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Integrated Prediction and Assessment of Underwater Noise for Commercial Vessels

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iii. Nomenclature

B is the bulk modulus (or elastic property) of the medium, in N/m^2

B is the number of blades on a propeller

$C_{\varepsilon 1}$, $C_{\varepsilon 2}$ and $C_{\varepsilon 3}$ are coefficients

D is the cylinder diameter in metres of a marine diesel engine

D is the propeller diameter in m

DP is the direct path type of ray path

G_b is the turbulent production due to buoyancy

$\mathrm{G_f}$ is the grid flux

G_k is the turbulent production

$G(\boldsymbol{x}, \boldsymbol{y}, t, \tau)$ is Green's Function

$H(f)$ is the Heaviside Function

I is the intensity of the sound

$\boldsymbol{I}$ is the identity matrix

I_{ref} is the reference sound intensity

L_0 is the reference length

L_{WB} is the Sound Power Source Level, in dB re 10^{-12} W

M is the Mach number

M_r is the Mach number of a fixed point on the blade

$\dot{M}_i$ is a result from differentiation of the Mach number at a fixed point at the source time

P is the power input from the machinery

P_{ij} is the compressive stress tensor

R is the range of the receiver from the sound source, in m

R_0 is the reference location; typically 1 metre from the nominal source in hydroacoustic applications

RL is the received level at range R (in m), in dB re 1μPa/1Hz

RBR is the refracted-bottom-reflected type of ray path

RSR is the refracted-surface-reflected type of ray path

RSRBR is the refracted-surface-reflected-bottom-reflected type of ray path

S is the surface area

S_k and S_ε are strain rate parameters

SIL is the sound intensity level in dB re 20μPa in air and 1μPa in other media such as water

SL is the value at 1m from the source, in dB re 1μPa/1Hz/1m

SPL is the sound pressure level in dB re 20μPa in air and 1μPa in other media such as water

$\boldsymbol{T}$ is the viscous stress tensor

T_{ij} is the Lighthill stress tensor

TL is the transmission loss in dB re 1μPa

V is the ship speed in knots

V_t is the propeller tip velocity

V_0 is the reference speed

W_{rad} is the radiated acoustic sound power

χ is the porosity

$\mathbf{a}$ is the face area vector

b is the number of piston rings for one piston

c is the speed of sound, in m/s

c_1 is the speed of sound in the initial medium in m/s

c_2 is the speed of sound in the new medium in m/s

c_L is a constant, usually taken as 2

c_v is a constant, usually taken as 6

dS is for integration over the surface in question

$d\Omega$ is an element of the surface area of the sphere $r = c(t - \tau)$

d_τ is for integration at source time

f_i is the net force per unit volume exerted by any external mechanical forces that may be acting on the fluid

$\boldsymbol{f}_g$ is the body force due to gravity

$\boldsymbol{f}_r$ is the body force due to rotation

$\boldsymbol{f}_p$ is the porous media body force

$\boldsymbol{f}_u$ is the user-defined body force

$\boldsymbol{f}_\omega$ is the vorticity confinement body force

g is gravitational acceleration in m/s^2

g_i is a component of gravitational acceleration

j is the number of cylinders of a marine diesel engine

k is the turbulent kinetic energy

k is the number of the harmonic (a whole number) for engine noise

kW is the rated power of a machinery item

l_i is the local force per unit area on the fluid in the i direction

$\dot{\mathrm{m}}^*{}_{\mathrm{f}}$ is the face mass flow rate

n is the rotational speed in rps of a marine diesel engine

n_s is the rotational speed of the engine in rpm

m indicates if the engine is a two- or four-stroke engine

m_0 is the surface density of the structure to which it is attached

n_i is a direction cosine

p is the sound force per unit area, or sound pressure, typically given in µPa

p_{ref} is the reference pressure, in Pa

p_v is the vapour pressure, in Pa

p_0 is ambient static pressure, in Pa

$p^{'}$ is acoustic pressure, in Pa

p_{ij} is a pressure stress tensor, including normal and viscous shear stresses

$p^{'}_L$ is the acoustic loading pressure, in Pa

$p^{'}_Q$ is the acoustic pressure due to quadrupoles, in Pa

$p^{'}_T$ is the acoustic thickness pressure, in Pa

q is the rate at which new mass is created per unit volume

r is the distance between x and y

$\hat{r}_i$ is a unit radiation vector

$\hat{r}_j$ is a unit radiation vector

rpm is the given rotational speed of a machinery item

rpm_0 is the rated rotational speed of a machinery item

s_{α_i} is the source or sink in the ith phase

u is the fluid velocity in the x-direction in m/s

$\bar{v}'$ is acoustic particle velocity in m/s

$\mathbf{v}^*{}_\mathrm{f}$ is the known boundary velocity

$\mathbf{v}_\mathrm{f}$ is the boundary velocity

$\boldsymbol{v}$ is the velocity

$\boldsymbol{v}_g$ is the grid velocity

v_i' is a component of acoustic particle velocity in m/s

v_i is a component of particle velocity in m/s

v_j is a component of particle velocity in m/s

v_n is the local normal velocity of the source surface in m/s

$\dot{v}_n$ is the source time derivative of v_n

w is the gross weight in kg of a machinery item

*(**x**, t)* are the observer space-time variables

*(**y**, τ)* are the source space-time variables

z_p is the number of pistons in the engine

z_z is the number of valves for one piston

α_i is the phase volume fraction

γ_f is the Rhie-Chow-type dissipation

γ_M is the dilitation dissipation

δ_{ij} is the Kronecker delta

$\delta(f)$ is the Dirac delta function

ε is the turbulence dissipation rate

η is the damping loss factor

θ is the local angle between normal to the surface and radiation direction at emission time

θ_i is the angle of incidence

θ_t is the angle of transmission

λ is the wavelength, in m

μ_t is the turbulent viscosity

ρ is the density of the medium, in kg/m^3

ρ_0 is the mean density of the fluid

ρ' is acoustic density fluctuation

$\rho_0 c$ is the characteristic impedance of air

σ is the radiation efficiency of the structure

σ is the cavitation number

σ_k and σ_ε are the turbulent Schmidt numbers

φ is the velocity potential

$\overline{\Box}^2$ is the D'Alembert, or wave operator

∇ is the gradient operator

Δ is the displacement of a vessel

$[\,]_{ret}$ denotes evaluation at the retarded time, i.e. sound speed is not infinite

$\otimes$ is a tensor product

$\langle v^2 \rangle$ is the space averaged mean square velocity

iv. Acronyms and Abbreviations

ACCOBAMS – Agreement for the Conservation of Cetaceans of the Black Sea, Mediterranean Sea and Contiguous Atlantic Area

AIS – Automatic Identification System, fitted on commercial ships

AMG - Algebraic Multi-Grid Linear Solver (in CFD)

ANSI – American National Standards Institute

ARU – Autonomous Recording Unit

ASCOBANS – Agreement on the Conservation of Small Cetaceans of the Baltic and North Seas

ATOC - Acoustic Thermometry of Ocean Climate

BDC – Bottom Dead Centre (in an engine cycle)

BEM – Boundary Element Method

BRF – Blade Rate Frequency

BSPA - Baltic Sea Protected Area

BV - Bureau Veritas (Classification Society)

CAROS – Co-operative Arrangement for Research on Ocean Science

CFD – Computational Fluid Dynamics

CFR – Cylinder Firing Rate (in engines)

CIS – Cavitation Inception Speed

CLT – Contracted and Loaded Tip propeller

CMS - Convention on the Conservation of Migratory Species of Wild Animals (also known as the Bonn Convention)

COA – Closest Observed Approach

COTS – Commercial Off-The-Shelf equipment

CPA – Closest Point-Of-Approach of a vessel or animal to a given point

CPB – Costa Propulsion Bulb

CPP – Controllable Pitch Propeller

CR – Critical Ratio, for the increased energy required to be heard in a masking condition

CTD – Conductivity / Salinity, Temperature and Depth / Pressure profile (detailing these properties of the water)

dB - Decibel

DBDBV - Digital Bathymetric Data Base Variable resolution

DES – Detached Eddy Simulation (in CFD applications; see above)

DNS – Direct Numerical Simulation (CFD solution method which doesn't account for turbulence)

DNV - Det Norske Veritas (Classification Society)

DP - Direct Path

DSC – Deep Sound Channel (or SOFAR)

DWT – Deadweight Tonnage

EC - European Commission

EEDI - IMO's Energy Efficiency Design Index

EIA – Environmental Impact Assessment

EIONET - European Topic Centre on Biological Diversity

EFR – Engine Firing Rate

EPA – Environmental Protection Agency (US Government Department of Energy)

ESA - Endangered Species Act (in the US)

FEA - Finite Element Analysis

FEM - Finite Element Method

FFT – Fast Fourier Transform

FPP – Fixed Pitch Propeller

FRP – Fibre-Reinforced Plastic

FRV – Fisheries Research Vessel

F-WH - Ffowcs-Williams Hawkings equation

GDEM – Generalized Digital Environmental Model of global climate data

GEBCO - General Bathymetric Charts of the Ocean

GES - Good Environmental Status

GL - Germanischer Lloyd (Classification Society)

GPS – Global Positioning System

HELCOM – Helsinki Commission (Baltic Marine Environment Protection Commission)

HFO – Heavy Fuel Oil

HPC - High Performance Computer facility at University of Strathclyde

HVAC – Heating, Ventilation and Air Conditioning

Hz - Hertz

IACMST – Inter-Agency Committee on Marine Science and Technology

IACS – International Association of Classification Societies

ICES – International Council for the Exploration of the Sea

ICRW – International Convention for the Regulation of Whaling

IFAW – International Fund for Animal Welfare

IMO – International Maritime Organisation

INSEAN - The Italian Ship Model Basin

IONC - International Ocean Noise Coalition

ISO – International Organisation for Standardization

ITTC – International Towing Tank Conference

IUCN – International Union for Conservation of Nature

IWC – International Whaling Commission (established under ICRW)

JAMP - The OSPAR Joint Assessment and Monitoring Programme

JNCC – Joint Nature Conservation Committee (UK Government)

LAA – Lighthill's Acoustic Analogy (looks at generation of noise – usually in aeroacoustics)

LAB - Laboratori D'Aplicacions Bioacústiques (at UPC)

LES – Large Eddy Simulation (in CFD applications)

LFA – Low Frequency Active Sonar (also known as SURTASS – LFA in the US)

LHS - Left-hand side

LIDO - Listening to the Deep Oceans

LNG – Liquefied Natural Gas

LOSC – UN's Law of the Sea Convention

LR - Lloyd's Register of Shipping (Classification Society)

MARPOL 73/78 – Marine Pollution (International Convention for the Prevention of Pollution From Ships, 1973 as modified by the Protocol of 1978)

MCA - Maritime and Coastguard Agency (in the UK)

MCR – Maximum Continuous Rating of an engine

MFR - Moving Frame of Reference approach

MIZ – Marginal Ice Zone

MMPA – Marine Mammal Protection Act (in the US)

MoD – Ministry of Defence (in the UK)

MPA – Marine Protected Area

MSFD - EU Marine Strategy Framework Directive

M/V - Motor Vessel

NASA - National Aeronautics and Space Administration (in the US)

NATO – North Atlantic Treaty Organization

NAVOCEANO - U.S. Naval Oceanographic Office

NBS – New Blade Section propellers

NCR – Non-Continuous Rating of an engine

NGO – Non-Governmental Organisation

NMFS – National Marine Fisheries Service (part of NOAA, US)

NOAA –National Oceanic and Atmospheric Administration (U.S. Department of Commerce

NRC - National Research Council (in the US)

NRDC – Natural Resource Defence Council (in the US)

OLP - Over-Lapping Propeller system developed by Kawasaki Shipbuilding Corporation

OSPAR Commission – Oslo and Paris Conventions (Administrator of the Oslo and Paris Conventions for the protection of the marine environment of the North-East Atlantic)

Pa - Pascal

PBCF – Propeller Boss Cap Fins

PCAD – Population Consequences of Acoustic Disturbance

PCT – Propeller Cap Turbine

PE – Parabolic Equation modelling

PSSA – Particularly Sensitive Sea Area

PTS – Permanent Threshold Shift of hearing range

RAM – Range-dependant Acoustic Model (A FORTRAN code using Parabolic Equation modelling)

RANS – Reynolds-Averaged Navier-Stokes (CFD solver method)

RBR – Refracted Bottom-Reflected propagation

RHS - Right-hand side

RL – Received Levels of sound

RM - Rotating Mesh approach

RMS – Root Mean Square (statistical averaging value)

RPM – Revolutions per minute

RPS - Revolutions per second

RSR – Refracted Surface-Reflected propagation

R/V – Research Vessel

SA - Spalart-Allmaras (one-equation CFD turbulence model)

SAC – Special Area of Conservation

SCN – Simplified Compensative Nozzle

SEA – Statistical Energy Analysis

SEL – Sound Exposure Level

SIL - Sound Intensity Level

SILENV - Ships Oriented Innovative Solutions to Reduce Noise and Vibrations, European project

SIMPLE - Semi-Implicit Method for Pressure-Linked Equation (in CFD)

SL – Source Level of noise, typically at 1m from source

SMRU – Sea Mammal Research Unit

SNR – Signal-to-Noise Ratio, at a specified level required for an animal to hear a sound

SOFAR – Sound Fixing and Ranging Channel (or DSC)

SOLAS – International Convention for the Safety of Life at Sea

SONAR – Sound Navigation and Ranging

SPL – Sound Pressure Level (in dB re 1µPa in water)

SST – Shear Stress Transport model (two-equation CFD turbulence model)

SVP – Sound Velocity Profile

TDC – Top Dead Centre (in an engine cycle)

TEU – Twenty-foot Equivalent Unit (shipping containers)

TKE – Turbulent Kinetic Energy

TL – Transmission Loss (usually in dB per m or km)

TS -Time-step

TTS – Temporary Threshold Shift of hearing range

UN – United Nations

UNCED – United Nations Conference on Environment and Development

UNCLOS – United Nations Convention on the Law of the Sea

UNICPOLOS - United Nations Open-Ended Informal Consultative Process on Oceans and the Law of the Sea

UoS - University of Strathclyde, Glasgow

UPC - Universitat Politècnica de Catalunya

URANS - Unsteady Reynolds-Averaged Navier-Stokes

URN - Underwater Radiated Noise

VoF - Volume of Fluid

WDCS – Whale and Dolphin Conservation Society

WIF - Wake Improvement Fin, used with overlapping propeller system

WMO – World Meteorological Organization

WOA05 – World Ocean Atlas 2005

WOA09 - World Ocean Atlas 2009

Abstract

The main aim of this study is to address the need for a commercial ship underwater radiated noise prediction and assessment methodology that could be used in the early stages for new designs. A detailed literature review of current state of knowledge on ship underwater noise sources and potential impacts is carried out. This review also looks in detail at the available noise prediction and propagation techniques, to assess their suitability for use in early stage design. The review also focuses on the related areas of underwater ambient noise, and ship noise regulation and reduction.

Following this, a numerical approach for commercial ship noise and propagation prediction is proposed and tested a case study vessel. Field measurement data for the vessel is used to validate and test the approach, and several variations. Some investigation is also carried out for the prediction of machinery noise tonal frequencies. Suggestions are provided for the use of empirical approaches where use of the numerical approach is not viable or appropriate. Impact assessment of the predicted spectra is then addressed.

An assessment tool is developed, with several key purposes. Firstly it allows input of key vessel, propeller and machinery parameters to allow empirical prediction of spectra and overall noise levels, and machinery tonal frequencies, which can be compared to predicted or measured spectra. Secondly, an extensive database of marine wildlife species, their conservation status, typical habitat regions, hearing and vocalisation frequency ranges and recorded responses to ship noise, has been compiled. The tool allows this to be filtered or highlighted by operational area, conservation status and frequency range, so that those species which are likely to be affected by a particular vessel can be identified. The potential impacts can thus

be assessed in a goals-based approach. Thirdly, the predicted, estimated or measured spectra can be compared with existing regulation and suggested threshold noise levels for a more rules-based approach to impact assessment. Use of these impact assessment approaches are demonstrated using a case study of one of the vessels.

Finally, some discussions and conclusions on the main findings of the work are presented.

Chapter 1 - Introduction

1.1 Chapter Overview

This chapter will detail first the General Perspectives (§1.2) of the issues covered in this thesis, and then define the problem to be addressed (§1.3). Further, it will provide an overview of the structure and layout of this thesis (§1.4).

1.2 General Perspectives

It is quoted in Fang et al. (2013) that currently, 90% of international trade is seaborne. Cargoes of all natures are transported by sea in a myriad of commercial vessels, and without it, international trade as we know it would not exist. Aside from this, the marine environment is being industrialised in a multitude of ways, including fisheries, renewable and non-renewable energy extraction, resource exploitation, leisure and defence. Many of these industries have seen continual expansion and rise in profits, and trends suggest that this is set to continue. Many emerging, developing and developed nations are heavily supported by marine-based industries. However such exploitation of the vast marine environment cannot come without a price. This price is being paid by the environment itself and those who inhabit it, in the form of pollution. Pollution of the marine environment can take many forms, such as air emissions, toxic contaminants, displaced organisms, physical damage and energy emissions. The focus of this project is pollution through the introduction of energy into the marine environment; specifically sound energy. More specifically, it will focus on the underwater noise emitted by commercial vessels in transit.

1.3 Problem Definition

Ships, waterborne vessels, and the offshore and subsea industries in general have a huge impact on the marine environment, with the potential to cause significant damage. The majority of these negative impacts, such as carbon emissions and waste disposal, are already well understood and positive action is being taken to control and minimise these aspects through a number of international regulatory instruments, specifically by the International Maritime Organisation (IMO) and Marine Pollution (MARPOL) regulations. The addition of significant levels of underwater or noise pollution as it can be called, has been investigated since the Second World War; however much of this research has been carried out with regards to Naval and Defence applications and requirements, rather than from any concern regarding its impact on marine life. These sounds can take various forms, including ship radiated incidental noise, sonar and seismic air guns, offshore drilling and pile-driving, and underwater explosions. Due to a recent move into polar exploration and shipping, this has also grown to include icebreaking vessel noise, and ship-ice interaction noises. It is often said, however, that the noisiest 10% of ships contribute 50% to 90% of overall noise pollution (Frank 2009); therefore the scope for improvement is significant. The exact source of the data to support this claim is unclear however it still provides an interesting view. Typically, it is referred to as "noise" when it represents unwanted sound or “wasted” energy as a by-product of normal operation, rather than the intentional creation of sound as in the case of sonar.

Ship underwater radiated noise from commercial vessels has three principle components: propulsion unit noise, machinery noise and hydrodynamic noise. The majority of its power is focussed in the low frequency ranges especially up to around 1000Hz. Incidental radiated ship noise from transit events is also a principal contributor to ambient noise levels at lower frequencies, between 5 – 500Hz, both from near and distant shipping activity (Urick 1983).

It is anticipated that global commercial shipping in all sectors will continue to see growth, in terms of both total tonnage of individual vessels, and of fleet sizes in general. This will only serve to increase the ambient underwater noise levels attributable to shipping. According to predictions presented in Fang et al. (2013):

The total tonnage of tankers is expected to grow at 1.7-1.8 times, compared to bulk carriers, containerships and LNG, which are expected to grow between 1.8 and 3 times over the next two decades.

Only in recent years has underwater noise begun to be recognised as a potentially serious threat to wildlife, in particular marine mammals and fish, with conservation groups and government departments highlighting the issues, and calling for further research into relevant fields. Serious international regulations such as those stating allowable threshold noise limits and special conservation zones could still be a long way off. Regulatory instruments in their current state are largely unsuitable for regulation of ship noise as a source of pollution, as many identify pollution to be specifically substances or contaminants rather than energies. It is also impossible to create new international regulations without political agreement worldwide, therefore options are limited to localised requirements set by flag states, or states with power over their territorial waters (Scott 2004), (Simmonds et al. 2004). However such regulations are limited in their ability to tackle the wider issue, as noise propagates freely through the world's oceans without boundary, making the regulation of it very much an international concern. Nevertheless, the idea has been initiated in the European Commission's Marine Strategy Framework Directive (2008/56/EC) which includes "noise" under its definition of pollution both as sound pressure energy and as particle velocity in Article 3.8.

Relatively little is known about either the use of noise by marine animals, or the impacts of exposure to different noise types and levels on their behaviour and physiology. There is also little data publically available on typical ship radiated noise levels, ambient underwater noise levels in different geographical locations, and models for near- and far-field propagation of radiated noise in shallow and deep water areas. All of these aspects are important as they contribute to the final received sound intensity and frequency perceived by marine animal. Little research has so far been carried out focussing on the more passive form of noise pollution known as ship radiated noise; most attention has been turned on the more deliberate noise impacts associated with sonar and airgun arrays. These intermittent impulse noises are believed to have a significant impact on some marine animal species in the very short term, however the potential short and long term impacts of the lower intensity, lower frequency, more continual ship radiated operational noises are less clear.

The main reason for this lack of comprehensive data on ship radiated noise lies in the inherently large costs associated with its acquisition. Running costs for a ship alone can amount to several thousands of pounds per day, and coupled with, amongst many other things, the hydrophone arrays, analysis equipment and the high level of tests required, the costs quickly mount up, even for relatively simple radiated noise measurements. Unfortunately, delaying investigation of a vessels acoustic characteristics until the ship is fully designed and built leaves very little room for alteration and improvement, suggesting that accurate models need to be developed, based on a suitable understanding of the composition of ship noise spectra at different ship speeds and loading conditions. The nature and mechanics of noise propagation have been the most researched area in this field, and theoretical models exist for noise transmission, attenuation and spreading mechanisms in a variety of conditions.

