Specialist Committee on Hydrodynamic Noise

Final Report and Recommendations to the 27th ITTC

1 OVERVIEW

This report summarizes the work of the Specialist Committee on Hydrodynamic Noise for the 27th ITTC.

1.1 Membership and Meetings

The 26th ITTC appointed the following members to serve on the Specialist Committee on Hydrodynamic Noise:

- Herbert Bretschneider
  HSVA, Germany
- Johan Bosschers (secretary)
  MARIN, Netherlands
- Gil Hwan Choi
  Hyundai HI, Korea
- Elena Ciappi (chair)
  CNR-INSEAN, Italy
- Theodore Farabee
  NSWCCD, USA
- Chiharu Kawakita
  Mitsubishi Heavy Ind., Japan
- Denghai Tang
  CSSRC, China

The committee held four meetings at the following locations:

- Rome, Italy at INSEAN on March 1-2, 2012
- Wuxi, China at CSSRC on November 6-8, 2012
- Ulsan, South Korea at Hyundai HI on September 26-27, 2013
- Wageningen, Netherlands at MARIN on April 29-30, 2014

1.2 Recommendations of the 26th ITTC

The 26th ITTC recommended the Specialist Committee on Hydrodynamic Noise for the 27th ITTC to address the following activities:

(1) Create an overview of the characteristics of hydrodynamic noise sources (including machinery and equipment, e.g. sonars) and its influence to marine environment.

(2) Create an overview of existing national and international regulations regarding hydrodynamic noise.

(3) Check the existing methods and develop relevant guidelines for performing both model and full scale noise measurements.

(4) Identify scale effects in prediction of hydrodynamically generated noise (flow noise, cavitation noise....).

(5) Examine the possibilities to predict full scale values (correlation and operational requirements).

2 INTRODUCTION

The underwater radiated noise of ships can be important for various reasons. For naval vessels the underwater radiated noise is part of the signature requirements with respect to threats. High underwater noise levels may also influence fish behavior, which has resulted in noise requirements for fishery research vessels. Nowadays, there also is an increasing concern regarding the adverse influence of underwater noise, including shipping noise, on marine wildlife. Reduced ship traffic in a bay in Canada, following the events of 11 September 2001, resulted in a decrease of especially the low-frequency underwater noise levels while simultaneously a decrease was measured of stress hormones of whales within that bay (Rolland et al. 2011). Compared to decades ago, an increase in low-frequency deep-ocean ambient noise levels has been measured (Andrew et al. 2002, McDonald et al. 2006) which can be related to the increase in number of ships (Ainslie 2011). This has resulted in a wide variety of scientific, political and technical activities including studies to review measures by which underwater noise of commercial vessels can be reduced (Renilson, 2009).

Underwater noise emission of vessels can be grouped according to Urick (1983) and Ross (1976) into three major classes:

- Machinery noise comprising propulsion and auxiliary components.
- Propeller noise caused by flow phenomena related to propeller operation and interaction with the vessel hull.
- Hydrodynamic noise caused by flow of water along the ship hull and behind the vessel.

The noise exciting mechanisms in each class may be of different kind. Examples of noise that are of a mechanical origin include rotating unbalance, gear teeth loading, combustion processes and bearing friction. Fluid flow phenomena like cavitation, turbulence, vortex shedding, displacement and lift are a source of both near field pressure fluctuations and radiated noise.

Measurement hydrophones respond to pressure fluctuations which can be due to underwater sound, propagating with the speed of sound in water, or due to ‘pseudo sound’ caused by the turbulence passing over the hydrophone. Additionally, for flow over sonar systems, the pseudo sound can be a significant source of sonar self-noise and for flow over non-rigid surfaces, the pseudo sound can result in radiation of sound by exciting flexural vibrations of the surfaces.

With respect to discussions of noise emission from ships, use of the term ‘hydrodynamic noise’ is both too restrictive and, more importantly, misleading and will be replaced in the following by the term ‘underwater radiated noise’ or in short ‘noise’.

3 REVIEW OF NOISE SOURCES (INCLUDING SCALE EFFECTS)

3.1 Measured Noise Levels of Ships on Noise Ranges

A listing of the main underwater noise sources for ships is provided in Table 3.1. This listing provides information on the frequency range and impact to both the ship and environment of each source. A brief summary of noise sources for large and medium sized commercial vessels is presented below.

Large commercial vessels produce relatively loud and predominately low-frequency sound. Broadband source levels are generally in the range of 180 to 195 dB (re: 1μPa) with maximum levels in the frequency range of 10 to 125 Hz resulting from propulsion system generated noise. Individual vessels produce unique acoustic signatures and these signatures may change with ship speed, vessel load, operational mode and implementation of noise-reduction measures.

Table 3.1 Underwater Noise Sources for ships

Noise source	Frequency range	Impact to environment	Impact to the ship
Propeller noise non-cavitating tonal components	BPFs	Low/ medium	Depend on ship
Singing propeller	100 Hz – 2 kHz	high	high
Propeller non-cavitating broadband	1 Hz – 20 kHz	low	low
Propeller cavitating tonal	BPFs	high	high
Propeller cavitating broadband	10Hz - 20kHz	high	high
Propeller-hull interaction	BPFs and structural NF	low	high
Cavitation on appendages	100 Hz – 20 kHz	medium	medium
Wave breaking	100 Hz – 10 kHz	low	low
Slamming	1 Hz – 100 Hz	low	low
Sea water cooling systems	100 Hz – 10 kHz	medium	medium
Main engines	1 Hz – 500Hz	medium	high
Driving systems	10 Hz – 1 kHz	low	medium
Auxiliary engines and systems	10 Hz – 2 kHz	low	medium
Active sonar military	100 Hz – 50 kHz	high	Medium
Active sonar echo-sounder	10 Hz – 30 kHz	low	low
Active sonar navigation	10 Hz – 100 kHz	low	low
Airguns	1 Hz – 100 Hz	high	low

Most of the acoustic field surrounding large vessels is the result of propeller cavitation causing ships at their service speed to emit both low-frequency tonal sounds, which can be heard over great distance, and high-frequency noise (up to 20 kHz) close to the vessel. Less intense, but potentially significant levels of radiated noise can result from onboard machinery (engine room and auxiliary equipment). Hydrodynamic flow over the ship's hull and hull attachments is also a potentially important broadband sound-generating mechanism, especially at higher ship speed. The far field underwater noise levels are furthermore influenced by water depth and the variation of sound speed with depth which influence propagation losses. The presence of the free surface leads to the Lloyd-mirror interference pattern which depends on the submersion of the source.

Large vessels are loud sources in both offshore (in shipping routes and corridors) and coastal waters (mainly in traffic lanes, waterways/canals or ports). Due to their loud and low-frequency signatures, large vessels are the dominant source of low-frequency background noise in many marine environments worldwide.

Medium sized vessels such as tugboats, crewboats, supply ships, research vessels, and many fishing vessels typically have large and complex propulsion systems, often including bow-thrusters. Typical broadband source levels for small to mid-size vessels are generally in the range of 165 to 180 dB (re: 1μPa). Most medium-sized ships are similar to large vessels in that most of the sound energy is low-frequency (<1 kHz). While broadband source levels are usually slightly lower for medium-sized vessels than for the larger commercial vessels, there are some exceptions (e.g., as a function of age or maintenance of the ship), and medium-sized ships can produce noise of sufficient level and frequency to contribute to marine ambient noise in some areas. There is concern that mid-sized vessels spend most of their operational time in coastal or continental shelf waters, and hence overlap in time and space with marine animals, many of which occupy these waters for the important purposes of breeding and/or feeding.

Arveson and Vendittis (2000) present a set of very detailed noise measurements of a modern cargo ship. Extensive radiated noise measurements were made of the M/V Overseas Harriette, a bulk cargo ship (length 173 m and displacement of 25,515 tons) powered by a direct drive low speed diesel engine, which is a design representative of many modern merchant ships. The spectral levels of noise generated by the vessel at various speeds and propeller rotation rates are shown in Figure 3.1.

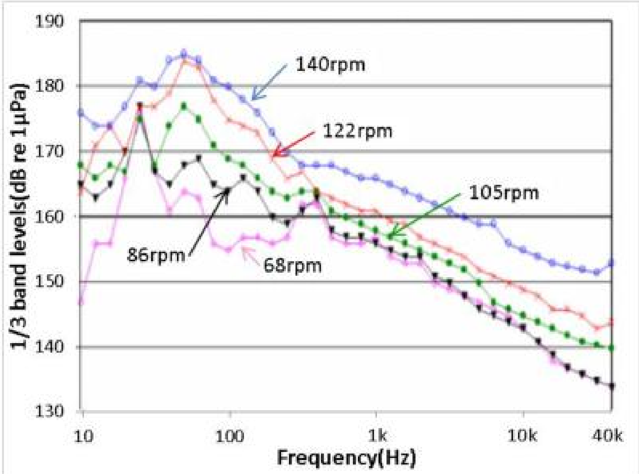

Figure 3.1 Spectra for a bulk cargo ship at various speeds and propeller rotation rates (modified from Arveson and Vendittis, 2000).

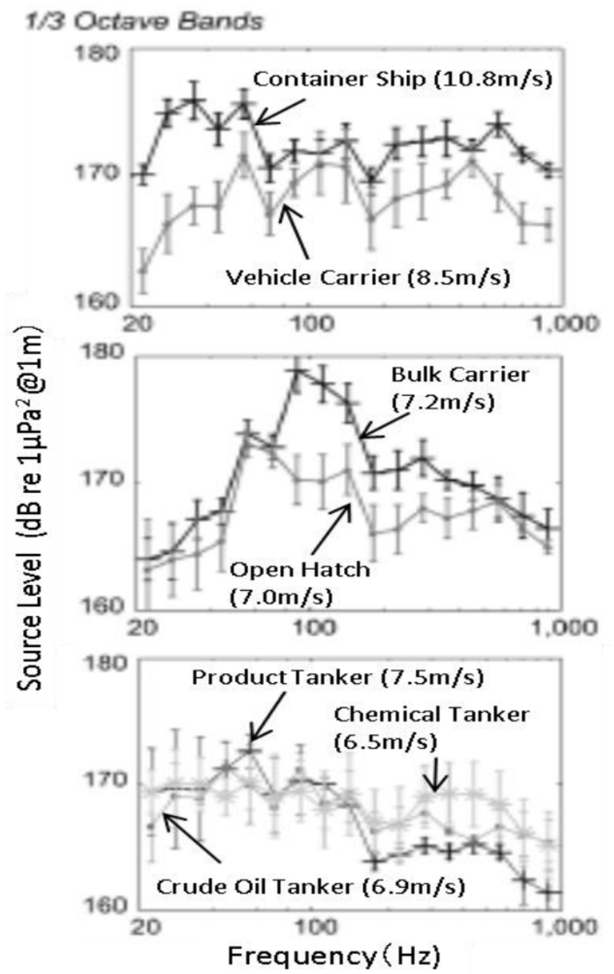

Figure 3.2 Typical noise levels for different types of ships (modified from McKenna et al., 2012).

McKenna et al. (2012) present measured source levels for several types of ship: (a) container ships and vehicle carriers, (b) bulk carriers and open hatch cargos, and (c) three types of tankers. Figure 3.2 shows the 1/3 octave band source levels with mean and standard errors. Figure 3.3 shows the broadband (20 to 1,000 Hz) source level for these ships as a function of ship speed. There is significant differences in both source level and spectral characteristics of underwater noise amongst the ship types for which measurements were made.

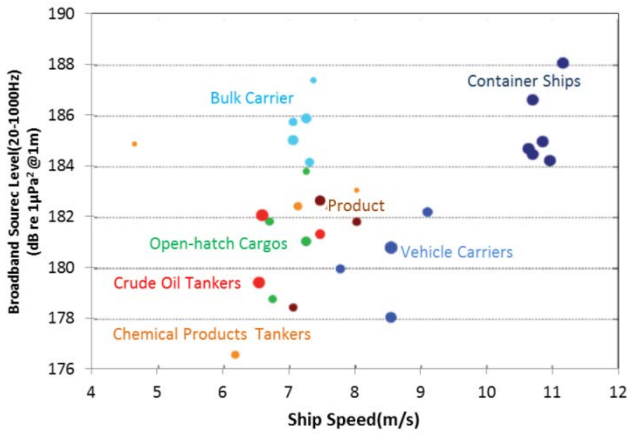

Figure 3.3 Broadband ship source level versus speed for measured ships. Bubble color signifies ship-type. Bubble size represents the relative size of the ship, measured as GT. (modified from McKenna et al., 2012).

3.2 Hydrodynamic Noise Sources

3.2.1. Non-cavitating Propeller Noise

The noise radiated from a non-cavitating propeller is caused by fluctuating hydrodynamic forces generated on the propellers which can be of two types, discrete frequency (tonal), and continuous spectrum (broadband). Discrete frequency forces are caused by the action of a propeller operating in the presence of upstream non-uniform wakes. The frequency of discrete forces correspond to the blade frequencies f=nz (#blades x shaft rotation rate) and generally do not exceed 20 Hz (first 3 harmonics). Continuous spectrum forces are generated as a result of upstream flow disturbances or turbulence generated on the blade surface. Low frequency continuous spectrum hydroacoustic forces are caused when the hull turbulent boundary layer on the vessel surface is ingested into the propulsor. High frequency continuous spectrum hydroacoustic forces are caused when the local boundary layer, formed on the blade surface, passes over the trailing edge of the blade.

The sound pressure level of a non-cavitating propeller is less intense and of less impact compared to a cavitating propeller. The features of cavitating and non-cavitating propeller noise spectra are illustrated in Figure 3.4 (Fréchou and Dugué et al., 2000).

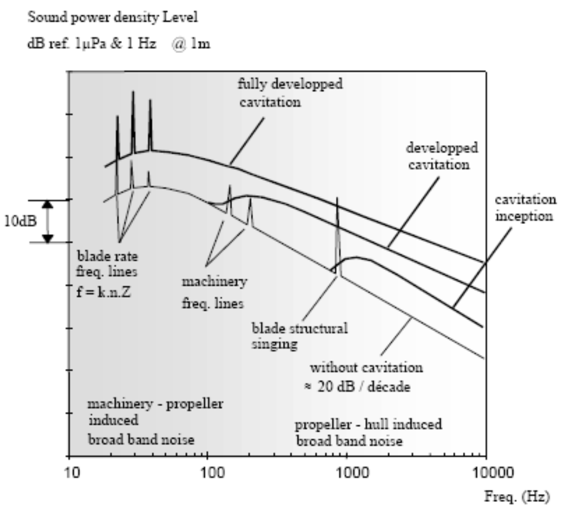

Figure 3.4 Sound pressure level radiated by cavitating and non-cavitating propeller.

The radiated noise data of Arveson et al. (2000) discussed earlier show high-level tonal frequencies from the ship's service diesel generator, main engine firing rate, and at harmonics of blade rate due to propeller cavitation. At low ship speeds, tonal components from the ship's service diesel generator contribute almost all of the radiated noise power of the ship. At higher speeds, propulsion-related sources dominate the ship's radiated noise. In this case the propeller is heavily cavitating and blade rate harmonics are an important sources of radiated noise.

In order to understand the physics of non-cavitating propeller noise, hydroacoustic test facilities -especially large quiet high speed water tunnels- are essential tools. However, because the dimensions of the cavitation tunnel test section are limited, there exists a limiting low frequency below which meaningful acoustic measurements cannot be obtained. Below this frequency propeller noise can only be measured, or inferred, using indirect methods. One method of assessing discrete line (tonal) noise of a propeller is to measure the fluctuating forces of the propeller and then predict the noise generated by a force of that magnitude applied to the water. Investigations of this type have been conducted in the GTH (Fréchou and Dugué et al., 2000) and at the DTRC laboratories (Jessup, 1990).

