between observer and source, the index i represents a variable in the x_i direction, M_r is the

Mach number of a noise source projected in the observer direction. Moreover, the terms U_n , L_i and T_{ij} may be interpreted as the monopole (thickness), dipole (loading) and quadrupole (volume) sources respectively.

After further development, it was proposed an alternative approach to the FW-H equation, by considering a permeable (or porous) data surface (PDS), surrounding the vicinity of an impermeable surface in arbitrary motion. The PDS accounts the nonlinearities (quadrupole term) of the flow due to the movement of the impermeable surface. This is the so-called *permeable FW-H equation*. A general time domain solution of equation 2.17 was proposed by Brentner & Farassat (1988) and Brentner & Farassat (1998), where the monopole (Eq. 2.21) and dipole terms (Eq. 2.22) were derived (Formulation 1A):

$$p'_T(\vec{x}, t) = \frac{1}{4\pi} \int_{f=0} \left[\frac{\rho_0 \dot{v}_n}{r(1-M_r)^2} + \frac{\rho_0 v_n \hat{r}_i \dot{M}_i}{r(1-M_r)^3} \right]_{ret} dS + \frac{1}{4\pi} \int_{f=0} \left[\frac{\rho_0 c v_n (M_r - M^2)}{r^2 (1-M_r)^3} \right]_{ret} dS \quad (2.21)$$

$$p'_L(\vec{x}, t) = \frac{1}{4\pi} \int_{f=0} \left[\frac{\dot{p} \cos \theta}{cr(1-M_r)^2} + \frac{\hat{r}_i \dot{M}_i p \cos \theta}{cr(1-M_r)^3} \right]_{ret} dS + \frac{1}{4\pi} \int_{f=0} \left[\frac{p(\cos \theta - M_i n_i)}{r^2 (1-M_r)^2} + \frac{(M_r - M^2) p \cos \theta}{r^2 (1-M_r)^3} \right]_{ret} dS \quad (2.22)$$

In addition, to compute the quadrupole terms of noise, Brentner (1997) developed the Formulation Q1A as follows:

$$p'_Q(\vec{x}, t) = \frac{1}{4\pi} \int_{f>0} \left[\frac{K_{r1}}{c^2 r} + \frac{K_{r2}}{cr^2} + \frac{K_{r3}}{r^3} \right]_{ret} dS \quad (2.23)$$

Here, we have:

$$K_{r1} = \frac{\ddot{Q}_{rr}}{(1-M_r)^3} + \frac{\ddot{M}_r Q_{rr} + 3\dot{M}_r \dot{Q}_{rr}}{(1-M_r)^4} + \frac{3\dot{M}_r^2 Q_{rr}}{(1-M_r)^5} \quad (2.24)$$

$$K_{r2} = \frac{-\dot{Q}_{ii}}{(1 - M_r)^2} - \frac{4\dot{Q}_{M_r} + 2Q_{\dot{M}_r} + \dot{M}_r Q_{ii}}{(1 - M_r)^3} + \frac{3 \left[(1 - M^2) \dot{Q}_{rr} - 2\dot{M}_r Q_{M_r} - M_i \dot{M}_i Q_{rr} \right]}{(1 - M_r)^4} + \frac{6\dot{M}_r Q_{rr} (1 - M^2)}{(1 - M_r)^5} \quad (2.25)$$

$$K_{r3} = \frac{2Q_{MM} - (1 - M^2) Q_{ii}}{(1 - M_r)^3} - \frac{6Q_{M_r} (1 - M^2)}{(1 - M_r)^4} + \frac{3Q_{rr} (1 - M^2)}{(1 - M_r)^5} \quad (2.26)$$

$$Q_{ij} = \int_{f>0} T_{ij} dz \quad (2.27)$$

$$Q_{rr} = Q_{ij} \hat{r}_i \hat{r}_j \quad (2.28)$$

where Q_{ij} is the quadrupole surface source tensor, M_r is the Mach number of the source in the radiation direction ($M_i \hat{r}_i$).

Hence, after establishing the theoretical background related to Ffowcs Williams - Hawkings acoustic analogy, a brief discussion of the features of underwater propeller noise will be developed in the next section, since such topic is one of the main subjects of the present study.

2.4 Underwater Propeller Noise

Since the publication of the acoustic analogies and the formulations concerning the aerodynamic noise from jets and propellers, some studies have been developed about the acoustic behavior of aircrafts and aeronautical devices, such as Farassat (1975), Brentner & Farassat (1988), Brentner (1997) and Brentner & Farassat (1998). A valuable conclusion developed about the nature of the radiated sound in air was that under subsonic flow conditions, the acoustic field is strictly dominated by the noise generated due to body surface interactions with fluid, i.e., thickness and loading noise components, leaving aside the quadrupole (non-linear) term which plays a relevant role just at transonic or supersonic regime and requires a huge computational effort to be calculated (IANNIELLO *et al.*, 2013); (BRENTNER; FARASSAT, 2003).

Nonetheless, not long-ago, propeller underwater noise has been the focus of some researchers concerned about the singularities of such highly turbulent flow characterized by very low rotational Mach number $M_R = \omega R / c_0$ (where ω is the rotational velocity, R is the blade radius and c_0 is speed of sound in the undisturbed medium). As stated by Ianniello (2016), underwater M_R typically varies between 0.01 and 0.02, whereas the range of aeronautical values is 0.3-0.5 for regular subsonic operation conditions.