Ambient noise measurements differ in that the costs are associated primarily with the time-scales involved in such measurements. Ambient noise is known to vary temporally, and is also variable with geographical location, and the physical aspects of that location. Already it can be seen that the issue of radiated ship noise is a highly complex one, and modelling of spectral levels would involve substantial numbers of variables; however the problem is exacerbated by a lack of information on what is or is not “acceptable” for the marine environment.

With regards to data specifically on marine animal hearing and behaviour, it is extremely difficult to collect suitable data on animals living in the wild, and any tests on captive species, aside from taking years of training and interaction, is likely to involve only a very small number of species in an artificial environment, which is unlikely to yield representative results for the whole species. The results of these captive subject tests also tend to contain some bias, as the subject will receive a reward for a correct identification of a sound, and may err towards being more liberal, in order to increase the number of rewards received. Many behavioural aspects, and some hearing aspects, are also highly dependent on the biology of the individual, their age, the season, habituation to noise and previous experience of ships, amongst many other factors, highlighting the potential for variability even amongst a single species. The potential effects of noise on different animals also depend on an extensive list of variables, and can be very difficult to accurately assess or measure. It is generally agreed that with respect to shipping noise, the main types of impact on marine mammals will be masking, avoidance and behavioural changes and in more extreme cases, temporary threshold shift (TTS) in hearing ability.

1.4 Structure of the Thesis

The structure of the thesis is summarised briefly below.

- Chapter 2 will detail the research questions being answered in this study, and the specific aims and objectives intended to achieve this

- Chapter 3 will review the literature currently available on ship underwater radiated noise and other relevant topics

- Chapter 4 will outline the approaches adopted for the execution of this study

- Chapter 5 will provide details of the numerical modelling developed to predict non-cavitating propeller and hydrodynamic noise, along with the empirical methods used for machinery tonal prediction

- Chapter 6 will present the application of the numerical model to an LNG Carrier as a validation case study and prediction results, along with the empirical prediction results

- Chapter 7 will discuss impact assessment of ship radiated underwater noise on marine wildlife, and provide a case study as an example

- Chapter 8 will provide a discussion of the contributions to knowledge of the thesis, outlining how the aims and objectives have been achieved. It will also suggest areas for future research

- Chapter 9 will provide some concluding remarks

1.5 Chapter Summary

This chapter has provided general background details to the subject, before defining the nature of the problem to be addressed in the following chapters. The structure of the thesis has been outlined for greater clarity.

The next chapter will detail the specific research questions being addressed. It will also outline the aims and objectives to be achieved.

Chapter 2 - Research Questions, Aims and Objectives

2.1 Chapter Overview

This chapter will outline the research questions being addressed in this study (§2.2), followed by the specific aims and objectives of the work (§2.3).

2.2 Research Questions

In order to add clarity, the purpose of this study is abbreviated into the research question posed below:

"Can the underwater radiated noise of a commercial ship be predicted and assessed using information available during the early design stages of a new build?"

2.3 Aims and Objectives

The main aims of this study are to address the need for a ship radiated underwater noise prediction model, and to develop methodologies to address this need. The most suitable prediction methods for the different aspects of ship underwater noise will be selected, and then developed using full scale underwater and onboard measurement data for validation. It is also important to have way to assess the potential impact of the noise, and therefore a means for carrying out such assessments will also be addressed.

The specific objectives proposed to achieve these aims are defined below:

- To review the available literature on ship radiated underwater noise sources, modelling, impact, regulation and other relevant areas, with particular reference to how they may be applied in this work

- To develop a numerical prediction methodology for the propeller, using field underwater measurement data for validation

- To gain a better understanding of the importance of cavitation noise in relation to underwater noise

- To develop a methodology for the prediction of machinery noise, using field measurement onboard measurement data for validation

- To test the performance of the prediction methodologies for a commercial vessel, to establish the capabilities and limitations of the approaches

- To develop a means of assessing the potential impact of the ship radiated underwater noise on marine wildlife

2.4 Chapter Summary

This chapter posed the main research question to be addressed, before outlining the main aims and objectives of the project.

The next chapter will review the available literature on ship radiated noise, and other relevant subjects.

Chapter 3 - Literature Review

3.1 Chapter Overview

This chapter will introduce the fundamentals of acoustics (§3.2), before reviewing the currently available literature on ship noise sources and signatures (§3.3) It will then review topic relevant to this study in the form of ambient noise (§3.4) and underwater noise propagation (§3.5). A detailed discussion of the available modelling techniques will follow (§3.6). The effects of underwater noise on marine wildlife will be discussed (§3.7), before closing the review by focussing on regulations aspects (§3.8) and noise mitigation (§3.9). The chapter will close by identifying the research gaps which exist in current knowledge and state of the art (§3.10).

3.2 Fundamentals of Acoustics

Acoustics is the branch of physics that deals with sound and sound waves, and is hence a very wide and varied field; however the basic principles are the same throughout, regardless of the application. These fundamentals are briefly outlined below, with particular focus on those aspects which relate to hydroacoustics and ship radiated underwater noise. There is also a comprehensive body of literature dealing with this subject. The interested reader should refer to the classic texts by Ross (1976), Urick (1983), or more modern publications by Ainslie (2010) and Lurton (2010).

Sound travels as a longitudinal wave, where the energy is transferred as the medium through which it is travelling alternates between areas of compression and rarefaction in the direction of travel. These changes in pressure are the signals which are detected by the receiver as sound. The nature of the sound detected will depend on the frequency, amplitude and intensity of the sound at the source, as well as the way in which it has been propagated. The amplitude of the sound wave will represent the peak pressure per cycle, so higher amplitude will result in a louder, stronger sound than lower amplitude.

The speed of sound, which will vary depending on the medium through which the sound is travelling, and which is an important factor in the propagation of sound, is found using the equation shown below in (3.1):

$$c = \sqrt{\frac{B}{\rho}} \tag{3. 1}$$

Where: c is the speed of sound, in m/s

B is the bulk modulus (or elastic property) of the medium, in N/m^2

ρ is the density of the medium, in kg/m^3

Due to the differences in the mechanical properties of air and water, the speed of sound in air is typically around 343 m/s, whereas in water it will around 1500 m/s. The speed of sound can also be found if the properties of the sound wave are known as shown in equation (3.2):

$$\boldsymbol{c = \frac{\lambda}{f}} \tag{3. 2}$$

Where: λ is the wavelength, in m

f is the frequency of the sound, in Hz

The frequency or frequencies of the sound typically given as a value or ranges in Hertz (Hz) are also very important factors especially in the propagation of sound. Lower frequency sounds with longer wavelengths will tend to travel much further without great losses, whereas higher frequencies with shorter wavelengths will be absorbed much more easily, and hence will be propagated over shorter distances. Ship radiated sound is composed of a large range of different frequencies, at different levels, arranged into a sound spectra, which is unique to the vessel and the operational conditions it is currently in. Ship radiated sound signatures and their composition are discussed in more detail in Section 3.3 Ship Noise Sources and Signatures.

The intensity of the sound is another very important parameter, and is typically defined as the amount of acoustic power or total energy carried per unit area in the direction in which the sound is travelling; it is also proportional to the square of the acoustic pressure as shown in equation (3.3):

$$I \propto p^2 \tag{3. 3}$$

Where: I is the intensity of the sound

p is the sound force per unit area, typically given in μPa

To simplify the problem of defining sound intensity in a way which would relate it to the differing sensitivities to different frequencies of sound by humans and other creatures, the dimensionless, logarithmic scale known as the Decibel (dB) scale was introduced. This scale allows for the sound pressure level (SPL), shown in equation (3.4), and sound intensity level (SIL), shown in equation (3.5) to be defined in a non-dimensional and therefore comparable manner: if both are quoted in dB then they can be said to be equivalent. Both of these values depend on a reference value for the medium in which the sound is travelling; without these reference values, the dB level stated is effectively meaningless.

$$SPL = 20log(\frac{p}{p_{ref}}) \quad (3.4)$$

Where: SPL is the sound pressure level in dB re 20µPa in air and 1µPa in other media such as water

p is the measured sound pressure

p_{ref} is the reference pressure

$$SIL = 10log(\frac{I}{I_{ref}}) \quad (3.5)$$

Where: SIL is the sound intensity level in dB re 20µPa in air and 1µPa in other media such as water

I is the measured intensity

I_{ref} is the reference intensity

Using the logarithmic decibel scale makes it easier to deal with and visualise larger ranges, and can be used to describe perceptual levels or differences in levels, as the logarithmic scale mimics the way in which biological auditory systems operate. There are however issues with the use of the decibel scale (Chapman & Ellis 1998).

Sound pressure level is often used to report the sound level of a given noise, either at the "source" or at a more distant "receiver". This will give a single dB value for the noise, which will represent a summation of all the contributory levels at the different frequencies present. Doubling the pressure of a sound will be represented as an increase in SPL of 6dB. The table below illustrates how increases in Sound Pressure relate to the dB scale:

Table 3.1 - Variations in Sound Pressure Level with Varying Sound Pressure (André et al. 2009)

Increase in Sound Pressure	Increase in Sound Pressure Level (SPL)
1x	+0dB
2x	+6dB
10x	+20dB
100x	+40dB
1000x	+60dB
10000x	+80dB

It should be noted that due to the differences in the way sound propagates in air and water, different reference pressures are used, and hence the quoted dB values in air and water cannot be directly compared. A general rule of thumb when comparing sound intensity is shown below in equation (3.6) as derived in Simmonds et al. (2004):

$$dB_{water} = dB_{air} - 62dB \qquad (3.6)$$

To give a rough comparison, a research vessel may have a source sound pressure level of around 120 dB re 1μPa, which would equate to around 60 dB re 20μPa, the noise level experienced in a typical office. In comparison, a large commercial vessel may emit around 180 dB re 1μPa at source, which would equate to around 120 dB re 20μPa or the sound of an aircraft taking off.

Sound Exposure Level (SEL) is also sometimes used as a comparison value for noise; typically when assessing relative impact on marine wildlife, as it takes into account the duration of exposure to a given noise, as shown in equation (3.7). It should be noted that this equation is only valid when the averaging time for the SPL (rms) is equal to the duration, or when the sound is stationary.

$$SEL = SPL(rms) + 10log(Duration) \qquad (3.7)$$

Where: SPL is measured in dB re 1μPa

Duration is typically measured in seconds

3.2.1 Weighted Spectra

Stating just the SPL level can give misleading impressions of the impact of the sound, as humans and other creatures do not have the same sensitivity to sounds of different frequencies. The typical human hearing range is said to be between 20-20,000Hz, with peak sensitivity around 2,000-4,000Hz. The audiogram shown in Figure 3.1 below is an example of typical human hearing sensitivity and range. An audiogram is a graphical representation of hearing ability at different frequencies, and they are widely used in many medical and scientific applications.

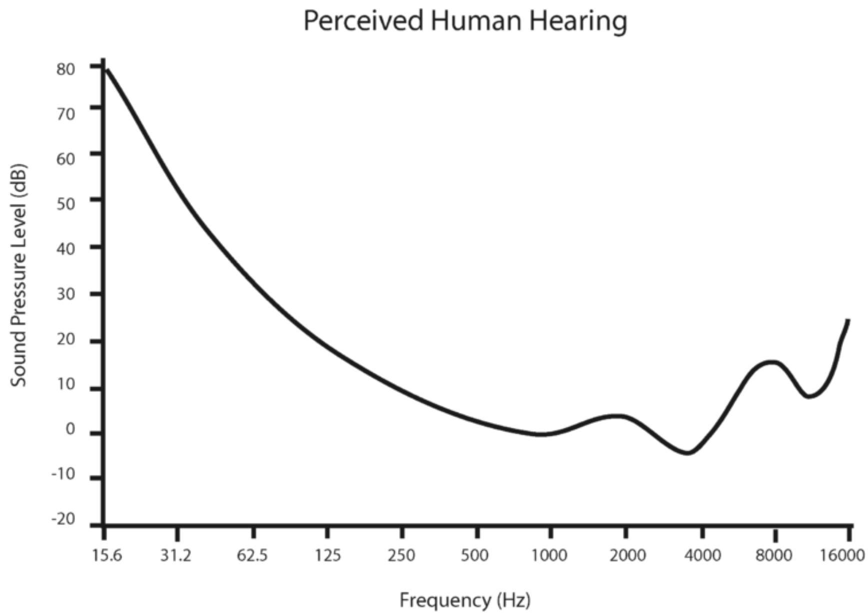


Figure 3.1 - Typical Human Audiogram

This sensitivity is reflected in the A-weighted dB scale, which takes into account these limits and adjusts the reported SPL level to better reflect the likely affect of the noise on humans, and is typically used for quieter sounds. The formula in (3.8) below is used to calculate the relative A-weighted value:

$$A(f) = 20log_{10}\left[\frac{12200^2 f^4}{(f^2+20.6^2)(f^2+12200^2)(f^2+107.5^2)^{0.5}(f^2+737.9^2)^{0.5}}\right] + 2 \quad (3.8)$$

Where: f is the frequency in Hz

B and C weightings for humans are intended for medium and loud sounds respectively. Such weighting does not yet exist for marine animals, however the application of such weightings has been suggested and researched by Dr J. R Nedwell et al at Subacoustech Acoustic Research Consultancy, in the form of a dB_{ht} (species) (Nedwell et al. 2007). This will reflect the natural sensitivities of different species to sound at different frequencies, and aim to relate specific species weighted SPL levels to observed behavioural and auditory effects.

A more generic M-Weighting has also been proposed by the Noise Exposure Criteria Group from the Acoustical Society of America. The idea behind the M-Weighting system is based on the same principles as the C-weighting system for human hearing, whereby given sound pressure levels are adjusted to reflect sensitivity to and perception of sounds of different frequencies. Although there are many different species of marine animals, they have been divided into 5 separate hearing function groups, as it is assumed that species in the same group will have a similar peak sensitivity range, which is assumed to be linked to their ranges of key vocalisation. Table 3.2 below presents these different hearing function groups. The upper and lower frequency limits are based on value for the most sensitive species in each group.

Table 3.2 - M-Weighting Functional Hearing Groups (Southall et al. 2007)

Functional Hearing Group	**Estimated Auditory Bandwidth**	**Genera Represented (# species/sub-spec.)**	**Frequency Weighting Network**
Low-frequency cetaceans	7 Hz to 22 kHz	*Chiefly Baleen Whales, including the Bowhead Whale, Right Whale, Gray Whale, Humpback Whale, Minke Whale and Blue Whale* (13 species/sub-spec.)	M_{lf} (lf: low-frequency cetacean)
Mid-frequency cetaceans	150 Hz to 160 kHz	*Chiefly oceanic dolphins and toothed whales, including the Bottlenose Dolphin, Common Dolphin, Right Whale Dolphin, Killer Whales, Pilot Whale, Sperm Whale, Beluga Whale, Narwhal, Cuvier's Beaked Whale* and *Bottlenose Whale* (56 species/sub-spec.)	M_{mf} (mf: mid-frequency cetaceans)
High-frequency cetaceans	200 Hz to 180 kHz	*Chiefly porpoises and small dolphins, including the Harbour Porpoise, River Dolphin and Hector's Dolphin* (18 species/sub-spec.)	M_{hf} (hf: high-frequency cetaceans)
Pinnipeds in water	75 Hz to 75 kHz	*Including the Fur Seals, Sea Lions, Bearded Seal, Common Seal, Gray Seal, Harp Seal, Monk Seals, Elephant Seal, Leopard Seal* and *Walrus* (41 species/sub-spec.)	M_{pw} (pw: pinnipeds in water)
Pinnipeds in air	75 Hz to 30 kHz	Same genera as pinnipeds in water (41 species/sub-spec.)	M_{pa} (pa: pinnipeds in air)

The corresponding sound level is then calculated from the formula below:

$$M(f) = 20log_{10}\left[\frac{{f_{high}}^2 f^2}{\left(f^2 + {f_{low}}^2\right)\left(f^2 + {f_{high}}^2\right)}\right] \qquad (3.\ 9)$$

3.2.2 Acoustic Sources

Underwater sound or indeed any physical mechanism that generates acoustic pressures typically occurs as one of the three dominant source types: a 0-order monopole source, a 1st-order dipole source, or a 2nd-order quadrupole source. The diagrams below illustrate these three main types of source:

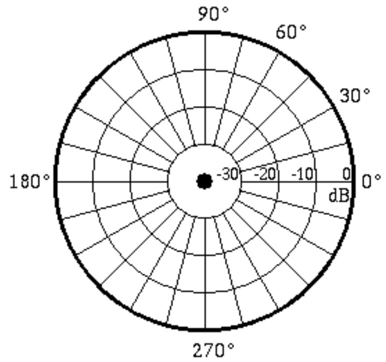


Figure 3.2 - Monopole Source (Bogdan Petriceicu Hasdeu National College 2006)

Monopole sources tend to be simple and omni-directional, and arise from volume or mass fluctuations. This is the most dominant type of source, and represents cavitation and other similar phenomena. Cavitation is a phenomena where the pressure in a given area, for example on the low pressure side of a propeller blade, falls below vapour pressure, and the water effectively boils, creating bubbles or cavities, which then collapse, causing a shockwave.

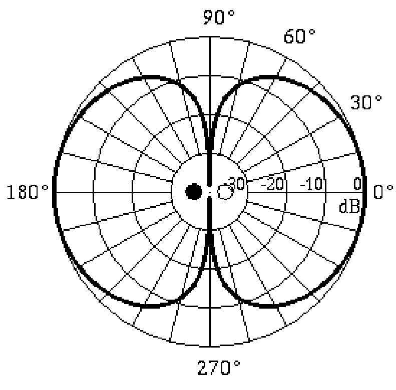


Figure 3.3 - Dipole Source (Bogdan Petriceicu Hasdeu National College 2006)

Dipole sources will typically have a cosine directional pattern and are dominant where no monopoles exist. They represent the fluctuating forces and vibratory motions of unbaffled rigid bodies, and can be used as a series of "sources and sinks" to model flow around a solid body in a fluid.

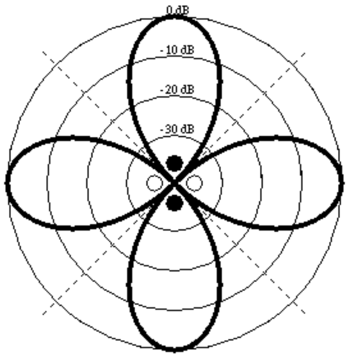


Figure 3.4 - Quadrupole Source (Bogdan Petriceicu Hasdeu National College 2006)

Quadrupole sources cover turbulent fluid motions in a fluid, to give moments and shear stresses. Unlike monopole and dipole sources, which can only occur at a fluid boundary, quadrupole sources can also occur within the fluid itself.

In his classic paper on jet noise prediction Lighthill (1952) suggested that there were three ways in kinetic energy could be turned into acoustic energy, as listed in Ross (1976):

1. by forcing the mass in the a fixed region of space to fluctuate
2. by forcing the momentum in a fixed region to vary, i.e. by exerting a fluctuating external force on it
3. by forcing the rates of momentum flux across fixed surfaces in space to vary, as by turbulent shear stresses in space

These three mechanisms are consistent with the above representations of sources, whereby the first two can only occur at a boundary, while the third can occur within the flow. These ideas will be more fully discussed later.

3.2.3 Underwater Noise

The underwater environment is filled with a myriad of different sounds, from many varied and diverse sources. This body of noise is made up of the natural ambient and biological noises of the specific location, transient and permanent industrial noise, and many other incidental noise sources (Hildebrand 2009). The ambient noise is discussed in more details in Section 3.4 Ambient Noise.

The industrial noises also take many forms. Many of these sounds will propagate over significant distances and hence will also add to the ambient background noise in more distant regions. The figure below gives an overview of some of the many different sources of underwater noise, and the key frequencies in which they are dominant:

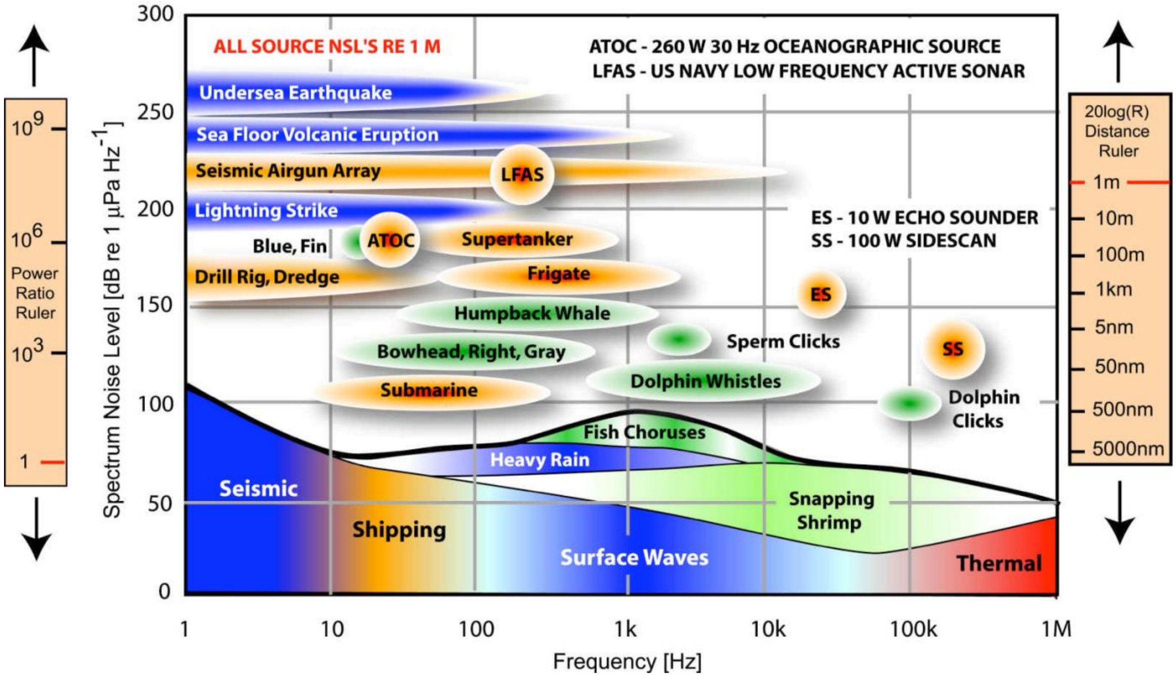


Figure 3.5 - Sources of Underwater Sound (IACMST 2006) Seiche (© Seiche Ltd. 2006) reproduced with kind permission from Seiche and Professor Rodney Coates

Figure 3.5 clearly shows the substantial range of frequencies covered by the many different anthropogenic sound sources introduced into the marine environment by the many facets of the marine industry.