Higher frequency propeller noise can generally be investigated in testing facilities at model scale providing that the facility has low enough background noise. A number of similarity conditions have been proposed and evaluated (Fréchou and Dugué et al., 2000, and Levkovsky, 2002) for predicting full scale noise levels based on propeller noise measurement made in a cavitation tunnel. 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 P_s 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):

$$f_{FS} = f_M \cdot \frac{n_{FS}}{n_M}$$

$$L_{FS} = L_M \cdot \left(\frac{n_{FS}}{n_M}\right)^5 \cdot \left(\frac{D_{FS}}{D_M}\right)^7 \cdot \left(\frac{r_M}{r_S}\right)^2 \cdot k$$

$$\text{or:} \quad G_{FS} = G_M \cdot \left(\frac{n_{FS}}{n_M}\right)^4 \cdot \left(\frac{D_{FS}}{D_M}\right)^7 \cdot \left(\frac{r_M}{r_S}\right)^2 \cdot k$$

where subscript FS 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 is also described by Fréchou and Dugué et al. (2000).

3.2.2. Cavitating Propeller Noise

The simplest description of the mechanisms of propeller cavitation noise is the noise generated by the volume acceleration of a single bubble of which the dynamic behavior can be described by the Rayleigh-Plesset equation (Blake 1986). The equation has been extended and studied in much detail (Brennen, 1995 and Leighton, 1994) and the noise spectrum of the collapse of a single bubble has been described by Fitzpatrick and Strasberg (1956). Up to the point of collapse bubble dynamics are well predicted using potential flow assumptions. However, the dynamics of bubble collapse are very complicated with energy dissipated by sound radiation, heat conduction and viscosity, and rebounds of the bubble occurring in the presence of non-condensable gas.

The noise spectrum from a prototypical cavitating propeller has been described, for instance, by Løvik (1981) and Brown (1976) as illustrated in Figure 3.5. The figure shows a low frequency region in which tonals are present at harmonics of the blade passage frequency. A broadband hump is present of which the centre frequency is proportional to the reciprocal of the typical duration time of the large scale cavity dynamics.

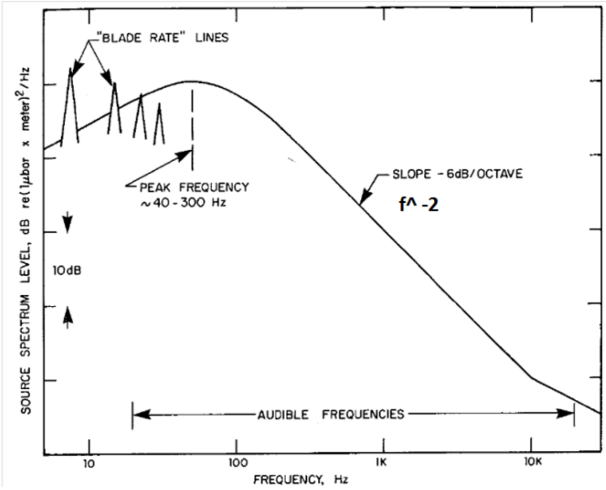

Figure 3.5 Stylistic power spectral density of cavitating propeller noise (adapted from Brown, 1976).

For frequencies below the centre frequency, Fitzpatrick and Strasberg's (1956) theoretical analysis for a single bubble which yields a spectral density increasing with the fourth power of the frequency is thought to apply. The high frequency region is determined by the collapse of individual bubbles and the spectrum decreases with the reciprocal of the frequency squared. As bubble collapse is cushioned by the presence of gas, the magnitude of the spectrum level in this region also decreases with increasing gas content. Additionally, the compressibility of the fluid influences the radiated noise in this region. The high frequency slope of the power spectral density generally decreases according to f^{-2} which corresponds to a decreases of 6 dB/octave (for constant bandwidth). In the stylistic spectrum by Løvik (1981) several regions are distinguished at high frequency which are also discussed in the report of the Cavitation Committee of the 18th ITTC (1987). However, only part of the noise spectrum of a cavitating propeller can be attributed to single bubble dynamics with an important portion arising due to the collective behavior of bubbles (Omta, 1987; Wang and

Brennen, 1994) and, for very high frequencies, bubble-bubble interaction (Hallander and Bark, 2002).

For ships with fully developed propeller cavitation, the spectral levels scale approximately with the ship speed to a value between the fifth and sixth power (Ross, 1976), $L_s \propto 10\,\log_{10} V^6$. Near cavitation inception, a higher speed dependency can be found, see e.g. Blake (1986).

Scale effects relevant for cavitation observations and hull pressure measurements also apply to cavitation noise measurements. Geometric similarity is usually satisfied but complete kinematic similarity, i.e. similarity of the velocity vectors, is usually difficult to obtain due to differences in Reynolds number which lead to differences in the ship wake. Tests are performed using kinematic similarity for the mean velocity implying identical mean thrust coefficients. The influence of wake scaling on hull pressure fluctuations has been reported, see e.g. Schuiling (2011) and Johannsen et al. (2012), but its influence on the radiated noise levels is not known.

Hydrostatic pressure variations are only similar if the Froude number is identical. For cavitation tunnel testing, this condition is usually not satisfied and similarity of cavitation number is specified for a selected location in the propeller disc. Nuclei are required for cavitation inception and while nuclei similarity is hard to achieve it is not strictly necessary. In model scale measurements nuclei can be generated through the application of leading edge roughness, changing the gas content in the flow, or by bubble injection through electrolysis or injection of supersaturated water.

Two specific similarity parameters that are relevant for noise tests are gas content and Mach number. Gas content will influence the collapse of cavitation bubbles due to a cushioning effect and, to a smaller effect, has an influence on the speed of sound: increasing the gas content will lead to reduction in sound speed which changes the Mach number and the acoustic impedance of the fluid. The gas content should be kept to a minimum in model scale testing since too high of a gas content leads to a reduction of noise levels at high frequencies (Løvik, 1981, Bark, 1985). It is remarked though that at full scale the gas content may change significantly, due for example to breaking wind waves and waves generated by the ship. Mach number expresses the similarity of compressibility effects which are responsible for the conversion of hydrodynamic energy to acoustic energy. While Mach number may influence the high frequency part of the noise spectrum, the consequences of dissimilarity of this parameter is unknown. The same holds for the ratio of acoustic wave length to ship and propeller length scales (acoustic compactness) which influence reflection and diffraction.

A detailed description of the extrapolation procedure for propeller cavitation noise is presented in Section 7.2. The Cavitation Committee of the 19th ITTC (1990) reports that the mean deviations between predicted noise levels from model tests and full scale measured levels are in the order of 3 to 5 dB with the remark that it is not fully recognized if this is representative for the best agreement obtained. There is a clear lack of published detailed validation studies between model scale tests and full scale trials related to propeller cavitation noise.

The inception of tip vortex cavitation is known to be severely influenced by the size of the viscous core of the vortex and therefore by the Reynolds number. Due to the reduced Reynolds number at model scale, cavitation inception is delayed by a certain factor, usually expressed with the ratio of Reynolds numbers:

$$\left(\frac{\sigma_{i,fs}}{\sigma_{i,ms}}\right)=\left(\frac{\mathrm{Re}_{fs}}{\mathrm{Re}_{ms}}\right)^m$$

Empirical values for the exponent m have been obtained by comparing model scale experiments with full scale trials and were reviewed by the 21st ITTC Cavitation Committee. The mean value is approximately 0.35. More recently, Shen et al. (2009) have shown that the exponent m is a function of Reynolds number and is smaller with increasing model scale Reynolds number.

Due to the delayed inception of vortex cavitation, alternative formulations have been proposed for the scaling of tip vortex cavitation noise. Blake (1986b) proposes a universal semi-empirical scaling formulation generated for bubble, sheet, and vortex cavitation. The formulation is discussed by Baiter (1989) who concludes that more detailed understanding of the physics is required in order to understand the consequences of dissimilarity of cavitation inception. Oshima (1990) found a good correlation between full scale and model scale predictions for noise levels due to a cavitating vortex if dissimilarity in cavitation number is applied using a value that scales with the Reynolds number to the power 0.15. Bosschers (2010) suggests that the dissimilarity of cavitation inception influences the size of the cavitating vortex for cavitation numbers a bit beyond inception. For well-developed tip vortex cavitation, the cavity size becomes independent of the viscous core size suggesting that noise measurements can be performed at cavitation number identity.

3.2.3. Singing Propeller

Sometimes propellers produce high-pitch squeaking noise, mainly in non-cavitating conditions, due to a phenomenon termed singing. Often the spectra of underwater radiated noise, hull vibration, and onboard airborne noise exhibit sharp lines belonging to one or more of the natural propeller blade frequencies, typically in the frequency range from 100 Hz to1.5 kHz. With increasing rotational shaft speed higher natural frequencies may appear in a stepwise manner due to the dynamics of a lock-in process. Vortex shedding at the trailing edge excites blade vibration, which can have a feedback on the shedding process (lock-in effect). Propeller singing is difficult to predict due to its dependence on unknown parameters, e.g. mechanical damping factor or details of trailing edge geometry. For example, not all blades may sing and it is not uncommon for only one or two propellers out of a series of geometrically similar ones to exhibit this phenomenon.

Singing during model testing of propellers is sometimes visible during cavitation observations (see Figure 3.6). Due to the low pressure in the core of the shedding vortices, cavitation starts and visualizes the vortices as white stripes parallel to the trailing edge of the blade.

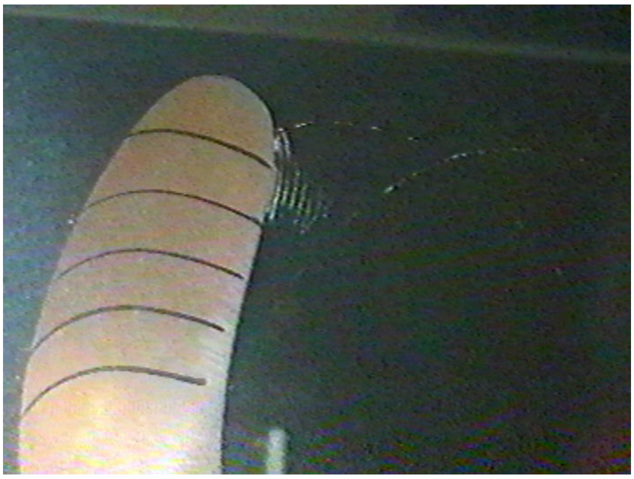

Figure 3.6 Singing model propeller (HSVA).

Propeller singing is characterized by one (or more) very high amplitude distinct tones that cause annoyance for passengers and crew, reduces detection and classification range for navy vessels, decreases the performance of seismic and fishery research vessels, and may lead in extreme cases to propeller fatigue failure. Often the problem can be mitigated by application of an appropriate modification ("Anti-singing Edge") of the suction side trailing edge geometry of the blades, in the radial range from 0.5R to 1.0R where R is the propeller radius.

3.2.4. Flow noise, including wave breaking and slamming

Flow noise is the noise generated by the flow around the ship hull which includes the turbulent boundary layer pressure fluctuations, wave dynamics and bubble generation, see Figure 3.7. In general, these sources generate less noise when compared to cavitation noise and machinery noise unless extensive noise mitigation measures have been applied such as on naval vessels.

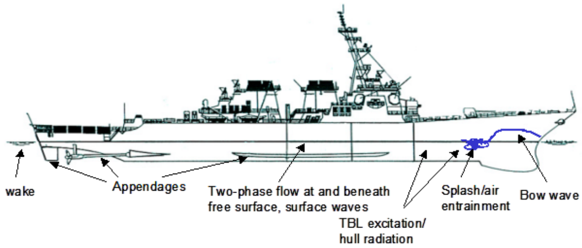

Figure 3.7 Surface ship flow noise mechanisms.

The pressure fluctuations due to the turbulent boundary layer is a rather inefficient (quadrupole) sound source when considered in isolation, but it can become more efficient in the presence of a rigid or especially a flexible surface such as the hull plating of which the vibrations generate sound (Blake 1986). The radiation efficiency of the hull plates is strongly influenced by fluid loading and by the presence of ribs and stiffeners. Both spatial and temporal characteristics of the turbulent boundary layer pressure fluctuations need to be taken into account for the excitation of the hull vibrations. Unsteady surface pressure measurements have been performed by Goody et al. (2007) on the surface of a ship model hull in a towing tank. The results compare well with an empirical model and, for low frequencies, with computational results using a Reynolds Averaged Navier-Stokes Statistical Model. Similar measurements have been performed by Ciappi et al. (2009) and De Jong et al. (2009). The scaling parameters are strongly related to Reynolds number and include the boundary layer displacement thickness and the wall shear stress. In addition, hull conditions are critical.

Wave breaking with its generation of air bubbles in water is a noise source which has been studied in detail for e.g. breaking waves in a coastal zone (Deane 1997). Most of the noise is caused by oscillating air bubbles and clouds of air bubbles with the noise depending on the amount of air entrained and the bubble size distribution. Individual bubbles will emit sound when they are formed, due to entrainment, splitting, coalescence, or under influence of external pressure fluctuations (Leighton, 1994), and the noise is therefore influenced by Froude number, Weber number, Reynolds number, turbulence intensity and water quality which complicate scaled model tests and computational predictions. An example of the noise generated by a breaking bow wave and stern wave of a ship model in a towing tank is given in De Jong et al. (2009).

Bow and stern slamming results from the impact of the fore or the aft sections of the vessel on the water surface. Speed and sea orientation are the main variables dictating the inception and severity of slamming. Slamming can cause global vibration (whipping) or local vibration of the part directly impacting the water surface. The phenomenon is important mainly for the fatigue life of the ship and for safety of passengers and crew. Global vibration involves the modal response of the whole ship, typically of the order of few Hz. As reference, the lowest order fundamental frequencies of a section of hull plating (between frames) is on the order of 100 Hz.

Although some international organizations report slamming as one of the sources of underwater noise, no evidence has been found in the technical literature to support this. It is worth noting that generally if seaway conditions are such that slamming occurs, a ship will slow down to prevent slamming and underwater noise will be dominated by noise from the rough seas.

The phenomenon of slamming can be accurately tested at model scale if the model is properly scaled to replicate global hydro-elastic effects and tested at the correct Froude number. It has been demonstrated that with this physical model it is possible to establish a perfect correlation between model and full scale in term of load bending moments and of the first bending modes of the ship. A detailed overview of the method and of the results so far achieved can be found in Hirdaris et al. (2014). When local response is considered, hydrodynamic loads (pressure and acceleration) can be measured on rigid models and the structural response calculated numerically or theoretically.

3.3 Other Noise Sources

3.3.1. Machinery Noise

3.3.1.1. Generality

Machinery noise originates as mechanical vibration of many and different parts of a moving vessel. There are three ways of noise transmission between a vibration/acoustic source, for example an engine, and the environment (Fischer, 2007). The first, which is the most important for underwater noise, is by structure borne noise transmitted via foundations, pipes, and couplings. The second way of noise transmission is by airborne noise. This is most important for people working near the noise source but the effect of this noise outside of the ship is very low. The last noise transmission way is via the exhaust gas chimney. This noise is most significant above the water surface.

Machine vibrations can originate in the following ways (Urick, 1983): *i)* unbalanced rotating shafts, *ii)* repetitive discontinuities, e.g. gear teeth, armature slots or turbine blades, *iii)* reciprocating parts, e.g. combustion in engine cylinders (piston slaps), *iv)* cavitation and turbulence in fluids flowing through pipes, pumps, valves, condensers, and *v)* mechanical friction as in bearings and journals

The first three of these sources produce a line component rich spectrum in which the noise is dominated by tonal components occurring at the fundamental frequency and harmonics of the vibration producing process. The other two give rise to noise having a continuous spectrum.

