A REVIEW OF METHODS FOR HYDRO-ACOUSTIC ANALYSIS OF NON-CAVITATING MARINE PROPELLERS

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SUMMARY

Underwater Radiated Noise (URN) emanating from surface and underwater marine platforms has become a significant concern for all the Nations in view of the global requirement to minimise the increasing adverse impact on marine mammals and fishes and maintain ecological balance in the 'Silent' ocean environment. Ambient noise level in the sea, in 10 to 300 Hz frequency band, has increased by 20 to 30 dB due to shipping (Wittekind, 2009). Marine propeller (in non-cavitating and cavitating regime) is a potential contributor to the ships noise and a lot of scientific research has been undertaken and considerable progress has been achieved in estimating the hydro-acoustic performance of marine propellers. In light of this, the scope of this paper is to review and critically examine the various methods used for estimating the hydro-acoustic performance of marine propellers, particularly in the non-cavitating regime, over the past many years. This review paper brings out the details, applicability, merits and demerits of various methods, extrapolation laws to obtain full scale results, scientific conclusion of all the know-how on this subject and the scope of further research as perceived by the authors. This paper also presents a numerical methodology to estimate the noise radiated by a DTMB 4119 model propeller in the non-cavitating regime in open water condition. The hydrodynamic analysis of the propeller was performed using commercial CFD software STARCCM+, closure was achieved using standard k-ε turbulence model and hydro-acoustic predictions have been performed using FWH acoustic analogy. The results compare very well with the published literature.

NOMENCLATURE

A_C	mean cavitation area on the blades (m^2)
A_D	propeller disk area (m^2)
BPF	Blade Passage Frequency (Hz)
c	Speed of sound in water (m s^{-1})
CIS	Cavitation Inception Speed
DTMB	David Taylor Model Basin
DTT	Depressurised Towing Tank
FWHE	Ffowcs-Williams Hawkings Equation
D	Propeller diameter (m)
F	frequency (Hz)
H_{Dist}	distance of hydrophone from model propeller in cavitation tunnel (m)
L_S	Sound Pressure Level (dB ref 1µPa)
n	Propeller rpm (rpm)
P	Pressure (N m^{-2})
SPL	Sound Pressure Level
URN	Underwater Radiated Noise
V_{Tip}	blade tip velocity (m s^{-1})
V^i_{Tip}	rotation rate of start of tip vortex (RPM)
V	Ship speed (m s^{-1})
V_A	Speed of advance (m s^{-1})
Z	Number of propeller blades
υ	Kinematic viscosity (N s m^{-2})
ρ	Density of water (kg m^{-3})

1. INTRODUCTION

International shipping is the 'Life Blood of World Trade' with over 80% of global trade by volume and more than 70% of its value being carried on board ships. United Nations Conference on Trade and Development (UNCTAD) had forecasted world seaborne trade to increase by 2.8% in 2017 (Hoffman and Sirimanne, 2017). To meet these growing demands, bigger, faster and economical ships are required. In addition to commercial shipping activities, military surface and underwater platforms, which have now become an important arm in the Naval Defence, have also been increasing in numbers and potential capabilities. All these marine platforms generate considerable amount of 'unwanted sound' or 'noise' which contributes to the ambient noise in the sea particularly in the low frequency bands. URN is a persistent and pervasive pollutant and furthermore, due to its spatial and temporal variability, it represents a particular challenge for marine conservation, management and planning (McWhinnie *et al.*, 2017). Studies have reported an increase in low frequency ambient sea noise by an average rate of about 1/2 dB per year (Ross, 2005) which is attributable to the growing fleet of ships. Marine propeller is as an important contributor to the ships noise and radiates noise in the non-cavitating as well as cavitating regime. Quantitative and qualitative analysis of marine propeller radiated noise (hydro-acoustics) has been an area of interest and challenge for researchers since many years. Propeller noise, in particular non-cavitating noise, has emerged as a focus area in view of the changing designers perspective from measuring a propellers cavitation noise to estimating and delaying its cavitation inception speed (CIS) (R Kinns, Peake and Kessissoglou, 2015) thereby leading to propeller operations in the non-cavitation regime.

Considering this present scenario, the scope of this review paper is to critically examine the various methods available for estimating the hydro-acoustic performance of a marine propeller, particularly in the non-cavitating regime of operation, at various stages of its design. The scope of this paper would include introduction to the propeller noise as a growing nuisance, physics behind generation of non-cavitating and cavitating noise, detailed analysis of methods for estimating the hydro-acoustic performance including their merits and demerits, scaling laws for extrapolation to full scale values, authors conclusion and scope of further research in this field as perceived by the authors. This paper also presents the results of a numerical study undertaken to estimate the noise radiated by a DTMB 4119 model propeller in the non-cavitating regime in open water condition. The hydrodynamic analysis of the propeller was performed using commercial CFD software STARCCM+, closure was achieved using standard k-ε turbulence model and hydro-acoustic predictions have been performed using FWH acoustic analogy. The hydrodynamic and hydro-acoustic predictions compare very well with the published literature.

2. PROPELLER NOISE – A NUISANCE

The noise produced by a marine propeller, in terms of both its intensity and its spectral content, has been of considerable importance to surface and underwater platform designers for many years. In fact, it is the aerodynamic propeller noise, which was much earlier accepted as a nuisance for environment (as early as 1950s with the introduction of turbo-jet powered aircraft) and extensively researched and laid the basis for most of the principles and studies in hydro-acoustics. This marine noise has got serious implications on the source platform as well as the marine environment and these have been elaborated in the succeeding paragraphs.

2.1 CONTRIBUTION TO SHIPS NOISE

Propeller is one of the potential contributor to the ships Underwater Radiated Noise (URN) in the low frequency range (less than about 100 Hz) (Wittekind and Schuster, 2016). The ships URN is generated due to machinery noise (main propulsion and auxiliary), propeller noise (non-cavitating and cavitating noise) and hydrodynamic flow noise, each varying differently in strength with ship speed. As illustrated in Figure 1 below, at low speeds the dominant contributor is machinery noise which generally increases slowly in level with ship speed and at higher speeds the propeller noise dominates the URN, particularly for speeds greater than CIS. Flow noise may be a contributor to URN in the mid-speed range but is not a controlling source at any speed (ITTC, 2017).

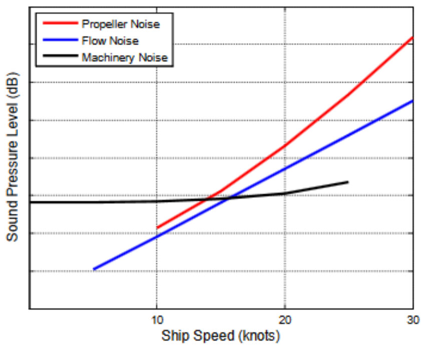

Figure 1: Illustration of various underwater ships noise contributors with ship speed (ITTC, 2017)

2.2 PROPELLER NOISE: PHYSICS INVOLVED

A marine propeller operates in a highly three-dimensional non-uniform wake field behind the ship's hull. As the propeller rotates to generate the required thrust, it is subjected to unsteady force, which leads to generation of noise. The propeller noise can be classified into cavitating and non-cavitating noise depending upon the regime of its operation. In view of the increased demand for high operating speeds and high propeller loads, avoiding cavitation had become practically impossible to achieve and hence, study of propeller cavitation and noise had been the focus area for ship designers and a lot of published literature on this is available in open domain. In the case of Merchant ships plying at low economical speeds (less than CIS), deeply submerged vehicles (where cavitation does not occur) and specialised ships (e.g. oceanographic research and survey ships), the study of non-cavitating noise is very important.

2.2 (a) Mechanism of Noise Generation

There are four principal mechanisms by which a propeller generate pressure waves in water and hence give rise to a noise signature. These are (Carlton, 2007):

a) The displacement of the water by the propeller blade profile – Thickness noise
b) The pressure difference between the suction and pressure surfaces of the propeller blade when they are rotating – Loading noise
c) The periodic fluctuation of the cavity volumes caused by operation of the blades in the variable wake field behind the vessel.
d) The sudden collapse process associated with the life of a cavitation bubble or vortex.