According to Carlton (2007), there are five main mechanisms by which a propeller generates underwater noise: The displacement of the water by the blade; the pressure gradient between the suction and pressure sides of the blade; the flow over the surface blade; the periodic fluctuation of the cavity volumes caused by the propeller operation; and the sudden collapse process due to cavitation bubble or vortex. The latter two causes occur only when propeller is under cavitating regime. It is important to notice that such causes are related to both surface and volume noise terms of the FW-H equation.

Carlton (2007) also explains that the broadband noise spectrum of a typical marine propeller comprises components derived from inflow turbulence into the propeller and various boundary layer and edge effects, such as vortex shedding and trailing edge noise. This latter component of propeller noise may be the least well understood of the broadband noise mechanisms, since it involves a detailed knowledge of the flow around the trailing edge of the blade section. We also may highlight that the role of viscosity within the boundary layer is a crucial parameter in estimating the levels of radiated noise produced by the marine propeller.

The components of the propeller noise also comprise a series of periodic components (tones) at blade rate and its multiples, together with a spectrum of high-frequency noise due to cavitation and blade boundary layer effects (CARLTON, 2007). In this sense, Figure 2.12 presents a radiated cavitating propeller noise spectrum based on a third-octave band analysis (which is a common approach in propeller noise analysis); the sound pressure levels are referred to 1 μ Pa reference pressure.

Figure 12 – Radiated cavitation noise spectra measured outside a hull at full-scale

Source:Carlton (2007)

Considerable contribution to the assessment of underwater propeller noise has been made by authors such as Ianniello *et al.* (2012), Ianniello *et al.* (2013), Ianniello (2014), Ianniello & Bernardis (2015), Ianniello (2016) and Seol *et al.* (2002), where a wide understanding about the acoustic behavior of marine propellers was developed through the use of diverse numerical tools based on the FW-H analogy.

Ianniello *et al.* (2013) give a detailed panorama of underwater noise assessment during the past years. By the way, such work provides a valuable benchmark on the parsing of propeller noise in uniform flow (open-water condition), in a scaled model and full-scale ship as well as the development of several methodologies to measure and predict hydroacoustic variables of interest by experiments and numerical simulations, and discussion about criteria of adequate turbulence models and meshing uncertainty.

Furthermore, Ianniello *et al.* (2013) also showed that there is an important contrast between the noise generated by subsonic aerodynamic and underwater flows, in relation to non-linear noise terms. Regardless of the low rotational blade speed, to adequately measure the noise radiated from a marine propeller, one cannot discard the contribution of the nonlinear noise sources represented by the turbulence and vorticity fields, therefore requiring the computation of the FW-H quadrupole source terms.

On the other hand, it is important to highlight the study of Zimmerman & Chadwell (2005), where the authors investigated strategies of lowering AUV acoustic self-noise. One approach adopted in such study was the using of a propeller with high thrust per rotation speed, which in turn would decrease the propeller speed, yielding in a minoring of the hull induced vibrations as well as the radiated acoustic noise.

Numerical Scheme

As we saw in Chapter 2, the acoustic analogies developed by Lighthill and Ffowcs Williams-Hawkings described the noise by an extensive mathematical formulation. On these wise, robust numerical tools are needed to calculate the noise by means of the aforementioned analogies. This chapter provides the main numerical procedures to account the desirable hydrodynamic and hydroacoustic variables needed to evaluate the propeller underwater noise. The numerical problem may be divided in 2 parts: the first one is the definition of hydrodynamic field due to propeller operation by means of steady state numerical simulations in order to provide initial conditions to further transient hydroacoustic simulations (the second part), where the Ffowcs Williams-Hawkings porous formulation will be used to account the propeller underwater noise coupled with the Detached Eddy Simulation (DES) as turbulence model.

3.1 Governing Equations

Let us consider again the fluid motion in the continuum approximation. The fundamental conservation laws for a Newtonian fluid in a turbulent flow are the continuity and momentum equations:

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u_i)}{\partial x_i} = 0 \quad (3.1)$$

$$\frac{\partial(\rho u_i)}{\partial t} + \frac{\partial(\rho u_i u_j)}{\partial x_j} = \frac{\partial \tau_{ij}}{\partial x_j} - \frac{\partial p}{\partial x_i} \quad (3.2)$$

In most applications, the acoustic pressure amplitude is considerable tiny if compared to the mean pressure, and sound propagation may be studied by linearized equations. Therefore, as specified by Howe (2002), an equation establishing the pressure p' may be obtained by considering the hypothesis of fluid homentropy:

$$p_0 + p' = p(\rho_0 + \rho', s) \approx p(\rho_0, s) + \rho' \left(\frac{\partial p(\rho_0, s)}{\partial \rho} \right)_0 \quad (3.3)$$

From this perspective, the pressure-density derivative might be assessed at the undisturbed values (ρ_0, p_0) , exhibiting the dimension of velocity. As a result, the speed of sound may be defined as the square root of such derivative:

$$c_0 = \sqrt{\left(\frac{\partial p}{\partial \rho} \right)_s} \quad (3.4)$$

As the entropy s is held constant at the value assumed in the undisturbed flow, the implication that arises is the neglecting of losses due to heat transfer between the fluid particles by viscous and thermal diffusion during the propagation of a sound wave, in other words, the motion of a fluid particle is adiabatic.

Thereby, from equations 3.3 and 3.4, a linear relationship between pressure and density that numerically describes the hypothesis of a compressible fluid is developed as:

$$p' \approx c_0^2 \rho' = c_0^2 (\rho - \rho_0) \quad (3.5)$$

The equation 3.5 was implemented in the CFD code by means of an equation of state (EOS). This equation is important because it accounts the small adiabatic acoustic pressure fluctuations in the fluid domain and, thereupon, the underwater noise may be adequately modelled.