With reference to underwater radiated noise, the machinery onboard a ship can be divided roughly into 2 categories, namely:

- Machinery for the Main Engine Propulsion: Diesel Engines geared or directly drive, Diesel-Electric, Steam and Gas Turbines Gas turbine-electric

  For this category noise from reduction gears, bearing and journals etc. are included. The typical frequency range of noise from main engine propulsion is from a few Hz up to 1 KHz.

- Auxiliary Machinery:

  Pumps, purifiers, electrical generators, fresh water generators, heaters, coolers, oily water separator, auxiliary steam boilers, steering gears, air conditioning machines, refrigerator machines, cargo winches, cranes, air compressors, air tanks, oil tanks, water tanks, bow thrusters, stabilizers, firefighting installations, lifeboat engines, filters, and many others

  Noise emission from auxiliary machinery covers the range 10 Hz to 5 KHz.

3.3.1.2. Characteristics of noise induced by machinery

Diesel engines direct drive and geared. Typical propulsion noise contributors included the diesel engines and the reduction gears. The dominant noise of diesel engines is normally due to *"piston slap"* (Ross, 1976; Arveson and Vendittis, 2000). Other main characteristic vibration frequencies visible in the noise spectrum are those due to the cylinder firing rate, crankshaft, engine valves, and piston rings. Because diesel engine rpm varies according to propulsion demand, these signature components occur at frequencies that depended upon ship speed.

The majority of large ships are propelled by a low speed, 2-stroke diesel engine directly driving a single propeller. These engines work at low revolutions (70 to 120 - 140 rpm) and are heavy. Due to the size, the engines are rigidly connected to both the ship hull and the propeller shaft resulting in significant vibration below 100 Hz.

Other diesel-powered ships employ medium speed, 4-stroke diesel engines, which connect to the propeller shaft via a reduction gear. Typical speeds of these engines are 300 to 1,000 rpm. The engines can be rigidly or resiliently mounted.

The noise emission of medium speed engines can be separated in two bands. The lower band covers the range between 6 Hz and approximately 150 Hz. The noise in this range is generated by mass forces of the moving pistons, conrods and crankshafts and by gas forces arising from the internal combustion process and exhibits distinct frequencies which are integer or half integer multiples of the shaft rate frequency. For the higher frequency band, engine noise is broader band, excited by the internal combustion process, and thump noise of pistons, gear wheels, and valves.

Vibration levels produced by medium and high speed diesels are typically higher than that produced by low speed diesels. Diesel vibration source levels usually scale as $(\text{power/weight})^2$ (Nelson et al. 2000); therefore heavy low speed diesels have lower source levels due to their lower power to weight ratio. Reduction gears of medium speed engines may generate noise at much higher frequencies, up to 1 kHz and possibly higher.

Diesel Electric. The noise signature of diesel-electric ships typically contain energy contributions from the diesel generators and from the electric propulsion motors in combination with the frequency converters (synchroconverter or a cycloconverter). The levels of electric propulsion motor noise, and the frequencies at which they occur, vary by ship and with propulsion shaft rpm. Noise sources for electrical machines can be mechanical (angular and parallel shaft misalignment, dynamically unbalance rotors, loose stator lamination, bearing), and electromagnetic (phase unbalanced, slot opening, input current waveform distortion, magnetic saturation etc.). Even large direct drive electric motors are quiet if compared with reduction gears and piston engines.

For diesel-electric systems, the diesel generators operate at a constant rpm and therefore their noise characteristics are not dependent on ship speed. Moreover when used as a genset they are usually elastically mounted. The same consideration holds when a gas turbine is used as a generator.

Turbines gear drive. Propulsion turbines, turbine generators, and reduction gears are the dominant sources of propulsion system noise on steam turbine equipped ships. Propulsion turbine and reduction gear related noise components occur at frequencies related to propulsion shaft rpm (typically up to 1 kHz). Gas turbine driven vessels are generally quieter than their diesel counterparts. This is primarily because this machinery is rotary instead of reciprocating and hence vibration levels – both tonal and broadband – are lower for comparable power-to-weight ratios. Furthermore, the tones produced by a gas turbine are much higher due to their higher rotation rate, which can be as high as 3,600 rpm or 60 Hz. A comparison of representative diesel and gas turbine vibration levels is provided in Figure 3.8.

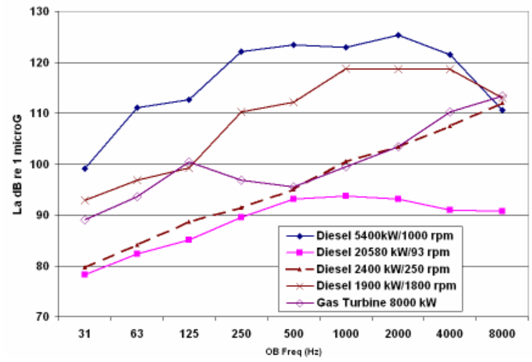

Figure 3.8 Source vibration levels for Diesel and Gas Turbine (Fisher and Brown, 2005).

Auxiliary Machinery. Noise components from rotating auxiliary machinery and other shipboard equipment also contributes to a ship's overall noise signature, but usually at lower levels than the propulsion systems. A typical frequency range for vibration spectra for auxiliary machinery is 1 Hz to 5 kHz.

Problems of underwater noise radiation from auxiliary machinery is only reported from navy surface vessels and submarines which have very low noise levels and requirements.

Sea water cooling pumps. Sea water cooling pumps are mainly of a centrifugal type and the impellers cause tonals at impeller blade passage rate and related harmonics. Source levels in the pipes close to the pump reach up to 180 dB (re: 1µPa) for non-cavitating condition and can be more that 200 dB in cases of impeller cavitation.

Mitigation of blade tonals can be achieved within the pump by increasing the tip clearance of the impeller and accepting a reduced efficiency. Another measure is to introduce fluid silencers up- and down-stream of the pump.

Bow and Stern Thrusters. Bow and stern thrusters are mainly horizontal axis tunnel type impeller systems and are strong noise sources. Most of the noise from thrusters is caused by cavitation on the impeller blades. The cavitation causes direct radiated noise and also structure-borne noise which propagates through the hull structure and can radiate as underwater noise. The spectrum of thruster noise is broadband with energy covering a very wide frequency range. Specialized thruster types such as azimuths, pumpjets etc. have different noise characteristics compared to conventional thrusters. (Lloyd's Register Consulting 2013).

Both blade form design modifications and improved inflow to the impellers can reduce the cavitation volume. However, due to design limitations, such as support structures and rather high loadings of conventional thrusters, cavitation is nearly inevitable.

3.3.1.3. Solutions and recommendations for machinery noise reduction

It is generally recommended that structural-acoustic measurements be made onboard to identify the main noise sources and associated transmission paths. Some solutions that should be adopted to reduce machinery vibration and noise, derived from the technical literature and discussed in the IMO/MEPC.1/Circ.833, are hereafter summarized:

- Provide passive modification of the engine bed section in order to change the mobility at the source location;
- Decouple machinery from the hull by proper design of resilient mounting and use of two stage isolation systems;
- Provide elastic coupling between engine and gear box;
- Use double hulls outboard of the engine room;
- Place noisier equipment towards the centerline of the ship;
- Provide high quality mechanical finishing and perform maintenance regularly;
- Use high quality diesel-electric motors;
- Use flexible pipe and hose.

3.3.2. Sonars

3.3.2.1. Active Military Sonars

Active military sonars (AMS) pose perhaps the most significant acoustic impact to the ocean environment and receive the most press and public discourse. The environmental impact of an AMS depends significantly on the sonar's purpose since this determines the sonar's frequency range, source strength, and mode of operation (pulse duration, etc.).

The NRDC report (Jasney, et al.) titled "Sounding the Depths II: The Rising Toll of Sonar, Shipping and Industrial Ocean Noise on Marine Life" provides a thorough listing of AMS systems in use, or in development, by NATO countries which includes information on the military name of the sonar system, a general categorization of the sonar frequency range, and the platform carrying the sonar. Although information on source strength and mode of operation is not provided it can generally be deduced based on the purpose of the sonar system. While a similar listing for non-NATO countries was not found, it is reasonable to assume comparable systems are employed.

The operational purpose of the AMS dictates the sonar's frequency range and source strength. A majority of active military sonars are used for anti-submarine warfare purposes and thus operate in the low (~100 to 1kHz) to medium (~1kHz to 8 kHz) frequency range so that signal strength is not significantly impacted by acoustic absorption which increases with frequency. Sonars operating in the high frequency range (~ 8 kHz and higher) are generally used as navigational aids or for mine hunting where interest is in detecting the presence of objects at shorter ranges and where higher (spatial) resolution is needed.

In terms of environmental impact, those that pose the greatest impact are ones operating at low frequency (nominally 100 to 1 kHz) for which there is little propagation loss other than that due to spreading from the source. An important issue regarding potential environmental impact is the purpose of the sonar and the platform on which it operates. For example, while submarines carry powerful sonars that operate at low frequency, they are seldom used since they serve as a beacon indicating its presence and location. This is in contrast to sonars on military surface ships which are used more often since operation does not significantly increase knowledge of the ships position beyond what is readily determined by other means. For example, SURTASS is a sonar system that includes a low frequency active capability that can be continuously operated as the ship "sweeps" the ocean searching for underwater vehicles. It is reported that this system operates in the 100 to 500 Hz frequency range with an effective source strength of up to 235 dB. Additionally, active sonar systems can be deployed from helicopters (dipping sonars) and can be dropped from various types of aircraft (sonobuoy).

3.3.2.2. Active Sonar Echo-Sounder & Active Navigation Sonar

Active sonar echo-sounder and navigation sonar systems are discussed together since they are closely related and are part of a broader group of general purpose active sonar systems. These sonars typically operate at lower power levels and in the medium to high frequency range and are not considered to pose environmental issues.

Active sonar echo-sounders include sonars termed depth sounders and fathometers. Such sonars operate in the medium to high frequency range depending on where and how they are used. The method of operation is generally to emit an acoustic pulse downward and measure water depth based on time of flight of the bottom reflected return pulse. They generally operate at relatively low source levels to reduce issues with multiple reflections and at higher frequencies to provide higher accuracy in determining depth.

Fish finders operate similarly to echo-sounders except that the intent is for the acoustic pulses to reflect off fish instead of the ocean bottom. They also operate at higher frequencies to provide discrimination and at low source levels as to not adversely disturb the fish that are trying to be located. It is noted that the frequency of operation is potentially set at a frequency that provides maximum acoustic reflection for the fish of interest. Fish finders are used both commercially and recreationally.

Searchlight sonars, which includes side-scan sonars, and acoustic cameras are examples of high frequency sonar systems used for the purpose of imaging underwater objects. These sonars generally operate at lower source levels to reduce issues with multiple reflections and at quite high frequencies to provide high resolution capabilities. Acoustic Doppler current profilers have become common instruments for high accuracy measurement of speed, either of vehicles on which they are mounted or of currents passing over them. Water speed is measured based on the Doppler frequency shift of pulses reflected back from particulates in the water. To obtain highly accurate measurements of speed, they typically operate at high frequencies.

A final type of ship-board sonar system includes those used for underwater acoustic communications. They typically operate in the medium frequency range and have low to medium source strengths. This category includes systems used for voice communication or as underwater acoustic modems. Most often these systems are used for communication between surface vessels and submerged vehicles.

3.3.3. Airguns

Currently almost all marine seismic surveys use arrays of airguns as a noise source for seismic signals. An airgun is a twin piston steel cylinder charged with high-pressure air (up to 200 bar). After triggering by an electric signal the airgun suddenly releases the compressed air to the lower outside pressure causing a transient high pressure peak like from explosives.

The peak pressure reaches values of about 230 dB (re: 1µPa at 1m), with a spectrum that is of broadband type. Most airgun noise occurs in the range below 1 kHz with increasing levels at lower frequencies with a maximum typically below 100 Hz.

At the beginning of a seismic survey the airgun arrays are initially operated from low pressures and stepwise increase pressure to the operating pressure - so called soft start - to ward of marine animals.

4 NOISE REGULATION

4.1 Influence of Noise on Marine Environment

The Marine Environment Protection Committee (MEPC) of the International Maritime Organisation (IMO) stated the following. “Most marine animals produce and receive sounds for critical life functions such as communicating, foraging, evading predators, and navigating. Much as human rely heavily on their vision for most activities, most marine animals rely on sound for survival and reproduction. Scientific investigations of many marine animals (including mammals, fish, and even some invertebrates) have shown that the production and reception of sounds are critical to various aspects of their life histories. Human-produced sound has the potential to interfere with various important biological functions of marine animals. The range of resulting adverse impacts is highly dependent on characteristics of the sound source, the environment where the sound occurs, and the animals receiving the sounds. Marine animals such as large whales, many fish, and some seals and sea lions are particularly vulnerable to adverse impacts from incidental shipping noise because they primarily use the same low frequency sounds as that generated by commercial ships for such things as communication and/or to perceive their environments” (IMO/MEPC 58/19).

4.2 Noise regulation

The problem of anthropogenic noise emissions in the sea has been assessed only in recent years. This problem has been analysed mainly at a regional level, in particular for restricted areas where there is a higher concentration of species of marine mammals or fishes. The national and international regulations reviewed to date often do not address underwater noise quantitatively in the sense of specifying acceptable underwater source levels but instead restrict activities that are determined to harass or harm marine animals.

4.2.1. International Framework

At an international level there are several associations which deal with the protection of marine mammals. In some of their regulations or treaties they cover underwater sound. In the following a short review of some of these regulations as they relate to underwater noise is presented.

United Nations Convention on the Law of the Sea (UNCLOS). The most widely recognized set of international regulations that can be applied to underwater noise are those derived from the United Nations Convention on the Law of the Sea (UNCLOS). Although this document (UNCLOS, 1982) is more than 200 pages and addresses a very wide range of law-of-the-sea issues, neither the word "noise" nor "sound" (as in underwater sound) appear. Invoking that UNCLOS grants the right of individual governments (states) to regulate anthropogenic underwater noise within their sovereign waters derives from careful inference of wording of Articles *1(1)(4)* and *192*. *Article 1(1)(4)* defines *"pollution of the marine environment*" as "*.. the introduction by man, directly or indirectly, of substances or energy into the marine environment, including estuaries, which results or is likely to result in such deleterious effects as harm to living resources and marine life…".* *Article 192*, under the subheading of *"Measures to prevent, reduce and control pollution of the marine environment"* states in part that *"States shall take, individually or jointly as appropriate, all measures consistent with this Convention that are necessary to prevent, reduce and control pollution of the marine environment from any source, using for this purpose the best practicable means at their disposal and in accordance with their capabilities, and they shall endeavor to harmonize their policies in this connection"*. Hence, by recognizing underwater sound as a pollutant by virtue of the *"substances or energy*" wording in *Article 1(1)(4)*, then *Article 192* grants each nation the authority to *prevent, reduce and control such pollution*, etc.

The International Maritime Organization (IMO). The IMO /MEPC approved the inclusion of 'noise from commercial shipping and its adverse impacts on marine life' as a '*new high-priority item*' (IMO/MEPC 58/19, 2008) and established a Correspondence Group with the specific task to: '*identify and address ways to minimize the introduction of incidental noise into the marine environment from commercial shipping (...) and, in particular, develop voluntary technical guidelines for ship-quieting technologies as well as navigation and operational practices*'. Hence, the work of the Correspondence Group is confined to the development of *non-mandatory technical guidelines* but was not instructed to develop a regulatory framework for this issue.

In the reports IMO/MEPC 59/19 (2009) and 60/18 (2009), the Corresponding Group stated that noise in the low frequency range of 10 Hz to 1 kHz has the biggest impact on the marine biodiversity. Great interest also existed regarding the 50 Hz peak of ship noise, which is always present and especially notable at low speeds although the main source of this peak was not fully clear. Different noise control technologies were discussed for propeller, machinery and hull silencing and it was estimated that an overall reduction of about 20 dB in noise can be achieved through optimization of machinery and propeller noise mechanisms.