This research presented here focuses on the underwater noise radiated by surface shipping during normal operational activity, rather than the intentional use of noise in airguns or sonar. More specifically, it focuses on surface vessels during normal transit, and will not address particular cases such as manoeuvring procedures.

Surface ship radiated noise signatures are made up of a large number of different components; vibrations, interactions and responses as well as many direct sound sources, and so requires detailed analysis of their composition and overall spectra, as well as the actual dB level.

In order to gain the most accurate representation of a vessels noise properties and impacts, a series of data sets is required, on all aspects of the system. These are listed below:

- *The vessels underwater noise signature* - this is affected by all aspects of the ships design, and therefore an accurate representation of this will require details of the vessels structure and layout, installed systems and machinery, operating conditions and speeds, and properties of the flow around the vessel. This signature will have both a broadband component giving a general overall SPL, as well as very narrow tonal peaks at specific frequencies associated with propulsion and machinery operational frequencies. It is therefore important to identify all the key noise sources of the vessel

- *The operational area of the vessel* - the area of operation will dictate the way in which the ship radiated noise is propagated, the typical ambient noise that already exists in the area, whether or not the vessel is operating in ice conditions, and the main marine animal species that are likely to be affected by the presence of the vessel

- *Marine animal information* - the required data includes the typical habitats of marine animals, their use vocalization behaviour and use of noise in everyday life, their hearing ability and frequency ranges, and typical responses to given noise at different distances from the source

All this information can then be used in large variety of models and formula, some more accurate than others, in order to predict the noise radiation and impact resulting from the vessels transit. These many models and prediction methods published within the public domain, and their advantages and limitations, will be discussed in various sections of this review. Key gaps and therefore potential areas for significant development will also be identified throughout, and summarised in Section 3.10 Research Gaps.

3.3 Ship Noise Sources and Signatures

Shipping noise is known to be a dominant part of ambient underwater noise levels at lower frequencies, up to around 600Hz, with a peak at around 100Hz (Urick 1983). According to Arveson & Vendittis (2000), shipping noise can increase the ambient noise level in a given area by up to 20-30 dB re 1 μPa and in deep water areas where ambient noise measurement data covers a number of decades, it has been suggested that increases of around 3 dB / decade have been observed since the 1950's (McDonald et al. 2006). It should be noted that these finding were based measurement made in deep water off the coast of California, and hence this trend may not be reflected for different areas. In fact, shipping noise now tends to exceed natural wind noise, even at high sea states (Okeanos: Foundation for the Sea 2008). As mentioned in Ross (2005) and demonstrated by United Nations (2011), this increase is attributed not only to the rapidly growing fleets of ships, but also to increase vessel lengths and service speeds causing higher source levels of radiated noise, despite advances in machinery and propulsor efficiencies. These increases in ambient noise level arise not only from ships transiting locally through an area, but also from the far-field propagation of distant shipping noise, particularly the lower frequency components of the ship noise spectra. There is less data available for shallow water areas, and hence the trends in these areas are less clear. There would appear to be a need for study of these trends, so that the impact of shipping on shallow waters coastal areas can be better understood. In a recent study by

Renilson Marine Consulting Pty Ltd (2009) commissioned by The International Fund for Animal Welfare (IFAW), it was suggested that:

> "IFAW has identified that a reduction in hydroacoustic noise of 3 dB for vessels which exceed mean noise levels of 175 dB, by one standard deviation (16% of vessels), would result in a reduction of 40% in the area ensonified to 120 dB ..... It also identified that a 6 dB reduction would reduce the corresponding area by 60%. Therefore, great gains can be made by reducing the noise output from the noisiest vessels."

Apart from relating to military or fisheries research applications, these noise levels have not been viewed as a particularly important aspect of the vessels design or operation, and few meaningful efforts have been made to accurately measure and analyse these signatures, or to take serious steps towards reducing them. This appears to be the case in spite of the significance of this issue, and the apparent gains that could be achieved through addressing it.

The diagram below gives an indication of the sources of ship noise which may impact the underwater noise spectra, and their approximate frequency ranges, originally from Crocker (1997):

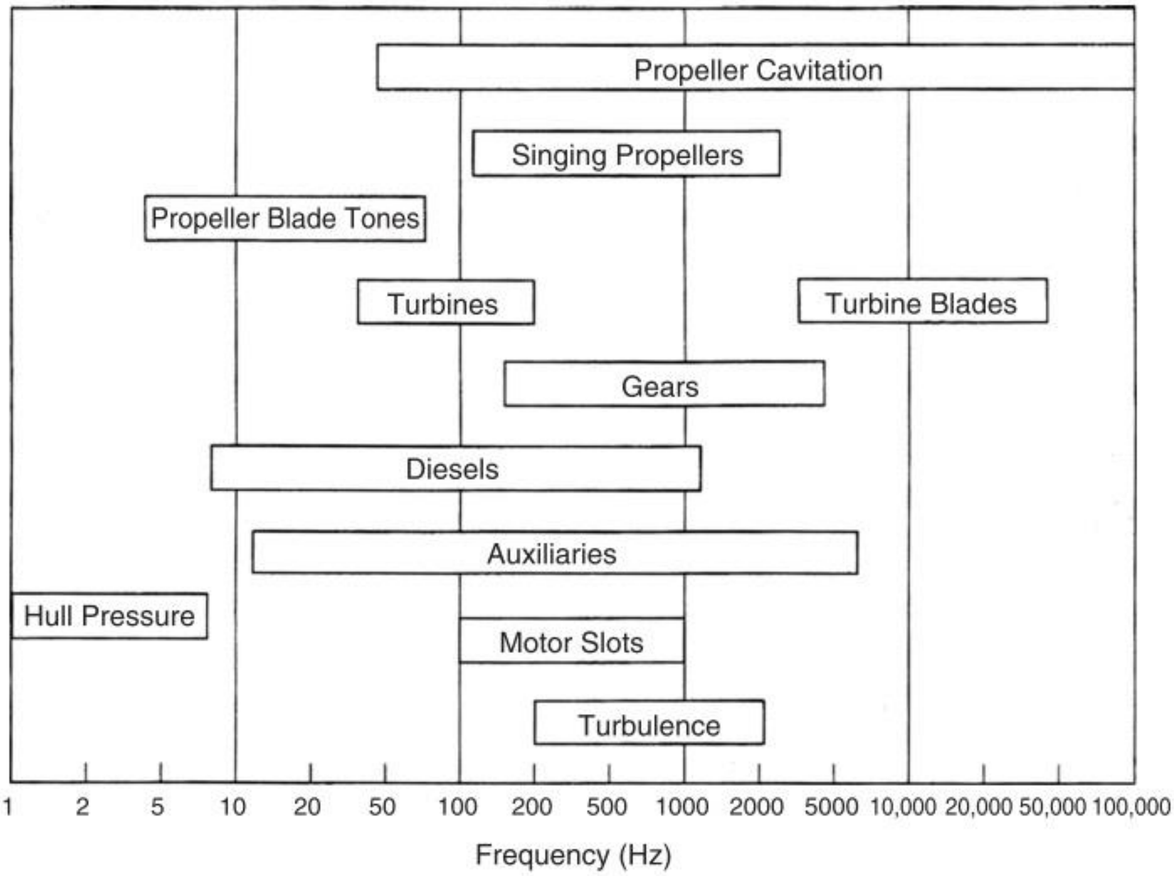


Figure 3.6 - Frequency Ranges of Noise Radiated By Onboard Noise Sources (Fischer & Collier 2007)

It can be seen that these sources in combination have influence over a very wide frequency range, although the majority of the acoustic energy is in the lower frequencies (Richardson et al. 1995). Figure 3.7 below gives an overview of the relative influence of different noise sources at different speeds. As can be seen, at lower speeds the noise comprises of contributions from machinery, propeller and flow noise. The machinery noise often takes the form of distinct tonal peaks at specific frequencies relating to system rotation or operational frequency and its harmonics. These machinery noises are usually only significant below the cavitation inception speed (CIS) of the propeller, i.e. below the point at which the local vapour pressure of the liquid rises above its local ambient pressure, causing the water to change phase to a gas.

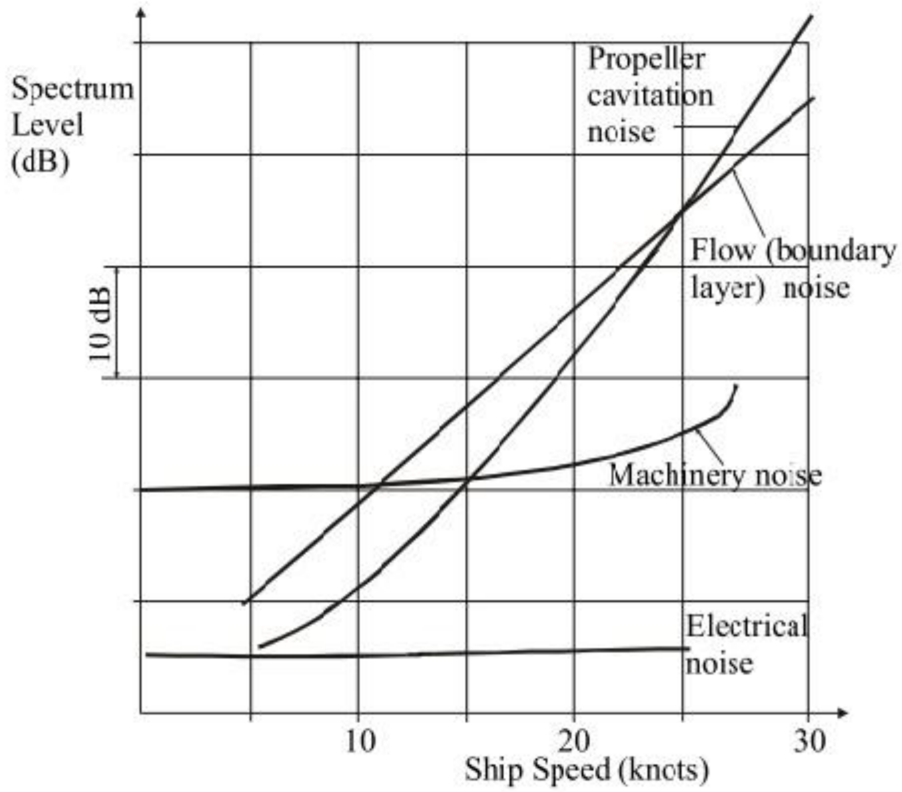


Figure 3.7- Relative Influence on Underwater Noise of Different Noise Sources (Carlton 1994)

As can be seen from the figure above, at higher speeds, i.e. above the cavitation inception speed (CIS), propulsor-induced noise becomes dominant. This takes the form of both a broadband continuous spectrum created by the randomly collapsing cavities / bubbles, and some tonal peaks relating to propeller "blade rate" and its harmonics. There are also a number of less dominant noise sources over the whole spectrum, which depends on an extensive number of factors, such as hull- and structure-borne vibration noise, resonant responses, installed systems and hydrodynamic noise. In very specific cases such as anti-submarine frigate, every measure is taken to reduce the noise and vibrations (Carlton 1994). All of these different noise sources will be discussed in the sections below.

When considering onboard noise levels, and human factors associated with noise, it is also typical to give some attention to airborne noise sources. These can arise from conversation, incidental noise such as ship's whistles, and hydrodynamic "splash" noises, as well as the components of machinery noise and vibration transmitted through the air. For underwater noise levels, however, airborne noise can be considered negligible, due to both the dB levels involved, and the difficulty in transmitting this noise to the underwater at an angle of incidence suitable for significant propagation. Airborne noise sources and propagation are therefore considered outside the scope of this research. It should be noted that in very specific cases, such as "silent" vessels, airborne noise from machinery transmitting to the hull should also be considered, with some examples outlined by Bernard et al. (2009).

3.3.1 Non-Cavitating Propeller Noise

Propeller noise is caused by the movement of the propeller blades though the water and the resulting pressures in the blades themselves. This movement also causes turbulence in the propeller wake; another source of noise. Propellers operate within the non-uniform inflow which is caused by the presence of the hull in front of the propeller, and this has a significant influence on the acoustic behaviour of the propeller, as well as efficiency etc. This noise is mostly broadband in nature, and will tend to increase with increasing rotation speed and hence propeller loading.

Other sources of propulsion noise include propeller excitation from the inflow of the wake into the propeller race, propeller "beats" associated with the rotational speed of the propeller shaft, and propeller-induced hull and rudder excitation, also known as pressure pulses. Where a vessel has two or more propellers, some interaction may also occur between the out of the different propellers.

Where an alternative propulsor, such as waterjets, or an azimuth propulsor system is used, these sources will have very different radiated noise properties. There is little information available in the literature regarding the comparative acoustic performance of these installations. However, the noise radiated by these installations specifically is outside the scope of this research.

3.3.2 Cavitating Propeller Noise

The phenomena known as cavitation, or more specifically inertial cavitation, is effectively the "cold-boiling" of liquid in an area where the static pressure of the liquid has fallen below the saturation pressure of that liquid. This phenomena and the mechanisms involved are widely discussed in the literature, with Ross (1976) and Brennen & Ceccio (1989) being just two examples. Bubbles of liquid vapour form, known as cavities. The bubble cavities fluctuate periodically creating tonal components, and then collapse, emitting significant amounts of energy as they do, which creates broadband noise and which also has the potential to erode any solid surfaces they come into contact with. Marine propellers are particularly prone to cavitation, as the rotation of the blades causes a pressure differential between the leading and trailing sides of the blades. There are a number of different types of cavitation which can occur in marine propellers, depending on the region of propeller being affected, which includes sheet, hub vortex, tip vortex, root and bubble cavitation. Figure 3.8 below, originally from Wijngaarden (2012), shows tip vortex cavitation with a bit of sheet cavitation, and how it impacts on the rudder blade.

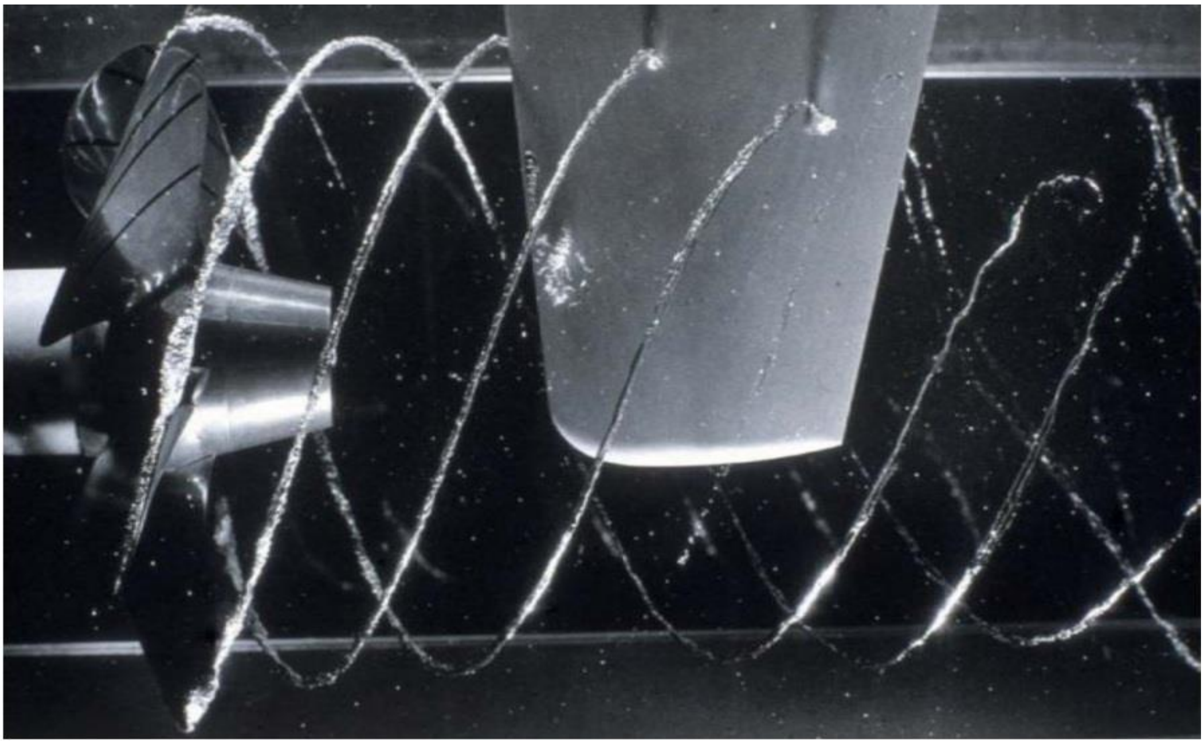


Figure 3.8 - Propeller Tip Vortex Cavitation and Impact on Rudder (Abrahamsen 2012)

The increase in underwater radiated noise sound pressure levels will vary depending on many different variables.

The cavitation number is a dimensionless value which can describe the potential for a flow to cavitate. Each scenario has a critical value after which cavitation inception will occur. This cavitation number is calculated as shown in the formula below:

$$\sigma = \frac{p - p_v}{\frac{1}{2}\rho_l U^2} \qquad (3.10)$$

Where: σ *is the cavitation number*

p *is the static pressure in Pa*

p_v *is the liquid vapour saturation pressure in Pa*

ρ_l *is the liquid density in kg/m*^{3}

U *is the incoming flow speed in m/s*

Alternatively, the cavitation number can also be calculated based on the rotational speed of the propeller using the below formula:

$$\sigma_n = \frac{(p_0 - p_v)}{\rho n^2 D^2} \qquad (3.\ 10)$$

Where: p_0 *is the total static pressure in Pa*

p_v *is the vapour pressure in Pa*

ρ *is the liquid density in kg/m*^{3}

n *is the propeller revolutions per minute*

D *is the propeller diameter in m*

To examine the cavitation characteristics, a "cavitation bucket" diagram, as shown below in Figure 3.9, is plotted for the propeller, in order to define the boundaries of cavitation-free operation:

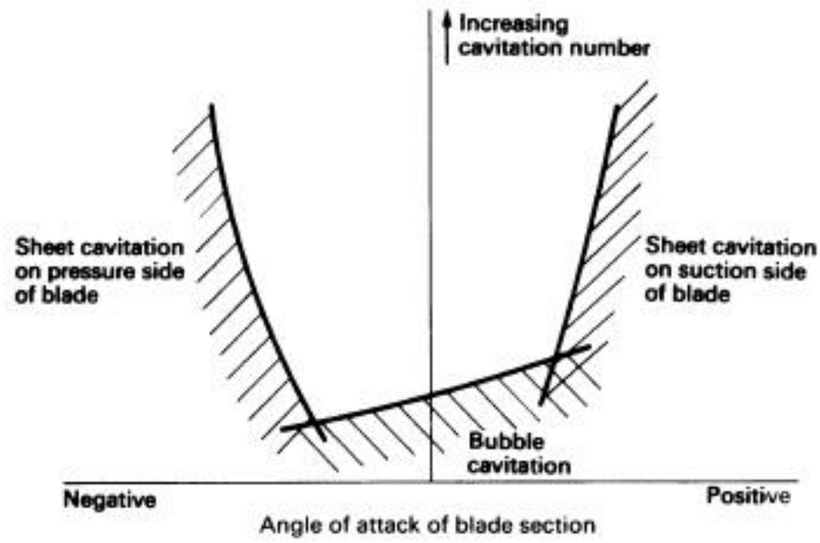


Figure 3.9 - Cavitation Bucket Diagram Example for a Propeller Section

(http://www.boatdesign.net/forums/boat-design/speed-vs-cavitation)

Figure 3.10 below shows the hypothetical radiated noise against propeller RPM for two vessels with different CIS. It demonstrates the potential gains which could be made by improving the CIS if only by a small amount, and also the gains which could be achieved by operating at a slightly lower speed.