3.2 Turbulence Modeling

To solve the turbulent flow variables, one may adopt a methodology for turbulence modeling as a means to well calculate the large eddy scales and modeling the small ones, due to the very time-consuming approach of directly resolve the Navier-Stokes

Equations for all turbulent scales. The main turbulence models adopted in acoustic purposes are the URANS (Unsteady Reynolds Averaged Navier-Stokes), LES (Large Eddy Simulation) and hybrid RANS-LES such as DES (Detached Eddy Simulation).

3.2.1 RANS Turbulence Model

The Reynolds Averaged Navier-Stokes Equations (RANS) turbulence models are one of the most widely used in computational applications of fluid dynamics, where statistical averaging is based on the so-called Reynolds decomposition, in which the flow variables $\Phi(\vec{x}, t)$ are broke-down into a time-mean value $\bar{\Phi}(\vec{x}, t)$ and a fluctuating component $\Phi'(\vec{x}, t)$, such as:

$$\Phi(\vec{x}, t) = \bar{\Phi}(\vec{x}, t) + \Phi'(\vec{x}, t) \quad (3.6)$$

Hence, to obtain the time-averaging form of the original governing equations, the averaging and fluctuating components of each variable must be properly replaced and the time average of the resulting equations must be taken. In this sense, according to Moukalled *et al.* (2016), the Reynolds averaged forms of conservation equations of continuity, momentum and energy are obtained as:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \bar{\mathbf{v}}) = 0 \quad (3.7)$$

$$\frac{\partial (\rho \bar{\mathbf{v}})}{\partial t} + \nabla \cdot (\rho \bar{\mathbf{v}} \otimes \bar{\mathbf{v}}) = -\nabla \bar{p} + \left[\nabla \cdot (\bar{\sigma}_{ij} - \rho \overline{\mathbf{v}' \mathbf{v}'}) \right] + \bar{f}_b \quad (3.8)$$

$$\frac{\partial (\rho c_p \bar{T})}{\partial t} + \nabla \cdot (\rho c_p \bar{\mathbf{v}} \bar{T}) = \nabla \cdot (k \nabla \bar{T} - \rho c_p \overline{\mathbf{v}' T'}) \bar{S}^T \quad (3.9)$$

where $\mathbf{v}$ is the fluid velocity, f_b are the body forces, p is the fluid pressure, T is the fluid temperature, k is the thermal conductivity of the fluid, c_p is the specific heat at constant pressure of the fluid, and the term $\rho \overline{\mathbf{v}' \cdot \mathbf{v}'}$ is the Reynolds stress tensor (τ^R).

The above equations are similar to the original conservation formulae, with the exception of the additional averaged products of the fluctuating components due to the non-linear terms. Such method introduces new unknowns ($\overline{\rho \mathbf{v}' \cdot \mathbf{v}'}$ and $\rho c_p \overline{\mathbf{v}' T'}$), whose determination is the core of the turbulence modeling approach.

One direct modeling approach of the Reynolds stress tensor is based on the Boussinesq's turbulent viscosity hypothesis which assumes τ^R as a linear function of the mean velocity gradients:

$$\tau^R = -\rho \overline{v'v'} = \mu_t \left[\nabla \vec{V} + (\nabla \vec{V})^T \right] - \frac{2}{3} [\rho k_e + \mu_t (\nabla \cdot \vec{V})] I \quad (3.10)$$

with μ_t as the turbulent eddy viscosity and $k_e = \frac{1}{2} \overline{v'v'}$ as the turbulent kinetic energy. Such approximation transforms the problem of determining the Reynolds stress components into computing μ_t and k_e . Similarly, the turbulent thermal flux ($\rho c_p \overline{v'T'}$) may be determined by means of the turbulent thermal diffusivity (k_t) such as $\rho c_p \overline{v'T'} = k_t \nabla T$.

Some turbulence models that consider the Boussinesq hypothesis have been developed to calculate the turbulent viscosity. Nevertheless, according to Moukalled *et al.* (2016) none of them is universally applicable to all flow conditions. In this sense, one of the most widely used two-equations turbulence model is the $k - \omega$, developed by Wilcox (1988) and revised by Wilcox (1994) and Wilcox (2008). In such turbulence model, ω is the specific turbulence dissipation which represents the rate of conversion of turbulence kinetic energy into internal energy per unit volume and time:

$$\omega = \frac{\varepsilon}{k_e C_\mu} \quad (3.11)$$

where ε is the turbulent kinetic energy dissipation rate and C_μ is a model constant.

Further modifications to k -model have improved its adverse pressure gradient performance yielding to the Shear Stress Transport (SST) $k - \omega$ developed by Menter (1994), that stated the linear relationship between the shear stress and the turbulent kinetic energy in the boundary layer region (Bradshaw's hypothesis). Nowadays, the $k - \omega$ SST has become one of the most used RANS turbulence models in CFD applications.

3.2.2 Detached Eddy Simulation

Detached eddy simulation (DES) is an unsteady and hybrid turbulence modeling approach first proposed by Spalart *et al.* (1997) for prediction of turbulent flows at high Reynolds numbers. DES combines features of Reynolds-Averaged Navier-Stokes (RANS) simulation in the attached boundary layer flow and Large Eddy Simulation (LES) in the regions of separated flow far from the wall. Squires (2004) declares that

the application of LES to complete configurations of fully turbulent flows is quite costly due to the high resolution required in the boundary layer region. Therefore, a hybrid approach is needed to modeling the boundary layer region with lower computational cost.