Clearly, the first two causes are associated with the propeller in either its cavitating or non-cavitating state, but are non-cavitating effects only. The latter two causes are cavitation-dependent phenomena, and therefore occur only when the propeller is experiencing cavitation.

The noise generating mechanisms for a propeller operating in non-cavitation regime consists of tonals (due to thickness effects, steady and fluctuating component of thrust and torque) and broadband noise (due to vortex shredding or laminar boundary layer instability noise, surface turbulence and inflow turbulence) (Richard and Munjal, 1980) (Parchen, 2000).

It has been widely stated and accepted that the thickness and loading noise are radiated as monopole and dipole source respectively (Jenkins, 1988). But Ianniello in 2014 during numerical studies observed that the thickness component acts as a monopole just in the propeller disk plane, but behaves in a different manner out from it (Ianniello, 2014).

2.2 (b) Classification of Propeller Noise

On the basis of regime of operation, propeller noise can be divided into non-cavitation and cavitation noise. In addition, propeller noise is divided into three sources of noises due to propeller loading, thickness and turbulence. At lower Mach numbers, noise due to turbulence is generally ignored and total noise is calculated by propeller loading and thickness (Sezen, Dogrul and Bal, 2016). As per the frequency distribution/spectral representation, propeller noise is divided into discrete frequency noise/tonals/rotation noise (all types of noise emitted at the shaft and/or blade rate frequencies and their harmonics) and broadband noise (all other type of noise that arise due to the boundary layer around the blades).

2.3 EFFECT OF UNIFORM AND NON-UNIFORM FLOW

Propeller noise usually includes a series of periodic parts or tones at blade rate and its multiples. These periodic unsteady forces impose discrete tonal noise at BPF together with a broadband noise spectrum caused by turbulence interaction with blade and vortex shedding at the trailing edge and the tips. A small-scale part of turbulent eddies in the wake cause unsteady blade forces. Besides, the boundary layer separation and blades vortex shedding also causes fluctuating forces. Shedding vortex will happen at the area of trailing edge and tip of rotating blades. Induced pressure pulses by the propeller may be considered as one of the important sources in the SPL. Moreover, whereas the propeller operates in the heavy-load condition, the boundary layer around the blade may separate at the stagnation point in the suction side. The flow separation and vortex shedding are completely unsteady events which will impose oscillating pressure on the propeller. So, the underwater propeller will produce noise, even in uniform inflow (Pan and Zhang, 2010). In the non-uniform flow the physics becomes even more complicated.

2.4 IMPACT ON MARINE LIFE

The propagation of ships radiated noise increase the ambient noise in the ocean and creates short and long term adverse effects on the marine life. It has been observed that noise in the low frequency range of 10 Hz to 1 kHz has the biggest impact on the marine biodiversity (Bertschneider *et al.*, 2014). An increase in ambient noise by 20 dB, as is common from natural sources, would typically lead to a decrease in detection range of about a factor of 10 for marine animals (Cato, 2014). In view of the increasing global concern on minimising the harmful effects of ambient noise on the marine organisms, various international organisations/authorities have undertaken studies to assess the potential adverse impact of noise on marine life (Southall, 2004) (Gotz *et al.*, 2009) (Cato, 2014) (Audoly *et al.*, 2016) which has motivated statutory/regulatory bodies to develop implementable technologies (IMO, 2009), regulations & design procedures (Leaper *et al.*, 2008), (Jokat, 2009), (Bosschers and Wijngaarden, 2010), (Renilson, Leaper and Boisseau, 2013), (Park, 2014), (Bertschneider *et al.*, 2014), (IMO-Noise Working Group, 2014), (AQUO Consortium, 2015), (Fradelos, 2016), (Prins *et al.*, 2016) aimed towards designing quieter ships.

2.5 PROBLEM FOR MERCHANT AND WARSHIPS

This hydrodynamic induced noise and vibrations has been an issue of relevance for both the Navy and civil naval engineering. In case of Naval platforms, noise governs the detectability, survivability and operational performance while for merchant ships, it is important from on-board habitability and safety considerations. Also, for naval ships, often sensors on board such as sonar or acoustic positioning systems require a low noise level of the vessel itself in order to have an adequate signal to noise ratio. With considerable progress achieved in controlling Structure Borne Noise (SBN) control of non-cavitation propeller noise of Naval vessels has become very important (Parchen, 2000). Special vessels like oceanographic research vessel require low radiated noise for minimum interference with the acoustic sensors.

3. CAVITATION AND NON-CAVITATION NOISE SPECTRUMS

The description of a general spectrum of a cavitating and non-cavitating propeller is discussed in succeeding paragraphs.

3.1 CAVITATING PROPELLER NOISE SPECTRUM

Cavitating noise from a propeller can be due to various types of cavitation including in the order of importance sheet cavitation on propeller blades, vortex cavitation from propeller tips and propeller hub and bubble cavitation (Hynna, 1986). Figure 2 below shows a general noise spectrum for a cavitating propeller. It has been observed that sheet cavitation radiates sound from 5Hz to more than 10 kHz. The low frequency noise (region I and II) is caused by the fluctuations of the sheet cavitation volumes possibly represented by a large bubble that acts as an acoustic monopole. On the other hand, high-frequency noise (region III and IV) is caused by sheet cavity collapse or by shock wave generation (Seol, Suh and Lee, 2005).

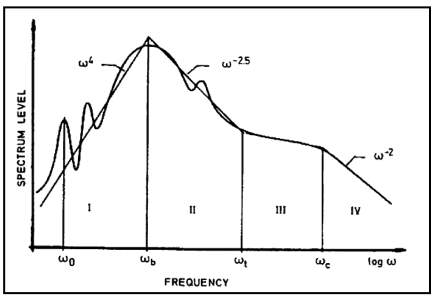

Figure 2. General noise spectrum of a Cavitating marine propeller

3.2 NON-CAVITATING PROPELLER NOISE SPECTRUM

The propeller without cavitation is a low frequency source and its contribution at 2000 Hz or higher is very low compared to other ship noise sources (Salinas, Rizzuto and Audoly, 2015). In the non-cavitating propeller spectrum, it is possible to identify distinct tones and broadband noise (Audoly, Rizzuto and Audoly, 2015). Figure 3 below shows a general noise spectrum for a non-cavitating propeller in 1/3 octave bands. It shows that at high frequencies (f > 1 kHz), turbulent trailing edge noise and inflow turbulence noise are approximately equally important. At very low frequencies on the other hand (f < 31.5 Hz), it appears that the blade-rate tones dominate the spectrum. Between both regimes inflow turbulence noise dominates. In this simulation the author assumed that vortex shedding does not contribute significantly to the radiated noise. However, when it does it often manifests itself in the frequency region just below the region where the trailing edge noise obtains its maximum (Parchen, 2000).

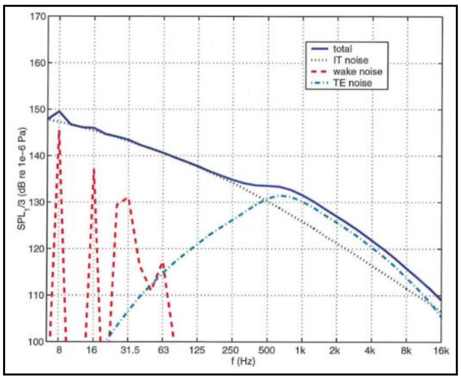

Figure 3. General noise spectrum of a Non-cavitating marine propeller (Parchen, 2000)

3.3 INCREASE IN NOISE FROM NON-CAVITATING TO CAVITATING REGIME

Several numerical and experimental studies have been conducted to estimate the increase in the noise from non-cavitation to cavitation regime. As per numerical analysis by SSPA, Sweden (using FLUENT), a narrow sheet cavitation on blades increased broadband noise by 15-20dB and tonal noise by 5-10dB when compared with the non-cavitating case, agreeing well with the sea trial observation where an increase of 20dB broadband noise was attributed to the developed cavitation on blades (Hallander and Da-Quin, 2013). Further, a numerical study done in 2015 showed that the difference between overall SPL in non-cavitation and cavitation inception conditions is approximately in the range of 5 to 20dB for each frequency. This range of difference is related to the increasing of propeller rotational speed from 900 to 1400 rpm. In comparison overall SPLs under cavitation development condition with both other conditions, the difference between the overall SPL is in the range of 10 to 30dB (Bagheri, Mehdigholi, *et al.*, 2015). Similar numerical studies have also been undertaken under the EU funded AQUP project (Salinas, Rizzuto and Audoly, 2015). Experimental measurements done in the DWB at MARIN for a cavitating and a non-cavitating propeller, at a condition just beyond CIS, showed that for frequencies above 1 kHz the noise levels for the cavitating propeller were more than 10dB above the noise levels for the non-cavitating propeller (Bosschers *et al.*, 2013).