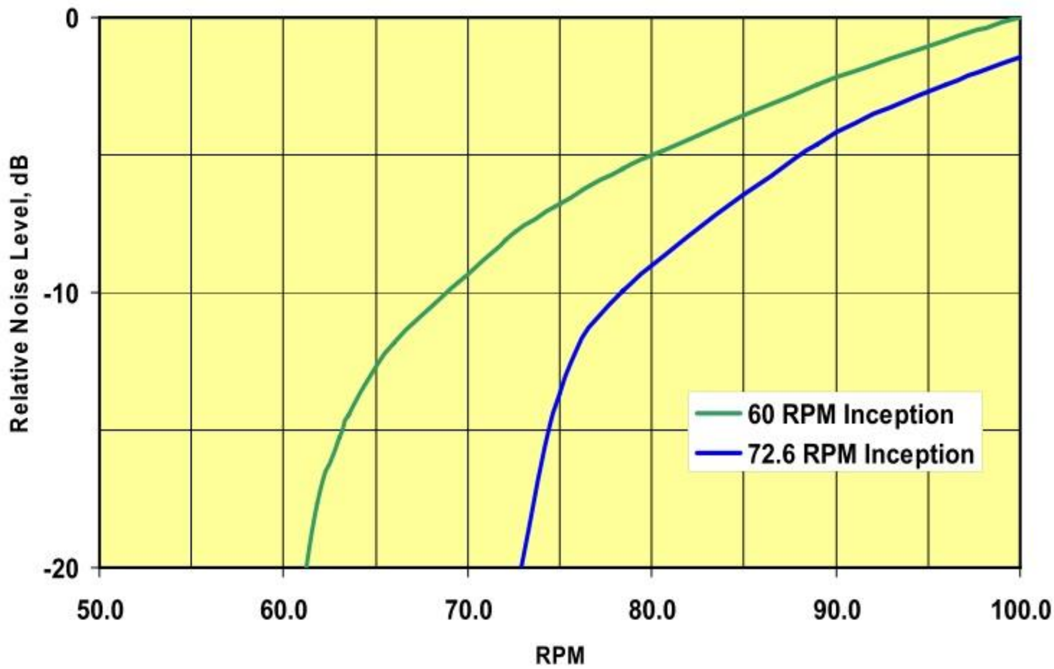


Figure 3.10 - Relative Noise Level Vs Propeller RPM for Vessels with Different CIS (Spence et al. 2007)

By far the most dominant aspect of propulsion noise is propeller cavitation noise; in fact when cavitation occurs, it is generally the most dominant underwater noise source in the whole ship system. This is largely due to the fact that the cavitating propeller is located in the water, and therefore propagates noise much more effectively into the surrounding area. It also causes significant increases in the amount of turbulence and other non-linear sources, which can become dominant in the far field. Cavitation noise can also cover a very wide range of frequencies, typically 50-100,000Hz, and has both tonal and broadband attributes, although it is generally only considered dominant in the lower frequencies (Southall & Scholik-Schlomer 2007). The tonal values are associated with the blade rate frequency of the propeller and its harmonics, while the broadband noise arises from the non-uniform collapse of the cavitation bubbles (Vassenden & Lovik 1981). There are also many different forms of cavitation as indicated above, including sheet, bubble, tip vortex, cloud and hub vortex cavitation. Blade rate cavitation and propeller "singing" phenomena tones are related to the propeller modes and vortex shedding. Tip vortex cavitation can be a major noise source and is often mode dominant than sheet, and also tends to be more broadband and higher frequency in nature. However the sheet cavitation should not be neglected when considering the underwater radiated noise especially for higher power vessels. It is less of an issue for moderate or lower power vessels and tends to be more focussed in the lower frequencies. Due to the stronger tip vortex with bigger bubbles observed in a highly loaded propeller, it will have lower frequency and higher amplitude noise compared to a less loaded propeller.

These types of cavitation all arise in different areas of the propeller, for different reasons and at different times during operation. Consequently, given the very non-uniform nature of hull wakes on commercial vessels, it is very difficult to design a non-cavitating propeller for a vessel operating above around 8 - 9 knots. Banks et al. (2013) presents data on current commercial vessel operating profiles, and specifically speed distributions. It can clearly be seen that despite recent trends in reducing vessel speeds for emissions, economics dictate that the majority are still operating at speeds which would correspond to cavitation conditions therefore speed reduction alone is not a suitable measure for reducing this noise source. Designing entirely cavitation free propellers will be very detrimental to their propulsive efficiency therefore a measured level of cavitation on the propeller blades is required. The Burril diagram (Carlton 1994) presents limits for acceptable limits for the amount of cavitation coverage on the propeller blades, with approximately 10% being a typical limit. Above 25%, thrust breakdown may occur and again needs to be avoided. The aim is thus to gain a good understanding of the mechanisms and design aspects associated with the cavitation phenomena, in order to improve propeller design for delayed and improved cavitation performance. It should be noted that it is not only in Naval applications that significant improvement in cavitation performance have been achieved; the cruise ship industry has also made notable progress in this area, with some vessels now having a CIS of around 13-14 knots (Vie 2013).

3.3.3 Machinery Noise

The huge complexity of large commercial vessels means there is a vast array of machinery, equipment and systems installed, each of which contribute in some way to the machinery noise of the ship. Clearly, the largest pieces of equipment such as the main engines, turbines and generators will form the most dominant sources of onboard noise (Nilsson 1978). As stated by Fischer & Brown (2005), their radiated noise is typically a combination of tonal peaks associated with rotating or reciprocating frequencies, with some contribution to broadband aspects associated with the vibrations of the equipment being propagated to the water. Other sources of machinery noise arise from the Heating, Ventilation and Air Conditioning (HVAC) systems, gearboxes and turbochargers, pumps, and other auxiliary systems as observed from onboard measurements in Zoet et al. (2010b). A significant body of research has been carried out regarding machinery and onboard noise, over a long period of time, and hence there is a wealth of information available in the literature, such as (Junger 1987), (Filcek 2006), (Zinchenko 1957) and (Fischer et al. 1983). The sample spectra seen below, taken from the bulk cargo vessel M/V Overseas Harriette clearly indicates the very tonal nature of the lower frequency section of the noise spectrum associated with machinery noise, and also how these peaks and their harmonics can be attributed specifically to different items of machinery. The two spectra shown below represent operation at 68 and 140 RPM, with keel-aspect underwater measurement data presented in narrow bands:

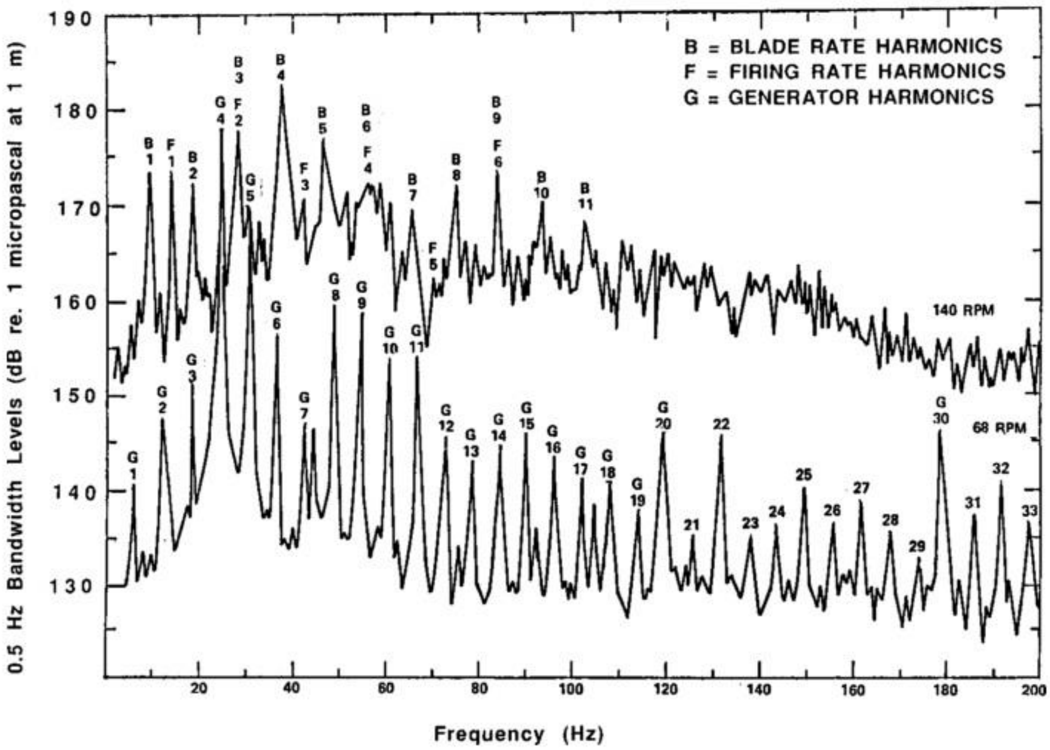


Figure 3.11 - Sample Machinery Noise Spectra (Arveson & Vendittis 2000)

Some manufacturers will provide information of the noise emissions of their equipment however this is not the typical. The direct noise of a particular machinery noise source is very difficult to measure once it is installed, because the measurement is always made with the supporting structure and any isolation mounting attached, and this will affect the noise properties measured. It is nevertheless imperative that accurate information on the different noise sources is obtained. A good knowledge of the noise properties of a given piece of equipment in different operational conditions, and at different speeds and loadings, allows the designer to use appropriate mounting and isolation techniques to minimise the transmission of this noise both as airborne noise and as structure-borne vibration. This information is also used in the prediction of the vessels overall noise signature, and therefore the better the noise properties of the equipment are understood, the more accurate the estimated noise signature can become.

Once a vessel is in service, onboard measurements can be carried out to estimate the noise properties of the various sources. Outboard underwater noise measurements can also be made to gain the overall noise signature, however it can be very difficult to separate the contributions of individual sources in this overall spectra. Such measurements can be used to validate models and estimations of both the noise radiated by different equipment, and the overall spectra of the vessel, including the tonal peaks arising from different sources.

The electrical supply and propulsion systems on board a ship, although serving different purposes, are often hard to separate in terms of the equipment associated with them, and hence they are commonly known collectively as the Power Plant (Woud & Stapersma 2002). The following section will discuss the different power plant options typically used in commercial vessels, and their associated acoustic properties. Detailed diagrams showing the different components and typical layouts of the different system types can also be found in Society of Naval Architects and Marine Engineers (1992), which also includes a chapter on noise sources and control of shipboard noise.

Diesel Direct Drive

Mechanical power plant systems are typically made up of three key sections: the prime mover, the transmission and the propulsor. In direct drive diesel systems, the low-speed engine runs at a sufficiently slow rpm that it can be directly connected to the propeller via a propeller shaft. These systems are large and heavy, but also reliable and efficient. The electrical supply is often provided by a shaft generator and diesel generator sets. As a fixed pitch propeller is often used, additional systems may be required with the shaft generator, to deal with the variations in voltage frequency which will arise. Key sources of noise will include tonals and structure-borne noise from the auxiliary engine and diesel engine, and tonals from the generator and propeller shaft(s). A significant proportion of commercial vessels are propelled by 2-stroke diesel engines, which typically run at lower speeds, with an RPM of around 70 to 120 (Okeanos: Foundation for the Sea 2008). Due to their size and weight, they cannot be mounted on resilient isolating foundations, but are instead secured directly to the ship structure (SILENV Consortium 2012a). This means that there is a direct propagation path for noise and vibration created by the vessel to the structure, and a considerable amount of this will be further propagated into the water. It is therefore typical to see clear tonal peaks in the low frequency section of the underwater radiated noise spectra associated with the engines as shown in Fischer & Brown (2005). Their contribution to the underwater radiated noise spectrum of a vessel is typically seen chiefly in the form of tonal peaks which correspond to the rotational speed of the engine, and its harmonics, as can be predicted as shown in the paper. Diesel systems in general are stated to be the most noisy propulsion system options. It might be that current focus by the IMO on emissions from shipping, and the options for alternatives or improvements outlined by the Royal Academy of Engineering (2013) will also act as a secondary impact to improve propulsion plant noise characteristics, with improvements in engine efficiencies and moves to use alternatives to Heavy Fuel Oil (HFO) and diesel. An example of such a system and the noise sources is presented below:

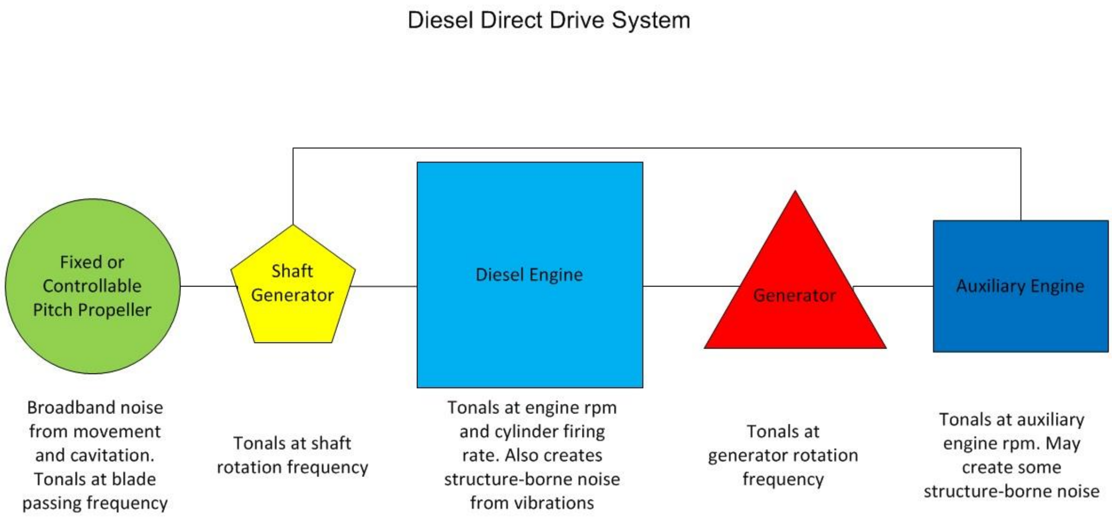


Figure 3.12 - Diesel Direct Drive System Components and Noise Sources

Diesel-Geared Drive

Diesel-geared drive systems, the medium- or high- speed diesel engines run at RPM values which are above the operational range of the propellers, and hence the shaft speed needs to be reduced in the transmission stage by a gear box (Woud & Stapersma 2002). The medium-speed engines are used are in the smaller commercial vessels with lower power output requirements, while the high-speed engines are particularly suited to use in fast ferries, patrol boats and the like. The electrical supply is again provided by a shaft generator and diesel generator sets; however as controllable pitch propellers are more common, the issues of voltage variation are eliminated. Key noise sources will be as for the direct drive system, with additional tonals associated with the gearbox. The majority of the remaining commercial vessels not using a diesel direct drive system are powered by smaller and lighter 4-stroke diesel engines. These will be run at higher speeds, of RPM around 500. Their smaller size and weight means they can be mounted on resilient isolation mountings which are discussed in Andreau (2010) and are more often used in smaller research and luxury vessels (Salm et al. 2013). These help to damp the vibrations of the engine, and reduce the amount of noise and vibration which is propagated to the ships structure and hence to the water, reducing the tonal component of underwater noise, and also improving onboard noise performance. For this reason, the contributions of a resiliently mounted 4-stroke engines are much less distinctive in underwater noise spectra, however they are still considered noisier than other propulsions systems. Again, an example of a typical system with its key noise sources is presented:

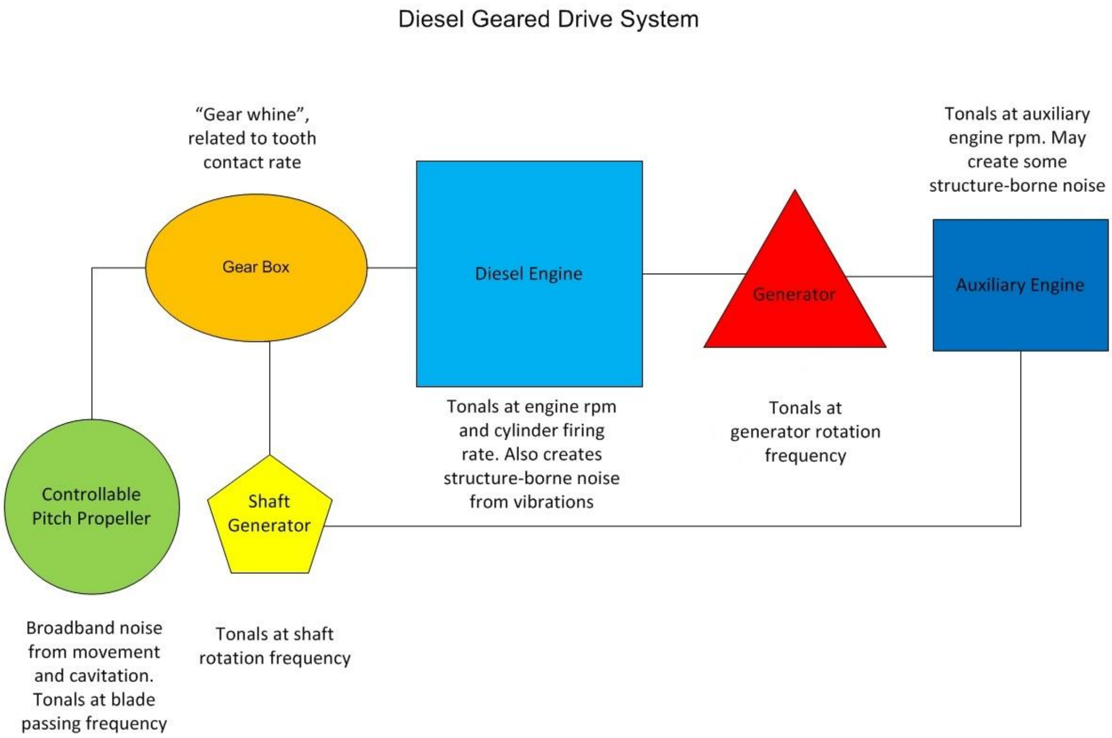


Figure 3.13 - Diesel-Geared Drive System Components and Noise Sources

The majority of the remaining commercial vessels are powered by smaller and lighter 4-stroke diesel engines. These will be run at higher speeds, of RPM around 500. Their smaller size and weight means they can be mounted on resilient isolation mountings. These help to damp the vibrations of the engine, and reduce the amount of noise and vibration which is propagated to the ships structure and hence to the water. For this reason, the contributions of 4-stroke engines are much less distinctive in underwater noise spectra.

Gas or Steam Turbine Geared Drive

Gas turbine systems are much less common however they tend to be used in Naval and some ferry and cruise vessels, where their high power-to-weight ratio is a key factor (Royal Academy of Engineering 2013). Steam turbines are very uncommon in modern commercial vessels, except in very specific application such as LNG carriers and Naval vessels. This mostly due to their lower fuel economy, low power density, and high capital cost, however as discussed by the Royal Academy of Engineering (2013) they may become a more popular choice in the future. They are however well suited to use in LNG Carriers, as the natural boil-off from the LNG cargo can be used as fuel. The high rotation speed of the turbines mean a gearbox is always required. Steam turbine units are however well known for being quiet, with low levels of vibrations (Fischer & Brown 2005). Any noise which is observed will tend to arise from any auxiliary engines and generators and the gear box, rather than from the steam turbine or boiler. Therefore from a pure underwater noise performance consideration, these systems are a very appealing option. The diagram below illustrates the main components of such a system, as well as the key noise sources:

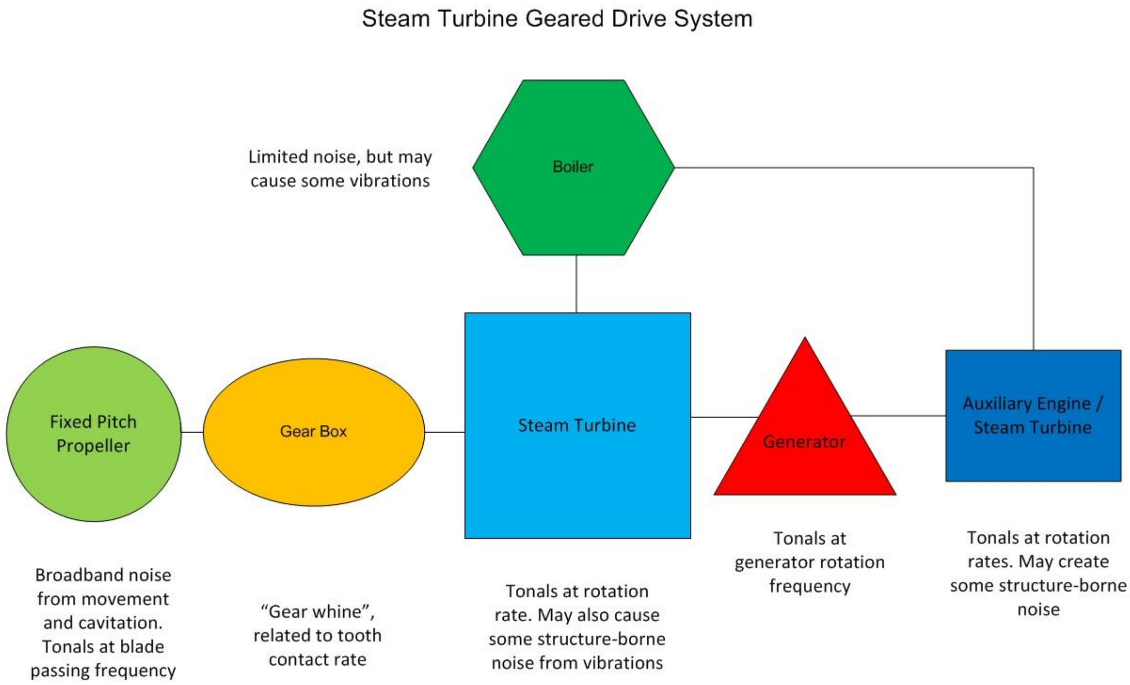


Figure 3.14 - Steam Turbine Geared Drive System Components and Noise Sources

Electric Drive

In electrical systems, the prime mover is used to drive a generator, which is used to power an electric propulsion motor.