However, RANS approaches present the disadvantage of underestimating the solution in high Reynolds number separated flows, because its intrinsic statistical models are designed and calibrated on the basis of averaged parameters of thin turbulent shear flows composed by “standard” eddies that are not representatives of typically separated flows structures.

Wagner *et al.* (2007) observes that the purpose of time-dependent techniques such as LES or DES is to account only the largest scales of motion, i.e., the most significant noise-generating scales. On the other hand, one may expect the problem of missing noise due to unresolved scales to be exacerbated by the use of global hybrid RANS–LES methodologies. In addition, the authors emphasized that the coupling of DES with sound prediction methods has not progressed far, but the confidence in its capabilities to reproduce the dominant eddies of a separated flow is good enough to be optimistic.

3.3 Computational Hydroacoustics Scheme

This section briefly provides some details of numerical simulations mainly regarding the FW-H porous formulation, applied to compute the non-linear noise terms by permeable surface integrals and few numerical requirements intrinsic to CFD simulations, as it plays the role as input of hydroacoustics analysis.

3.3.1 FW-H Porous Formulation

The *Farassat's formulation 1A* presented in equations 2.21 and 2.22 is typically applied in general subsonic source regions to predict far-field noise. Nonetheless, for subsonic rotating blades, the *Dunn-Farassat-Padula Formulation 1A*, developed by Farassat *et al.* (1987) and based on the modification of Farassat's Formulation 1A and on Hubbard (1991), will be used to compute the acoustic pressure due to thickness and loading terms, as well as the equation 2.23 (Formulation Q1A) to calculate the non-linear quadrupole term. Hence, the whole formulation of the FW-H Porous approach is given by :

$$p'_T(\vec{x}, t) = \frac{1}{4\pi} \left\{ \int_{f=0} \left[\frac{\rho_0 v_n [r \dot{M}_r + c (M_r - M^2)]}{r^2 (1 - M_r)^3} \right]_{ret} dS \right\} \quad (3.12)$$

$$p'_L(\vec{x}, t) = \frac{1}{4\pi} \left\{ \left(\frac{1}{c} \right) \int_{f=0} \left[\frac{\dot{L}_i r_i}{r (1 - M_r)^2} \right]_{ret} dS + \int_{f=0} \left[\frac{L_r - L_i M_i}{r^2 (1 - M_r)^2} \right]_{ret} dS + \right. \\ \left. + \left(\frac{1}{c} \right) \int_{f=0} \left[\frac{L_r [r \dot{M}_r + c (M_r - M^2)]}{r^2 (1 - M_r)^3} \right]_{ret} dS \right\} \quad (3.13)$$

$$p'_Q(\vec{x}, t) = \frac{1}{4\pi} \int_{f>0} \left[\frac{K_{r1}}{c^2 r} + \frac{K_{r2}}{c r^2} + \frac{K_{r3}}{r^3} \right]_{ret} dS \quad (3.14)$$

It is worth mentioning that the FW-H formulation is used in its porous and non-convective versions, i.e., the surface of integration is the porous surface S that will account both the linear and non-linear noise terms generated by the rotation of the impermeable surface (propeller blades) and propagated throughout the volume $\forall$ (Figure 13). The porous surface S used in this study was based on Ianniello *et al.* (2013) and Lloyd *et al.* (2015), i.e., a cylinder open upstream and downstream. The lateral surface of the cylinder surrounds the propeller and its turbulent wake in order to avoid the crossing of the wake through the integration surface and to avoid the generation of spurious noise.

Figure 13 – Scheme of impermeable and permeable FW-H surfaces

Source: Author

The correct choice of the permeable surface is quite important to achieve good results. Ianniello & Bernardis (2015) developed a study that showed the effect of a closed cylindrical permeable surface on the underwater noise generated by a marine propeller. According to the authors, as the receptor moves downstream to the propeller wake, the contribution provided by the cap located behind the propeller becomes dominant and heavily affects the overall signature of the propeller radiated noise. On the other hand, the permeable surface composed just by the lateral face of the cylinder provided good results.

3.3.2 Numerical Requirements

The FW-H approach can provide accurate far-field noise prediction by using a second order discretization scheme, both spatial and temporal, embedded with reasonable choices for mesh resolution and time-step size. It only predicts the propagation of sound in free space and does not include effects such as sound reflection, refraction, or material property changes. The input of FW-H model needs time-accurate data to predict the near field flow, as well as an adequate fine mesh used for accurate prediction of the boundary layer or other requirements for turbulence models. Moreover, the computational domain for this model does not need to include the location of the receivers, because the predicted noise field is calculated by surface integrals over S (Figure 13).

The computational mesh used in this study will follow a discretization that must comply with the minimum requirements of 20 cells per wavelength (defined by the higher frequency of the noise spectra) or 10 cells per diameter of the vortex core. Hence, the typical dimension Δm of a mesh element is given as

$$\Delta m = \min \left(\frac{c}{20f_u}, \frac{D_v}{10} \right) \quad (3.15)$$

where f_u is the upper limit of the analyzed frequency spectrum and D_v is the vortex diameter.

The discretization of time on this model takes into account the requirements of the International Towing Tank Conference (ITTC, 2011) that suggests a minimum of 200 time-steps (Δt) per revolution. In addition, to ensure temporal and spatial fidelity so as to enforce numerical stability and accurate description of the resolved scales of motion, especially considering the LES-based requirements of DES model, it is also important

to achieve an adequate convective Courant-Friedrich-Lewy number (CFL), that takes into account the fluid velocity (u) and the dimension of a mesh element (Δm):

$$CFL = u \frac{\Delta t}{\Delta m} \quad (3.16)$$

3.3.3 Simulations Setup

The CFD simulations (to both E779A and Mod5 propellers) were developed in the commercial software Star CCM⁺, by means of the Finite Volume Method coupled with an implicit unsteady solver and a 2nd order temporal discretization in the transient simulations. In addition, a total of 20 inner iterations were used for each time step.