4. FACTORS AFFECTING PROPELLER NOISE

4.1 WAKE

Propellers usually operate in the ship's stern, where the inflow wake generates periodic and fluctuating pressure, due to which, as a result, noise and vibration occurs. It is very important to study the cavitation behaviour for a propeller in the desired wake field since wake field has got very strong influence on cavitation noise (Sharma, Mani and Arakeri, 1990) as well as the thrust generated (Spence and Fischer, 2014). In the specific case of a 4 bladed propeller, it was found that by inserting the wake inflow, sound strength increased by about 10% (Ghassemi, 2016). The effect of wake on the propeller open water characteristics has also been examined (Ghassemi, 2009) (Hayati, Hashemi and Shams, 2012).

4.2 MATERIAL

Composite marine propeller, due to its potential advantage of reduced weight and noise reduction, has increasingly attracted the attention of researchers. It has been shown that composite propeller typically has a weight only one-third of the weight of conventional Nickel-Aluminum Bronze (NAB), reduces the propeller noise typically by 5 dB (by virtue of being thinner at the tip than metal propellers) and reduces the fuel consumption by up to 15% by using the hydro-elasticity to optimize propulsive efficiency (Marsh, 2004). Estimation of the hydro-elastic, cavitation, radiated noise and efficiency of composite propellers has been undertaken in the past (Paik *et al.*, 2013), (Taketani *et al.*, 2013), (Hong *et al.*, 2017).

4.3 GEOMETRICAL PARAMETERS

The geometrical parameters of a propeller play an important role in the noise generation and propagation. It has been observed that skew is very effective in reducing the pressure amplitudes and thus noise over a very large frequency range (Burnside, Kana and Reed, 1979), (Hynna, 1986) and this is possible by reducing the summation over the span (Jenkins, 1988). Skew affects the propeller open water characteristics (Ghassemi, 2009) but it also appears to be one of the few parameters that can be tuned in order to reduce the radiated noise (Ayris, 2016) without affecting the propeller thrust or propeller efficiency (Parchen, 2000). Geometrical parameters blade design, skew, pitch distributions, rake, warp, etc had already been identified as the tools available to a designer to reduce noise (Spence and Fischer, 2014). A study was undertaken to numerically estimate the effect of rake and skew on the hydrodynamic performance and noise levels of a propeller (Gorji, Ghassemi and Mohamadi, 2017). Ayris in 2016 observed that rake had no notable impact on sheet cavitation performance (Ayris, 2016) but it has a strong influence on open water performance (Hayati, Hashemi and Shams, 2012).

4.4 INFLUENCE OF FREE SURFACE

The free surface acts as a pressure release boundary and it has been observed that levels at low frequencies (~ 3-80 Hz (Baudin and Mumm, 2015)) can be reduced markedly by the free surface of the sea, depending on propeller immersion (R. Kinns, Peake and Kessissoglou, 2015). This is called Lloyds Mirror Effect and this influence (reduction) increase with increasing frequency (ITTC, 2017). The Lloyd's Mirror effect (LME) is produced by interference between the direct-path and the sea surface phase-reversed reflection of a sound as observed at a receiver resulting in a frequency-dependent interference pattern.

5. STUDY OF PROPELLER HYDRO-ACOUSTIC PERFORMANCE

Investigation of marine propeller noise has been a challenging area of study for researchers since many years. A total of 309 documents are indexed in Scopus (as on 11 Jun 18) with the keywords "Marine Propeller Noise" (period 2003-2018) but documents indexed, with the key words, "Marine Propeller Noise AND non-cavitating" are only 31. Thus as compared with the extensive amount of literatures on cavitation noise of propellers, works concerning the non-cavitation noise are hard to find (Rama, Bangaru and Suryananarayana, 2015). But an increasing trend in research in this field is observed (as shown in Figure 4 below). In order to make this scientific review more comprehensive, in addition to papers from Scopus indexed journals, papers/documents obtained from open source literature have also been included. The various methods of estimating propeller noise have been discussed in detail below.

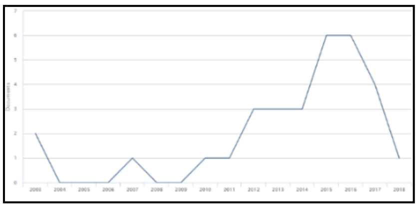

Figure 4. Research documents indexed in Scopus for the keywords "Marine Propeller Noise" AND "Non-cavitating" (ref: www.scopus.com)

6. EMPIRICAL FORMULAE

Empirical formulae are very useful in estimating the propeller noise at the initial design stage when the detailed information about the propeller geometry and hull parameters is not known. Farassat in 1981 discussed the linear acoustic formulas for calculating the rotating blade noise of helicopter rotors and propellers (Farassat, 1981). The formula proposed by Fraser (Fraser, 1986) is indicated as Equation 1 below. This formula gives SPL in dB (ref 1μPa) but does not differentiate between the non-cavitating noise and cavitating noise.

$$L_S = 10\log[\frac{D^6\,(60n)^6\,Z}{4}] - 6\; for\; f \le 100\; Hz$$
$$L_S = 10\log[\frac{D^6\,(60n)^6\,Z}{4}] + 34 - 20logf\; for\; f \ge 100\; Hz \qquad (1)$$

Gesret in 2000 (Gesret, 2000) proposed an empirical formula (Equation 2) based on full-scale measurements to estimate noise of a cavitating propeller. In his model, at full-scale, pressure fluctuation measurements were performed on the hull above the propeller during sea trials and then a monopole was used to modelise propeller source.

$$L_P\,(dB) \le 63 + 10.\log_{10}(\frac{Z.D^4.n^3}{f^2}) \qquad (2)$$

7. SEMI-EMPIRICAL METHODS

Semi-empirical methods to estimate the source strength of propellers were prescribed by Janssen and Buiten in 1973, Ross in 1976, Brown in 1976 and Nilsson in 1978 without taking into consideration the propeller geometry, wake distribution and ship structure above the propeller and hence these methods may not give accurate prediction of propeller noise for a new design ship (Hynna, 1986). De Bruijn et. al in 1986 presented a method to predict the propellers broadband cavitation noise (above the lowest blade rate frequencies, namely between 30 and 500 Hz), using the concept of an equivalent monopole to take into account the effect of cavitation volume variations (Salio, 2015). Gesret in 2000 proposed a semi empirical approach based on a statistical model to estimate noise of a cavitating propeller. This model used volume velocities calculated from pressure measurements and hydrodynamic as well as geometric characteristics of a given propeller in order to calculate the volume velocity of this propeller (Gesret, 2000). Browns semi empirical formula for estimating the broadband cavitation noise is shown in Equation 3 below (Ekinci, Celik and Guner, 2010).

$$L_S(dB) = 163 + 10\log\left[\frac{Z.D^4\,n^3}{f^2}\right] + 10\log\left[40.\frac{A_C}{A_D}\right] \qquad (3)$$

Takinaci and Taralp (2013) developed a semi empirical formula (similar to Equation 1) for estimating cavitation noise (including sheet and tip vortex cavitation) shown in Equation 4 below (Takinaci and Taralp, 2013).

$$L_S(dB) = 163 + 10\log\left[\frac{Z.D^4\,n^3}{f^2}\right] + 10\log\left[40.\frac{A_C}{A_D}\right] + K_{Tip}\log\left[\frac{V_{Tip}}{V_{Tip}^i}\right] + 10\log[H_{Dist}] \quad (4)$$

The value of the coefficient K_{Tip} is normally taken to be 60 but for deeply submerged propellers a value of 80 is suggested. Equation (3) is valid for f_p<10 kHz where the center frequency f_p lies at the peak of the broadband noise spectrum.