The IMO/MEPC.1/Circ. 833 (7 April 2014), with a view to providing guidance on the reduction of underwater noise from commercial shipping, and following a recommendation made by the Sub Committee on Ship Design and Equipment, approved the annexed "*Guidelines for the reduction of underwater noise from commercial shipping to address adverse impacts on marine life*", MEPC 66/17 (2013).

These non-mandatory Guidelines are intended to provide general advice about reduction of underwater noise and focus on the primary sources of underwater noise such as associated with propellers, hull form, onboard machinery, and operational aspects. Moreover, a specific section addresses the use of numerical tools for noise prediction indicating that CFD can be used to predict the flow characteristics around the hull and appendages, thus providing the wake field in which the propeller operates and propeller analysis methods, such as lifting surface theory, or CFD, can be used for predicting cavitation. Finally, mention is made that Statistical Energy Analysis (SEA) and Finite Element (FE) methods can be used to solve the vibro-acoustic problem at high and low frequency, respectively.

Other Organizations. Declarations regarding the impact of shipping noise have also been made by many other international organizations, among them, the Convention on the Conservation of Migratory Species of Wild Animals (CMS), the International Union for the Conservation of Nature (IUCN), the International Whaling Commission (IWC), the International Council for the Exploration of the Sea (ICES), the International Fund for Animal Welfare, the Whale and Dolphin Conservation Society (WDCS), and the International Ocean Noise Coalition.

4.2.2. Regional and National Framework

The European Union (EU). In 2004 the European Parliament adopted a Resolution on the environmental effects of high-intensity active naval sonar. This Resolution calls upon the European Union (EU) and its Member States to adopt a moratorium on the deployment of high intensity active naval sonar until a global assessment of their cumulative environmental impact on marine mammals, fish and other marine life has been completed.

The recent EU Marine Strategy Framework Directive (2008/56/EC) specifically mentions the problem of noise pollution and provides a legal framework for addressing this issue. The Directive represents the first international legal instrument to explicitly include anthropogenic underwater noise within the definition of pollution (Article 3 (8)), which needs to be properly mitigated in order to achieve the good environmental status (GES) of European marine waters by 2020 (Article 1).

The Directive identifies 11 environmental descriptors to achieve the GES, and the 11th reads: "the introduction of energy, including underwater noise, is at levels that do not adversely affect the marine environment". Moreover, the EU Commission Decision of September 2010 provides the following descriptor (11.2) for 'continuous low frequency noise' (as generated by shipping): "*Trends in the ambient noise level within the 1/3 octave bands 63 and 125 Hz (centre frequency) (re 1μPa RMS; average noise level in these octave bands over a year) measured by observation stations and/or with the use of models if appropriate*". With this Directive, enforced from 2014, underwater noise is an issue of great relevance and all member states are obliged to provide an evaluation of the "good status" of their seas based on those descriptors. The directive is discussed in detail by Tasker (2010), Piha (2012) and VanderGraaf et al. (2012).

Starting in 2012, two multinational collaborative projects are partly funded by the 7th Framework Programme of the European Commission with the goal to develop tools to investigate and mitigate the effects of underwater noise generated by shipping. These projects are SONIC (www.sonic-project.eu) and AQUO (www.aquo.eu).

The Convention for the Protection of the Marine Environment of the North-East Atlantic (the "OSPAR Convention"). The OSPAR Convention is the current legal instrument guiding international cooperation on the protection of the marine environment of the North-East Atlantic. Work under the Convention is managed by the OSPAR Commission, made up of representatives of the Governments of 15 Contracting Parties and the European Commission, representing the European Community. While programs and measures relating to questions of fisheries management and shipping cannot be adopted by the OSPAR Commission, issues concerned with the impact on biodiversity are drawn to the attention of the competent authorities and relevant international bodies. In particular, the OSPAR Commission has an Agreement of Cooperation with the IMO.

OSPAR published a report (OSPAR, 2009a) on the impact of noise considering many different noise sources. Specific publications refer to shipping noise (OSPAR, 2009b; OSPAR, 2011) and to the impact of small touristic vessels (OSPAR, 2008).

The Agreement on the Conservation of Small Cetaceans of the Baltic and North Seas (ASCOBANS). ASCOBANS was concluded in 1991 under the auspices of the Convention on Migratory Species and entered into force in 1994. In February 2008, an extension of the agreement area came into force which changed the name to "Agreement on the Conservation of Small Cetaceans of the Baltic, North East Atlantic, Irish and North Seas".

In Resolution N. 5 "Effects of Noise and of Vessels "(4th meeting of the parties to ASCOBANS 2003), Parties and Range States are requested to introduce guidelines on measures and procedures for seismic surveys, and invited to conduct further research into the effects on small cetaceans of: vessels, particularly high speed ferries; acoustic harassment devices such as those used in fish farms and elsewhere; offshore extractive; and, other acoustic disturbances.

The Agreement on the Conservation of Cetaceans of the Black Sea, Mediterranean Sea and Contiguous Atlantic Area (ACCOBAMS). The ACCOBAMS is a cooperative tool for the conservation of marine biodiversity in the Mediterranean and Black Seas. Its purpose is to reduce threats to cetaceans in Mediterranean and Black Sea waters and improve our knowledge of these animals. ACCOBAMS was concluded in the auspices of Convention on Migratory Species in 1996 and entered into force in 2001.

In 2010, under resolution 4.17, guidelines to address the impact of anthropogenic noise on cetaceans in the ACCOBAMS area were adopted and a working group was established that will focus on the mitigation of noise impact issues resulting from sonar, seismic surveys, coastal and construction works, and maritime traffic including commercial shipping.

In 2012 the ACCOBAMS and ASCOBAMS noise working groups joined. The working group has produced among other things a review of various international guidelines (Maglio, 2013).

United States. The United States (US) Congress passed three cardinal pieces of legislation that form the framework for protecting marine mammals and marine ecosystems from, in part, the harmful effects of anthropogenic noise. These are the Marine Mammals Protection Act (MMPA, 1972), the Endangered Species Act (ESA, 1973), and the National Environmental Policy Act (NEPA, 1969). While each piece of legislation is intended to address separate environmental issues, the actions taken to implement them are often overlapping contributing greatly to a complex mosaic of legal requirements.

The MMPA established a moratorium on the taking of marine mammals in U.S. waters and explicitly defines "take" to mean "to hunt, harass, capture, or kill" any marine mammal or attempt to do so. The inclusion of harassment in the definition is wide reaching and includes potential adverse effects due to anthropogenic noise. However, exceptions to the moratorium can be made for particular activities. Enforcement of the MMPA is divided between the Department of Interior's US Fish and Wildlife Service (UWFWS) and the Department of Commerce's National Marine Fisheries Service (NMFS), which is under the National Oceanic and Atmospheric Administration (NOAA).

The ESA provides for the conservation of species that are endangered or threatened throughout all or a significant portion of their range, and the conservation of the ecosystems on which they depend. Similar to the MMPA, the NMFS and the USFWS share responsibility for implementing the ESA.

The NEPA requires full disclosure of possible environmental impact, alternatives and mitigation measures for any federal actions and thus has direct impact on military activities in the ocean.

Other US agencies that also have a regulatory or enforcement role related to anthropogenic sound include: the Marine Mammals Commission (MMC); the Minerals and Management Service (MMS, under Department of Commerce) and the US Navy.

Moreover, the US Coast Guard is one of the five Armed Services of the US and enforces a wide range of maritime safety, security and environmental policies of the US. Issues related to habitability concerns due to ship-borne noise levels would in part be handled by the Coast Guard.

It is noted that while there are numerous and wide ranging environmental laws within the US that can be used to regulate underwater anthropogenic noise, enforcement of these laws is greatly hampered by the lack of quantifiable metrics. Wording of regulatory laws are necessarily in the form of phrases such as "to take" and "to hunt, harass, capture, or kill". As related to enforcement of underwater noise regulations the terms "harass" and "kill" are most applicable but to date there is no clear definition, or even understanding, of the character of anthropogenic noise that results in harassment, or even at times resulting in death. For example, the reason(s) for the mass beaching of mammals that is often observed is strongly debated one group attributing the cause to military activities (sonar use) and another to biological irregularities due to either natural events or presence of (other) oceanic pollutants.

4.3 Noise standards of ICES and DNV

ICES methodology. The International Council for the Exploration of the Sea (ICES) derived noise limits for research vessels (Mitson, 1995). The ICES proposal considers the cod audiogram and selects the noise limit by considering the lowest point of the curve at 30 dB over the threshold sensitivity. The curve was interpreted as the limit over which behavioral effects (escaping) started to appear. The selected point at 200 Hz represents the frequency of maximum sensibility for cod, the corresponding level was set as the limit for radiated underwater sound from a research vessel, free running at 11 knots, at a target distance of about 20 m. The noise limit at 1 m was obtained applying the spherical dispersion law.

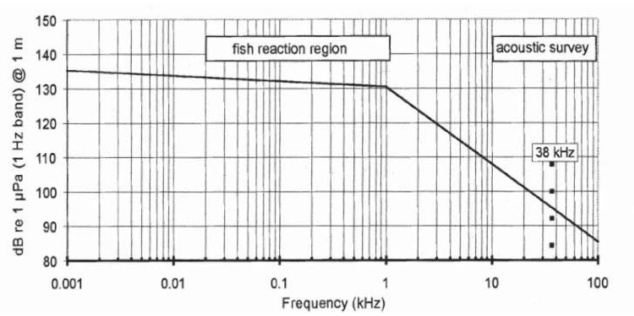

Figure 4.1 Proposed underwater radiated noise limit at 11 knots free running for all vessels used in fisheries research (from Mitson, 1995).

To provide an allowable underwater noise source level (SL) spectrum the ICES procedure uses the prior discussed level at 200 Hz as a reference value and provides two simple power-law relationships, one for lower frequencies that passes through the 200 Hz value and one for higher frequencies (see Figure 4.1). The slope of these lines is selected to generally follow the frequency dependency of measured underwater ship noise. The lower frequency relationship covers the frequency range of 1 Hz-to-1 kHz and the higher covers from 1 kHz-to-100 kHz and are given as:

$$SL = 135 - 1.66\log_{10}\left({}^{f_{\mathrm{Hz}}}\!/_{1\ \mathrm{Hz}}\right)$$

$$\text{for } 1\ \mathrm{Hz} \le f \le 1\ \mathrm{kHz}, \ \text{ and}$$

$$SL = 130 - 22\log_{10}\left({}^{f_{\mathrm{kHz}}}\!/_{1\ \mathrm{kHz}}\right)$$

$$\text{for } 1\ \mathrm{kHz} < f \le 100\ \mathrm{kHz}$$

where, SL is the underwater noise source level given in dB re: 1 µPa/Hz at 1 m.

DNV SILENT Class Notation. DNV (DNV, 2010) has recently issued new rules to ensure low underwater sound emissions from ships. This is the first attempt made by a Classification Society to fix limits for underwater noise radiated from commercial ships.The rules apply to vessels which need a low environmental impact and/or to ships which operate with hydro-acoustic equipment. In particular five cases are taken into account, each with a different limit curve: i) Acoustic (ships involved in hydro-acoustic measures); ii) Seismic (ships involved in seismic surveys); iii) Fishery; iv) Research; and, v) Environmental (any vessel which require controlled environmental noise emission).

The curves for the mentioned categories report maximum allowable noise levels (in dB re: 1µPa at 1m) versus frequency (1/3 octave resolution). In the case of the acoustic, fishery and environmental categories two different curves are given depending on the operational conditions of the ship. The curve relative to research vessels corresponds to the ICES one except for the format (third octave bands instead of narrowband (1 Hz) and the shape of the curve for frequencies below 25 Hz, which contains less restrictive limits.

5 PROPELLER NOISE PREDICTION METHODS

The noise produced by a propeller has been of considerable importance to warship designers and military strategists for many years. In recent years this subject has been of more importance for merchant shipping and is likely to maintain increased importance in the future with the consideration of its influence to marine environment. Propeller noise comprises a series of periodic components, or tones, at blade rate and its multiples, together with a spectrum of high-frequency noise due to cavitation and various edge effects. With the development of computer capabilities, more efforts are paid in recent years to developing computational prediction methods.

5.1 Propeller non-cavitating noise

5.1.1. Prediction Methods of Propeller Discrete Noise

Predicting the low-frequency discrete spectrum of non-cavitating propeller noise has been the subject of research for many years. Most of the prediction methods are focused on obtaining the unsteady propeller forces or fluctuating pressures of the flow field using CFD calculations. In turn, these pressures and forces are used as the sources to acoustic methods of predicting the discrete tonal noise of a propeller.

Seol, Suh and Lee (2002, 2004, 2005) present a numerical method to predict non-cavitating tonal noise of an underwater propeller. The noise is predicted using a time-domain acoustic analogy. The flow field is analyzed with a potential-based panel method and the time-dependent pressure is used as input to the Ffowcs Williams–Hawkings (FW-H) formulation to predict the far-field acoustics. In the study, the dominant noise sources of the propeller are identified and used as the basis for proper noise control strategies. In a similar way, Sharma and Chen (2013) put forward a numerical approach for predicting tonal noise for counter-rotating rotors. In this study, Reynolds Averaged Navier-Stokes (RANS) simulations are used to obtain near field description of the noise sources.

Gennaretti, Testa, and Bernardini (2012) proposed a novel frequency-domain formulation for the prediction of tonal noise emitted by rotors. It is derived from the Farassat (boundary integral) Formulation 1A for the time-domain solution of the FW-H equation, and represents noise as harmonic responses to body kinematics and hydrodynamic loads via frequency-response-function matrices. The method has been used to analyze a marine propeller working in non-uniform inflow. This approach is particularly suitable for noise control purposes in view of the definition of the relationship between noise harmonics and blade control variables.

Most methods for predicting marine propeller noise are for far field and as such various higher-order terms are neglected in the numerical solutions. However, in the near field, the higher–order terms play important roles. The solutions of blade rate underwater noise induced by the unsteady force of a marine propeller in the time domain is described in the paper by Kehr and Kao (2004). The method can be used not only for computing far field acoustic pressure, but also for near field pressures.

Tonal noise of a propeller can also be assessed using the measured fluctuating forces of the propeller. In GTH (Fréchou et al., 2000), an unsteady thrust dynamometer is integrated in the shaft close to the propeller hub. The sensor is a piezoelectric crystal that provides a high stiffness and mounts on the shaft centreline with steel hemisphere to be insensitive to side forces and bending. The system is able to measure very low thrust fluctuations ($\triangle T/T<<1\%$).

Jessup (1990) conducted unsteady force measurements in the DTRC 24-inch water tunnel. Three-bladed propellers were operated behind 3-, 6-, 9-, and 12-cycle wake screens generating blade rate and multiple axial wake inflow harmonics. A six-component unsteady propeller dynamometer was used. The upper limit of frequency response of the measurement system was around 800 Hz in thrust and torque and around 200 Hz for the side force components.

5.1.2. Prediction Methods on Propeller Broadband Noise

Propeller broadband noise has been an extensive aeroacoustic research topic for decades, both experimentally and theoretically. In recent years, the increasing computational capabilities led to the extension and application of various approaches to marine propellers.

Howe (1978) presented a detailed review of the various theories of predicting trailing edge noise and categorized the methods into three groups: (1) theories based on the Lighthill's acoustic analogy; (2) theories based on the solution of special problems approximated by the linearized hydrodynamics equations; and, (3) ad hoc models involving postulated source distributions for which strengths and types are empirically determined.