The grid generation models used to discretize the computational domain was the *Trimmed Mesher*, which develops unstructured hexahedral cells, and the *Prism Layer Mesher*, that generates orthogonal prismatic cells layers next to wall surfaces in order to capture the boundary layer effects. In this sense, a total of 35 prism layers were used in the blades and hub surfaces to reach $y^+ \approx 1$, where y^+ is the dimensionless wall distance.

The rotational movement of the propeller operation was modeled by means of the sliding mesh approach, in which one grid domain rotates relatively to a stationary domain. As the solution advances in time, the cell alignment changes within the domain and the governing equations are solved in each cell zone and the fluxes are calculated across the interfaces between the domains.

The compressibility of water was modeled by means of a User Defined EOS (Equation of State) model that considers the relation described in equation 3.5. Regarding the propagation of sound, the FW-H acoustic model implemented in Star CCM⁺ code does not include effects such as sound reflections or refractions.

The porous surface used in this study is an open cylinder modeled far from the interface between the two grid domains (static and rotating) in order to improve the interpolation of hydrodynamic variables. The surface dimensions were determined by following the parameterization of Lloyd *et al.* (2015), which prescribed that the porous surface diameter and length were, respectively, $1.25D_P$ and $3D_P$, where D_P is the propeller diameter.

Table 1 lists some physical models and other setups used in the numerical simulations of both propellers (E779A and Mod5). It is worthy mentioning that a steady state simulation was initially developed until reach the convergence. The Menter's $k-\omega$ SST model was used to improve the initial conditions and speed up convergence of the

further transient simulations with DES approach.

Table 1 – Simulations setup for the E779A and Mod5 propellers cases

Viscous Regime	Turbulent
Turbulence Model	DES with $k-\omega$ SST and DDES formulation
Temporal discretization	2nd order upwind
Convection scheme	Hybrid-BCD
Equation of State	User Defined Density
Wall treatment	All $y+$ wall treatment
Time	Implicit Unsteady
Maximum Inner iterations	20
Time-steps per propeller revolution	256 (E779A)
	720 (Mod5)
Number of propeller revolutions	10
Acoustic model	Ffowcs Williams-Hawkings (FW-H)

Source: Author

The final number of propeller revolutions was determined by assessing the acoustic signals of the FW-H model. The first revolutions are neglected in order to eliminate spurious transient measurements and the FW-H model just accounts the final ones. The simulations developed in this study considered a total of 10 propeller revolutions, neglecting the first 4.

It is important to clarify some points about the simulation's setup listed in table 1. The Equation of State was modelled according to equation 3.5, i.e., a linear relationship between the fluid pressure and density. The number of time steps per propeller revolutions for the E779A case was chosen in accordance with the prescribed by Ianniello *et al.* (2013). Nevertheless, to the Mod5 propeller case, it was chosen the way described in ITTC (2011), in order to have a more detailed temporal discretization of the propeller revolution.

3.3.4 The Mod5 Propeller

Mod5 is a 7-bladed high-skewed propeller (Figure 14) designed by the Hydrodynamics Laboratory (LabHidro) at the Brazilian Navy Nuclear Development Directorate (DDNM), based on the methodology proposed by Sbragio (1995) in which the propeller geometry is developed by means of the Lifting Line and Lifting Surface theories and the Biot-Savart law to modeling the vortex shedding. The LabHidro Mod5 propeller, whose main characteristics are showed in Table 2, was originally projected to operate in an AUV hull (Figure 15) based on a 1:1.588 scaled DARPA Suboff model (GROVES *et al.*,

1989), whose parameters are listed in Table 3. A brief description of the design of Mod5 propeller is presented in Neto *et al.* (2018).

Figure 14 – Mod5 propeller geometry

Source: Author

Table 2 – Mod5 propeller parameters

Diameter	0.1889 m
Hub diameter	0.0378 m
Number of blades	7
Rake	0°
Skew	20°
Pitch/diameter ratio	1.216
Chord length 0.7R	0.03658 m
Rotation rate (n)	1008 rpm
Advance ratio (J)	0.9194

Source: Author

Figure 15 – AUV hull with Mod5 propeller

Source: Author

Table 3 – Scaled DARPA Suboff AUV parameters

Length overall	2.743 m
Maximum hull radius	0.16 m
Volume displacement	0.177 m ³

Source: Author

3.3.5 Boss Cap Geometries

The divergent geometries to be investigated are parameterized with respect to the hub radius r_h according to:

$$r_a = \left(1 + \frac{a}{10}\right) r_h, \quad a = 0, 1, 2, 3 \quad (3.17)$$

where r_a is the outer radius of divergent boss cap named *Div-a*. Figure 16 depicts the variations of boss cap profiles, in a way that the elliptical cap follows the DARPA Suboff stern contour, as described in Groves *et al.* (1989), whereas each divergent profile is generated by the revolution around the propeller axis of following curve:

$$y(x) = \left(\frac{\frac{a}{10}}{L_c^2}\right) x^2 + r_h \quad (3.18)$$

where L_c is the cap length.

Figure 16 – Boss cap profiles

Source: Author

Figure 17 shows the different configurations of Mod5 propeller with each boss cap geometry that will be further analyzed with a view to determine its influence on propeller hydroacoustic features.