Takinaci and Taralap in 2013 developed an empirical prediction model of broadband noise for marine propellers. The model is composed of two components: firstly, the empirical prediction of the frequency domain broadband noise, and secondly, modulation of the noise in the time domain (Takinaci and Taralp, 2013).

Kim et. al. in 2016 developed a semi-empirical formula (Equation 4) for the tip vortex cavitation formation noise based on experimental tests of 9 model propellers (Kim *et al.*, 2016).

$$L_S\,(TVC_{contribution}) = -1.5\,(\log(St) + 0.7)^2 + 31 \quad (4)$$

Where:

$$St = f_{vs}.\frac{D_{core}}{\sqrt{(2\pi n_m)^2 + (V_m)^2}}$$
$$D_{core} = \alpha.\delta_{thick}$$
$$\delta_{thick} = 0.154.c.Re^{-1/7}$$

Taking this into account the D Ross formula was modified by Kim et. al. as Equation 5 (Kim *et al.*, 2016):

$$L_S\,(TVC_{contribution}) = 10logZ + 60logD + 60logn - 20logf - 1.5\,(\log(St) + 0.7)^2 + 31 \quad (5)$$

8. EXPERIMENTAL METHODS

Model-scale experiments involving noise measurements of propellers are usually performed using one or more hydrophones mounted in the test facility in which the propeller is tested. This facility could be a Cavitation Tunnels or a Towing Tank. For measuring cavitation noise, the model test conditions should satisfy the same propeller working conditions (namely: Propeller loading K_T and Cavitation number σ) as predicted for the full-scale ship. An important concern in these facilities is the intrusive influence of the restricted boundaries which needs to be 'filted away' from the measured results (Tani *et al.*, 2016). The specific requirements with respect to similarity will be discussed within the scope of the respective facilities.

8.1 CAVITATION TUNNEL (CT)

The Cavitation Tunnels are the most preferred facilities for these cavitation tests and noise measurements. In order to get accurate measurements it is very important to set the flow around the propeller similarly to how it is set in a full-scale ship. In order to measure noise, an acoustic chamber (box made of steel plates stiffened by the frames in order to minimize the vibration and inner walls coated with damping materials to prevent noise reflections) equipped with one/more hydrophones is generally fitted below the test section.

8.1 (a) Types of cavitation tunnels

Big size tunnels that permit to locate inside a full hull model and in the same time model correctly the cavitation are a better choice for these tests. Examples of these big size tunnels are the Grand Tunnel Hydrodynamique (G.T.H.), France (Frechou et al. in 2001) and INSEAN Large Circulating Channel (Felli in 2011), SSPA large cavitation tunnel and HYKAT cavitation tunnel. Nevertheless, the small and medium size Cavitation Tunnels (e.g. UNIGE cavitation tunnel, NSTL cavitation tunnel) are also utilised for acoustic measurements taking into account the propeller hull interaction only as a non-uniform flow (wake) upstream of the propeller using wake screens or dummy models (Zhu et al in 1978, Yuasa et al.in 1986, Ukon et al. in 1987, Bark et al in 1987, Sharma et al. in 1990 (Sharma, Mani and Arakeri, 1990), Wills in 1990, Sevik in 1996, Sasajima et al in 1986 (Sasajima, Nakamura and Oshima, 1986), Atlar et al. in 2001 in Emerson Cavitation Tunnel (ECT), Wang et al in 2006 in ECT, Park et al. in 2009 (C. Park *et al.*, 2009), Arazgaldi et al in 2009 (Arazgaldi, Hajilouy and Farhanieh, 2009), Bertetta et al. in 2011), (Kowalczyk and Kaiser, 2013), (Rama, Bangaru and Suryananarayana, 2015).

8.1 (b) Test Methodology

The procedure and guidelines for test set-up, test conditions, instrumentation, background noise measurement, noise data acquisition and processing and uncertainty analysis for cavitation noise measurement has been laid down by ITTC (*ITTC – Recommended Procedures and Guidelines - Model-Scale Propeller Cavitation Noise Measurements*, 2017). These could be followed for measuring non-cavitation noise also.

8.1 (c) Similarity laws for hydro-acoustics tests

The similarity laws for performing hydro-acoustic tests on a ship model with propeller in a cavitation tunnel were discussed by Frechou et al in 2001 (Fréchou *et al.*, 2001).

8.1 (d) Advantages

- A large size CT can accommodate a scaled hull model also. Hence, the hull-propeller interaction can also be accurately modelled.

- Permits easy visualisation of cavitation phenomenon.
- Both cavitation and non-cavitation noise can be measured (Kowalczyk and Kaiser, 2013).

8.1 (e) Disadvantages

- Absence of free surface that is known to contribute, although less than the cavitation, to the whole acoustic field as well the reverberation of tunnel's walls (Fréchou *et al.*, 2001), (Bagheri, Mehdigholi, *et al.*, 2015).
- Need to establish the correlation factor for reverberation in the CT.

8.2 DEPRESSURISED TOWING TANK (DTT)

Non-cavitating and cavitating propeller noise can be measured in a DTT. Two such facilities known in open domain are DTT facility at China Ship Scientific Research Centre (CSSRC), China and DTT (renamed to DWB in 2012) facility at MARIN, Netherland. The 240m*18m*8m DWB at MARIN has been extensively used for cavitation and radiated noise measurements since 1972. In order to reduce carriage noise rubber wheels were mounted on the towing carriage and the vibrations of the construction for power and data transfer between the remotely operated towing carriage and the basin were mitigated (Bosschers *et al.*, 2013).

8.2 (a) Test Methodology

The test methodology is similar to that of a cavitation tunnel (as mentioned at Para 8.1(b)). Since this facility has a free surface, the influence of free surface on the reverberation and the noise measurements is required to be assessed and, if necessary, corrected for with an acoustic calibration test (*ITTC – Recommended Procedures and Guidelines - Model-Scale Propeller Cavitation Noise Measurements*, 2017).

8.2 (b) Similarity laws for hydro-acoustics tests

In addition to Propeller loading K_T and Cavitation number σ similarity (as in case of a CT), Froude number scaling is required due to the presence of free surface.

8.2 (c) Advantages

- Hull propeller interaction can be accurately modelled by placing the scaled hull model.
- The ambient pressure in the basin can be lowered from atmospheric pressure to a pressure as low as about 30 mbar in order to perform cavitation tests. Hence, both non-cavitating and cavitating conditions can be tested.
- Due to the presence of free surface, the influence of the ship wave pattern is automatically and properly captured if the tests are performed at Froude number similarity. The wave pattern can have a significant influence on both the velocity and pressure field of the ship wake field in which the propeller operates.
- Performing the tests at Froude number similarity also implies that the variation of the hydrostatic pressure in the propeller plane is correctly modelled, although this aspect becomes only important when cavitation is present over a large range of blade positions.

8.2 (d) Disadvantages

- The reflection by the basin walls (reverberation) and effect of free surface on the measured noise needs to be assessed and taken into consideration in order to correctly interpret the results. Impulse response measurements conducted at DWB, MARIN established the measured cavitation noise can directly be interpreted as the far field radiated noise for a (model scale) frequency greater than 1 kHz, but below 1 kHz the reflections of the basin somewhat influence the noise measurements and the measured values cannot be not easily converted into full scale values (Bosschers *et al.*, 2013).
- The measured noise levels could be influenced by the background noise of the test set-up and the facility. Hence acoustic characterisation of the facility is required and background noise is to be measured. Corrections to the measured SPLs depending on the difference between the measured and background noise are made (*ITTC – Recommended Procedures and Guidelines - Model-Scale Propeller Cavitation Noise Measurements*, 2017). The measured values should be discarded if this difference is less than 3dBs.