Modern commercial ships using electric systems tend to have an integrated electric power plant for powering both the propulsions and the auxiliary systems. The prime mover can be any of the above mentioned mechanical systems. This system offers flexibility in terms of arrangements, and equipment size and location, and hence greater control is afforded over the noise and vibration characteristics. This type of drive also allows the designer a choice in propulsors, including fixed and controllable pitch propellers and podded propulsion. The diagram below gives an example of such a system, along with its key noise sources:

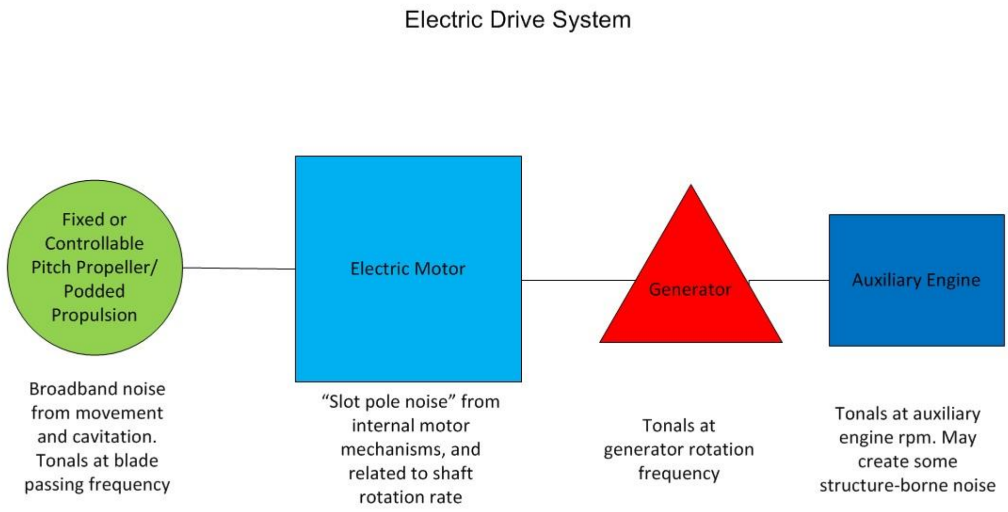


Figure 3.15 - Electric Drive System Components and Noise Sources

Hybrid Drive

Hybrid drive systems are made up of a combination of mechanical and electrical methods, for example with a mechanical system for the propulsion along with a DC electric drive, and a separate AC electrical system for the power supply (Bucknall 2013). The main noise sources are again highlighted in a sample system show below:

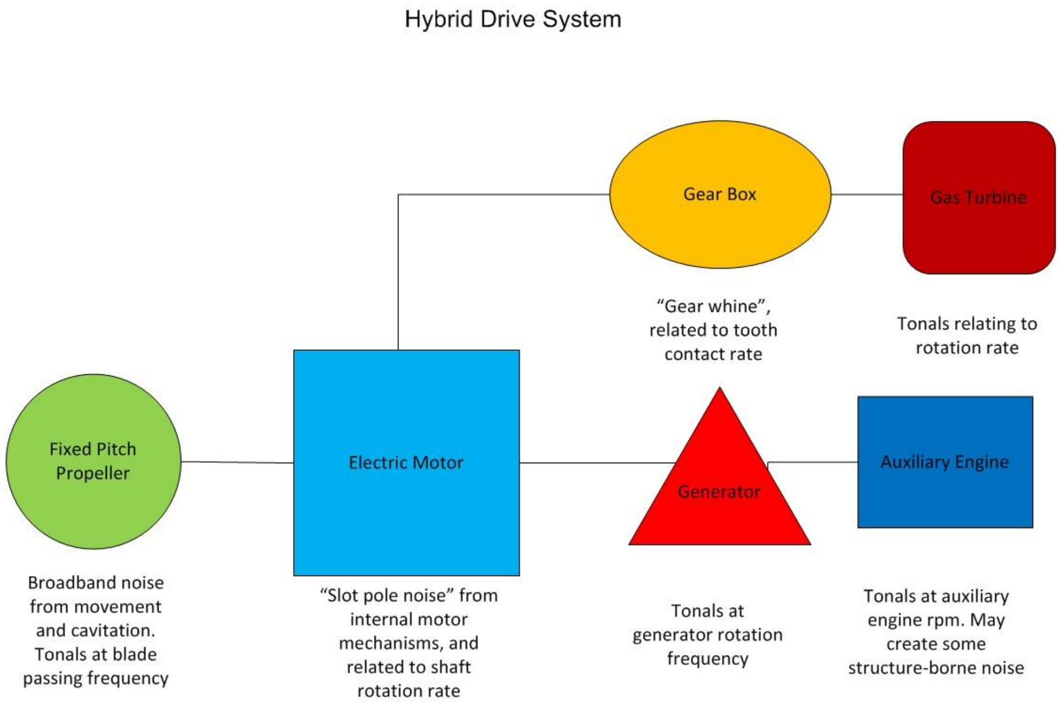


Figure 3.16 - Hybrid Drive System Components and Noise Sources

A small number of ships are powered by alternative propulsion and powering methods, whose noise contributions may need to be considered on a case by case basis, as there is very little information available in the literature for these systems. Further research should also be carried out on the relative acoustic performances of the future propulsion approaches (Royal Academy of Engineering 2013).

Onboard auxiliary systems installed on a ship will vary greatly depending on the type and purpose of a ship. All will have some acoustic properties associated with them, however in general, these smaller and less highly powered systems will have little impact on the underwater noise of the vessel as mentioned by Grelowska et al. (2012). The exception may be in cases where the systems include sea-connections, or where there is less dominant noise from the larger power plant system.

The Heating, Ventilation and Air Conditioning (HVAC) system for example can be a significant source of noise in onboard areas, however this noise will not tend to propagate through to the water surrounding the hull.

There are many aspects of the operation of the vessel which could impact on the noise and vibration characteristics of the vessels installed machinery systems. The first of these is ship speed. Although there is a speed dependence of machinery noise in relation to the main engines, some machinery items such as the generators will run to supply hotel load requirements, which will be independent of speed. Hotel load here is defined as the base electricity requirement for the ship, including powering equipment, lighting, heating etc. The variations in hotel load requirements will also therefore have an impact on the machinery noise contributions from generators etc, however as these variations are very difficult to predict, especially at an early stage, they will be disregarded. It should also be noted that generally, such machinery items only make a very minor contribution to ship underwater noise hence this omission of noise variations will not be detrimental to the results. Onboard data gathering for in-service vessels may be a way to establish trends and typical fluctuations, not only in terms of hotel load but also general ship operating profiles. The loading on the propeller will also vary the noise emitting from the main engines, which could also be assumed to have some speed dependence, amongst other factors discussed previously and below. The cargo or ballast loading of the ship could also have an impact on the propeller and engine loading and emitted noise observed. Some trends in operating profiles, taken from operational data are presented by Banks et al. (2013). Another significant contributor to propeller loading variations and general noise emissions are weather conditions and sea state as researched in detail by McKenna et al. (2013). The more severe they are, the greater the loading on the propeller and therefore the main engine is likely to be, as it will need to do more work to continue propelling the vessel forward at the same speed.

3.3.4 Vibration and Structural Response

All moving elements onboard, such as machinery and equipment, will cause the adjacent and supporting structures to vibrate. This vibration is then transmitted through the structure, causing the hull and appendages to vibrate, creating noise both onboard and underwater. Airborne noise from both the machinery and the structural vibrations will also cause further noise and vibrations to occur in the vessel, although to a much lesser degree than through direct contact vibration. The diagram below illustrates these connections:

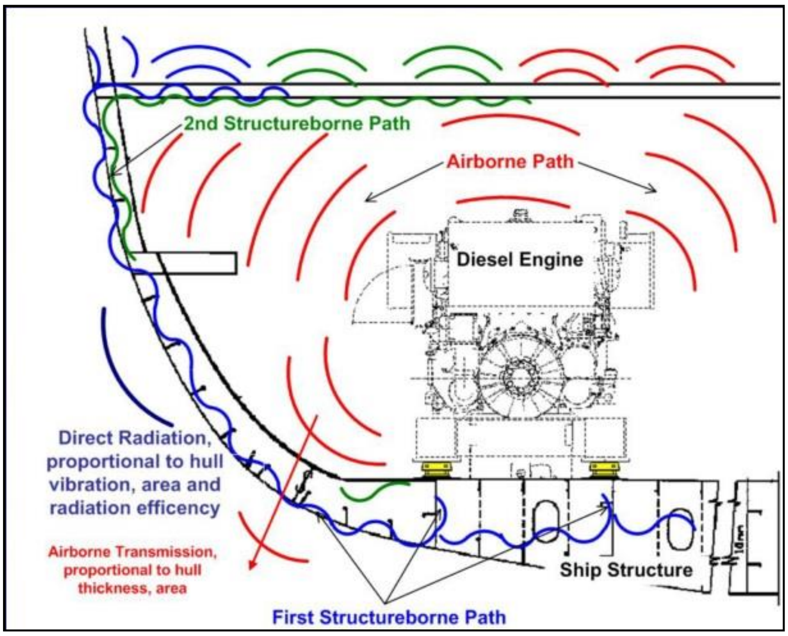


Figure 3.17 - Structural Vibration Transmission Paths (Fischer 2004)

In severe vibration cases, and at certain frequencies, the vibration can also cause resonance responses in some parts of the structure. This creates elevated levels of noise, and can also cause structural damage through fatigue and deformation, and should therefore be avoided wherever possible. Considering this, it is important to understand the excitations and responses of the vessel in order to achieve designs for low vibration (Zoet et al. 2012). Unintentionally "tuning" a vessel to an unsuitable natural frequency can significantly amplify fairly low vibrations from onboard sources, unnecessarily adding to the noise generated by the vessel. However as the paper acknowledges, this is a complex problem which is still being researched, and is also something which should be considered for the design from the outset. As a greater body of full scale onboard measurement data become available, so the understanding of vibration sources and propagation will increase, meaning that such research also of great importance.

As well as the onboard sources of vibration, water flow around the hull, appendages and openings all cause the hull to vibrate. The cavitation and very turbulent flow around the propeller can also creates a significant amount of vibration both to surrounding hull structure and also back along the propeller shaft and into the ship if resonance occurs. The impact of these additional outboard sources of vibration should also be considered when investigating the noise and vibration properties of the vessel, both locally and as a global entity, and in particular, the natural frequencies of the system should be avoided.

3.3.5 Hydrodynamic Noise

Hydrodynamic noise covers any noise arising from ship-water interactions, and tends to be the least dominant source of ship radiated noise; nevertheless it cannot be disregarded for smaller vessels. This can include flow noise associated with the turbulent boundary layer, "splash" and noise arising from breaking bow and stern waves, resonant excitation of the hull and external structure, and cavitation occurring at struts and other appendages where these are present. Hull features such as thruster tunnels and sea-chests will also have an impact on flow noise. However as stated by Jong et al. (2009), while the control of propeller noise is well understood, there is a shortfall of knowledge of the mechanisms that govern the noise due to the flow around the hull of a ship. The paper therefore investigates the mechanisms and concludes that the noise if mostly generated by bubbles creating as the surface waves break, and to turbulent excitation of the hull. The nature of these mechanisms was observed to change with vessel speed. This is often modelled using a CFD code, as the distribution of hydrodynamic pressure around the hull surface can be used to represent the hydrodynamic noise properties of the vessel at a given vessel speed.

3.3.6 Ship Underwater Noise Signatures

The sources discussed above are then combined into a ship noise signature or spectra, which relates to the corrected Source Level (SL) sound pressure values of the vessel over a given frequency range, at a particular set of operational conditions. The example spectra below in Figure 3,.18, taken from McKenna et al. (2012) gives an example of the spectra for a range of modern vessel types. It can be seen in this figure that as discussed previously, the majority of the underwater noise radiated from a commercial vessel is focussed in the lower frequency ranges. It is precisely these lower frequencies that are of interest when considering shipping noise, as these are the frequencies that are likely to propagate greater distances with less attenuation. They should also therefore be the focus of efforts aimed at reducing the impacts of ship and shipping noise on marine wildlife. At lower speeds, it can be seen that there are also some noticeable peaks in the lower frequency ranges, which may be due to machinery noise. As the speed increase, the broadband hump at lower frequencies, typical of propeller and cavitation noise influences, becomes more prominent. At the higher frequencies, where the noise is consistently due to the broadband noise from the propeller, the shape remains broadly similar, with only the associated sound pressure levels increasing with speed.

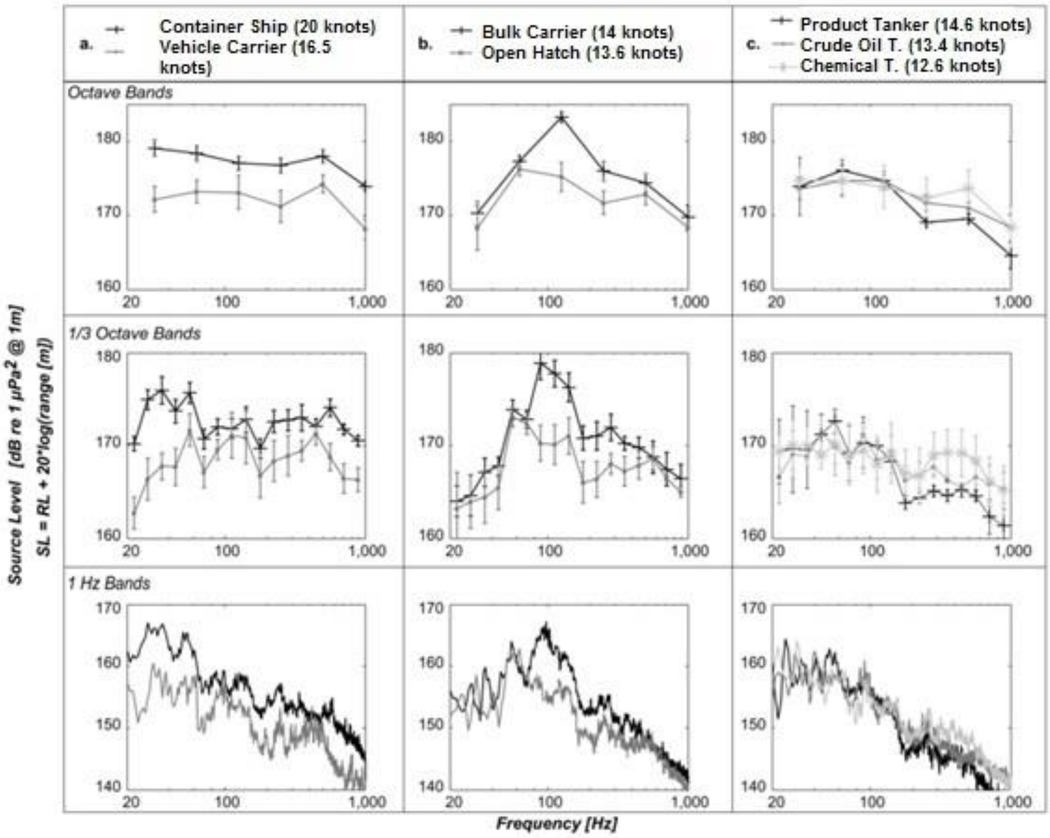


Figure 3.18 - Modern Commercial Ship Noise Spectra at Octave, 1/3 Octave and 1Hz Bands

At lower speeds, it can be seen that there are some noticeable peaks in the lower frequency ranges, which may be due to machinery noise. As the speed increase, the broadband hump at lower frequencies, typical of propeller noise influences, becomes more prominent. At the higher frequencies, where the noise is consistently due to the broadband noise from the propeller, the shape remains broadly similar, with only the associated sound pressure levels increasing with speed.

A ship's underwater radiated noise will also have a directivity associated with it. Below is an example of a directivity pattern, in this case for propeller cavitation noise in the lower frequencies for the vessel travelling at 14 knots. An increase of 5-10dB may be expected for typical large commercial vessels for stern-aspect noise, i.e. facing the propeller location, over bow-aspect noise, i.e. away from the propeller location, as observed by McKenna et al. (2012), as the hull itself creates a barrier between the main noise sources of the propeller and machinery and this area. This will not be the case for more unconventional vessels such as Ro-Ro ferries. The lack of symmetry between the port and starboard sides is also typical, as this will depend on, in this case, the direction of rotation of the propeller. Plots incorporating the machinery noise elements are also likely to show some non-symmetry where different machinery items are installed off the centreline of the vessel.

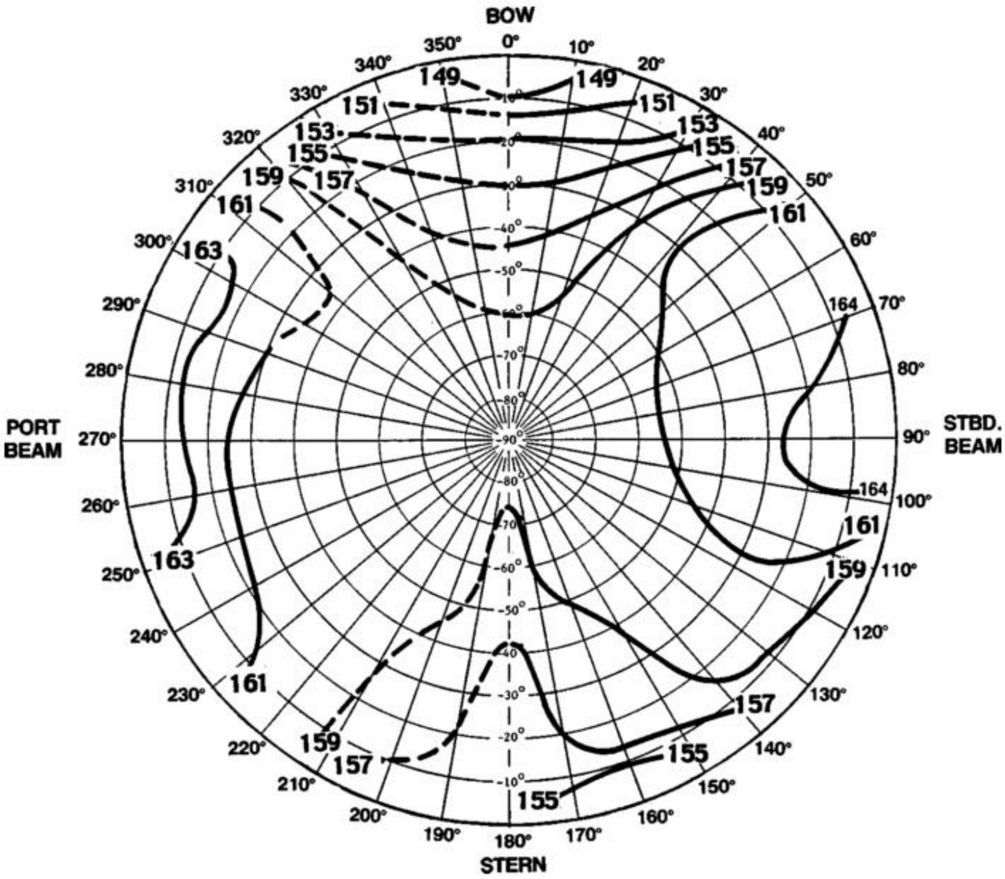


Figure 3.19 - Directivity Pattern for MV Overseas Harriette, Propeller and Cavitation Noise (Arveson & Vendittis 2000)

3.4 Ambient Noise

The term ambient noise refers to the existing typical background noise level, and is a measure of the Sound Pressure Level (SPL) at a given location. It is often used as a reference level against which to assess new noise sources in the same location. In the case of underwater ambient noise, it serves as a reference for several reasons; it can be used as a baseline value against which to compare the source level noise of a transiting vessel, it gives an indication of the typical noise level to which marine animals residing in the area are accustomed, and it also give a value above which to assess the vessel noise in terms of the critical ratio. The critical ratio is a dB value which indicates how much louder a noise source has to be above the ambient level for it to be detected by a marine animal species, at a given frequency or range of frequencies. It is therefore important to have an idea of the typical ambient noise levels in a ships area of operation when assessing the underwater radiated noise impact.

The ambient noise can be influenced by a very wide range of different factors, including:

- Geographical effects and location
- Effects arising from the nature of the water
- Natural effects such as thermal noise and underwater seismic activity
- Biological noise
- Meteorological effects

The ambient noise curves shown in the figure below, based on measurements carried out by Wenz, give an indication of these different ambient noise sources, and their associated sound pressure levels and frequency ranges (University of Rhode Island 2013).