Casper and Farassat (2002, 2004) proposed a trailing edge noise prediction method 'Formulation 1B', which is a solution of the loading source term of the FW-H equation, and validated it against measurements from a NACA 0012 aerofoil in a low Mach number flow. Such time domain methods allow a total decoupling of the acoustic signal from the aerodynamics. As such, the input for acoustic predictions can use experimental measurements or computational fluid dynamics (CFD) solutions. The authors (Farassat and Casper, 2012) also derived a new formula named Formulation 2B for the prediction of broadband noise that can be applied to rotating blades and airframes, etc.

For prediction of propeller broadband noncavitating noise, CFD simulations combined with acoustic methods can be applied. Kato (2011) applied calculations from a Dynamic Smagorinky Model (DSM) of a Large Eddy Simulation (LES) to the Helmholtz equation to solve for the acoustical pressure. Both solvers are designed for parallel computations with up to one million processing cores. Validation studies for basic flows show that the fully-resolved LES can predict fluid flow with an accepted level of accuracy. Sound pressure spectra radiated from a small seven bladed industrial fan was reasonably predicted with such methods as compared with experimental results.

Chen et al. (2012) developed a method for broadband noise prediction for hydrofoils or marine propellers. The turbulent flow around a propeller is calculated as a time series by LES and then the broadband noise is predicted using the FW-H equations. The method was validated against measurements for a hydrofoil and a model propeller (210 mm diameter). The predicted propeller broadband noise showed reasonable agreement with the measurements. Chen et al. (2013) used the method to calculate the broadband noise of five different hydrofoils. The relationship between hydrofoil broadband noise and the hydrofoil thickness and camber distributions were discussed. An optimized hydrofoil was presented for which the broadband noise is about 4 dB lower than for a foil of the NACA series.

Kellett, Turan and Incecik (2013) used a CFD-based unsteady RANS hydrodynamic prediction approach, coupled with the FW-H equation for ship radiated underwater flow noise modelling. The commercial CFD software StarCCM+ was used for flow field simulation. In the paper a variety of modelling setups were considered, such as propeller representation modelling, with or without free surface etc., to ascertain which should be modelled for different applications and required levels of prediction accuracy. The hydroacoustic behaviour of a marine propeller in a non-cavitating open water condition is examined in Ianniello et al. (2013), by coupling a RANS hydrodynamic solver to a hydroacoustic code designed to implement different solution forms of the FW-H equation. The numerical results suggest that the underwater pressure field seems to be significantly affected by flow nonlinearities, while the contribution from the linear terms (the thickness and loading noise components) is dominant only in very limited region of space. Similar conclusions are drawn in Ianniello et al. (2014a) when considering the underwater radiated noise from a complete ship scaled model in a steady course. Moreover, the effect of scattering from the hull surface is also highlighted. In Ianniello et al. (2014b) the methodology has been applied to a large ferry and satisfactory validation by comparison with full scale data was obtained.

5.2 Propeller cavitating noise

Propeller cavitation noise is generated by two kinds of acoustical mechanisms. The low frequency range is characterised by tonals at harmonics of the blade rate frequency and a broadband hump due to the large scale cavity dynamics whereas the noise in the high frequency range is due to the collapse of bubbles. Moreover, the noise level depends on the type of cavitation on a propeller. For example, back sheet cavitation and tip vortex cavitation have different noise signatures. Great effort has been put into predicting cavitation itself using computational methods and significant advances have been made of which some are already being applied in industry. However, up to now, computational prediction of cavitating flows is still a difficult problem in hydrodynamics, especially for the cases of instantaneously cavitating vortices or for the process of cavitation collapse. Computational methods are being applied to translate the cavitation dynamics to radiated noise but the possibilities and limitations for accurate noise predictions needs to be further assessed.

5.2.1. Prediction Methods of Propeller Sheet/Cloud Cavitating Noise

Kamiirisa (1998) developed a simulation method on the basis of bubble dynamics for predicting sheet cavitation noise of a marine propeller. For the low frequency range, after obtaining the variation of cavitation volume during one revolution of a propeller by applying computing techniques such as lifting surface theory, the sound pressure level is calculated using the FW-H equation. In the paper the cavitation is considered as a large spherical bubble. The high frequency range is constructed by summing up radiated noise from each cavitation bubble which occurs at the end of cavitation after the collapse of sheet cavitation. The size distribution of cavitation bubbles is proposed to be represented by the beta distribution. The trailing thickness of sheet cavitation is treated as uniform along a radial direction of the propeller blade. The simulation result compares favourably to full scale measurement for the SEIUN MARU full scale propeller.

Salvatore and Testa (2006) developed an integrated hydrodynamics/hydroacoustics approach for marine propeller sheet cavitation noise. The hydrodynamics model is based on a boundary element method (BEM) that is a potential flow formulation. A sheet cavitation model using a surface tracking approach is applied to estimate the transient cavity pattern on the surface of propeller blades. Propeller cavitation noise is studied through a general hydroacoustics formulation based on the FW-H equation. Hallander et al. (2012) presented work of the EU project SILENV and compared results of underwater radiated noise of a propeller obtained from different prediction methods, including URANS/FW-H, SYSNOISE/BEM, lifting-surface/FW-H, and a semi-empirical approach with sea trial data for an LNG ship under cavitating conditions.

Seo and Lele (2009) investigated cloud cavitation and cavitation noise on a hydrofoil section. The density based homogeneous equilibrium model and high-order numerical methods based on a central compact scheme were employed to resolve the cloud cavitation phenomena and the pressure waves generated by cloud cavitation. The governing equations are the compressible RANS equations for the gas/vapour-liquid mixture. Two-phase flow physics is treated by a linearly combined equation of state allowing the compressibility effects in liquid and gas phases, and application of the one-equation Spalart-Allmaras turbulence model is applied. Overall, the simulated pressures agreed well to measurement.

Salvatore et al. (2009) presented a numerical prediction method for non-cavitating and cavitating noise of propellers operating in a wake field. Propeller hydrodynamics is described by a BEM coupled with a nonlinear unsteady sheet cavitation model. Hydroacoustic models are based on a standard Bernoulli equation for incompressible flows and on the FW-H equation with a Transpiration Velocity Model to account for blade cavitation effects. While fair agreement between results from the two formulations is found for a non-cavitating propeller configuration, quantitative differences are observed for cavitating flow noise predictions between the Bernoulli and FW-H models. Numerical uncertainty in the evaluation of cavity pattern could have a strong impact on prediction of radiated noise levels.

5.2.2. Prediction Methods of Propeller Tip Vortex Cavitating Noise

With the hypotheses that: 1) there is strong dependence between averaged diameter of tip vortex and propeller noise emission, and 2) tip vortex diameter is a function of blade tip loading and static pressure (cavitation number), researchers at DNV use a tip vortex index “TVI” for predicting tip vortex noise and acoustic pressure (Raestad, 1996). The acoustic pressure is assumed to be a function of the volume acceleration of the tip vortex cavities from each blade. DNV established a database for inboard propeller noise on twin-screw passenger ships, including ferries and cruise liners (Raestad, 1996). Using the TVI method, predicted results gave quite good agreement with measured inboard noise data. The experience of DNV from high-powered cruise ships indicates that the noise generated by propeller tip vortices is much more important for inboard noise than was previously realized. A semi-empirical method similar to the TVI method has been developed by Bosschers (2013) for the prediction of the low frequency broadband hull pressures and far field underwater radiated noise of twin screw vessels. In general, acceptable agreement with experimental data of model scale tests and full scale trials is obtained.

Pustoshnyy et al. (2012) discussed the physical aspects of marine propeller broadband noise based on available publications, as well as experimental data obtained by KSRI. From the research, the authors discussed the TVI method and the effect of parameters used in TVI. The paper proposed that cavitating tip vortex in non-uniform flow may be a source of broadband pressures, especially when vortex bursting takes place, and that circulation gradients at the blade tip during its motion in non-uniform velocity field may be considered an important parameter of broadband pressure effect.

Park et al. (2009) numerically analyzed tip vortex cavitation behaviour and sound generation of a hydrofoil. A numerical scheme combining an Eulerian flow field computation and Lagrangian particle trace approach was applied to simulate tip vortex cavitation. The flow field was computed using a hybrid method which combines a RANS solver with a Dissipation Vortex Model. The trajectory and behaviour of each cavitation bubble were computed by Newton's second law and the Rayleigh－Plesset equation, respectively. Calculated volumes of the cavitation bubble and the trajectory were used as input to noise prediction methods. A bubble noise model, which assumes that cavitation bubbles behave as monopole sources, was used to calculate the noise. The relationship of cavitation inception, sound pressure level, and nuclei size was studied at several cavitation numbers. The study showed that cavitation inception of smaller nuclei is more sensitive to the change of cavitation number, and cavitation noise level due to cavitated smallest nuclei has the most influence on overall tip vortex cavitation noise.

6 SURVEY ON NOISE MEASUREMENTS

As a means to understand the activity of ITTC members and associated industries in hydrodynamic noise as it relates to the shipping industry, the committee developed a questionnaire designed to survey activity in the areas of full scale and model scale noise measurement methods. As the work in this area is principally empirical, it was felt soliciting information related to measurement methods would best establish the baseline for activities related to hydrodynamic noise.

The questionnaire, "Questionnaire on Noise Measurements Methods", consisted of two parts: questions related to full scale measurements, and questions related to model scale measurements. The Full Scale Noise Measurement Methods section consisted of five major groups: Site Information, Full Scale Propeller, Hydrophone Information, Data Acquisition and Processing, and Correction Methods. The Model Scale Measurement Methods section was divided into six major groups: Facility Information, Model Setup, Hydrophone Information, Definition of Test Conditions, Data Acquisition and Processing, and Scaling Methods. The responses that were received to the questionnaire are provided in tabular form at the end of this report. The survey was conducted on-line by posting the questionnaire to a common survey webpage which provided results in various collated forms. It is noted that a few responses were received as scanned versions of hand annotated questionnaires.

Summary of Responses. All ITTC member organizations along with industrial and academic groups known to be involved in hydrodynamic noise work were invited to participate in responding to the questionnaire. Organizations from a total of 14 countries provided completed responses for at least one of the two parts to the questionnaire resulting in a total of 11 responses to the full scale measurements section and 18 responses to the model scale measurements section. Following is a listing of the countries from which completed responses were received with the number of responses provided in brackets following the country name with the first number being the number of responses to the model scale measurements portion and the second number the number of responses to the full scale measurements portion: China (2,0), France (0,1), Germany (1,1), Iran (1,0), Italy (2,1), Japan (3,2), Korea (2,2), Netherlands (1,2), Norway (1,0), Russia (1,0), Spain (0,1), Sweden (2,0), Turkey (1,0), and USA (1,1)

6.1 Full Scale Noise Measurement Method

A total of 11 responders from 8 countries completed the Full Scale Noise Measurement Methods portion of the questionnaire.

Testing is generally done using either fixed and/or mobile measurement equipment with the majority being of the latter. It is noted that the three fixed sites also employ mobile equipment and were those that predominantly support measurements of military ships. Additionally, five organisations reported using on-board measurement equipment, four of which use such equipment in addition to fixed and/or mobile range equipment. Approximately half of the facilities utilize a hydrophone array and there is a nominally even number who deploy the measurement hydrophones from the sea bottom, from a buoy, or from a vessel. Four organisations reported measurements capabilities in water depth exceeding 100 m which may (arbitrarily) be considered deep water.

Attention to hull and propeller surface conditions in support of acoustic testing varied amongst organisations. Nominally half report that hull conditions are checked, that propeller conditions are checked and that the propeller surface is polished prior to testing.

All organisations use commercially available omni-directional hydrophones that generally provide measurement capabilities to a frequency of 100 kHz or greater. Each organisation but one reports the type of hydrophone calibration procedures used. Hydrophone arrays generally amounted to the use of 3 hydrophones with two organisations reporting using arrays consisting of 10 or more hydrophones.

Nearly all organisations report making noise measurements up to a frequency of approximately 50 kHz and as low as 10 Hz. All but two report performing instrumentation calibrations either and/or before testing. The definition of acoustic source used during testing varied amongst the organisations and only 4 organisations reported using some type of source localisation technique. A number of different data reporting formats are used with many organisations using multiple formats. The formats ranged from one-third octave band levels to narrow-band levels, each with or without being range corrected. For the question of ‘expected uncertainty for measurements’, all 5 who provided an answer gave a value of 3 dB or less. And, for the question on ‘confidence level on the quality of the test results’, based on a scale of 1-to-10 (10 being very confident) all 9 responses were at a level of 7 or greater with none being higher than 9.

Regarding whether measured levels are corrected for background noise and propagation-related issues, 6 out of 10 reported measurements are corrected for background noise, 8 out of 10 do not correct for environment-related propagation losses (i.e., absorption), 8 report that a $20\log_{10}(r/1m)$ adjustment for range correction is made. Only 3 report making an adjustment to correct for free-surface reflection. For the question of 'expected uncertainty for predicted noise source level' the 5 who responded gave a value of ± 3 dB or less. And, for the question of 'confidence level on the quality of correction methods' based on a scale of 1-to-10, of the 8 who responded the highest value was 9, the lowest 5, and the median being 7.

While relatively low values of uncertainty were given for both the measurement of full scale sound pressure levels and prediction of noise levels, as would be expected from a properly implemented measurement regime, a few notes of caution are warranted regarding estimates of ship noise levels for conditions other than those for which measurements are made. Specifically, the physics of underwater noise generation is critically dependent on issues related to the ship condition and its performance in a given seaway. Additionally, propagation of ship noise to a specific observation point is dictated by propagation issues related to the condition of the free-surface, sound speed variations in the water column, and conditions of the sea bottom. The combination of these generation and propagation issues make estimations of ship noise in other locations, based on measurements at an acoustic range, subject to an uncertainty bound appreciably greater than the combined individual uncertainties given for the measurements or predictions.

It is felt that the questionnaire related to full scale noise measurement methods provided a good review of the activities within the shipping community. There is clearly a wide range of approaches, methods and procedures for making underwater noise measurements with the variations being driven in large part by the objectives of the testing and the ocean site available for such measurements.

6.2 Model Scale Noise Measurement Methods

In total 18 model basins from 12 countries responded to the questionnaire. The distribution of the type of facility used for noise measurements is shown in Figure 6.1. The majority operate a closed jet-type cavitation tunnel of which the length and width of the test-sections are presented in Figure 6.2. The large size facilities (width 2 m or larger) use a full ship model for the wake generation while the smaller size use a dummy model with wire screen or a wire screen alone.

The ship wake distributions used is presented in Figure 6.3. Almost half of the organisations are simulating both a model scale wake and a full scale wake in their facility. The wake field is measured by pitot tubes and/or LDV while 5 respondents use PIV as well.

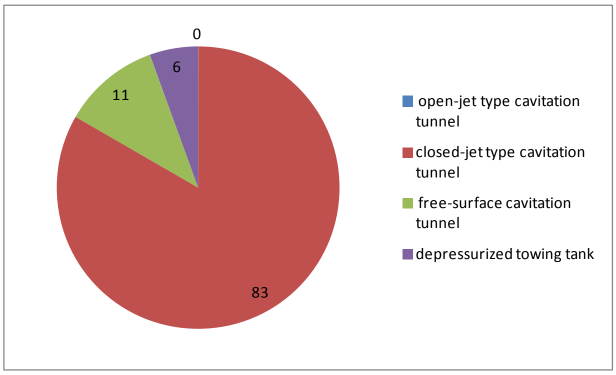

Figure 6.1 Distribution in [%] of facility type.

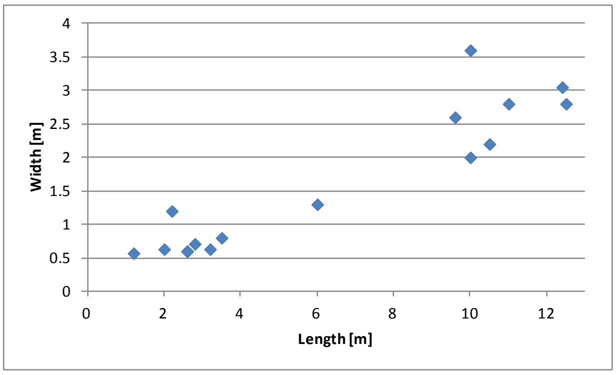

Figure 6.2 Variation in length and width of the test-section of the cavitation tunnels.