8.3 ATMOSPHERIC TOWING TANK

Limited attempts have been made to measure the propeller noise in atmospheric tank due to the incomplete modelling that leads normally to non-cavitating condition. In addition, the towing carriage is a strong source of background noise. The choice of such facility is justified by the need to produce non-cavitating acoustic propeller data for CFD validation purpose and can be of interest when the propeller is generally non-cavitating, contributing to distinguish the non-cavitating part of the noise produced by a cavitating propeller. The interaction with the hull is also better modelled in towing tank (Haimov *et al.*, 2016). Effect of free surface on the reverberation and the noise measurements should also be taken into consideration, as in the case of DTT.

8.3 (a) Test Methodology

The testing methodology is same as that followed for DTT (mentioned at Para 8.2(a)).

8.3 (b) Similarity laws for hydro-acoustics tests

These are also same as that for a DTT (mentioned at Para 8.2(b)).

8.3 (c) Advantages

DTT is a cost and maintenance intensive facility which is not easily accessible to all. The atmospheric towing tank can emerge as an economical option offering similar advantages as a DTT (in terms of hull modelling, presence of free surface and Froude number similarity).

8.3 (d) Disadvantages

In addition to the issues brought out in section 8.2(d), one major disadvantage of this facility is that generating only non-cavitating conditions is possible.

9. FULL SCALE MEASUREMENTS

Full scale measurements could be undertaken in the acoustic ranges or in open sea at a specified location. But, it is generally, difficult to accurately isolate the propeller noise due to the presence of other noise sources. Certain full-scale noise measurements were done by SSPA in 1986 on two propellers designed for an oceanographic research ship in order to validate the results of the cavitation tunnel (Sasajima, Nakamura and Oshima, 1986). Conducting these tests is very costly and challenging and is subject to the availability of a vessel.

10. NUMERICAL METHODS

Numerical methods have become popular, robust and convenient with the advent of fast and high capacity computing and are relatively less cost and time intensive when compared to experiments. But at the same time, it is very important to correctly model the underlying physics and understand the idealisations and assumptions to obtain meaningful predictions. These numerical methods need to be validated with experimental results in order to verify the modelling and analysis procedure. Numerical methods for hydro-acoustic analysis of marine propellers fall into two categories: namely the direct methods and indirect/hybrid methods.

10.1 DIRECT METHODS

In a direct method, the noise is determined together with the flow field, and the numerical scheme is adopted to properly achieve the required assessment of the different flow scales (turbulence and acoustic wave scales). Hence, the noise sources (sound generation) and sound propagation is solved simultaneously in a single numerical method. In principle, a direct numerical simulation (DNS) which resolves all flow scales, would provide the most complete picture, but the currently available computational resources do not allow to perform such a simulation, especially at the Reynolds numbers of full-scale ships. Then, an alternative is a large-eddy simulation (LES), where a filtering process is applied so that only the dynamically important flow scales are resolved, and the smallest ones are modelled, or a Reynolds averaged Navier-Stokes method (RANS), where only the largest flow structures are taken into account through an averaging procedure, or even a suitable combination of the different approaches, as the detached-eddy simulation (DES). In solving these methods we require accurate and robust spatial discretization, optimized time discretization and non-reflective boundary treatments. Since these methods solve the full compressible unsteady Navier-Stokes equations, they spend lots of time and energy in practical configurations due to the computational costs. Owing to these considerations, only a few researchers have employed DES or LES to predict underwater cavitation-induced noise radiated from propeller and NACA section (Kim *et al.*, 2016).

10.2 INDIRECT / HYBRID METHODS / TRANSPORT TECHNIQUES

The indirect methods are carried out in two steps: firstly we resolve the flow field variables (hydrodynamic analysis) and capture the acoustic sources (in terms of physical quantities responsible for production of sound) and then propagate the sources to an observer location (hydro-acoustic analysis). The acoustic sources could be propagated employing an acoustic analogy (e.g. FWH equation) or computational methods (e.g. Acoustic wave model). Hybrid methods have the advantage that they allow acoustic evaluations at receiver locations where the Navier-Stokes pressure field does not permit accurate predictions (due to grid coarseness or close proximity to boundaries), or outside of the computational domain itself. Among these indirect approaches, FWH acoustic analogy has been used more widely to predict underwater noise radiated from cavitating and non-cavitating propellers (Ianniello and Bernardis, 2015) (Noughabi, Bayati and Tadjfar, 2017). A review of techniques employed to carry out these two distinct steps is carried out in succeeding paragraphs.

11. STEP 1: HYDRODYNAMIC ANALYSIS

The numerical techniques to carry out hydrodynamic analysis have matured from conventional methods like lifting surface theory (Kim, Ki-Han and Kobayashi, 1985) (Breslin and Andersen, 1994) and potential based panel methods (Lee, 1987) (Seol *et al.*, 2002) (Seol, Suh and Lee, 2005) (Ryu *et al.*, 2015) to modern advanced techniques like Computational Fluid Dynamics (CFD) (Kulczyk, Skraburski and Zawiślak, 2007), (Prakash M.N. and Nath, 2012), (Morgut, 2012), (Boumediene and Belhenniche, 2016), (Gorji, Ghassemi and Mohamadi, 2017). A detailed review of all these methods has already been attempted by many researchers.

12. STEP 2: HYDROACOUTIC ANALYSIS

12.1 DIFFERENTIAL METHODS – BERNOULLI EQUATION BASED APPROACH (UNIFIED HYDRODYNAMIC AND HYDOACOUSTIC FORMULATION)

This approach is considered the most novel method developed for hydro-acoustic analysis of marine propellers wherein theoretical investigations of the noise of cavitating propellers are typically performed under inviscid–flow assumptions by determining the pressure from the potential velocity field by means of the Bernoulli theorem. The first step of this approach is determination of velocity potential on the body, by a boundary integral equation approach. Then, the integral representation for the potential yields the potential distribution in the field and the Bernoulli theorem gives the corresponding acoustic pressure. The differential formulation, boundary integral solution and mathematical details of this method have been discussed in detail by Testa in his PhD thesis in 2008 (Testa, 2008).

12.2 INTEGRAL METHODS: FWH ANALOGY

In this method, we evaluate sound sources on the control surfaces and then propagate the acoustic information to the far-field using analytic solutions. The acoustic analogy proposed by Ffowcs-Williams Hawkings (impermeable and permeable formulations), originally developed for aero-acoustic analysis in 1969, is now being extensively employed in hydro-acoustics also. As reported by Testa et al in 2005 (Testa, Salvatore and Ianniello, 2005), the earliest attempt to use this analogy for estimating cavitating marine propeller noise was made by Salvatore et al in 2003. The FWHE has proved to be the most suitable and convenient numerical approach for hydroacoustic analysis of a marine propeller (Salvatore and Ianniello, 2003) and more robust and efficient than Bernoulli method (Testa, 2008). This method has been extensively used by several researchers (Sezen, Dogrul and Bal, 2016), (Noughabi, Bayati and Tadjfar, 2017).

12.2 (a) Ffowcs-Williams Hawkings Equation (FWHE)

FWH acoustic analogy (Williams *et al.*, 1969) extends the Lighthills equation to predict noise originating from the presence of a turbulent flow (Williams *et al.*, 1969). The FWH theory includes surface source terms (thickness or monopole sources and loading or dipole sources) in addition to the quadrapole-like sources introduced by Caridi (2007). This equation is presented as follows (FWH 1969):

$$\frac{\partial^2 p'}{c_0^2 \partial t^2} - \nabla^2 p' = \frac{\partial^2}{\partial x_i x_j}\left[T_{ij} H(f)\right] - \frac{\partial}{\partial x_i}\Big(\left[P_{ij} n_j + \rho u_i (u_n - v_n)\right]\delta(f)\Big) + \frac{\partial}{\partial t}\left(\left[\rho_0 v_n + \rho(u_n - v_n)\right]\right)\delta(f) \tag{6}$$

The terms at the RHS of Equation (6) are named quadrapole, dipole and monopole sources respectively, p' is the source pressure level at the far-field, ($p'=p-p_0$)), c_0 is the far-field sound speed and T_{ij} is the Lighthill stress tensor (defined by Equation 7).

$$T_{ij} = \rho u_i u_j + \delta_{ij}(p - \rho c_0^2) + \tau_{ij} \tag{7}$$

The first term on the RHS of Equation. (7) is the turbulence velocity fluctuations (Reynolds stresses), the second term is due to change in pressure and density and the third term is due to the shear stress tensor.