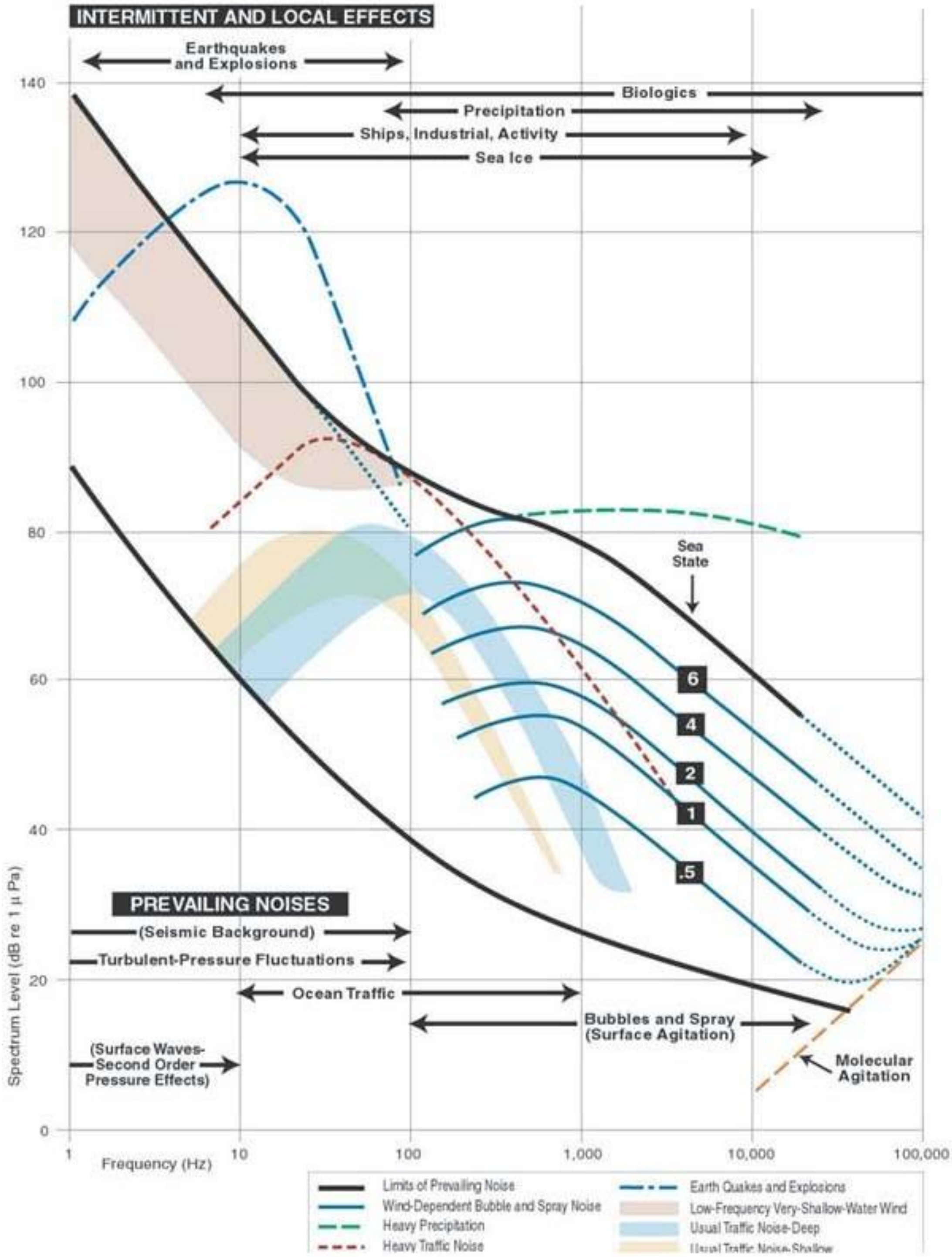


Figure 3.20 - Sources of Ambient Underwater Noise (University of Rhode Island 2013)

Ambient noise measurements are typically made with static underwater hydrophone arrays monitoring a given location over a significant duration of time. This data then incorporates diel, seasonal and meteorological variations in the noise levels measured. Such data is available from a number of sources, with historical averaged data for specific locations worldwide.

A key project in this field is currently underway at Universitat Politècnica de Catalunya, and is sponsored by a number of industry partners. The main aim of "Listening to the Deep Ocean Environment" or LIDO is "long-term monitoring of Geo-hazards and Marine Ambient Noise in the Mediterranean Sea and the adjacent Atlantic waters". Using a series of permanent seafloor observatories in a number of locations around the Mediterranean Sea and Atlantic, they can record real-time acoustic data on both general ambient noise levels and how these are affected by the presence of anthropogenic activity, and also on biological sounds by species of marine animals living in the area. This data will be used for a number of different key research areas including marine animal population and migration investigation, characterisation of marine animal bioacoustics, anthropogenic noise impacts on ambient noise and marine animals, and possible effect on ambient noise from climate changes. Further information is available on the project website (Laboratori D'Aplicacions Bioacustiques 2010). Some further work on acoustic contamination recently carried out by the LAB (Laboratori D'Aplicacions Bioacústiques) at Universitat Politècnica de Catalunya was to acoustically map the Spanish coastline. The data collected during measurements made throughout 2007, and displayed in an interactive format on the project website from Universitat Politècnica de Catalunya (2009) shows vessel transits with associated noise levels, and also distributions of marine animal species in the area. Such information and research is invaluable for developing and extended the current state on knowledge in these key areas, and allowing for increased understanding into the potential impacts of increasing anthropogenic noise on marine animals.

Data on the various factors which can affect ambient noise, such as average weather and sea conditions are typically monitored in important shipping area, and such information can be easily gained from shipboard weather monitoring systems.

For bathymetry data, there are a number of resources such as the ETOP01 system which provides a 1 arc-minute global relief model, with "ice surface" and "bedrock" versions available. GEBCO (General Bathymetric Charts of the Ocean) also produced gridded bathymetry data sets, both from 2008 data, on either a 30 arc-second or 1 arc-minute grid. For more up-to-date information, NASA hosts a Digital Bathymetric Data Base Variable resolution (DBDBV) with data from the U.S. Naval Oceanographic Office (NAVOCEANO) with data on a variety of different grid resolutions. This data can be accessed from the website (Goddard Space Flight Centre NASA 2011). Data on water temperature and salinity variations, as well as chemical compositions, at different depths, for different global areas can be accessed via the online World Ocean Atlas resource at (NOAA 2009).

As discussed before, distant shipping noise also makes a significant contribution to ambient noise across much of the world's oceans. Unlike the ambient noise spectra above 300Hz, which tend to correlate well with meteorological effects, the spectra below 200Hz seems almost independent of these effects, and the range of 10-100Hz tends to be associated almost exclusively with distant shipping noise in most ocean areas (Ross 1976). This dominance has also been confirmed experimentally, where measurements of vertical arrival angles showed that the majority of sound below 200Hz originated from ray path no greater than 20 degrees from horizontal (Axelrod et al. 1965).

Detailed discussions on the subject of ambient noise and its prediction can be found in (Heitmeyer et al. 2003), (Carey & Evans 2011), (Dahl et al. 2007), (Wenz 1962) and (Ross 2005).

3.5 Underwater Noise Propagation

Once the noise signature of a vessel has been established, the next issue is to predict how it will propagate through the water, and the potential received levels of sound (RL) that would be experienced by a marine animal or fish at a given distance from the source. The propagation of noise in water depends on a vast array of variables, and can therefore be extremely complicated to predict without the use of assumptions and known data. These variables will be related to both the radiated noise, especially with regards to its spectrum, key frequencies and intensity, and also to the properties of the water and the geography of the location in which the noise is being propagated.

3.5.1 Environmental Influences

Looking first at the properties of the radiated sounds, it is known that lower frequency sounds can propagate for much greater distances with little attenuation, while higher frequency sounds will be lost much more quickly. Factors such as the source depth and directionality of the sound will also affect the propagation, especially near the water surface, as interference from the Lloyd's Mirror effect, discussed later in Section 3.6 Modelling Techniques, and general loss of sound at the surface due to the pressure release boundary will greatly reduce the intensity and spectra of the noise.

The properties of the water will have a very prominent effect on the way in which the radiated noise is transmitted. Salinity, temperature and pressure are all key variables, with the latter two being largely depth-dependant, but the first two also being dependant on temporal scales, both diel and seasonal. Several empirical formulae by Urick (1983) exist that aim to predict the speed of sound using known values of temperature, salinity and water depth, which are discussed in more detail in the next section.

In these equations, as depth and temperature vary, as will occur as you go deeper into the ocean the speed of sound will be affected. These variations in sound speed combine to give a sound velocity profile (SVP), which will vary with depth, and this will define the propagation behaviour of sound energy. Speed of sound in water is typically around 1500 m/s, but will vary depending on these variables and can be as low as 1440 m/s in some areas. Figure 3.21 below gives an indication of how the speed of sound varies globally.

The physical and geographical properties of the water can also dramatically affect the way in which sound is propagated through the water. Bathymetry is the study of underwater depth of bodies of water such as oceans and lakes. Bathymetry and the type of material on the sea bed will both have significant effects on propagation, as sedimentary material will tend to absorb sound more easily, while deeper water will tend to have very different sound properties to shallow water. The nature of the seabed composition affects the propagation of sound especially in shallow water. This effect is not considered specifically within this work, as modelling is carried out for deep water, however information on the sedimentary composition of sea beds globally can be found using "Marine Geology Data: Seafloor Surficial Sediment Descriptions (Deck41)" (NOAA 1970). GEBCO charts can be used for bathymetry data, although this is not available for all regions

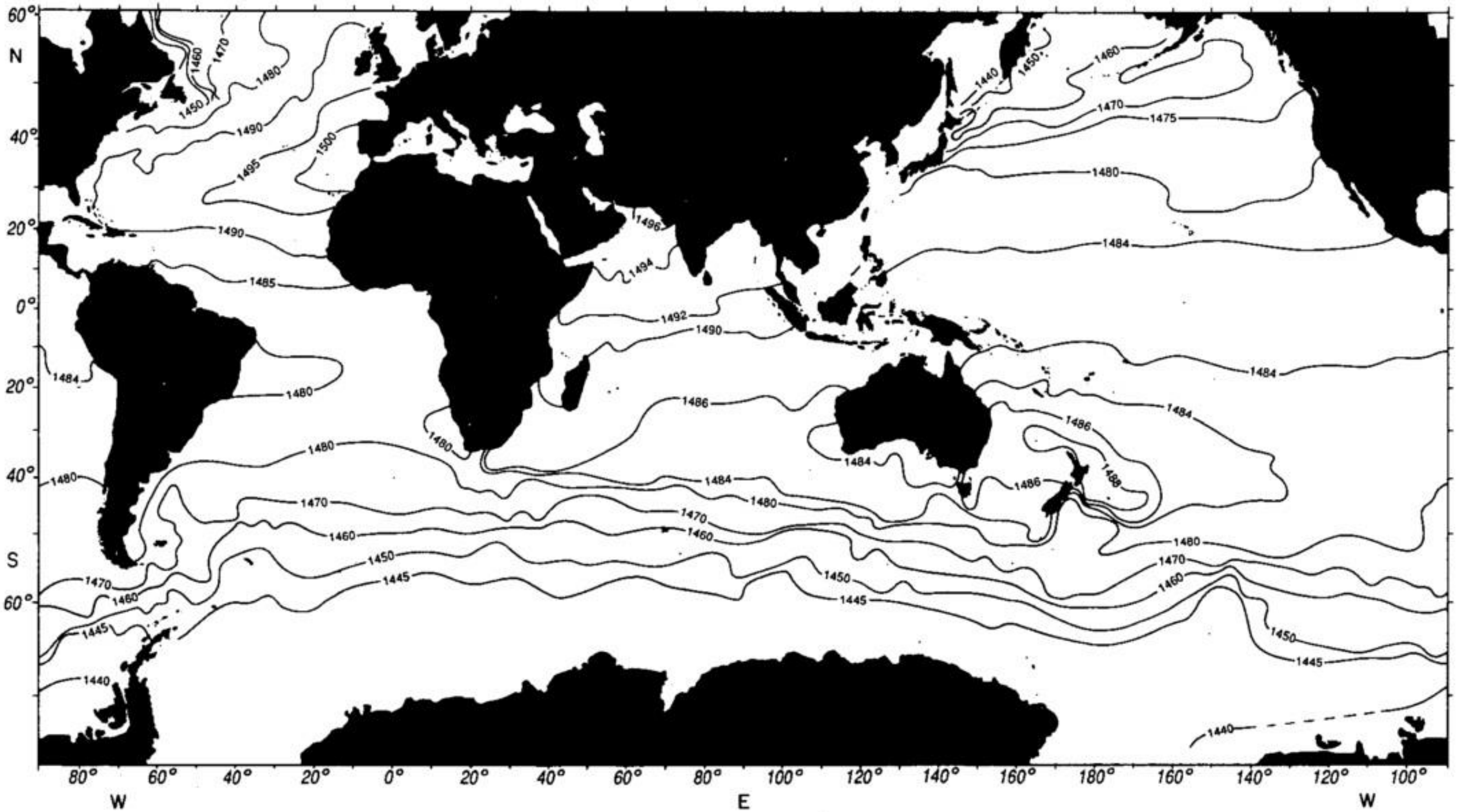


Figure 3.21 - Global Variation of Sound Speed in m/s on the Axial Surface (Munk & Forbes, 1989)

Given the nature of these variables of salinity, temperature and pressure, the Sound Velocity Profile (SVP) will not only vary for different geographical locations, but as stated before, is dependent on time of day and season. Looking first at temperature, decreasing temperature will decrease the speed of sound. Warmer weather at the surface will tend to give a steeper negative gradient in the near the surface, however the temperature in the deeper waters of any ocean will remain almost constant at around 4^{o}C, where the effect of pressure will then cause the speed of sound to increase with depth, with most influence below the value of minimum sound speed. Salinity also experiences most variation nearer the surface, and lower salinity will again cause a lower speed of sound. This is again only dominant up to the minimum speed of sound point, after which salinity will tend to be almost constant, and the SVP will be mainly dictated by the effects of pressure.

Figure 3.22 below gives an example of a typical SVP for deep open ocean in the mid-latitudes:

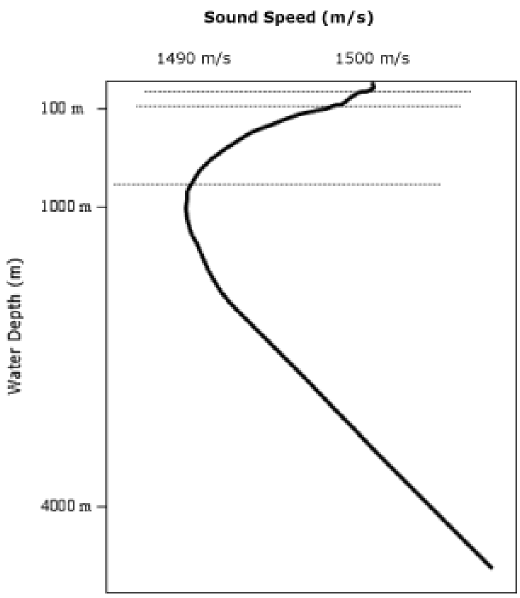


Figure 3.22 - Typical Sound Velocity Profile in Sea Water (University of Rhode Island 2002) (© University of Rhode Island)

The nature of the SVP will dictate the occurrence of different sounds layers in the ocean, and also the presence of sound channels, which often increase the propagation of sound dramatically. The sea is typically divided into 4 distinct layers (Urick 1983):

1. Surface Layer – this is susceptible to daily and local changes
2. Seasonal Thermocline – as the name suggests, this will vary depending on the season
3. Main Thermocline – this is also known as the Sonic Layer, and will typically contain the minimum sound velocity. In some cases, mainly during the winter seasons, the seasonal layer will combine with the main layer as the variation in temperature will be greatly reduced
4. Deep Isothermal Layer – this extends to the sea bed and will remain the same temperature all year round, typically at about $4^{o}C$ / $39^{o}F$ in all ocean areas

Figure 3.23 below illustrates these layers, with relation to the same typical SVP:

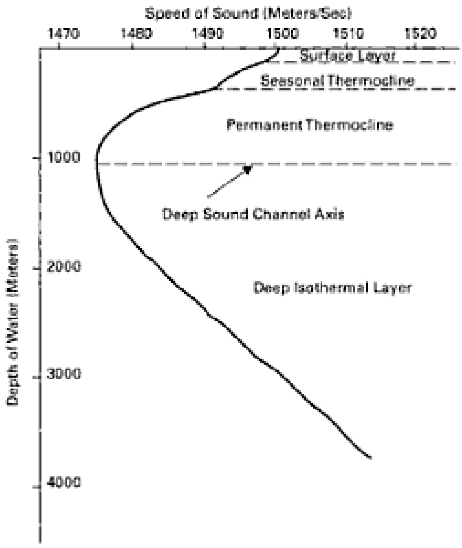


Figure 3.23 - Typical SVP in Sea Water Indicating Layers (Payne 2006)

The relative thicknesses and even the occurrence of these layers is dependent on season, time of day, meteorology and even latitude. The sound channels that occur tend to be related to these layers:

1. The Deep Sound Channel (DSC), sometimes known as the SOFAR (Sound Fixing And Ranging) channel, will occur in the deep isothermal layer, typically at a depth of around 1km, where the sound velocity is at a minimum
2. Shallow water channels will occur where noise is “trapped” within the surface layer
3. Mixed layer channels can occur where the seasonal and main thermoclines combine due to mixing and similar temperatures, and sound can also then become “trapped“ within this mixed layer
4. Half channels will occur in special cases such as Polar regions where there is a more linear sound velocity profile with a minimum at the surface and a maximum at the sea bed

Other major factors are sea state and general weather conditions. A rough surface will increase transmission losses more than in calm water, while the presence of wind and rain will affect the overall ambient noise. Some of these variations have specifically been observed by McKenna et al. (2013).

3.5.2 Speed of Sound Prediction

In order to accurately model the propagation of sound in any medium, the speed of sound in that medium needs to be well understood. In water, an average speed of sound is typically taken as 1500 m/s, compared to an average of 343 m/s in air. In the oceans, the speed of sound will be affected by temperature, salinity, water depth, water chemical composition, variations in biota, and many other variables as discussed above. This results in a sound velocity profile with speed variations according to these variables over depth.

Several empirical formulae exist that aim to predict the speed of sound using known values of temperature, salinity, water depth and pressure; four of these are presented below:

1) Developed by C. C. Leroy, taken from Leroy (1969):

$$c = 1492.9 + 3(T-10) - 6\times10^{-3}(T-10)^2 - 4\times10^{-2}(T - 18)^2 + 1.2(S-35) - 10^{-2}(T-18)(S-35) + \frac{D}{61} \tag{3. 12}$$

Limits: $-2 \leq T \leq 24.5$

$30 \leq S \leq 42$

$0 \leq D \leq 1{,}000$

2) Developed by V. A. Del Grosso, taken from Grosso (1974):

$$C_{STP} = C_{000} + \Delta C_T + \Delta C_S + \Delta C_P + \Delta C_{STP} \tag{3. 13}$$

Where: $C_{000} = 1402.302$

$\Delta C_T = 0.501109398873\times10^{1}T - 0.550946843172\times10^{-1}T^2 + 0.221535969240\times10^{-3}T^3$

$\Delta C_S = 0.132952290781\times10^{1}S + 0.128955756844\times10^{-3}S^2$

$\Delta C_P = 0.156059257041\times10^{0}P + 0.244998688441\times10^{-4}P^2 = 0.883392332513\times10^{-8}P^3$

$\Delta C_{STP} = -0.127562783426\times10^{-1}TS + 0.635191613389\times10^{-2}TP + 0.265484716608\times10^{-7}T^2P^2 - 0.159349479045\times10^{-5}TP^2 + 0.522116437235\times10^{-9}TP^3 - 0.438031096213\times10^{-6}T^3P - 0.161674495909\times10^{-8}S^2P^2 + 0.968403156410\times10^{-4}T^2S + 0.485639620015\times10^{-5}TS^2P - 0.340597039004\times10^{-3}TSP$

3) Developed by H. Medwin, taken from Medwin (1976):

$$c = 1449.2 + 4.6T - 5.5 \times 10^{-2}T^2 + 2.9 \times 10^{-4}T^3 + (1.34 - 10^{-2}T)(S - 35) + 1.6 \times 10^{-2}D \quad (3.14)$$

Limits: $0 \leq T \leq 35$

$0 \leq S \leq 45$

$0 \leq D \leq 1{,}000$

4) Developed by K. V. Mackenzie , taken from Mackenzie (1981):

$$c = 1448.96 + 4.591T - 5.304 \times 10^{-2}T^2 + 2.374 \times 10^{-4}T^3 + 1.340(S - 35) + 1.630 \times 10^{-2}D + 1.675 \times 10^{-7}D^2 - 1.025 \times 10^{-2}T(S - 35) - 7.139 \times 10^{-13}TD^3 \quad (3.15)$$

Limits: $0 \leq T \leq 30$

$30 \leq S \leq 40$

$0 \leq D \leq 8{,}000$

Where: T is the temperature, in ^{o}C

S is the salinity, in parts per thousand

D is the water depth, in m

P is the pressure in kilograms per square centimetres gauge

In modelling generally, an average speed of sound is taken when estimating noise propagation.

3.5.3 Operating in Ice Conditions

Recent global thermal changes have caused significant ice melt in Polar Regions. This has opened up the possibility of shorter summer trading routes through Arctic regions, between Asia and the US. Where previously these areas would be completely covered year-round in thick sheet ice, during the warmer summer season, this ice now melts, making it possible for ice-strengthened vessels to pass between the thinner, less dense coverage of floes and icebergs. A detailed discussion on the subject and the challenges is presented by Brigham (2008).

The possibility of increased ice-water operation brings with it a multitude of design problems which all need to be addressed, especially in terms of the ship radiated noise impacts on these new environments. In 2002, the International Maritime Organisation (IMO) issued "Guidelines for Ships Operating in Arctic Ice-Covered Waters". These requirements were accepted and structured into formal regulations by the International Association of Classification Societies (IACS) in 2006, in the form of the "IACS Polar Class Rules". These cover a wide range of design issues relating to operation in ice waters including structural strengthening, increased machinery capabilities, and improved safety and survivability standards. All these design changes will have a significant impact on the noise and vibration performance of the vessel. The added structure in particular will change the vessels noise signature; however the effects may not give a net gain in terms of improved acoustic performance. Additional strengthening and protection for propellers is also likely to reduce their acoustic performance, in particular in relation to increased cavitation. Ship-ice interaction and ice loading on the hull and propeller will also add to both the vibrations inherent in the vessel, and also to the general noise level associated with the transit of the vessel. This is especially true for "icebreaker" vessels as discussed by Erbe & Farmer (2000), where the impact of ice-breaker noise on Beluga Whales was investigated and modelled.