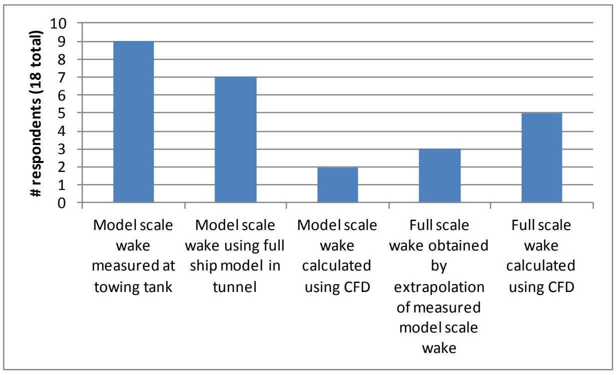

Figure 6.3 Ship wake distribution used (more than one answer possible).

All model basins use similar commercially available omnidirectional hydrophones of piezoelectric type from Brüel & Kjaer or Reson. Flush mounted hydrophones are used only supplementary. The specifications of these hydrophones (frequency range, dimensions, operating pressure and temperature range) developed for full scale measurements easily comply with the requirements of a cavitation test facility.

Most of the organizations (16) have specific hydrophone calibration procedures and conduct calibration before and/or after measurements, or regularly using a hydrophone calibrator. Only 2 organizations practice the more elaborate water tank calibration. During noise measurements the hydrophones are located mainly in acoustic chambers (7), outside walls or windows (4), flush mounted to walls or windows (4), on a rake respective in the flow (3) or mounted at the bottom of a basin (1). The dimensions of the location where hydrophones are mounted vary with the test section dimensions from small or medium sized to large scale facilities.

The purpose of noise measurements is predominantly the determination of cavitation noise (17), but includes non-cavitating noise (13) and cavitation inception (10). Typically one or two hydrophones (7) are used, but in case where array measurements are made (3) up to 56 hydrophones are used. The hydrophones are generally located at the propeller plane (11), downstream (8) and/or upstream (6). Noise measurements of nearly all model basins (17) are supported by background noise measurements to determine, and allow correction for, facility dependent noise levels and by supplementary investigations like cavitation observation (17), cavitation inception (13), hull pressure pulses (10) and others (3).

The propeller load condition is chosen according to the design point (11) or the model test result (8) corrected to full scale (16). The test condition is mostly adjusted by the thrust coefficient K_T (14) and rarely by the advance ratio J (3) or torque coefficient K_Q (1). The reference point of the cavitation number varies from shaft center line to 0.95R at 12:00. Depending on the type of test facility the typical water speeds are in the range of 1 to 3.5m/s (free surface) or 4 to 8m/s (closed jet). Most of the organisations monitor the water quality (14), mainly by measurement of the dissolved oxygen content (10).

For data acquisition, the measuring time is mostly more than 20 seconds (10), or 10 seconds (4). For the computation of spherical spreading loss or distance normalisation, most model basins (12) take the shaft centre as the location of the acoustical source of a propeller, while others use 0.7R (2), 0.8R (1) or 1.0R (1) at 12:00. The majority of the facilities (15) use anti-aliasing filters for signal conditioning.

The reported results of measurements are presented in the format of 1/3 octave bandwidth (14), narrowband normalized to 1 Hz bandwidth (13), 1/3 octave converted to 1 Hz bandwidth (4), normalization to 1 m distance (9) and harmonics (3).

About half of the organizations (7) perform uncertainty analysis of the noise signals according to ITTC general guidelines, while 2 organizations use their own uncertainty analysis method, and 8 organizations do not as yet perform uncertainty analysis.

More than half of the organizations (10) investigated the reverberation of their facility. Corrections for background noise are commonly applied. The distribution of signal corrections due to facilities is shown in Figure 6.4. The uncertainty level for model scale noise measurements is in the range 1 to 5 dB, while the confidence level is in the range of 5 to 10 (1 very uncertain, 10 very confident).

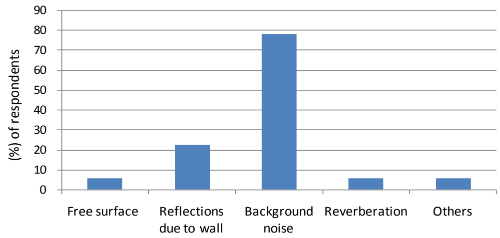

Figure 6.4 The distribution of corrections.

Thirteen organizations make full scale noise predictions using the model scale measurement, and most of them (8) use the ITTC-1987 extrapolation procedure or something similar. Only one facility reported to have a scaling method for tip vortex cavitation noise. Compared to the measured model scale noise data, there is a larger level of uncertainty for the scaling method which is in the range 2 to 10 dB.

The questionnaire related to model scale noise measurement provided a good review of the activities of the different facilities within ITTC members. For cavitation noise measurement at model scale, each facility has its own noise test procedure whiles similar methods for installation, test condition, data acquisition and scaling are used depending on the size and conditions of the facility. Some new approaches have been reported such as full scale wake simulation. While the limitations of the test facility in capturing the dynamics and noise of cavitation are acknowledged, there is a larger level of uncertainty in the noise scaling procedure than in the measured noise data. This implies that more validation against full scale data is necessary.

7 GUIDELINES FOR NOISE MEASUREMENTS

7.1 Full Scale Guidelines

Based on responses to the full scale measurements questionnaire and a review of existing full scale guidelines, both established and in development, it is recommended that the ITTC guideline for underwater noise measurement of full scale ships follow the standard ISO/PAS 17208-1:2012(E). This standard was drafted by international experts in the field of underwater ship noise measurements and provides general requirements for measurements in deep water. It is noted that guidelines for measurement of ship underwater noise in shallow water are currently being pursued by numerous international organizations but such standards will be slower to finalize due to the much more complex nature of measurements in shallow water where noise propagation issues related to bottom affects must be accounted for.

The ISO standard provides guidelines based on the Grade of measurement that is needed. Three Grades are addressed: Grade A–Precision Method, Grade B–Engineering Method, and Grade C–Survey Method. Following the guidelines for each Grade result in an achievable measurement uncertainty of 1.5 dB, 3.0 dB, and 4.0 dB for Grades A, B, and C, respectively and measurement repeatability of ±1.0 dB, ± 2.0 dB, and ± 3.0 dB, for each Grade, respectively. The standard specifically addresses the topics of: Instrumentation, Measurement Requirements & Procedures, Post – Processing, Measurement Uncertainty, and Reporting, each as related to the Grade of measurement to be obtained.

Considering the international interest and activities regarding measurement of underwater ship noise it is recommended that the Recommended Procedures and Guidelines developed by this Committee (7.5-04-04-01: Underwater Noise from Ships, Full Scale Measurements) be revisited and updated as necessary when further International Standards are established.

7.2 Model Scale Guidelines

The guidelines for model scale measurement of propeller cavitation noise are included in the ITTC Quality Manual as guideline no. 7.5-02-01-05.

Many aspects of cavitation noise measurements are related to other procedures such as:

- Procedure 7.5-02-03-03.1 on Model-scale cavitation tests. Part of this procedure is discussed by the Specialist Committee on Wake Fields of the 25th ITTC, 2008 and the Specialist Committee on Scaling of Wake Field of the 26th ITTC, 2011.
- Procedure 7.5-02-03-03.3 on Cavitation induced pressure fluctuations, model scale experiments. This procedure is discussed by the Specialist Committee on Cavitation Induced Pressures of the 23rd ITTC, 2002.

In the following only those aspects that are particularly related to noise measurements will be discussed. The noise measurements shall be supported by additional investigations like cavitation observation, cavitation inception and hull pressure pulse measurement.

Noise measurements in a test facility should in general be performed by means of hydrophones of piezoelectric and omnidirectional type. A compromise has to be found for sensitivity, usable frequency range, dimensions and directivity. The high pressure resistance of most hydrophones is higher than required in model test facilities, but only limited information is available for sub atmospheric pressure application. The measurement frequency range shall start below propeller blade rate and extend up to several 10 kHz, depending on the purpose of the noise measurement.

Typically at least one hydrophone should be located at the propeller plane. Additional hydrophone positions up- and down-stream, as well as abeam, should be included if feasible to augment acoustic testing. Hydrophones should preferably be installed one of the following ways:

- In a large or medium sized acoustic chamber below the test section
- Outside from walls or windows
- Flush to walls or windows
- To a rake in the flow
- Inside the basin

Hydrophone arrays enable noise measurements with high directivity to scan the model to identify local noise source regions and should be used if permitted by facility capabilities and testing budget. Examples of the setup of an array are provided by Abott et al. (1993), Chang and Dowling (2009), Park et al. (2009b), and Lee et al. (2012).

For every test condition the background noise of the facility has to be determined to check the quality of the acquired noise data and to correct the cavitation noise data if the difference between the two is less than 10 dB. Two procedures to measure background noise can be applied: Replacement of the propeller by a dummy boss, or increase of tunnel pressure to suppress propeller cavitation. Examples of background noise sources in a cavitation tunnel are pump cavitation, non-cavitating turning vane noise, test section turbulent boundary layer noise, impeller blade trailing edge noise and of course the propeller driving train (Etter and Wilson 1993). As reference, the speed dependency of the overall integrated background noise levels reported by Doolan et al. (2013) shows that at higher tunnel speeds the noise levels scale with tunnel speed to the power eight. Bosschers et al. (2013) provides a discussion of background noise mechanisms of a towing tank. It is noted that reductions in background noise of the propeller driving train can be achieved using a hydraulic turbine (Briancon et al., 2013).

The influence of testing environment on the noise transfer function needs to be determined in order to properly relate measured sound pressure levels to source levels at a normalized distance of 1 m. The noise transfer function can be measured by replacing the propeller with a calibrated noise source and examples are given by Briancon et al. (2013) and Seol et al. (2013). For reference, a detailed analysis of the noise field at 1, 5 and 10 kHz inside a small cavitation tunnel is given by Yamaguchi et al. (1996), showing that the noise distribution becomes more complicated with increasing frequency. An acceptable agreement from a qualitative point of view was obtained for the overall noise patterns between measurements and computational results obtained with a 2-D boundary element method.

The low frequency limit for valid acoustic measurements needs to be determined. It is typically defined as the frequency below which the noise field is determined by separate acoustic modes in the test facility. In the report of the 15th ITTC a formula is given for the number of modes in a test-section. A minimum mode number of one in each 1/3 octave band is required although other references suggest a minimum of three modes. While in room acoustics one defines the so-called Schroeder frequency which can be computed from the reverberation time, no similar information has been found in the literature for the reverberation time in cavitation test facilities.

The measured cavitation noise at model scale needs to be extrapolated to full scale. Frequency scaling is based on the Rayleigh formula for the collapse time and is given by:

$$\frac{f_{\mathrm{fs}}}{f_{\mathrm{ms}}} = \frac{n_{\mathrm{fs}}}{n_{\mathrm{ms}}}\sqrt{\frac{\sigma_{\mathrm{fs}}}{\sigma_{\mathrm{ms}}}}$$

For the noise levels, different scaling formula have been derived which have been presented by the Cavitation Committee of the 18th ITTC (1987) as:

$$L_{\mathrm{fs}} - L_{\mathrm{ms}} = 20\log_{10}\left[\left(\frac{D_{\mathrm{fs}}}{D_{\mathrm{ms}}}\right)^{z}\left(\frac{r_{\mathrm{ms}}}{r_{\mathrm{fs}}}\right)^{x}\cdot\right.$$
$$\left.\cdot\left(\frac{\sigma_{\mathrm{fs}}}{\sigma_{\mathrm{ms}}}\right)^{w}\left(\frac{n_{\mathrm{fs}}D_{\mathrm{fs}}}{n_{\mathrm{ms}}D_{\mathrm{ms}}}\right)^{y}\left(\frac{\rho_{\mathrm{fs}}}{\rho_{\mathrm{ms}}}\right)^{y/2}\right]$$

This formula is valid for noise in proportional band widths. For the case of constant band width, frequency scaling should be included as well and the powers of cavitation number and shaft rotation rate change. Assuming spherical spreading loss, we find for the above, that x= 1 is generally applied. For the other parameters in this equation, two different approaches for determining their values have been reported. One approach is based on linear acoustics (e.g. Strasberg 1977) while the other approach assumes constant acoustic efficiency which is considered to be more valid for higher frequencies (De Bruijn and Ten Wolde 1974, Levkovskii 1980). The corresponding parameters are given in Table 7.1. As the cavitation number is usually kept constant, it is only the scaling of the tip speed that is relevant.

Table 7.1 Coefficients used in the cavitation noise extrapolation for proportional band width.

	w	y	z
Linear acoustics	1	2	1
Constant efficiency	0.5	1.5	1

As a note of caution in the use of the above scaling procedure, due to the complex mechanism of propeller cavitation noise and the limitations of test facilities, it is impossible to achieve all similarities between model test and full scale. At the same time the environmental conditions of the test are often quite different from the full scale conditions. For example, there exist wall effects, blockage effects etc. in the model experiments. All these will result in errors in the measurement and are difficult to quantify. In order to quantify the uncertainty of the measurements and scaling procedure, further investigations and validations of such influences on the noise results are necessary.

8 SUMMARY AND CONCLUSIONS

It is well established that shipping noise is the major contributor to the increase of low-frequency ambient noise levels in the oceans over the last 40 years.

Various ship noise sources have been reviewed including equipment like sonar. Even though sonar is a very loud noise source it can be controlled by switching it off. Cavitation of marine propellers is recognised to be a very important source of underwater noise from ships. At high frequencies propeller cavitation is usually the most dominant noise source while low frequencies are dominated by machinery noise and cavitation noise. Which noise mechanism is most dominant depends on the amount of cavitation, type of machinery and applied noise reduction measures. Below the cavitation inception speed, ship noise is generally due to vibration and noise from main and auxiliary machinery equipment and the gearing box. Full scale noise measurements revealed that the broadband source levels of commercial ship noise spectra vary from around 165 dB for smaller size vessels to 195 dB for the larger vessels with the maximum amplitude concentrated in the frequency range between 10 and 125 Hz.

Because of these high levels, regulations are being considered or imposed. Most of the regulations regarding shipping noise are at a regional level and concern restricted areas where there is a higher concentration of species of marine mammals or fishes. At an international level, the Committee for the Protection of the Marine Environment (MEPC) of the IMO, had a correspondence group working on the topic, which developed a document recommending non-mandatory guidelines for the reduction of underwater noise from commercial shipping. The EU Marine Strategy Framework Directive specifically mentions underwater noise in its definition of good environmental status and 'requires' that ambient noise levels are monitored starting from 2014.

This increased interest in shipping noise has resulted in a commensurate increase in research on full scale measurements and model scale testing and computational prediction methods.

For shipping noise prediction, the committee has focussed on propeller noise prediction. Non-cavitating propeller noise can adequately be predicted by CFD methods like Large Eddy Simulation in combination with acoustic analogy. However, for commercial shipping, it is more important to predict propeller cavitation noise. Potential flow methods are capable of predicting the blade rate tonals of sheet cavitation while semi-empirical models have been developed for the broadband part of both sheet and tip vortex cavitation. Computational prediction of cavitation on propellers using CFD is getting more mature and is being applied in industry. CFD has the capability to capture all kinds of cavitation although the accurate prediction of, for instance, cavitating vortices is still very demanding. Predicting the resulting radiated noise using acoustic analogies is a promising approach. However, the possibilities and limitations for accurate noise predictions, which requires proper accounting for the collapse process of the cavitation need to be further assessed. This includes the upper frequency limit for which such predictions are valid. The full frequency range of interest for cavitation noise can be as high as 20 kHz.