Figure 17 – Mod5 propeller configurations

Source:Author

Numerical Results

4.1 Benchmark: INSEAN E779A Propeller

This section describes the application of the aforementioned numerical model in a widely used propeller geometry: The INSEAN E779A. An analysis of the propeller hydrodynamic performance and the corresponding flow field will be made for open water conditions. Afterwards, the acoustic pressure fluctuations will be calculated by using the Ffowcs Williams and Hawkings model.

The E779A propeller is a 4-bladed fixed-pitch propeller tested at the Italian Marine Technology Research Institute (INSEAN) in both numerical and experimental procedures. It is a Wageningen modified shape with a relatively simple geometry characterized by limited skew and rake. This model has been extensively used for validation of numerical simulations of propellers. The main particulars of E779A propeller are listed in Table 4 and a view of its geometry is presented in Figure 18.

Table 4 – E779A propeller parameters

Diameter (D)	0.227 m
Expanded blade area ratio	0.689
Number of blades	4
Rake	4°
Pitch/diameter ratio	1.1
chord length 0.7R	0,086 m

Source: Adapted from Ianniello *et al.* (2013) and Lloyd *et al.* (2015)

Figure 18 – E779A propeller geometry

Source: Author

The numerical simulations were performed to determine the flow pattern and the performance coefficients of E779A propeller in open water conditions, following the information provided by Lloyd *et al.* (2015)) and Ianniello *et al.* (2013), i.e., computational domain, temporal discretization and further information about the grid discretization process.

The computational domain was defined as a cylinder whose dimensions were parameterized by the propeller diameter based on the described in Lloyd *et al.* (2015). Figure 19 represents the computational domain divided in rotating and static regions, as well as the boundary conditions and the locations of the hydrophones (Probes 2 and 3). It must be emphasized that the orientation of the propeller adopted in this study was in a way that the velocity and outlet boundary conditions are inverted if compared to the domain used in Lloyd *et al.* (2015), in order to simulate the hub vortex such as in Ianniello *et al.* (2013).

Figure 19 – Computational domain and boundary conditions

Source: Adapted from Lloyd *et al.* (2015) (Not to scale)

The probes considered in this study were placed according to the location of the hydrophones 2 and 3 used by Ianniello *et al.* (2013), i.e., probe 2 is located 1.5 radii from the blade tip, and probe 3 is located 1.5 radii downstream from probe 2 (Figure 20).

Figure 20 – Location of the hydrophones (probes) 2 and 3 in Ianniello *et al.* (2013) simulations

Source: Adapted from Ianniello *et al.* (2013)

The uniform flow condition was developed by considering an advance ratio (Equation 4.1) of $J=0.88$, corresponding to an inflow velocity of $V=5\text{m/s}$ and a rotation rate of $n=25\text{ rps}$. The fluid (fresh water) properties were determined conforming to ITTC (2011), as summarized in Table 5.

$$J = \frac{V}{nD} \quad (4.1)$$

Table 5 – Fresh water properties

Temperature (°C)	25
Density (kg/m ³)	997.0476
Viscosity (Pa.s)	8.90E-04
Kinematic Viscosity (m ² /s)	8.93E-07

Source: ITTC (2011)

The turbulence modeling was based upon a DES- $k-\omega$ SST model to unsteady simulations, and $k-\omega$ model to steady state simulations. The computation of the acoustic pressure was made during 6 propeller revolutions, each one discretized by 256 time steps, according to Ianniello *et al.* (2013), and 20 inner iterations for each time step. The computational domain discretization was established by means of a non-uniform hexahedral grid (Figure 21) with higher density of elements in the tip and hub vortex regions in agreement with the prescribed in section 3.3.2.

Figure 21 – Detail of mesh near the propeller surface

Source: Author

The numerical results of hydrodynamic simulations were compared with the ones obtained numerically (CFD) and experimentally (EFD) by Ianniello *et al.* (2013). The curves of thrust (K_T) and torque (K_Q) coefficients (Equations 4.2 and 4.3) are showed in Figure 22, where the red marks are the results obtained by means of CFD simulations developed in the present study.

$$K_T = \frac{T}{\rho n^2 D^4} \quad (4.2)$$

$$K_Q = \frac{Q}{\rho n^2 D^5} \quad (4.3)$$

Figure 22 – Numerical prediction of performance coefficients of E779A propeller

Source: Author

Figure 22 shows that the values obtained by CFD to the performance coefficients of E779A propeller were quite closer to the reference. The difference between the values of K_T was 0.8% and of $10K_Q$ was 7.2%.

The pressure isocontours of the E779A propeller wake are compared in Figure 23. We can observe that the tip and hub vortices of both wakes are somewhat similar. Despite the difference of boss cap geometries used in this study and the used by Ianniello *et al.* (2013), the hub vortex of both cases presented reasonable adherence.

Figure 23 – Pressure isocontours at $J=0.88$

Source: Left: Ianniello *et al.* (2013); Right: Author

The accuracy of the FW-H model developed here was evaluated by comparing the obtained results with the data provided by Ianniello *et al.* (2013), i.e., the pressure fluctuation at probe 2 (Figure 24) and the linear and non-linear components from the noise calculated at probe 3 (Figure 25). A total of 10 propeller revolutions were developed in the unsteady simulations. Moreover, in order to eliminate the transient effects and achieve accurate results, the first four propeller revolutions were neglected. The pressure pulse analysis results in this study were performed in accordance with Lloyd *et al.* (2015), i.e., the FW-H signals were treated by applying a Kaiser-Bessel windowing function, and a fast Fourier transform algorithm to remove some high-frequency numerical noise.