As well as affecting the direct acoustic properties of the vessel, the presence of ice will also have a considerable effect on the propagation of noise; depending on the extent of the ice coverage, it effects propagation in opposing ways. A large ice sheet covering an entire area will greatly reduce the propagation of noise by absorbing a significant proportion of the sound pressure, while the presence of smaller scattered floes and icebergs in the marginal ice zones (MIZ) will reflect sound and hence increase the amount of sound transmission, although the signal is likely to be more altered due to the amount of interference and reflection occurring (Lynch et al. 2010). There are also a number of other factors which make the prediction of the effect of ice on noise propagation difficult, including the rapidly varying nature of the underside ice surface, and the potential importance of diffraction of sound around ice obstacles.

A number of empirical models exist based on under-ice propagation measurements however these tend to be severely limited seasonally and geographically. Many were also developed a significant amount of time ago, and are unlikely to still be valid given the dramatic changes taking place in these ocean regions due to global warming. Two of the best known empirical models are the Marsh-Mellen transmission loss model developed in 1963, and the Buck Arctic transmission loss model developed in 1981, both of which are based on seasonal measurement data in localised geographical locations. The Marsh-Mellen model (Marsh & Mellen 1963; Mellen & Marsh 1965) is based on measurements and observations carried out and the following transmission loss equation for long range and low frequency sound, below 400Hz, was proposed:

$$TL = 10logr_0 + 10logR + \alpha_s N_s \qquad (3.\ 16)$$

Where: r_0 *is the skip distance for the limiting ray in m*

N_s *is the number of surface reflections*

R *is the range in m, where* $R = r_0 N_s$

α_s *is the loss per bounce*

The Buck (1981) model has two equations, for long and short ranges of propagation, both for frequencies below 100Hz and in water deeper than 1,000m. They depend on variables such as range, frequency and standard deviation of ice depth, which is taken from charts also published by the author. These equations are based on linear regression fits to winter data collected by the author, and hence there is some doubt as to the applicability of the model to summer conditions, and for shallower water depths.

In several instances, ice-scattering coefficients have also been applied to existing numerical models for acoustic propagation. However numerical models specifically designed for underwater noise propagation in ice-covered regions are rare due to the complexity of the propagation in these areas. In spite of this, development of numerical models to accurately account for ice effects on noise propagation should be encouraged, as this problem grows in importance for future shipping.

The presence of ice will also impact the more general ambient properties of the area. The lower temperatures will give a different sound speed profile to those typically seen in more temperate waters. There is also a chance that the lower temperatures will slightly affect the cavitation characteristic of the propeller, hence affecting the noise this will create. The less densely-packed and thinner ice itself also increases the ambient noise levels of the area though cracking, and ice-ice interaction noise, meaning the critical ratio of the ship noise above this level will vary.

The final factor to consider is the relative sensitivity of the Arctic environment and associated wildlife to anthropogenic noise. Whereas species which live in areas close to shipping lanes or in more heavily industrialised areas may be more habituated to these noises, they will be entirely new to those living in previously un-navigable waters. This could increase the potential behavioural and physical impacts of the same levels of ship radiated noise compared to species in other areas.

However, Anders Jensen of the Institute of Marine Research in Norway, states that there could also be some positives to Arctic Shipping: shorter voyages would mean lower air emission levels, and the colder water temperatures could also reduce the problems associated with alien species carried in ballast water and on hulls. However as discussed by Lasserre & Pelletier (2011) there does not appear to be a surge in taking advantage of these possibilities. It should be noted that due to time constraints, propagation of sound under ice will not be included in the model developed here. It should however be considered as an area for future work.

3.6 Modelling Techniques

There is a wide variety of models available in the public domain for modelling different aspects of ship radiated noise and its underwater propagation. A number of these have been taken or adapted from the aeronautical industry, as the principles are very similar. The aerospace industry has recognised the significance of radiated noise and has for many years attempted to address this problem, particularly in relation to noise at airports and over residential areas. The relatively recent advent of powerful computational ability and advances in CFD codes has lead to a move from the more traditional and simpler empirical models to more accurate models covering increasingly extensive details of the issue. Both aspects will be covered in this section, as both simple and more complex modelling techniques have their applications.

3.6.1 Ship Underwater Noise Spectra Estimation

Some empirical methods exist for the direct prediction of a vessels underwater source level noise, which are typically developed using data for a small sample of vessels in service at the time of the research. These will then become increasingly inaccurate as the data becomes more outdated.

Some will only provide an average sound pressure level value rather than a spectral result, which would not be detailed enough for use within this work. For example, in his book on the subject of Underwater Noise Ross (1976) details two source level estimation formulae, for overall noise level in dB above 100Hz. These were popular for simple estimated during the WWII era and in the following year, and are based on trends observed in measurements carried out by Naval bodies around that time. This declassified data was gathered from American, British and Canadian naval underwater noise ranges just after World War 2, and covers naval vessels as well as civilian freighters, tankers, passenger vessels and cruisers. Both are based on relationships between ship size and noise, and ship speed and noise, and are presented below:

$$SL_1 = 112 + 50log\left(\frac{V}{10\ knots}\right) + 15log\Delta \tag{3. 17}$$

$$SL_2 = 134 + 60\ log\left(\frac{V}{10\ knots}\right) + 9log\Delta \tag{3. 18}$$

Where: SL is the source level above 100Hz in dB re1μPa at a reference distance of 1 yard

V is the ship speed in knots

$10\ knots$ is a reference speed

Δ is the displacement tonnage

It should be noted that these formulae are only said to be valid for vessels with a displacement tonnage of less than 30,000 tonnes. For larger ships, in this case defined as over 100m in length, a clear trend was observed between ship noise, and propeller tip speed and number of blades, which gave rise to the equation below:

$$SL_3 = 175 + 60log\left(\frac{V_t}{25\ m/s}\right) + 10log\left(\frac{B}{4}\right) \tag{3. 19}$$

Where: 25 m/s is the reference tip speed

B is the number of blades

V_t is the propeller tip velocity,

$$V_t = \pi nD$$

Where: n is the rotational speed in rps

D is the propeller diameter in m

These three empirical estimations, despite their disadvantages, can provide a fast estimation of sound pressure levels for a vessel using very basic data, and hence will be used within the data analysis model outlined in Chapter 4. It should be noted that these formula suggest general trends however they are not applicable to every case, for example vessels with a Controllable Pitch Propeller (CPP) will not have a direct correlation between vessel speed and SPL.

Others use base-line spectra which are averaged from selected ship spectra, which are then subtly varied based on some simple formulae. These again are restricted in the fact that they are based on a small sample of increasingly historical data. They are also unsuitable due to the difficulties in trying to extract the specific machinery noise contributions from the empirical spectra and incorporating this into the existing predicted spectra. By way of an example, the Ross model uses the idea that a vessels spectra is proportional to a base-line spectra with a constant of proportionality defined a by a power-law relationship on the ship speed and length; the equation below gives the model spectrum equation in its most widely used form, derived from the equations presented by Ross (1976):

$$S(f,V,L) = S_0(f) + c_v 10log\left(\frac{V}{V_0}\right) + c_L 10log\left(\frac{L}{L_0}\right) \qquad (3.\ 20)$$

Where: S_0 *is the base-line spectrum*

V_0 *is the reference speed*

L_0 *is the reference length*

c_v *is a constant, usually taken as 6*

c_L *is a constant, usually taken as 2*

Alternative spectral-based methods are discussed in the literature (Hazelwood & Connelly 2005), (Wales & Heitmeyer 2002) and (Oimatsu et al. 1996).

3.6.2 Noise Propagation Modelling

The following section will detail the many different areas of propagation modelling, through simple geometrical methods, through theoretical methods, to more complex numerical methods.

It should be noted that although they are not discussed here, a number of empirical methods (Marsh & Schulkin 1962) also exist for the approximation of sound propagation underwater. These however tend to be based on limited numbers of measurements from only a few specific geographical locations, and it is therefore difficult to justify their application to a wider range of scenarios. These models will not be used within this research and will therefore be neglected henceforth.

Significant details on sound propagation modelling can also be found in (Jensen et al. 2011) and (Etter 2003).

Geometric Sound Propagation Modelling

In this very simplified approach, it is simply assumed that the emitted sound signal weakens uniformly as it propagates away from the source. The level of noise which will be observed at the receiver is hence the original level, minus the uniform weakening, or transmission loss, as shown below in equation (3.21):

$$Received\ Level\ (RL) = Source\ Level\ (SL) - Transmission\ Loss\ (TL) \tag{3. 21}$$

Where: RL is the received level at range R (in m), in dB re 1μPa

SL is the value at 1m from the acoustic source centre, in dB re 1μPa^{2}m^2

TL is in dB re $1m^2$

The reference units above are currently being developed as part of a new underwater noise standard in ISO TC43. There are two main methods for estimating the transmission loss, suitable for slightly different scenarios, as illustrated below.

The spreading loss has been proven to vary with range according to a logarithmic scale, and is usually given as a number of decibels per distance doubled (Urick 1983). Modelling propagation solely based on spreading effects, without taking account of the influence of the environment, can lead to considerable inaccuracies, either over- or under-estimating the propagation loss, hence for more accurate results, there is a need for the attenuation and absorption effects to be considered as well. It is also inaccurate to assume that transmission losses are uniform across the entire frequency range; in fact, variations of up to 7dB can occur. However, as a preliminary estimation, this concept is acceptable. Some discussion is presented by Weston (1971).

Spherical Spreading

Spherical spreading represents uniform and omni-directional propagation away from the nominal source location. This is suitable for deep water or offshore applications, where it is appropriate to neglect any interference in propagation from either the water surface or the sea bed. For spherical spreading, the transmission loss is given as shown in (3.22):

$$TL_{sph} = 20\ log(\frac{R}{R_0}) \qquad (3.\ 22)$$

Where: R is the range of the receiver from the sound source, in m

R_0 is the reference location; typically 1 metre from the nominal source in hydroacoustic applications

Cylindrical Spreading

Cylindrical spreading assumes that the propagation of sound is more directional, with sound mostly spreading horizontally and very little being propagated vertically. This is more suitable for shallow water or channel applications, as it goes some way towards accounting for water surface and sea bed interactions.

For cylindrical spreading, the transmission loss is given as shown in (3.23):

$$TL_{cyl} = 15\ log(\frac{R}{R_0}) \qquad (3.\ 23)$$

Theoretical Sound Propagation Modelling

Many of the theoretical models outlined in this section were developed before the advent of accessible computational power, and hence cannot take into account all factors affecting the propagation of noise. They are relatively accurate within their respective ranges of applicability, and are therefore still widely used in many acoustics applications. They are particularly well suited to situations where details of the operational area are not available for more in-depth modelling. The figure below, adapted from Jensen & Krol (1975) by Etter (2003), summarises the different theoretical approaches to propagation modelling:

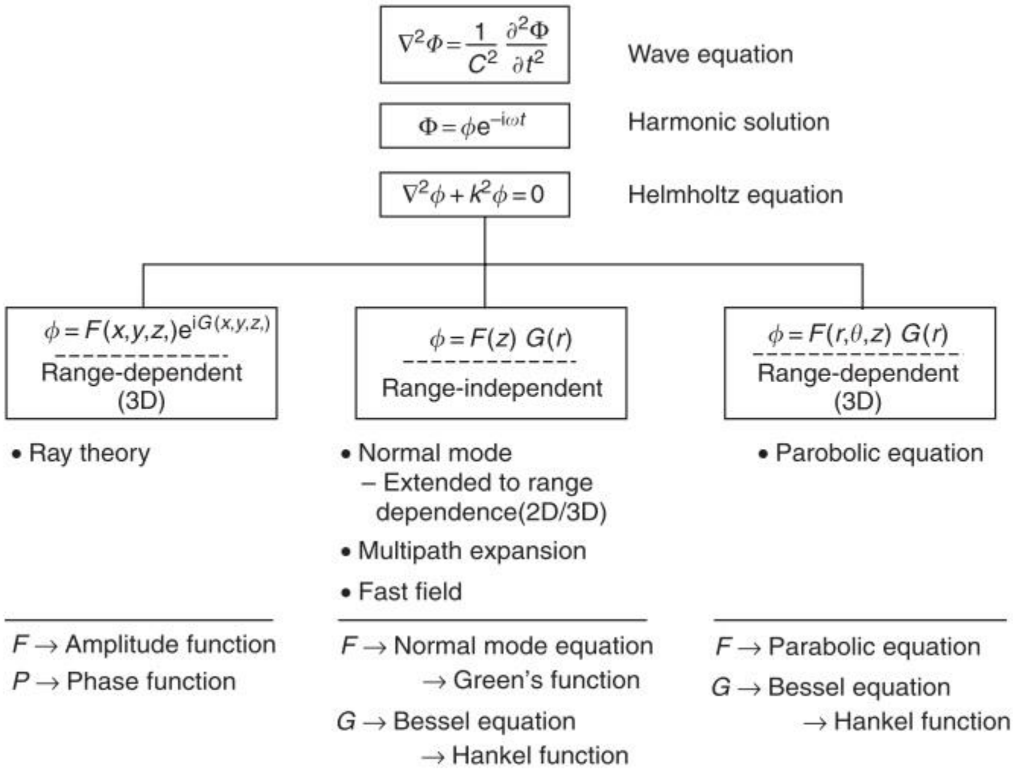


Figure 3.24 - Summary of the Relationships Between Theoretical Propagation Models (Etter 2003)

Multipath Expansion Models and Fast-Field models will not be discussed within this review, however the interested reader is referred to Etter (2003) for details.

Derivation of the General Wave Equation

There are a number of approaches that may be used for the derivation of the basic differential equation which describes the propagation of sound in a fluid. The derivation below gives an overview of the assumptions made and the resulting equation; a full derivation can be found in Ross (1976).

The wave equation is based on fundamental physical principles; namely satisfying the continuity equation for conservation of mass, applying the conservation of momentum given in Newton's Second Law of Motion, and taking the equation of state for fluids, or the stress-strain relationship.

The continuity equation in its acoustic form, and in its general form are given below in equations (3.24) and (3.25), the second varying from the first only in the addition of the source term, q.

Continuity equation - linear acoustic continuity in a region free of acoustic sources

$$\frac{\partial \rho'}{\delta t} + \rho_0(\nabla . \bar{v}') = \frac{\partial \rho'}{\partial t} + \rho_0 \frac{\partial v'_i}{\partial x_i} = 0 \tag{3. 24}$$

Where: ρ' *is acoustic density fluctuation*

ρ_0 *is fluid static density*

∇ *is the gradient operator*

$\bar{v}'$*is acoustic particle velocity*

$v_i{}'$ *is a component of acoustic particle velocity*

Continuity equation - linear continuity in a region containing acoustic sources

$$\frac{D\rho}{Dt} + \rho(\nabla . \bar{v}) = \frac{\partial \rho}{\partial t} + \frac{\partial(\rho v_i)}{\partial x_i} = q \tag{3. 25}$$

Where: ρ *is density fluctuation*

∇ *is the gradient operator*

$\bar{v}$ *is particle velocity*

v_i *is a component of particle velocity*

q is the rate at which new mass is created per unit volume

The momentum equation is again given in two forms below in equations (3.26) and (3.27), but this time they are more varying. It can be seen that equation (3.27) includes the viscous stresses as the stress tensor $\mathrm{p_{ij}}$ rather than pure pressure, p, even though Ross (1976) states these to be negligible in all practical calculations of fluid-dynamic noise. Equation (3.27) was used by M. J. Lighthill in his aero-acoustic analogy, published in 1952, which will be discussed later.

Momentum Equation - acoustic conservation of momentum for an ideal fluid with no external sources

$$\rho \frac{\partial \bar{v}}{\partial t} = -(\rho g \nabla z + \nabla p + \rho(\bar{v} \,.\, \nabla)\bar{v}) \tag{3. 26}$$

Where: ρ *is fluid density*

$\bar{v}$ *is particle velocity in m/s*

g is gravitational acceleration in m/s²

∇ *is the gradient operator*

p is pressure

Momentum Equation - rate-of-change of momentum in a region containing sources, in tensor notation, including viscous stresses

$$\frac{\partial(\rho v_i)}{\partial t} = -\frac{\partial p_{ij}}{\partial x_j} - \rho g_i - \frac{\partial(\rho v_i v_j)}{\partial x_j} + f_i \tag{3. 27}$$

Where: ρ *is fluid density*

v_i *is a component of particle velocity*

v_j *is a component of particle velocity*

p_{ij} *is a pressure stress tensor, including normal and viscous shear stresses*

g_i *is a component of gravitational acceleration, and* $g_i = -g\ gradz$

∇ *is the gradient operator*

f_i *is the net force per unit volume exerted by any external mechanical forces that may be acting on the fluid*

The state equation is the relationship between static pressure, density and temperature. Equation (3.28) below is for a fixed temperature, where the higher order terms of the power expansion of density to express pressure have been neglected.

State Equation - stress-strain relationship for an acoustic disturbance

$$p' = p - p_0 \doteq a(\rho - \rho_0) = a\rho' \qquad (3.\,28)$$

Where: p' *is acoustic pressure*

p *is pressure*

p_0 *is ambient static pressure*

ρ *is fluid density*

ρ_0 *is the mean density of the fluid*

ρ' *is acoustic density fluctuation*

a *is a coefficient, assumed to be a constant or a slowly varying function of position*

These are typically combined into the wave equation seen below, where acoustic pressure has been eliminated. Similar results can be obtained by eliminating acoustic density as seen in equation (3.29):

$$\nabla^2\rho' - \frac{1}{a}\frac{\partial^2\rho'}{\partial t^2} = 0 \qquad (3.\,29)$$

Ray Theory

Ray theory was developed to simplify propagation calculations. It calculates transmission loss based on ray-tracing methods which were developed for general physics application by the National Defence Research Committee in 1945; more details on the development of ray-tracing methods can be found in Spitzer (1945).

Using the Helmholtz Equation, two equations are obtained, which account for the ray geometry and the wave amplitudes. The geometrical acoustics assumption is applied; this assumes that the speed of sound will not vary much over one wavelength. Differential ray equations can then be derived by applying the above assumption to the ray geometry equation. It is typical to consider four types of ray paths: direct path (DP), refracted-surface-reflected (RSR), refracted-bottom-reflected (RBR) and refracted-surface-reflected-bottom-reflected (RSRBR).

Due to the assumption associated with the ray theory model, it is only suitable for short range and high frequency problems. This limitation renders the model unsuitable for the typically lower frequency and higher range problems associated with radiated shipping noise. It has however been applied in a number of impact assessment models for the close-range effects of smaller vessels such as whale-watching boats on marine animals in the immediate vicinity; as discussed by Erbe & Farmer (2000). Further discussion of the ray theory model and its applications can be found in Urick (1983) and Etter (2003).

Normal Mode Equations

Normal mode methods are associated with the solution of an integral form of the wave equation; practical solutions are typically achieved by assuming that the water characteristics and environment change with relation to water depth only, making this a range-independent method. They can be extended for range dependency, but with significant computational costs. Two equations result; one accounting for the standing wave in the depth direction, and one accounting for the travelling wave in the horizontal direction.

This method is much better suited to low frequency and longer range problems. A significant advantage of this method over ray tracing methods is that it allows for transmission loss at all receiver depths and ranges to be easily calculated with any combination of frequencies and source depths, without having to calculate each combination sequentially. The main disadvantage however, and the reason it is used less often in early stage indicative calculation of noise propagation, is the level of detail required on the bottom geometry, and the acoustic propagation characteristics of not only the water column, but also of the bottom sediment or rock layers. Without carrying out detailed measurements, it is unlikely that such information will be easily available, especially at early stages of a ships design. Further discussion on these methods can be found in Etter (2003) and an example of an application can be found in Wood & Humphrey (2012).

Parabolic Equations

In this method, the standard wave equation is re-written in the form of a parabolic equation (PE). This is achieved by assuming that the sound propagates at approximately either compression speed or shear speed, whichever is appropriate. The equation is then solved numerically; when the initial field is known, this can be done by a marching solution. A number of methods can be used and applied computationally to find an initial fields and solutions, including using normal mode methods. Detailed discussion of these approaches can be found in Etter (2003).

There are a number of examples where PE methods have been applied to underwater noise impact analysis, as a fast and efficient way of predicting the propagation of the noise in a given environment (Kongsberg Maritime Ltd. 2010).

Snell's Law and Lloyd's Mirror Effects

Snell's Law, also known as the Law of Refraction, is a formula which describes the behaviour of waves as they cross the boundary between two different isotropic media. The relationship covers the difference in wave directions before and after it crosses the boundary between the two media, and is typically given by the formula shown in equation (3.30) below:

$$cos\,\theta_t = \frac{c_2}{c_1} cos\,\theta_i \qquad (3.30)$$

Where: θ_t *is the angle of transmission*

c_1*is the speed of sound in the initial medium*

c_2 *is the speed of sound in the new medium*

θ_i *is the angle of incidence*

These differences in wave direction are related to the different sound velocities of the different media. In acoustics, it is more typical to see gradual rather than step changes, as sound velocities vary in property gradients such as temperature and pressure, as discussed previously. Clearly, such effects have to be considered for sound wave propagation, as interaction with both the sea bed and water surface will affect the transmission of the sound, due to significant changes in the sound speed. Both the irregular water surface and the seabed will tend to absorb higher frequency sounds, however depending on the nature of the seabed, lower frequency sounds will often be reflected. The nature of the sound arriving at the receiver will also vary from that initially leaving the source, depending on how different frequencies and wave paths are reflected, absorbed and transmitted by various interactions with the surface and seabed.