As a means to understand the activity of ITTC members and associated industries in hydrodynamic noise as it relates to the shipping industry, the committee developed a questionnaire designed to survey activity in the areas of full scale and model scale noise measurement methods.

Standards do exist for deep water noise measurements and the committee recommends that the ITTC guideline on ship underwater noise measurement follows the standard ISO/PAS 17208-1:2012(E). Standards for shallow water measurements are currently being pursued by various international organisations. The accurate prediction of ship source levels from shallow water measurements is difficult because the acoustic propagation from the source to the far field cannot easily be quantified. It is recommended that the ITTC guideline be revisited and updated when international standards for shallow water are established.

With respect to model scale measurements, several large cavitation tunnels have been built in the last few decades (or are being built) with specific requirements for noise measurements and which include the presence of an acoustic chamber. Progress has been made in model scale propeller cavitation tests, for instance in the use of full scale wake fields to improve the prediction of hull pressure fluctuations. However, few reports deal with the topic of model scale testing of cavitation noise. Specifically there is a lack of information on the accuracy of cavitation noise measurements and its extrapolation to full scale. The majority of the facilities apply the extrapolation procedure as published in the 1987 ITTC proceedings although coefficients vary. No specific extrapolation procedures are available for vortex cavitation noise. There is a large uncertainty in the extrapolated noise levels.

To evaluate the accuracy of numerical prediction methods and model scale tests in combination with extrapolation procedures, accurate full scale data is required in which the contribution of cavitation noise can be well distinguished from other noise sources.

9 RECOMMENDATIONS

The 27th Specialist Committee on Hydrodynamic Noise recommends adopting the following guidelines:

- 7.5-02-01-05 : Model Scale Noise Measurements
- 7.5-04-04-01: Underwater Noise from Ships, Full Scale Measurements.

The recommendations for future work are:

- Provide a procedure for model scale noise measurements.
- Check the existing methodologies regarding full scale noise measurements in shallow water and provide, if possible, guidelines. Establish communication with ISO working groups active on this topic.
- Update the overview of national and international regulations and standards regarding underwater radiated noise.
- Review the developments of prediction methods (experimental, theoretical and numerical) for propeller cavitation noise and of numerical prediction methods for noise propagation.
- Review uncertainties associated with model scale noise measurements and full scale noise measurements.
- Define a benchmarking test for numerical prediction methods and model scale noise measurements, preferably for a ship for which detailed full scale noise data is available to validate the results.

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Questionnaire on Full Scale Noise Measurements

COUNTRY	France	Germany	Italy	Japan		Korea		Netherlands		Spain	USA
ORGANISATION	DCNS	WTD 71	CETENA	MHI	Mitsui Lab.	KRISO	HHI	TNO	DMO	TSI	NSWC/CD
A: SITE INFORMATION											
A1: Measurement Methods											
Fixed location		•							•		•
On board system		•				•	•		•		•
Mobile equipment	•	•	•	•			•	•	•	•	•
Others					Hydrophone hull above the propeller						Various methods and facilities
A2: Do you have a hydrophone array?	No	Yes	Yes	No	No	No	Yes	Yes	No	Yes	Yes
A3: Facility Information											
Name		Aschau							Haringvliet, Heggernes		Santa Cruz, Autec, Seafac, Stafac
Location		Eckernförder Bucht							Holland, Norway		California, Andros Island (Bahamas), Ketchikan (Alaska), Andros Island
Water depth (m)		18- 22							14, 400		600, 1400, 500, 1400
A4: Hydrophones Deployment											
Mounted on seabed		•						•	•		
Deployed from buoy				•			•			•	
Deployed from vessel	•		•					•	•		
Others		on a trestle 1 m above bottom				Hull mounted					Bottom anchored vertical string of omni-hydrophones
A5:Depth of Hydrophones (m)	depend on location	17~21	0~320	5, 20			200	Max. 150	14, 20, 60	60	30~300
A6: Depth Position Measurement											
Pressure gauge	•	•	•								
Tape measure				•				•			
Others							Satellite information			From vessel	Permanent bottom anchor, depth tide compensated
A7: Minimum Sea-depth for Noise Measurements (m)	20	18	30	over 200			800	6		50	N/A
A8: Maximum Allowable Beaufort and Sea State numbers											
Beaufort								no limit for system deployed on sea bed		3	-
Sea State	3	2 - 3	2				2				4 *
A9: Typical Closest Distance between Noise Source and Reference Hydrophone (cpa).											
Horizontal distance (m)	30	0	100				50	100 m or ship length		80	200 for Beam aspect, 0 for Keel aspect
Vertical distance (m)	30	water depth	50				10	max 150		60	varies; typically 15 to 45 degree slant range

*: depends on ship and signature level of ship

COUNTRY	France	Germany	Italy	Japan		Korea		Netherlands		Spain	USA
ORGANISATION	DCNS	WTD 71	CETENA	MHI	Mitsui Lab.	KRISO	HHI	TNO	DMO	TSI	NSWC/CD
A10: What kind of environmental factors are measured during noise measurements?											
Water depth	•		•				•	•		•	
Water density											•
Current velocity							•				•
Water temperature							•	•			•
Temperature distributions			•					•			•
Sea state	•	•	•				•			•	•
Wind direction and velocity		•	•				•	•			•
Air temperature		•						•			•
Wave height and direction								•		•	•
Others (specify)										GPS location	
B: FULL SCALE HULL/PROPELLER											
B1: Do you check the surface condition of full scale propeller?											
Yes (specify when)	•	before meas. by the ship owner		diver's observation						According to vessel maintenance planning	Dockside prior to acoustic meas.
No					•	•	•	•	•		
B2: Do you check the hull surface condition?											
Yes (Please specify when) / No	Yes		No	diver's observation	No	Yes	No	No		According to vessel maintenance planning	Dockside prior to acoustic meas.
B3: Do you polish the propeller surface before noise measurements?	yes	no	no	yes	yes	yes	no	no		no	yes
C: HYDROPHONE INFORMATION											
C1: Hydrophone Manufacturer and Type	RESON TC 4032	B&K 8106, ITC		B&K 8105	B&K 8103, 8100	B&K 8103, 8105	B&K 8104	B&K 8106,8105, 8104		B&K	ITC (proprietary design for arrays). No. 8201
C2: Hydrophone Operation											
Omnidirectional (O), flush mounted (F)	O	O	O	O	F	O	O	O		O	O
Frequency range (Hz)	5 120k	4 100k	10 90k			- 200k	0.1 10k	5 160k		20 200k	5 56 k
Max. operating pressure level (atm or m)			100atm			40atm	100atm	150 m			up to 400 m
Operating temperature range (ºC)	-2 +55		-2 + 50			-40 +80	-30 +80	-40 +80			0 +25
C3: Sensitivity + Dimensions											
Hydrophone sensitivity (dB re V/µPa)	-170		-144			-214	-205	-174, -205		-170	- 186
Horizontal directivity ±...dB at ...kHz	Omni		Omni; +/-1.5dB at 50kHz			2dB at100kHz,					+/- 0.5dB at 56 kHz,
Vertical directivity ±...dB at ...kHz	270°; +/- 2dB at 15 kHz		Omni; +/- 2dB at 50 kHz			4dB at 100kHz					+/- 1dB at 10 kHz
Hydrophone diameter (m)	0.024		0.04	0.022		0.0095,0.022	0.021				0.05
Hydrophone length (m)	0.12		0.35	0.093		0.05,0.093	0,12				0.35

COUNTRY	France	Germany	Italy	Japan		Korea		Netherlands		Spain	USA
ORGANISATION	DCNS	WTD 71	CETENA	MHI	Mitsui Lab.	KRISO	HHI	TNO	DMO	TSI	NSWC/CD
C4: Calibration											
Do you have hydrophone calibration procedures?	yes	yes	yes	yes	yes	yes	yes	yes		yes	yes
How often do you calibrate? Before / After measurements, Regularly	B	R	B	B	B	B,A	B,A	B,A		R	R
How do you calibrate?	Hydrophone calibrator	lake, own procedure	Hydrophone calibrator	Water tank	Hydrophone calibrator	Hydrophone calibrator	Hydrophone calibrator	Hydrophone calibrator/ biannual factory calibration		Hydrophone calibrator	Acoustic at calibration facility; in-situ certification w/ acoustic source; electrical before/after testing
C5: Hydrophone Array											
Number of hydrophones installed	1	20	Max. 3	2	One or more if necessary	3-5	3	typically 3 (more possible)		2	10 (can vary depending on testing to be done)
Design frequency of hydrophone array if available (kHz)		1.6 - 10 kHz									is that of the hydrophone 56 kHz
D: DATA ACQUISITION & PROCESSING											
D1: Measurement Reporting Unit											
dB ref 1μPa/Hz at 1m	•	•	•	•	•	•	•	•			•
Others			Peak			RMS	RMS				1/3 octave band levels at 1 m
D2: Acoustical Source Center on Propeller											
0.7r above the shaft center in propeller disc				•							
Ship center			•				•				• (for cavitation)
Propeller center		•				•					
Halfway between engine room and propeller	•										
Others					Depends on application			Depends on application		Post-processing	Depends on application
D3: Frequency Range of Noise Measurements (Hz)	10 - 10 k	4 k - 80 k	10 - 50 k	100 - 80 k	- 20k	1 - 100k	0-100k	10 - 80 k		20-100 k	5 - 56 k
D4: Minimum Dynamic Range of Instrumentation (dB)			108		~ 40		102	144			Can resolve 1 dB SNR (do not report / use data w/less SNR< 3 dB
D5: When do you calibrate Instrumentation ? Beginning /End meas, Beginning/End each meas.	B	BE	B	B	BE	B	BE/EE	B/E			B/E
D6: Acquire phase signal of the propeller?	No	No	No	No	Yes	Yes	No	No		No	Yes
D7: Type of Signal Processing Filters											
Anti-aliasing filter	•	•	•		•	•	•	•		•	•
Others (specify)											Selected band-pass filtering
D8: Type of Sound Source Localization Methods											
Beamforming		•						•			
Crosspower spectrum								•			
Others (specify)							Parallel with over side test at sea				Hull scans (signal amplitude versus time or distance as ship passes)

COUNTRY	France	Germany	Italy	Japan		Korea		Netherlands		Spain	USA
ORGANISATION	DCNS	WTD 71	CETENA	MHI	Mitsui Lab.	KRISO	HHI	TNO	DMO	TSI	NSWC/CD
D9:Do you measure the vibration /acceleration of the hydrophone mount structure?	No	No	No	No	Yes	Yes	No	No		Yes	No
D10: Do you consider vibration /acceleration measurements during the evaluation of the noise signals?	No		Yes	No	No	No	Yes	No		No	No
D11: Presentation of Results											
1/3 octave		•	•		•	•	•	•			•
Narrowband normalized to 1 Hz	•	•	•	•	•	•	•	•			•
Harmonics						•					•
1/3 octave converted to 1Hz	•	•	•	•						•	•
Normalization to 1 m distance	•	•	•	•		•	•	•		•	•
Others							Time signal				Various requested formats
D12: Expected Uncertainty (dB) for Full Scale Sound Pressure Level Measurements	+/- 1		1.5		3		1-2				+/- 1.5 accuracy +/- 0.5 repeatability
D13: Confidence Level on the Quality of Test Results											
Scale of 1~10: 1 very uncertain, 10 very confident	8	9	7	8	7	7	9	8			9
E: CORRECTION METHODS TO OBTAIN THE NOISE SOURCE LEVELS											
E1: Do you correct the radiated noise measurement for ambient noise?	Yes	No	Yes	No	No	No	Yes	Yes		Yes	Yes
E2: Do you consider environmental parameters to predict the propagation loss?	No	No	No	No	No	No	Yes	Yes		No	No
E3: Do you measure propagation losses between sources and measurement location?	No	No	No	No	Yes	No	Yes	No		Yes	No
E4: Adjustment Factor for Propagation Loss											
20 $\log_{10}(r/r1m)$	•	•	•		•	•	•	•			•
Others (specify)						Numerical computation					standard high frequency absorption
E5: Correction for Free Surface Reflection				hemispherical diffusion				depends on application. Conversion to 'dipole source level'.			Average of levels from vertical string of hydrophones
E6: Correction for Bottom Mounted Hydrophones											
E7: Expected Uncertainty (dB) for Predicted Noise Source Level	+/- 3		3				1-2	3			+/- 3
E8: Confidence Level on the Quality of Correction Methods											
Scale of 1~10: 1 very uncertain, 10 very confident	6		5	7	7	7	9	8			7

Questionnaire on Model Scale Noise Measurements

COUNTRY	China		Germany	Italy		Iran	Japan			Korea		Netherlands	Norway	Russia	Sweden		Turkey	USA
ORGANISATION	CSSRC (2001)	SSSRI (1985)	HSVA (1990)	INSEAN (1978)	U. Genova (1979)	U. Sharif	MHI (1970)	JMUC (1980)	MEGURO (2006)	KRISO (2009)	HHI (1984)	MARIN (1973)	MARINTEK (1967)	KRYLOV (1961)	SSPA (1970)	Rolls-Royce	Istanbul TU	NSWC/CD (1991)
A: FACILITY INFORMATION																		
A1: Type of Facility																		
(a) Closed jet tunnel	•	•	•		•	•	•	•	•	•	•		•	•	•		•	•
(b) Free surface tunnel				•												•		
(c) Depressurized towing tank												•						
A2: Test Section Size																		
Length (m)	10.5	2.6	11	10	1.2	3.2	2.8	2.6	10	12.5	2.6	240	2.2	6	9.6	3.5	2	12.4
Width (m)	2.2	0.6	2.8	3.6	0.57	0.63	0.71	0.6	2	2.8	0.6	18		1.3	2.6	0.8	0.63	3.048
Height (m)	2	0.6	1.6	2.25	0.57	0.63	0.71	0.6	2	1.8	0.6	8		1.3	1.5	0.8	0.35	3.048
Diameter (for circular type) (m)													1.2					
Max. acceptable blockage (%)	15	60	25		20		20	14	10	10				10	20	25	40	
A3: Velocity and Pressure																		
Maximum velocity (m/s)	15	12	12	5	8.5	4	12	14	15	16.5	12.7	6	18	10	6.9	10	4	18
Minimum pressure (kPa)	10	0	25	3	10	20	20	9.81	10	20	9.8	2.5	10	30	20	10	100	3.5
Maximum pressure (kPa)	350	200	250	101.	101	200	100	196.	300	350	196	101	600	300	300	100	101	415
A4: Wake Generation																		
(a) Full ship model without additional flush mounted wire screen	•		•	•					•	•		•			•			•
(b) Dummy model without additional flush mounted wire screen		•		•					•			•						
(c) Combination of full hull model and flow liners																		
(d) Flow liners																		
(e) Full ship model with additional flush mounted wire screen				•											•			
(f) Dummy model with additional flush mounted wire screen				•			•						•	•		•		
(g) Wire screen					•		•	•	•		•						•	
(h) Shaft brackets and inclined shaft for twin screw ships					•													
A5: Wake Distribution																		
(a) Model scale wake measured at towing tank		•		•	•				•		•	•		•		•	•	
(b) Model scale wake using full ship model in tunnel	•		•	•					•	•					•			•
(c) Model scale wake calculated using CFD						•										•		
(d) Full scale wake obtained by extrapolation of measured model scale wake							•	•					•					
(e) Full scale wake calculated using CFD					•	•					•	•				•		

NB. A5b was introduced later following comment from HSVA and SSPA, answers correspond to answer a) in question A4.