Also, f is a function defined based on surface reference system where setting f=0 introduces a surface that embeds the external flow effect (f>0), and H(f) and $\delta(f)$ are Heaviside and Dirac delta functions, respectively. It has been observed that the noise signature predicted by the FWH approach does not exhibit a strong sensitivity to the shape of the wake used in the hydrodynamic analysis, when compared with Bernoulli based approach (Testa, 2008).

12.2 (b) Farassat's Solution to FWHE

There are various ways to evaluate the FWHE. Farassat formulations 1 and 1A (developed in 1975) are the solutions of the FWHE with surface sources only when the surface moves at subsonic speed. Formulation 1A has been recommended for undertaking acoustic predictions for helicopter rotors and marine propellers (Farassat, 2007). Formulation 1A is a simple linear time-domain formulation that can predict arbitrary shaped object in motion without the numerical differentiation of the observer time.

In the Farassat formulation, the pressure field is defined as (Equation 8):

$$P'(\vec{x}, t) = P'_T(\vec{x}, t) + P'_L(\vec{x}, t) \tag{8}$$

where P' is the acoustic pressure, P'_T and P'_L describe the acoustic pressure field resulting from thickness and loading, corresponding to the monopole and the dipole sources. The thickness and loading noise components are defined by Equation 9 & 10.

$$4\pi p'_T(x,t) = \int_{f=0}^{\cdot} \left[\frac{\rho_0 . \dot{v_n}}{r(1-M_r)^2} + \frac{\rho_0 v_n \hat{r_i} \dot{M_i}}{r(1-M_r)^3}\right]_{ret} dS + \int_{f=0}^{\cdot} \left[\frac{\rho_0 . c . v_n (M_r - M^2)}{r^2 (1-M_r)^3}\right]_{ret} dS \tag{9}$$

$$4\pi p'_L(x,t) = \int_{f=0}^{\cdot} \left[\frac{\dot{p}.\cos\theta}{c.r(1-M_r)^2} + \frac{\hat{r_i} \dot{M_i}.p.\cos\theta}{c.r(1-M_r)^3}\right]_{ret} dS + \int_{f=0}^{\cdot} \left[\frac{p.(\cos\theta - M_i n_i)}{r^2(1-M_r)^2} + \frac{(M_r - M^2).p.\cos\theta}{r^2.(1-M_r)^3}\right]_{ret} dS \tag{10}$$

Various researchers have solved the FWHE using Farassat 1A formulation (Bagheri, Seif, *et al.*, 2015).

12.2 (c) Impermeable FWH

In this form, sound sources are evaluated on the boundary of the object using surface integration. The volume sources are not taken into account. This method is generally used when the observer is in far-field, volume sources are negligible and source regions are acoustically compact. Typical applications are estimation of fan and propeller noise.

12.2 (d) Permeable FWH

In 1997, a new solving approach named porous or permeable formulation was developed by Di Francescantonio which played a leading role among the FWH-based integral resolution forms (Di Francescantonio, 1997). In this form, it is assumed that all sources are contained within control surface and hence, sound sources evaluated on the control surface. Scattering inside the control surface is also accounted for. This is generally used when the observer is in far-field and volume sources are non-negligible.

When a porous formulation is used, the non-linear term for sources located within the control surface are accounted for via the thickness and loading contributions. This also implies that for such a formulation the monopole and dipole contributions lose their physical meaning (Ianniello et al. 2012).

The porous method presented by Ffowcs-Williams & Hawkings (1969) allows one to account for the non-linear terms without the need for volume integration, offers short computational times, and does not require the flow solver to be modified. Contrary to a typical (impermeable) acoustic analogy where the noise terms are evaluated on the surface of the body, it does so on a permeable surface surrounding the object and flow features contributing to the radiated noise, such as cavities or wake.

From a practical point of view, however, the porous formulation is certainly the most suitable and effective way to solve the FWH equation. The difficulty lies in obtaining accurate input data on a surface far from the body-source and may require a high level computational capability in solving the corresponding hydrodynamic or aerodynamic problem (Ianniello and Bernardis, 2015).

12.2 (e) FWH method versus Bernoulli based methodology

A comparison between a simple Bernoulli-based methodology (rather usual for naval applications) and the FWH equation highlighting the theoretical and numerical consequences of the incompressibility assumption and the effects of the wake modelling showed that the acoustic analogy is more robust for noise prediction: it represents a physically consistent approach and exhibits many computational advantages with respect to the Bernoulli-based method (Testa *et al.*, 2008).

12.2 (f) Quadruple Noise Sources: Significance

The contribution from the quadrupole noise term (accounting for all possible nonlinearities taking place in the flow) to the total noise is generally neglected in all the hydro-acoustic studies because the propeller rpm is much less than the speed of sound in water (hence lesser acoustic efficiency) (Salvatore and Ianniello, 2003), (Testa *et al.*, 2008), (Salvatore, Testa and Greco, 2009), (Sezen, Dogrul and Bal, 2016). The contribution from the quadrupole term becomes important for several reasons. First, it fully describes the acoustic effect of the potential wake. In order to compare the FWHE and the Bernoulli approach exactly, non-linear terms should be included in both formulations. The non-linearities in both methods are not equivalent, that is, some non-linear effects described by the Lighthill tensor in the FWHE are not accounted for by the non-linear terms in the Bernoulli method. Furthermore, the inclusion of the quadrupole term would account for acoustic effects related to cavitating phenomena occurring in the flow– field, like cavitating tip vortices and hub vortices, and bubble cavitation (Testa, 2008). This common belief was challenged by Ianniello himself in 2014, when he stated that propeller noise is a nonlinear problem and hence the nonlinear flow noise sources play a dominant role independently of the low rotational speed of the blade (Ianniello, 2014), (Ianniello and Bernardis, 2015).

12.2 (g) Discrete and Broadband Noise

The various attempts made by researchers to estimate the discrete and broadband noise are:

- Discrete – Tonals and low frequency continuous spectrum (Kehr and Kao, 2004), (Kowalczyk and Felicjancik, 2015),
- Broadband – (Takinaci and Taralp, 2013)
- Discrete and broadband noise - (Kim *et al.*, 2016), (Noughabi, Bayati and Tadjfar, 2017)

12.2 (h) Uniform and Non-uniform flow condition

The various attempts made by researchers to estimate the noise in the uniform and non-uniform conditions are:

- Uniform condition– (Bagheri, Seif, *et al.*, 2015), (Lloyd, Rijpkema and van Wijngaarden, 2015)
- Non-uniform condition - (Seol *et al.*, 2002), (Seol, Suh and Lee, 2005) (Kowalczyk and Felicjancik, 2015)

12.2 (i) Non-cavitation and Cavitation noise

The various attempts made by researchers to estimate the noise in the non-cavitation (NC) and cavitation (C) noise are:

- NC and C noise of a PPTC (Noughabi, Bayati and Tadjfar, 2017),
- C noise of a PPTC- (Lidtke, Turnock and Humphrey, 2015)
- NC and C noise of DTMB 4119 propeller -(Seol *et al.*, 2002), (Seol, Suh and Lee, 2005), (K. Park *et al.*,

2009),(Bagheri, Seif, *et al.*, 2015), (Sezen, Dogrul and Bal, 2016)
- NC noise of E779 propeller - (Ianniello and Bernardis, 2015)
- NC and C noise of INSEAN E799 propeller - (Testa, 2008)
- C noise of DTRC 4148 - (Salvatore and Ianniello, 2003)
- C and NC of P5168 (DTMB propeller) - (Morgut, 2012)

12.2 (j) Turbulence modeling

Some of the turbulence models used by researchers are tabulated in Table 1 below.