Figure 24 – Pressure fluctuations at Probe 2

Source:Author

Figure 25 – Results of FW-H model at Probe 3

Source:Author

A brief analysis of the data showed in figures 24 and 25 leads us to conclude that the FW-H model is somewhat accurate by predicting not only the overall pressure fluctuations, but also the linear and non-linear terms separately in different positions along the propeller wake. Furthermore, as the results of pressure pulses are related to points near to propeller plane, Ianniello *et al.* (2013) stated that RANS pressure signals and FW-H noise results presented good agreement (Figure 26) to points at the nearfield.

Figure 26 – RANS pressure and FW-H noise at probe 2 (propeller plane)

Source: Ianniello *et al.* (2013)

4.2 Mod5 Propeller: Open-Water Results

Previously to numerical simulations, a set of cavitation tunnel tests were developed in the facilities of the Naval Architecture and Ocean Engineering Laboratory at the Institute for Technological Research (IPT). The IPT cavitation tunnel has a rounded corner squared cross section of 0.5m (Figure 27), in which the propeller is installed at the end of a support axis and connected to a thrust and torque dynamometer (See Silva-Junior *et al.* (2019) for full description of the experimental setup).

Figure 27 – Detail of the test section of IPT cavitation tunnel

Source: Silva-Junior *et al.* (2019)

The experiments were performed for each boss cap geometry (1 elliptical and 4 divergent) in a non-cavitating regime, in order to determine the K_T , K_Q and η (propeller efficiency – Eq. 4.4 curves as a function of the advance ratio J , which vary approximately from 0.4 to 1.12. Figure 28 shows some experimental configurations of Mod5 Propeller in the test section, equipped with Elliptical (left) and Div3 (right) boss caps, and table 6 presents some environmental data of the tests in the IPT cavitation tunnel.

$$\eta = J \frac{K_T}{2\pi K_Q} \quad (4.4)$$

Table 6 – Experimental tests environmental data

Water temperature	24°C
Atmospheric Absolute Pressure	700 mmHg
Water Relative Pressure	44.6 mmHg

Source: Author

Figure 28 – Detail of Mod5 Propeller in the test section – Elliptical (Left) and Div3 (Right)

Source: Author

The efficiency curves determined experimentally for each boss cap geometry are shown in Figure 29, as well as the steady-state numerical results (red marks), which were developed considering the flow regime of $J=0.92$. Table 7 presents the experimental and numerical results at $J=0.92$. In addition, a mesh convergence study was performed by changing the density of mesh elements around the blades and the vortex regions, in order to observe the behavior of propeller efficiency and chose the mesh that provides the best results with the lowest number of cells. Table 8 shows the results of the convergence study and the best mesh in each case (in bold and italic).

Table 7 – Experimental (EFD) and numerical (CFD) results ($J=0.92$)

	EFD			CFD		
	K_T	$10K_Q$	η	K_T	$10K_Q$	η
Elliptical	0.162	0.349	0.677	0.175	0.363	0.687
Div0	0.163	0.387	0.618	0.172	0.397	0.634
Div1	0.166	0.397	0.614	0.173	0.399	0.633
Div2	0.165	0.398	0.609	0.172	0.404	0.623
Div3	0.169	0.391	0.633	0.176	0.397	0.649

Source: Author

Figure 29 – Experimental and numerical performance coefficients of Mod5 for each boss cap geometry

Source: Author

Table 8 – Mesh convergence data (Cells in millions)

Elliptical		Div0		Div1		Div2		Div3	
η	Cells	η	Cells	η	Cells	η	Cells	η	Cells
0.711	31	0.693	30	0.656	31.3	0.641	32	0.737	32.2
0.688	37	0.652	36.3	0.644	37.6	0.628	38.4	0.68	38.7
0.687	44.6	0.634	43.2	0.633	45.2	0.623	46.2	0.649	46.5
0.687	53.7	0.634	51.9	0.633	54.3	0.623	55.4	0.649	55.6

Source: Author

CFD calculations presented over predicted numerical efficiency at $J=0.92$ for all boss cap geometries if compared to the experimental data. Nonetheless, as shown in figure 30, except to Div3, the tendency of decreasing efficiency as the boss cap divergence increases was preserved in both experimental and numerical predictions. Furthermore, Table 9 presents the percentage of deviation between the CFD and EFD values of Mod5 propeller.

Table 9 – Deviation between EFD and CFD results for Mod5 propeller

	K_T	$10K_Q$	η
Div0	5.58%	2.70%	2.59%
Div1	3.92%	0.51%	3.09%
Div2	4.30%	1.63%	2.30%
Div3	4.14%	1.55%	2.53%

Source: Author

Figure 30 – Behavior of propeller efficiency with the boss cap geometry variation($J=0.92$)

Source: Author

Silva-Junior *et al.* (2019) developed several tests with Mod5 propeller at IPT cavitation tunnel, employing the Particle Image Velocimeter (PIV) method to measure flow field at the propeller wake. The data provided by such study are used as benchmark to validate the vorticity field that is hereby calculated by transient CFD simulations, employing the DES turbulence model. Figure 31 shows the experimental (Left) and numerical (Right) vorticity fields of Mod5 equipped with elliptical boss cap at $J=0.92$, where it was possible to estimate the diameter of the tip-vortex: 10 mm to PIV field, and 9.6 mm to CFD field.