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A6: Wake Measurement																		
Do you perform wake field measurements	•	•	•	•	•		•	•	•	•	•	•	•	•	•	•		•
(a) Routinely		•	•	•			•		•		•		•	•		•		•
(b) only on sponsor's request	•		•		•			•		•		•			•			
A7: Methodology																		
(a) Pitot tubes	•	•	•			•	•	•	•	•	•	•	•	•	•			
(b) LDV	•			•	•				•	•						•		•
(c) PIV				•				•	•			•						•
A8: Components																		
1 component (axial)	•	•					•	•	•				•					
2 components		•																•
3 components		•	•	•				•		•	•	•			•	•		
B: MODEL SETUP																		
B1: Ship and Propeller Model																		
Ship model size (m)	6 - 10	11	8	5 - 6					7	6 - 9		7 - 12	1.2	4	7	3.5		6.1 - 12.2
Ship Model Material (FRP, GRP, Wood, Divinycell, Foam with rion skeleton)	GRP	W	W	W/FRP					FRP	W / FRP		W	Various	Foam	Div	Div		GRP
Propeller model size (mm)	250	250	250	250	200~250	300	250		250	250	250	230~250	250	200~300	200~250	250	150	300~460
Propeller Model Material	brass	Al	Brass	Bro, Al	various	steel	Al.		Al.	Al.	Al.	Bro.	Bro.	babbitt, duralumin	Bro.	Bro.	Bro.	Al.
B2: Appendages																		
Rudder	•		•	•			•		•	•	•	•	•	•	•	•		•
Stabilizer fins	•		•	•								•		•	•			•
Thruster openings		•	•							•		•		•	•			
Energy saving devices	•		•							•		•		•	•			•
Shaft brackets	•	•	•	•	•				•	•		•	•	•	•	•		•
Bilge keels	•		•	•								•		•	•			
Sonar dome	•		•	•					•	•		•		•	•			•
B3: Turbulence Stimulators																		
Ship hull (None, Studs, Trip Wire, Other)	N	D	N	S	O	N			S	N	N	S	N	N	T	N	S	S
Propeller (None, Carborundum, Paint)	N	N	N	N	N	N	N	P	N	N	N	C	P	N	P	N	N	N
B4: Propeller Manufacturing																		
Method (NC+ hands, NC only)	NC+	NC+	NC+	NC	NC+	NC	NC+	NC+	NC+	NC+	NC+	NC+	NC+	NC+	NC+	NC+	NC+	NC+
Accuracy (±, mm)	0.1	0.05	0.05	0.02	ITTC	0.05	0.05	0.05		0.02	0.05	0.05	0.02	0.02	0.05	0.05	0.05	0.075
B5: RPM measurement																		
Location (Dynamometer,Motor, Shaft)	D	D	D	S	M	D	S	S		D	M	S	S	S	S	S	S	S
Number of pulses / rev	1024	36	360	3600	100		1	tacho	360	1024	100	360	3600		100	720	1	15 or 60

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C: HYDROPHONE INFORMATION																		
C1: Hydrophone Type																		
Hydrophone manufacturer	B&K	B&K	Reson	B&K	Reson	B&K	B&K		B&K	B&K	B&K	Reson	B&K	B&K	B&K	B&K,PCB	B&K	Reson
Hydrophone type	8105	8104	TC 4032	8103	TC 4013	8103	8103		8103, 8104	8103, 8105	8104	TC 4014-1	various	8103	8103	8103, 112A22	8103	TC 4056
C4: Calibration																		
Do you have hydrophone calibration procedures?	Yes	Yes	Yes	Yes	Yes	Yes	Yes		Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	No
How often do you calibrate? Before /After measurements, Regularly	R	B	R	B	R	B, A	B		R	B	B, A	B	B		B	B, A	B, A	
How do you calibrate? Hydrophone calibrator / water tank	HC,WT	HC	HC	HC	HC	HC	WT		HC	HC	HC	HC	HC		HC	HC	HC	
C5: Hydrophone Location																		
Acoustic chamber (AC)	•	•	•						•	•								•
Outside wall or windows					•									•		•		•
Flushed wall or windows						•	•						•				•	
Rake / in flow				•	•										•			
Inside basin												•						
C6: Dimensions of Location																		
Height (m)	2	0.22	2.4		0.74				8	11.8	0.08			0.4				13.71
Width (m)	2.2	0.5	11		0.26				1.4	2	0.08			0.6				2.14
Depth (m)	9,5	0.22	2.8		0.08				1	1.5	0.068			0.35				1.15
Stand-off distance (mm)	1000	420	930		300				300~700					900				550
C7: Number + Position																		
How many hydrophones are installed?	1	1	1	5	1	5			32	48		2	3	1	3	various	2	56
Upstream of the propeller plane				1		4				1			1		1		1	
At propeller plane	1	1	1	1		1			2	2	1			1	1		1	
Downstream of the propeller plane				1	1		1		1	1			2		1	2		
Others				2					32			2				5~16		various
C8: Hydrophone Array																		
Do you use an array?	Yes	No	No	No	No	No	No		Yes	Yes	No	No	No	No	No	No	No	Yes
Planar type, combo random, spiral arm log spaced	PT								circle	CO								S-A
Hydrophone spacing (mm)	10,20,40								random	120~170								Other
What is the design frequency of the hydrophone array (kHz)	0.5~12.5								to 5	to 100								to 50
What is the half power spatial bandwidth?																		various
What is the aperture size? (m)										1.6 x 1.6								various

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D: DEFINITION OF TEST CONDITION																		
D1: Test Conditions																		
Noise test procedure is facilities own / others	F	F	F	F	F	O	F		F	F	F	F	F	F	F	F	F	O
The purpose of noise test is:																		
(a) Cavitation Noise	•	•	•	•	•	•	•		•	•	•	•	•	•	•	•	•	•
(b) Cavitation inception		•		•	•	•			•	•	•	•					•	•
(c) Non-cavitating noise	•	•		•		•	•		•	•	•		•	•	•		•	•
D2: Loading Condition																		
Design point / Model test result	D	D	M	M	D	M	D	D	D, M	D	D	M	D	M	M	D	D	M
Are the propulsion test data corrected to full scale?	Yes	Yes	Yes	Yes	Yes	Yes	No	Yes	Yes	Yes	Yes	Yes	Yes	No	Yes	Yes	Yes	Yes
Which parameters do you use to define the test condition?	Kt 1-Wt	Kt 1-Wt	Kt 1-Wt	Kt 1-Wt	Kt 1-Wt	J	Kt 1-Wt	Kt	Kt 1-Wt	Kt 1-Wt	Kt 1-Wt	J	Kt 1-Wt	Kt 1-Wt	J	Kq 1-Wq	Kt 1-Wt	Kt 1-Wt
Does the cav. number account for the stern wave height?	No	No	Yes	No	No	No	No	Yes	No	Yes		Yes	No	No	Yes	No	No	No
What is the reference point of the cavitation number?	0.8R 12:00	0.8R 12:00	0.8R 12:00	0.7R 12:00	SCL	0.7R 12:00	0.7R 12:00	0.9..0.95 r	0.7R 12:00	0.7R 12:00	0.7R 12:00	SCL	0.7R 12:00	SCL	SCL	0.7R 12:00	0.7R 12:00	SCL
Do you correct the tunnel pressure for the test section pressure drop?	No	No	No	No	No	Yes	No	No	Yes	No	No	No	No	Yes		Yes	Yes	Yes
Do you account for trim and sinkage?	No	No	Yes	No	Yes	No	No	No	No	No	Yes	Yes	Yes	No		Yes	No	Yes
D3: Typical Operational Range																		
Range of water speed (m/s)	3~8	1~8	5~8	1~3.5	5~6	0.6~3.6			3~10	7~12	2.5~5	2~3	2~5	4~8	4~9	4~5	2~3	5~18
Range of propeller speed (rev/s)	20~40	15~35	25~35	8~15	20~30	3~33	25		20~30	30~40	25	8~15	18~25	25~40	20~35	20~25	17~33	
Range of pressure at SCL (atm)	0.3~1.5	0.1~2	0.4~0.9		0.1~1				0.3~2	0.4~2.5	0.3~0.6	0.05~0.1	0.2~0.3	0.3~3		0.2	1.06	
D4: Water Quality																		
Do you monitor water quality?	Yes	Yes	Yes	No	Yes	No	Yes		Yes	Yes	No	Yes	Yes	Yes	Yes	Yes	Yes	Yes
What do you monitor?																		
(a) Total oxygen content														•				
(b) Dissolved oxygen content			•		•		•	•	•	•			•			•	•	•
(c) Nuclei size									•									
(d) Total air content	•	•																
(e) Dissolved air content												•			•			
Range of dissolved/total air or oxygen content (% Sat.,1.0atm)	< 85	40~60	70~90		40		50	30~50	30~80	30~70	36~40	25~40	25~30	30~40	30~40	30~40	60	60~80
Range of water temperature during noise tests? (°C)	8~25	0~30	18~24	17~22	15~25				17~23	10~25	10~17	10~18	15~18	17~25	15~25	20	10~20	21~35
D5: Vibration Measurements																		
Do you measure vibration during noise tests?	No	Yes	No	Yes	No		No		Yes	Yes	Yes	No	No	Yes	No	No	No	Yes
Before / Parallel / After the test		B		P		B			P	P	P			P				P
Above propeller / at shaft / at hull (H) / on the Tunnel (T)		P		S					various	H	T			H					

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D6: Test Conditions																		
Design condition(D), more than 1 (+)	D, +	+	+	D, +	+	+	+		+	+	+	D, +	D	D	+	D, +	+	+
D7: Background Noise Tests																		
Replacement of propeller by dummy boss, Increase of pressure	R	R	R	R	R	R	R		R	R, I	R, I	R, I	R, I	R	R	R	R	R, I
Before / After noise test	B	A	A	A	A	A	B		B, A	A	B, A	B, A	B, A	B, A	A	A	B	B
For every test condition?	Yes	Yes	Yes	Yes	Yes	Yes	Yes		Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes
D8: Supporting Investigations																		
(a) Cavitation observation	•	•	•	•	•	•	•		•	•	•	•	•	•	•	•	•	•
(b) Cavitation inception	•	•			•	•			•	•	•	•	•		•	•	•	•
(c) Hull pressure pulses		•		•	•				•	•			•		•	•	•	•
(d) Others				•	•													•
E: DATA ACQUISITION & PROCESSING																		
E1: Measurement Reporting Unit																		
in dB ref. ···	1μPa/Hz	1μPa/Hz	1μPa/Hz	1μPa/Hz	various		1μPa/Hz		1μPa/Hz	1μPa/Hz	1μPa/Hz	1μPa/Hz	1μPa/Hz	1μPa/Hz	1μPa/Hz	1μPa/Hz	1μPa/Hz	1μPa/Hz
RMS / Peak	RMS		RMS				RMS			RMS	RMS	RMS				Peak		
at 1m		1m	1m	1m						1m		1m	1m	1m	1m	1m	1m	1m
E2: Acoustical Source Center on Propeller																		
Shaft center line	•	•		•	•		•			•	•	•	•	•			•	•
0.7R 12:00 o'clock						•									•			
0.8R 12:00 o'clock			•															
1.0R 12:00 o'clock																•		
E3: Frequency Range of Noise Signals										•								
Lower frequency (Hz)	500	1000	1	800	100	10	10		10	1	20	100~1000	50	1 000	1	0	20	1000
Upper frequency (kHz)	80	80	100	100	30	50	40		20	100	12.8	100	20	80	22	20	12.5	50
E4: Acquire Phase Signal of the Propeller?																		
Acquire phase signal of the propeller	Yes	Yes	Yes	Yes	Yes	No	No		Yes	Yes	No	No	Yes	No	No	Yes	No	Yes
E5: Time Length of Model Noise Measurement?																		
How many revs or seconds do you Measure	20 s	30 s	10 s	> 500s	10 s				30 revs	20 s	30~60 s	10 s	20 s	30 s	150~300 s	7.5~30 s		> 10 s
E6: Check Reproducibility of the Noise Signals?																		
Check reproducibility of the noise Signals	Yes	Yes	Yes	Yes	Yes		No		Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes
E7: Type of Signal Processing Filters																		
Anti-aliasing filter	•	•	•	•	•		•		•	•	•	•	•	•	•	•		•
Others (specify)						•												•
E8: Sound Source Localization Method																		
(a)Beamforming	•	•							•	•								•
(b)Matched-filter array processing																		•
(c)Crosspower spectrum																		•
(d)None			•		•		•				•	•	•	•	•	•	•	
(e)Others (specify)				•*														

•*: near wake measurements with sound/pseudo sound filtering techniques

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E9: Uncertainty Analysis																		
(a) ITTC (general guideline)		•		•		•			•						•	•	•	
(b) Facility's own (specify)					•													•*
(c) None	•		•				•			•	•	•	•	•				
E10: Consider the Results of the Vibration/Acceleration Measurements?																		
Do you consider the results of the vibration/acceleration measurements? (specify)		•		• machinery noise filtering					• identify noise source location		•							•**
E11: Presentation of Results																		
1/3 octave	•	•		•	•	•	•		•	•	•	•	•	•	•			•
Narrowband normalized to 1 Hz	•	•	•	•			•		•	•	•	•		•	•	•		•
Harmonics				•						•			•					
1/3 octave converted to 1Hz	•		•									•					•	
Normalization to 1 m distance			•		•					•		•	•		•	•	•	•
Others: time signal (TS), narrowband (NB)					NB						TS							
E12: The Reverberation of Facility																		
Have you investigated the reverberation of facility	•		•	•					•	•	•	•		•	•			•
E13: What Corrections																		
(a) Free surface											•							
(b) Reflections due to wall	•									•	•			•				
(c) Background noise	•	•		•	•	•	•		•		•	•	•	•		•	•	•
(d) Reverberation	•									•								
(e) Others (specify)															•***			
E14: Uncertainty																		
Uncertainty (in dB) for model scale noise measurements	3	2	5		1 - 2				3		3	3 - 5		2	3		2	3
E15: Confidence Level on the Quality of Test Results																		
Scale of 1~10: 1 very uncertain, 10 very confident	7	8	8	6	6	5	7			8	8	7	6	10	9	7	8	5

•*: Uncertainty analysis depends on type of measurements (cavitating/non-cavitating). Generally uncertainty analysis based on standard procedures for signal processing (i.e., Bendat & Piersol's books on random signals).

•**: Evaluate vibration influence using coherence measurements (subtract coherent power)

•***: Correction to far-field based on calculations or in-situ reciprocity measurement

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F: SCALING METHODS																		
F1: Full Scale Noise Prediction																		
Do you carry out full scale noise prediction after noise measurements at model scale?	Yes	Yes	Yes	No	Yes		Yes		No	Yes	No	Yes	Yes	Yes	Yes	Yes	Yes	Yes
F2: Scaling Procedures																		
ITTC 1987		•	•		•					•					•	•	•	
Others (specify)	empirical						similar to ITTC 1987					de Bruijn and ten Wolde (1974)	own facility scaling method	own facility scaling method				•*
F3: Scaling Methods for Tip Vortex Cavitation Noise																		
Do you have special scaling methods for tip vortex cavitation noise?	No	No	No		No		No			No		No	No	Yes	No	No	No	No
F4: w, x, y and z Exponents for Scaling in ITTC 1987 (or similar) Formulation																		
w		0	1				0					0.5			1		0.5	
x		1	1				2					1			1		1	
y		0	2				3					1.5			2		2	
z		1.5	1				3					1			1		1	
F5: Uncertainty																		
Uncertainty (in dB) for your noise scaling procedure	5	2	5									5		3	10		3	5
F6: Confidence Level on the Quality of Scaling Method																		
Scale of 1~10: 1 very uncertain, 10 very confident	6	8	8				7			7		6	6	10	7	3	7	4

•*: Depends on type of noise mechanism. In-house cavitation relationships & non-cavitating scaling relationship