Paper	Turbulence Model	Remarks
Kulczyk et al in 2007	k– ω and k–ε	practically identical results obtained for the two models.
Morgut and Nobile, 2012	SST (Shear Stress Transport) turbulence model and BSL-RSM (Baseline-Reynolds Stress Model) turbulence model	BSL-RSM turbulence model provides only slightly better predictions
Lidtke et al 2015	k-ω SST URANS model	To capture tonal part of the noise
(Kowalczyk and Felicjancik, 2015)	two-equation SST k-omega	advantage over the k–ε model by its improved performance for boundary layers under adverse pressure gradients
(Rama, Bangaru and Suryananarayana, 2015)	LES	LES is chosen as viscous model because, it needs time dependent solution for hydrodynamic solution and it is not highly dependent to geometrical conditions
(Sezen, Dogrul and Bal, 2016)	k-ε	
(Noughabi, Bayati and Tadjfar, 2017)	SST k-ω	due to more accurate results in the cavitation flows around the propellers

12.2 (k) Propellers models used for hydro-acoustic study

The various propeller models tested for hydro-acoustic performance are listed below.
- Potsdam Propeller Test Case (PPTC) - (Lidtke, Turnock and Humphrey, 2015), (Noughabi, Bayati and Tadjfar, 2017)
- DTMB 4119 - (Seol *et al.*, 2002),(Seol, Suh and Lee, 2005), (Ekinci, Celik and Guner, 2010), (Bagheri, Seif, *et al.*, 2015) (Sezen, Dogrul and Bal, 2016)
- DTMB 4118 –
- DTRC 4148 - (Salvatore and Ianniello, 2003), (Ekinci, Celik and Guner, 2010)
- INSEAN E779A- (Testa, 2008), (Lloyd, Rijpkema and van Wijngaarden, 2015)
- P5168 – (Morgut, 2012)

12.2 (l) Validation of SPL obtained with FWH analogy

The first attempt to employ and validate the FWHE was undertaken by Salvatore and Ianniello in 2003 and results showed that noise predictions by the FWHE were in satisfactorily agreement with those obtained by using the Bernoulli equation (waveform of signatures were fully comparable and both the two models pointed out the major features of cavitation induced noise generation). However, some discrepancies between the two numerical results were present even at non cavitating flow conditions which required further investigations (Salvatore and Ianniello, 2003). Subsequently, the SPL predicted using FWHE have been compared with Bernoulli based methodology, CFD (Lloyd, Rijpkema and van Wijngaarden, 2015) and experimental results (Sakamoto, Kawakita and Kamiirisa, 2016).

12.3 ACOUSTIC WAVE MODEL

This method requires solving incompressible CFD and an acoustic wave model. It is required to solve an additional PDE for the sound propagation which can be done using specialized numerical methods. Acoustic sources are calculated from hydrodynamic pressures.

12.4 ADVANTAGES OF NUMERICAL METHODS

The advantages of numerical methods are as follows:-

(a) Freedom in the placement of hydrophones, which is somewhat restricted in typical experimental environments (Lloyd, Rijpkema and van Wijngaarden, 2015).

(b) Experimental pressure sensors may measure facility reverberations and/or model vibrations, which are subsequently difficult to separate from the recorded signals but this is eliminated in numerical simulations (Lloyd, Rijpkema and van Wijngaarden, 2015).

(c) Numerical methods are not as cost and time intensive as experiments.

(d) From cavitation tunnel tests the cavitation characteristics (length, area, volume etc.) cannot be precisely estimated. In this case, importance of the numerical method is of great value (Ekinci, Celik and Guner, 2010).

12.5 DISADVANTAGES OF NUMERICAL METHODS

Some of the difficulties with the numerical methods are as follows:-

(a) The ability of a Navier-Stokes solution to resolve pressure fluctuations far away from the source may be questioned, especially since the incompressibility assumption is invoked in maritime simulations (Lloyd, Rijpkema and van Wijngaarden, 2015).

(b) Another challenge is to develop a suitable grid, since the required resolution is both frequency dependent and spatially non-uniform (Lloyd, Rijpkema and van Wijngaarden, 2015).

13. SCALING PROCEDURE

Scaling procedures are essential to obtain full-scale noise levels of a propeller tested in an experimental facility at model scale. The guidelines for extrapolating cavitation noise have been clearly laid down by the Cavitation Committee of the 18th ITTC (1987) but procedures for extrapolating the non-cavitation noise are still not clearly understood and only some broad guidelines are available in the open domain.

13.1 CAVITATION NOISE

The scaling laws for extrapolating cavitation noise concern only differences in dimensions and operating conditions of the model and full-scale propellers and therefore do not correct for reverberation or dissimilarity in cavitation pattern and dynamics. The increase in noise levels from model to full scale is given by Equation 11 and the frequency shift (based on Rayleigh formula for the collapse time) is given by Equation 12:

$$\Delta SPL = 20log_{10}[(\frac{\sigma_s}{\sigma_m})^w(\frac{r_m}{r_s})^x(\frac{n_sD_s}{n_mD_m})^y(\frac{D_s}{D_m})^z] \tag{11}$$

$$\frac{f_s}{f_m} = \frac{n_s}{n_m}.\sqrt{\frac{\sigma_s}{\sigma_m}} \tag{12}$$

In the above, the subscripts s and m refer to full-scale and model-scale respectively and the increase in noise level is for proportional band width. The exponents x, y and z have different values depending on theoretical assumptions, test facility, range of Reynolds number applied and the model test method (*ITTC – Recommended Procedures and Guidelines - Model-Scale Propeller Cavitation Noise Measurements*, 2017).

13.2 NON-CAVITATION NOISE

For non-cavitating propeller trailing edge noise, as stated in Levkovsky (2002), scaling model test data to full scale levels will not provide an accurate prediction since the Cauchy number (Ch) and Reynolds number (Re) cannot be satisfied in the laboratory tests. According to the empirical relations between sound pressure Ps and blade tip speed U=nD, a similarity-based scaling method of predicting full scale sound pressure levels based on model scale experiments is suggested by Levkovsky (2002) (Equations 13 & 14):

$$\frac{f_s}{f_m} = \frac{n_s}{n_m} - \tag{13}$$

$$L_s = L_m[(\frac{r_m}{r_s})^2(\frac{n_s}{n_m})^5(\frac{D_s}{D_m})^7].\mathrm{k} \tag{14}$$

or

$$G_s = G_m[(\frac{r_m}{r_s})^2(\frac{n_s}{n_m})^4(\frac{D_s}{D_m})^7].\mathrm{k}$$

where subscript s and m mean full scale and model scale conditions, respectively, and G and L are power spectral density and spectral levels, respectively. Further, k=k(f,Re,Ch) is a frequency dependent coefficient to correct for the discrepancy between model and full scale conditions and is determined from statistical analyses of numerous test results of modern model scale and full scale propellers. A similar expression was also described by Fréchou and Dugué et al. (2000) (Fréchou *et al.*, 2001).

14. NUMERICAL STUDY

A numerical study has been undertaken to predict the noise of a DTMB 4119 model propeller (3D model shown in Figure 5 below) operating in the non-cavitating regime for the uniform flow condition. Flow around the propeller is solved with a commercial CFD software STAR-CCM+, while hydro-acoustic analysis is performed using Ffowcs Williams-Hawking (FWH) Equation. The numerical closure was achieved using k-ε Reynolds Averaged Navier-Stokes (RANS) model. The predicted hydrodynamic performance curves have been validated with the experimental results of Jessup et. al. (shown in Figure 6) and predicted sound pressure levels have been compared with the published numerical results of Seol et. al. (2002) (shown in Figure 7). All numerical computations are performed on a DELL workstation with 16 GB RAM, INTEL® XENON® CPU E5-1620 v4 @ 3.50 GHz and 1 TB HDD.