Figure 31 – Experimental (PIV) and numerical (CFD) vorticity field of Mod5 with elliptical boss cap ($J=0.92$)

Source: Left: Silva-Junior *et al.* (2019); Right: Author

It must be pointed out that CFD calculations have to present an adequate accuracy in order to provide a steadfast input to further hydroacoustic simulations. Hence, by comparing the experimental and numerical results of the propeller performance, it is reasonable to claim that CFD calculations present a satisfactory agreement with the experimental tests. Figure 32 shows the differences between the vorticity fields calculated numerically for each propeller configuration. As the divergence increases, the vorticity field in the hub vortex region presents a considerable variation, becoming more scattered if compared to Elliptical cap hub vortex.

Figure 32 – Numerical vorticity field of each boss cap geometry ($J=0.92$)

Source: Author

The pressure coefficient profile in the hub vortex region (Figure 33) and in the blades surface (Figure 34) also presents different behaviors as the boss cap geometry is changed. The pressure field in the hub vortex center is not so concentrated in the divergent caps if compared to the elliptical. It may suggest that the underwater noise emitted from these propeller configurations may present some difference. That is the point we will see in the foregoing section.

Figure 33 – Pressure Coefficient Contours ($J=0.92$)

Source: Author

The behavior of the pressure profile in the blade sections also showed some differences. From the blade root until the 0.3 blade section (Figure 34), the pressure distribution along the suction and the pressure sides of the blades was modified as the divergence increased. Nevertheless, to the region from the 0.3 blade section until the tip, no changes were observed in the pressure profile. These results are in accordance with the stated by Carlton (2007), where the author emphasized that the change in the divergence may affect the pressure distribution in the regions near the blade root, avoiding or retarding the occurrence of cavitation.

Figure 34 – Pressure coefficient distribution at 0.3 blade section ($J=0.92$)

Source: Author

4.3 Mod5 Propeller: Hydroacoustic Results

The computational domains in all numerical simulations were developed according to the scheme presented in Figure 19 (section 4.1). The porous surface, an open-ended cylinder surrounding the propeller and wake, was dimensioned in accordance with Lloyd *et al.* (2015), i.e., with a diameter of $1.25D_P$ and a length of $3D_P$, where D_P is the propeller diameter. Figure 35 shows the volume mesh generated to the Mod5 computational domain.

Figure 35 – Computational mesh of Mod5 simulations

Source: Author

In order to observe the stability of the numerical simulations by the behavior of pressure amplitudes at the nearfield, pressure pulse signals were collected at probe 2 (on the propeller plane) to each boss cap geometry, and presented in the figure 36 separated by surface and volume terms.

Figure 36 – FW-H Pressure pulse signals for each boss cap geometry at Probe 2

Source: Author

The amplitudes of the pressure signals presented a slight variation as the boss cap geometry changes. The amplitude of the volume term increased as the boss cap became more divergent. Nevertheless, as shown in figure 37, the surface terms showed an irregular behavior, decreasing the value to Div0 and Div3 caps, and increasing to Div1 and Div2 if comparing to elliptical boss cap.

The mechanism that may be inferred by analyzing figures 34 and 37 is that the divergent boss cap modifies the pressure distributions at the blade surface and, hence, the surface term of FW-H noise. On the other hand, the hub vortex is also altered by the divergence of the boss cap, in terms of its vorticity (Figure 32) and pressure distribution (Figure 33). Thus, the observed quadrupole noise term associated to the turbulence of the propeller wake has increased.

Figure 37 – Pressure amplitudes for surface and volume terms

Source: Author

On the other hand, the farfield noise results were obtained considering a probe located at a distance of 1m from the propeller plane and analyzed in the frequency domain, by means of 1/3 octave bands Sound Pressure Level (SPL), with the values represented in the central-band frequencies. Such methodology is often used in propeller acoustic analyses and allows the determination of the range of frequencies that the propeller operates with lower noise level. As shown in Figure 38, Div1 and Div2 geometries presented high noise level at most frequencies, especially in values above 500Hz, where the noise levels were almost 5dB higher than the Elliptical cap SPL. On the other hand, Div0 and Div3 caps produced quite similar SPL to the Elliptical one. One most highlight that to some frequencies, Div3 presented lower noise levels than the Elliptical geometry, notably at frequencies from 25Hz to 100 Hz and from 1000Hz to 2500Hz.

Figure 38 – 1/3 Octave bands SPL for each boss cap geometry

Source: Author

Concluding Remarks

The effect of the variation of boss cap geometry was addressed by means of the observation of propeller performance curves and the underwater noise calculated by using the Ffowcs Williams and Hawkings acoustic model, keeping the blades geometry and other propeller features invariable.

The propeller efficiency showed a general behavior of decreasing as the boss cap became more divergent. Div-2 geometry presented the lowest efficiency in both experimental and numerical results – an average of 10% decrease compared to the elliptical. On the other hand, a slightly increase in efficiency has occurred to Div-3 results. Even so, the numerical and experimental efficiency was almost 6% lower than the elliptical geometry.

As showed in section 4.3, the variation of the boss cap geometry induced a slight modification in the noise terms from the FW-H model. The quadrupole (volume) term amplitude expanded proportionally to the boss cap divergence. Nevertheless, the surface term presented an irregular behavior, decreasing to Div-0 and Div-3 geometries. The SPL of farfield results provided the frequencies in which the noise level varied, and interestingly, the lowest values to some frequency ranges were achieved by Div-3 geometry.

Hereupon, further studies about the flow pattern in the blades region and in the propeller wake are being developed (numerically and experimentally) for the purpose of explaining the mechanism of variation of propeller efficiency and radiated noise. The correlation between the propeller turbulent wake and the noise terms of FW-H model is a variable that must be studied profoundly and further experimental tests must be performed in order to validate the numerical results obtained in this study.

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