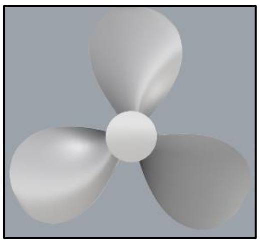

Figure 5. 3D model of DTMB 4119 model propeller

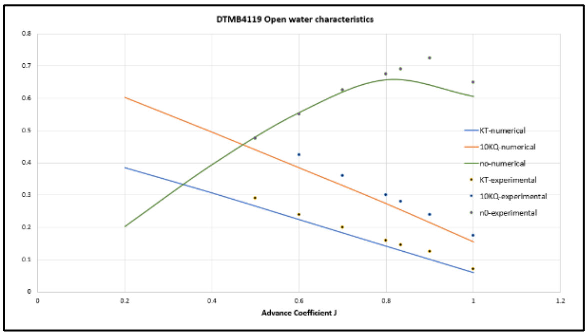

Figure 6. Comparison of numerical and experimental results of the open water characteristics of DTMB4119 model propeller

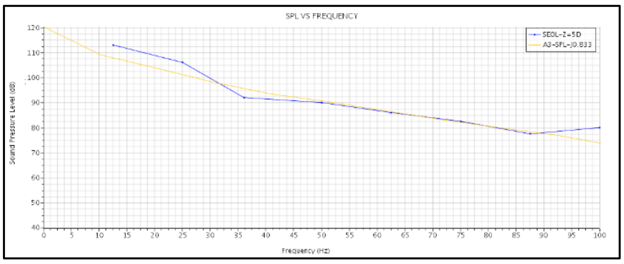

Figure 7. Comparison of predicted SPL (at 5D from propeller) with published numerical results

Based on the analysis of results of this study, the following conclusions were drawn by the authors:

(a) A numerical methodology has been established to predict the hydrodynamic and hydro-acoustic characteristics of DTMB4119 model propeller in open water condition.
(b) The numerical results and experimental data for the open water hydrodynamic characteristics of the model propeller show matching with an error of 2.88% in the open water efficiency at the design J of 0.833.
(c) The predicted SPL is compared with the numerical results of Seol et. al. and shows matching. The deviations in the SPL from the numerical results of Seol et. al. may be attributable to the difference in the methodologies adopted in the two studies.
(d) The SPL decreases as the distance of the observer/receiver from the propeller is increased.
(e) The Moving Reference Frame and Sliding Mesh methods of modelling the propeller and its rotation in CFD show almost similar results in terms of SPL at very low rpms (rps=10 in this study).
(f) Contribution of the thickness noise is negligible in comparison to that of loading noise at the receiver locations placed on the propeller hub axis for J=0.833.
(g) Monopole thickness noise radiates strongest towards the plane of blade rotation and the unsteady dipole loading noise has a strong radiation tendency towards the observer on the hub axis for J=0.833.

15. CONCLUSION

Design of new state-of-the-art propellers with low noise has emerged as an area of extreme relevance and challenge in the present scenario. It has been estimated that an overall reduction of about 20 dB in noise can be achieved through optimization of machinery and propeller noise mechanisms (Bertschneider *et al.*, 2014) which is considered significant. From this review paper, following salient conclusions can be drawn:-

1. Estimation of propeller noise has been a challenging yet interesting area of research since last 50 years.
2. Low noise propeller design options have been strongly recommended for the new ship designs. At the same time, it has also been found that design principles for cavitation reduction (for e.g. reducing pitch at the blade tips) can cause decrease of efficiency (IMO-Noise Working Group, 2014).
3. Propeller noise measurements are generally undertaken using scaled propeller models operated in hydrodynamic testing facilities like Cavitation Tunnel and Depressurised towing tanks. The conditions of similarity for each of these facilities are well established. These facilities can be used for measuring non-cavitation as well as cavitation noise of a scaled model for uniform and non-uniform flow conditions.
4. Scaling laws to extrapolate full scale propeller noise from model scale propeller noise of a cavitating propeller have been clearly laid down by the ITTC. But scaling laws to extrapolate full scale propeller noise from model scale propeller noise of a non-cavitating propeller noise are still not explicitly available.
5. The noise radiated by a propeller depends on various factors like wake field, geometrical parameters like rake angle, skew angle, blade tip shape, section shape, number of blades, propeller diameter, blade area ratio and pitch distribution and operating conditions like propeller rpm, advance coefficient and ships velocity/velocity of advance.
6. Experimental noise measurements have shown that advance coefficient has the maximum influence not only on the cavitation noise but also on the inception of cavitation.
7. Wake has a strong influence on the propeller cavitation and noise performance.
8. In the early days, potential-based panel method coupled with time domain acoustic analogy were used for non-cavitating propeller noise prediction. For cavitating noise prediction, the effect of cavitation in

terms of the time dependent cavity volume data is incorporated in the noise prediction method.

9. With the advent of fast and high capacity computing, numerical methods have gained popularity in ship hydrodynamics and hydro-acoustics.
10. Propeller in a uniform flow condition produces both monopole thickness noise and dipole Gutin noise. However, Gutin noise is negligible for underwater propellers. Under non-uniform inflow conditions, propellers produce dipole noise due to the unsteady loading on blade surfaces.
11. The quadrupole noise source term has been neglected in most of the studies since the rotating speed of the propeller is much lower than the underwater speed of sound and hence the contribution of the quadrupole source (noise generated due to turbulence) to the radiated noise has been neglected.
12. In the case of a ducted propeller, the effect of a duct on the radiated noise is small in the far field under non-cavitating situations since the noise directivities of single and ducted propellers are almost the same and only the high order Blade Passage Frequencies (BPFs) are influenced by the existence of the duct.
13. Monopole thickness noise, with its acoustic energy concentrated at its lower harmonics, is known to radiate strongest towards the plane of blade rotation. The unsteady loading noise is known to be dipole in nature, with a strong radiation tendency towards the observer on the hub axis.
14. It has been found that under non-cavitating conditions, unsteady loading noise / dipole noise dominates the overall radiated noise.
15. Propeller with greater Expanded Area Ratio (EAR) shows lower cavitation extension as compared to propeller with lower EAR.
16. An underwater propeller will radiate noise, even in uniform flow. It is called “self-noise” or “broadband noise”.

16. SCOPE OF RESEARCH

The increasing demand for fast, large, efficient, economic and ‘silent ships’ put pressure on industry, research establishments, and academic communities to develop improved techniques for noise reduction and better methods for its prediction. Considering all this, detailed research in the following areas needs to be undertaken:-

1. Diagnostic analysis of the effects of geometrical parameters like rake angle, skew angle, tip shape, section shape etc. on the non-cavitating noise performance of marine propellers. Modifications like incorporating ducts, increasing skew angle, proper pitch distribution, increasing blade area and number of blades and placing a foil on blade tip can help in reducing non-cavitating noise (Chekab *et al.*, 2013).
2. Mapping noise and efficiency for marine propeller designs – Knox et. al. in 2016 presented a methodology for exploring the trade-off space between maximising propeller efficiency and minimising far-field noise for a Wageningen-B series propeller. The geometry variation was restricted to the choice of pitch-diameter ratio, blade-area ratio, diameter and number of blades (Knox *et al.*, 2016).
3. Development of model test procedures to conduct acoustic measurements using a scaled propeller model in an atmospheric towing tank. This endeavour will also require acoustic characterisation of the towing tank facility in order to quantify the background and facility noise and develop correction factors for effects like wall reflection, free surface effect and tank reverberation.
4. Development of scaling laws to extrapolate non-cavitating noise from model scale to full scale propellers.
5. The common belief is that quadruple noise can be neglected in case of marine propellers due to low rotational speeds and the loading and thickness noise dominate the far-field radiated noise. Ianniello in 2014 observed deviations to this common belief and stated that this can be true for aeronautical cases but in case of marine propellers, the main geometrical features of a marine propeller blade (the very limited aspect ratio and the twist and thickness distribution along span) dramatically reduce the acoustic efficiency of the thickness (and, presumably, also the loading) noise component; thus, the well-known dominant role played by the FWH linear terms in air disappears underwater and the quadruple noise plays an important role in the far field noise (Ianniello, 2014). The understanding on this aspects needs further research.
6. A full uncertainty analysis of unsteady computations, for a larger number of grid and timestep combinations could be undertaken. Following this it will also be useful to examine the effect of changing the dimensions and location of the porous data surface, as well as comparing the pressure signals at a wider range of receiver locations in order to determine when best to use the FWH acoustic analogy method (Lloyd, Rijpkema and van Wijngaarden, 2015).

17. DISCLAIMER

The opinions presented herein are those of the authors and should not be construed as reflecting the views of any company or institution.

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