First described from an optics experiment in 1834 by Humphry Lloyd, the Lloyd's Mirror experiment comprised of a monochromatic slit light source being shone onto a glass surface from a small angle. The resulting pattern arising from interference between the direct light and reflected light is known as the Lloyd's Mirror Effect. In underwater acoustics, these effects are important when a sound source is located near the water surface; constructive and destructive interference occurs between direct path and reflected path sound waves. This particularly affects the lower frequency sounds, which can be eliminated almost entirely through these effects.

Numerical Sound Propagation Modelling and Source Prediction

The underlying theories to many of the current numerical models for underwater noise propagation were developed some time ago, many in connection with aeronautical applications. Nevertheless, it took the rise of accessible, fast and reliable computational ability to bring them to foreground of research and development in the field of hydroacoustics. These models tend to be much more complex, and require a greater level of detail regarding the vessel and its acoustic characteristics, however where these are available, the results that can be gained from these models can be accurate and realistic for a wide range of applications.

The following sections will focus on the theories which have led to the development of the numerical model based on the Ffowcs-Williams Hawkings Equation, which is the main model currently being researched and further developed for widespread use in hydroacoustics applications.

Lighthill's Aero-Acoustic Analogy

Using the same principles as discussed in the derivation of the general wave equation previously, Sir Michael James Lighthill, a pioneering applied mathematician, re-arranged the Navier-Stokes equations which describe the motion of compressible viscous fluids, into an inhomogeneous wave equation (Lighthill 1952). The principle considers a jet of air streaming into a quiescent medium, i.e. one that is assumed to be stable, and unlikely to change for a significant period of time. The general application of interest to him was to predict the noise generated by the jet of an aircraft turbojet engine. The laws of conservation of mass and momentum will therefore hold everywhere. Lighthill's equation is seen below in (3.31):

$$\Box^2 p' = \frac{\partial^2 T_{ij}}{\partial x_i \partial x_j} \qquad (3.\ 31)$$

Where: $\Box^2$ is the D'Alembert, or wave operator, whereby

$$\Box^2 = \left(\frac{1}{c^2}\right)\left(\frac{\partial^2}{\partial t^2}\right) - \nabla^2$$

∇ is the gradient operator

p' is acoustic pressure

T_{ij} is the Lighthill stress tensor as given below:

$$T_{ij} = \rho u_i u_j + P_{ij} - c^2 \rho' \delta_{ij}$$

Where: c is the sound speed in a quiescent medium

P_{ij} is the compressive stress tensor

δ_{ij} is the Kronecker delta, whereby $\delta_{ij} = 1$ when $i = j$, and otherwise $ä_{ij} = 0$

Unlike the many approximations and assumptions used in the derivation of the standard wave equation for sound in fluids, Lighthill's equation contains no assumptions, and hence the results obtained are exact. In this application, the behaviour of the fluid outside the jet is described by the manipulated wave equation on the left-hand side (LHS), whilst the behaviour of flow inside the jet is described by the inhomogeneous quadrupole source term on the right-hand side (RHS). This source term is equal to zero everywhere outside the region of agitated flow caused by the jet; within the turbulent flow the effects of this flow are replaced by the quadrupole sources.

There are often some clarifications required in regards to this Acoustic Analogy, as outlined below, and discussed by Farassat & Brentner (1998), whose own work in this field will be discussed later. There are important issues to note, and highlight the need for the acoustician to fully understand the problem they are undertaking to solve:

- The term "source" in relation to the jet noise is highly dependent on the way in which the conservation laws are manipulated, and the terms which are left on the right-hand side of the equation

- The acoustic analogy treats all quadrupole sources as spherically radiating, and refraction effects are dealt with through phasing of the quadrupoles within the jet

- The acoustic analogy provides exact values due to it containing no assumptions or approximations, however if used in conjunction with poor or incomplete flow field data, there is no guarantee that the results will be accurate. This then becomes an issue of the suitability of application of the acoustic analogy to different problems

Ffowcs-Williams Hawkings Equation

The fundamental Ffowcs-Williams Hawkings (F-WH) equation presented below was proposed by Ffowcs-Williams & Hawkings (1969). The Ffowcs-Williams Hawkings equation uses generalized functions to extend the application of Lighthill's Acoustic Analogy to the aerodynamic noise generated by rotating bodies such as helicopter rotors and fan blades. More recently this equation has also been applied to operation in other fluids, namely water, for the noise generated by propellers. In situations where detailed data on the turbulent phenomena in the near-field can be obtained, the Ffowcs-Williams Hawkings equation can also be used for broadband noise prediction. The equation in terms of generalized functions can be seen below in (3.32):

$$\bar{\Box}^2 p' = \frac{\bar{\partial}}{\partial t}[\rho_0 v_n \delta(f)] - \frac{\bar{\partial}}{\partial x_i}[l_i \delta(f)] + \frac{\bar{\partial}^2}{\partial x_i \partial x_j}\left[T_{ij} H(f)\right] \tag{3. 32}$$

Where: $\overline{\Box}^2$ *is the D'Alembert, or wave operator as defined above*

p' *is the acoustic pressure in the undisturbed medium, in this case* $p' = p - p_0 = c^2\rho'$

ρ_0 *is the density of the quiescent medium, or fluid static density*

v_n *is the local normal velocity of the source surface*

$\delta(f)$ *is the Dirac delta function*

T_{ij} *is the Lighthill stress tensor as defined above*

$H(f)$ *is the Heaviside Function*

The three terms on the RHS of the equation are then the thickness, loading and quadrupole source terms respectively. Most of the Analogy is now linear, with all non-linearity being collected in the Lighthill Stress Tensor. As the Ffowcs-Williams Hawkings equation is valid in the whole 3D space, it is common to use the Green's Function of the wave equation, defined below, to turn the equation into an integral form, allowing it to be solved numerically.

Green's Function is defined as:

$$G(\boldsymbol{x}, \boldsymbol{y}, t, \tau) = \frac{\delta(g)}{4\pi r} \, for -\infty < \tau \leq t$$

$$G(x, y, t, \tau) = 0 \, for \, \tau > t$$

Where: $g = \tau - t + \frac{r}{c_0}$

$r = |\boldsymbol{x}(t) - \boldsymbol{y}(\tau)|$

*(**x**, t) are the observer space-time variables*

*(**y**, τ) are the source space-time variables*

In most aerodynamic applications, the quadrupole source term is assumed to be negligible at low rotational mach numbers, and is usually only considered when the rotational speed approaches Mach 1. This was also typically accepted as the case for hydrodynamic applications, however recent research work at the Italian Ship Model Basin INSEAN has proved that these quadrupole sources cannot be neglected, and that they in fact have a significant contribution in hydrodynamic noise formation, especially in the presence of a rotating propeller (Ianniello et al. 2010a).

The assumptions used in this approach are that the fluid has constant density, temperature and speed of sound however this is not a reflection of reality. This assumption is suitable for calculations up to several hundred metres into the far field, as is used within this work, however after this point some account should be taken of the variation in these parameters. It has been suggested that the Ffowcs-Williams Hawkings equation could be combined with a ray tracing approach in the very far field to take into account these variations.

Farassat Formulation

In his 1975 Technical Report for NASA Farassat (1975) proposed a method of evaluating the Ffowcs-Williams Hawkings equation through a time domain formulation that can predict the noise of an arbitrary shaped object in motion, without the need for numerical differentiation of the observer time (Seol et al. 2005). These are general solutions for the equations which allow for the use of realistic geometry and kinematics of the rotors and propellers under investigation; the formulations were originally developed to assist in the prediction of aeroacoustic noise radiated by helicopter rotors and propellers. The two best known formulations, which were developed by F. Farassat and colleagues at NASA Langley Research Center, are Formulation 1 and 1A. These formulations, derived fully by Farassat (2007), account only for the thickness and loading terms; however later research by F. Farassat and K. Brentner also gave rise to Formulations Q1 and Q1A, which provide predictions for the quadrupole noise source contributions as well. These are fully derived by Farassat & Brentner (1988) and Brentner (1997) respectively. These equations can be seen in Appendix A. For full derivations and details of all the equations, the reader should refer to the quoted references.

These formulations have been developed as general solutions to allow the calculation of radiated noise for realistic blade geometries and kinematics of the system, and are valid for both the near- and the far-field. To allow for these properties, as discussed above, solution is carried out in the time-domain and then results are transferred to the frequency-domain using a Fast Fourier Transform (FFT). These formulations are now becoming again gaining popularity amongst researchers in many acoustics applications.

The Porous Formulation

Prior to computational advances which made it possible to calculate the contribution of the quadrupole sources or volume integrals, Ffowcs-Williams suggested the use of a porous or permeable surface to account for these contributions. In this approach, a porous source surface is placed within the flow, and the surface integrals are calculated on this surface. This means that the contributions of all the flow properties and non-linearities contained within the porous surface are accounted for, without the need for volume integration. The F-WH equation for the properties of the porous data surface can be seen below:

$$\begin{aligned}\Box^2 c^2 \rho' \equiv \Box^2 p' \\ &= \frac{\partial}{\partial t}[\rho_0 U_n]\delta(f) - \frac{\partial}{\partial x_i}[L_i \delta(f)] \\ &+ \frac{\partial^2}{\partial x_i \partial x_j}\left[T_{ij} H(f)\right]\end{aligned} \tag{3. 33}$$

Where:

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

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

and the other variables are as defined in above and in Appendix A.

3.6.3 Non-Cavitating Propeller Noise Modelling

There are a number of numerical methods which are typically used for the prediction of propeller noise, detailed here for use in non-cavitating conditions, however many could also be applicable in cavitating conditions.

Numerical Approaches

The Bernoulli Theorem can be used to calculate the hydrodynamic pressure field using a boundary surface equation. The velocity potential, ϕ, is found on the boundary surface, and using this, the velocity and pressure can then be found, using the Bernoulli equation. The same equation is used as an integral representation of the velocity potential, to find the acoustic pressure at any point in the far field. The Bernoulli Theorem for this application is as given below:

$$\frac{\partial\varphi}{\partial t} + \frac{1}{2}|\boldsymbol{u}|^2 + \frac{p}{\rho_0} = \frac{p_0}{\rho_0} \qquad (3.34)$$

Where: φ is the velocity potential

$\boldsymbol{u}$ is the fluid velocity in the x-direction

This approach is often used in naval applications however it can be affected, as other potential based approaches, by the approximations made. Nevertheless it can be used as a suitable input for a F-WH Solver.

Boundary Element Methods (BEM) are more typically used to analyse the scattering and sound deflection effects caused by propeller ducts and other similar installations. An example of this application can be seen in Seol et al. (2002) where the non-cavitation noise of a ducted propeller is predicted in various different operating conditions and configurations. BEM's are not considered appropriate for the general prediction of noise from a ship radiated into free space, as it cannot be realistically enclosed. Incidentally, it was also shown in this paper that the effects of using propeller ducts have negligible impact on the far-field acoustic properties of the propeller under non-cavitating operation. There is also a significant body of work using other potential theory approaches, which are generally thought to be suitable for more linear problems.

Nevertheless, the most popular and widely-developed approach currently in use by research facilities is the one based on the Ffowcs-Williams Hawkings equation discussed above. It is demonstrated in Testa et al. (2008) and other works that the F-WH based method can provide more accurate noise results if applied in a suitable manner and using good input data, and has been shown to be more robust and to have computational advantages over the Bernoulli-Based approach. In Seol et al. (2005), a hybrid method using a potential-based flow solver and the FWH equation for propagation is presented, as well as comparison against results obtained using other panel-based methods published in the literature. This particular example of a numerical approach to ship radiated underwater noise prediction appears to be the most promising, however a gap still exists in the development of a specific methodology for this, which has been validated using field measurement data. This gap will be addressed in this study. It is this approach which will form the basis of this investigation, and will be detailed more in the following chapter.

It has been suggested that the accuracy of results in such work might benefit from coupling with a more complex hydrodynamic solver such as the Large-Eddy Simulation (LES) or Direct Eddy Simulation (DES) solvers as opposed to a RANS-based solver. Initial investigation into coupling LES with the FWH equation is presented in Gong et al. (2012), with promising results. However for the purposes of this work, where accuracy must be balanced against run time and complexity, a URANS hydrodynamic solver is tested to establish whether it is sufficient.

Experimental Approaches

As an alternative to numerical modelling, it has also been shown that for some cases, a propeller model run in a cavitation tunnel, using either a hullform or a wake grid, can be used to sufficiently accurately measure the noise properties, so that they can be scaled to full size within a suitable margin of error. A study at Newcastle University's Emerson Cavitation Tunnel compared the noise data measured from a fixed pitch propeller model for a 100m fisheries research vessel to the full size controllable pitch propeller values, and correlation appeared to be good except in the very high frequency range (Atlar et al. 2001). Therefore, in order to allow investigation of a propeller's noise characteristics prior to building, cavitation tunnel and depressurized towing tank measurements provide a feasible alternative to theoretical modelling. This approach can be used for both a non-cavitating and cavitating propeller situation. Due to the sensitivity of the data being recorded, the test facilities carrying out such measurements are required to have particularly low background noise levels, low noise test rigs and to carry out suitable acoustic calibrations. This is discussed in detail, with recommendations presented in Bosschers et al. (2013).

It is difficult to state categorically whether experimental or numerical modelling is "better". Numerical approaches have the advantages of being more cost effective as they do not require models to be built, and provide large amounts of accurate data on the problem as a whole in terms of the flow field etc., however it typically requires large computation resources and can pose reliability issues. Physical testing meanwhile is much more reliable however it is limited by non-physical aspects and can have problems with background noise issues for achieving good results. It is also much more difficult to test a variety of configurations quickly and easily.

Empirical Approaches

Similarly to general modelling trends, several empirical models were developed for the prediction of propeller noise prior to the advent of accessible and high performance computational power. These models were based on a limited amount of measurement data, usually for a specific type of vessel, with varying levels of complexity. However, non-cavitating propeller noise is not typically addressed in isolation by these methods; rather it is predicted together with general vessel noise, or with cavitation noise.

3.6.4 Cavitating Propeller Noise Modelling

As mentioned previously, many of the approaches used for the prediction on non-cavitating propeller noise can also be applied to cavitating propellers, providing sufficiently accurate modelling is carried out to properly capture the cavitation phenomena.

Numerical Approaches

When addressing the occurrence of cavitation, the pressure fluctuations associated with this phenomenon, for the different varieties of cavitation that typically occur, have to be calculated during the hydrodynamic assessment of the near-field pressure distribution. In most examples, this is done using an unsteady BEM approach, where the occurrence of cavitation from a propeller operating in an unsteady inviscid flow is included by means of an additional model. This approach is detailed fully by Salvatore & Ianniello (2003). The hydroacoustic analysis is then carried out using the F-WH approach as before. No study has yet been found in the literature which uses a fully CFD-based approach to predict both the cavitation and acoustic performance of the vessel. Detailed CFD approaches are often used in the prediction of cavitation performance alone, although these models tend to be extremely complex. If a good approximation of the cavitation activity at a given speed can be achieved using a less complex model, then there could be a potential to combine this with a noise prediction approach as discussed above for cavitation-condition noise prediction. This gap in knowledge will also be investigated in this project.

Experimental Approaches

A large body of work has been carried out by the International Towing Tank Conference (ITTC) and others looking at improving the reliability of experimental prediction of propeller cavitation and hence noise, through addressing scaling issues and other procedural details. These are discussed in ITTC (1987) and Vassenden & Lovik (1981).

Empirical Approaches

In empirical approaches, a broadband source level value is typically estimated using a given formula.

The Gray & Greeley (1980) dipole source level formula is based on data from a sample population of single-screw merchant vessels which were around at the time the paper was written. It uses both monopole and dipole sources to describe the propeller radiation properties during cavitation. As stated in the paper,

"A model is developed for the acoustic source strength of blade rate line energy produced by single-screw merchant vessels. These source strengths are based on observed cavitation time histories on merchant vessels and on limitations imposed by considerations of propeller design procedures and ship vibration criteria."

The propeller-based formula can be seen below:

$$SL_D = 92 + 94logD + 10log(\sigma_{SL//D}) \qquad (3.\ 35)$$

Where: D is the propeller diameter in m

$10log(\sigma_{SL//D})$ *is the standard deviation of the uncertainty in source level characterised by its variance, in this case taken as 8dB*

This formula allows for a ball-park figure of the sound pressure level in cavitation condition to be approximated quickly. However, the disadvantages of this approach are that it gives only a single figure for the whole frequency range, and it is also based on ship data available pre-1980, which is very unlikely to be representative of current fleet characteristics.

There are also a few cavitation noise spectral models such as those proposed by Lovik and Brown which give a general spectral form based on frequency and slope values. The figure below shows an approximation of a cavitating propeller noise spectrum, proposed by Brown:

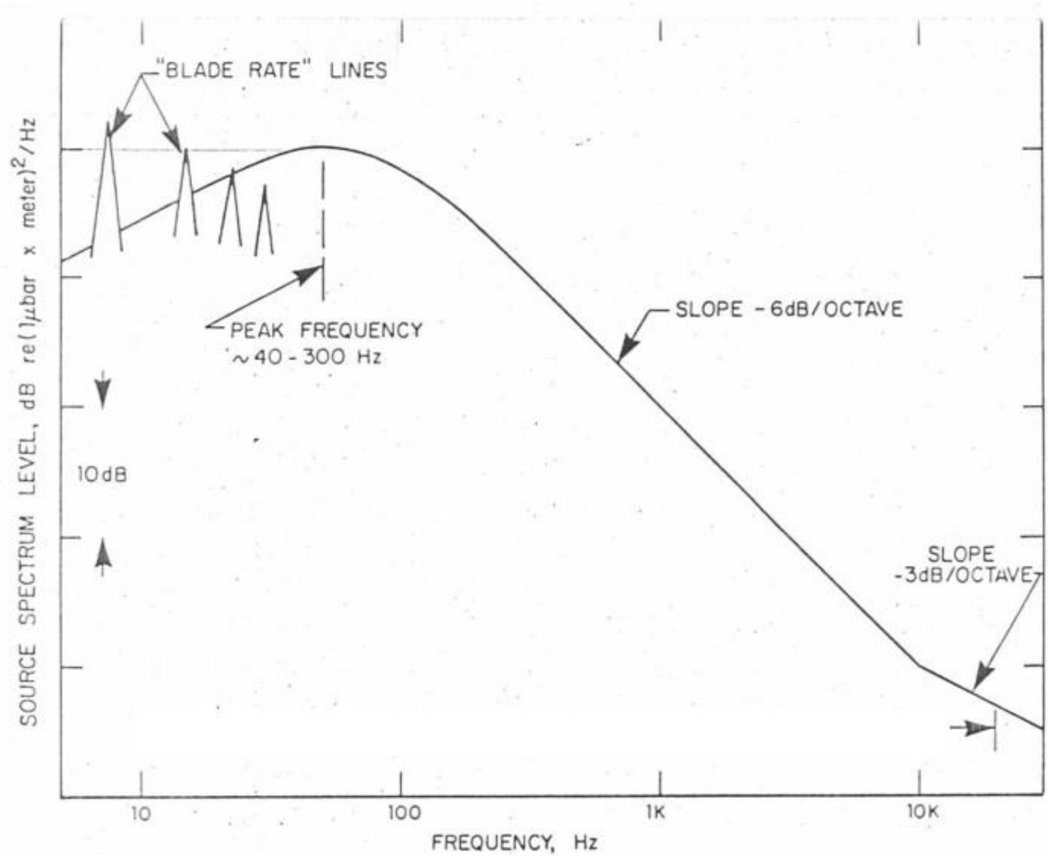
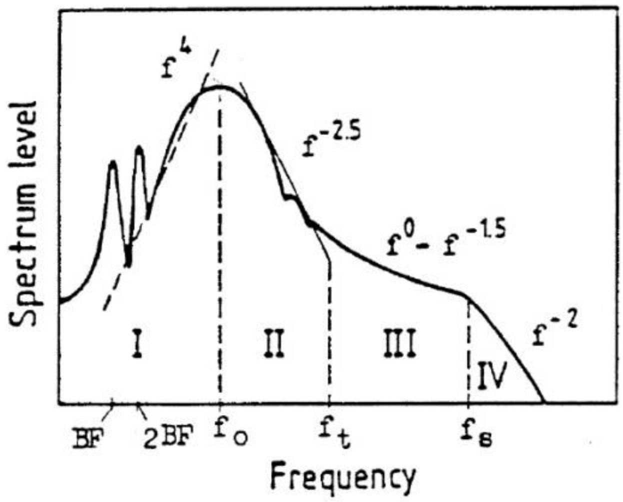


Figure 3.25 - Cavitating Propeller Noise Spectrum (Brown 1976)

In comparison, Vassenden & Lovik (1981) proposes different scaling rules for different frequency ranges, with a typical noise spectrum of a cavitating propeller being represented in the figure below:



p'^2 dependency	Slope for constant bandwidth		Slope for proportional bandwidth	
	Change dB /decade	Change dB / octave	Change dB /decade	Change dB / octave
$f^{+4.0}$	+40.0	+12.5	+50.0	+15
$f^{-1.5}$	-15.0	-4.5	-5.0	-1.5
$f^{-2.0}$	-20.0	-6.0	-10.0	-3.0
$f^{-2.5}$	-25.0	-7.5	-15.0	-4.5

Figure 3.26 - Cavitation Noise Spectra (Vassenden & Lovik 1981)

As all the above models tend to be severely limited in their applicability to current vessels, and in the range of situations for which they are valid, they will not be used within this work

3.6.5 Machinery Noise Modelling

This section will briefly outline some of the machinery noise modelling techniques available. It is not covered in detail as machinery noise is not the main focus of this study. However there is an extensive amount of research in machinery noise on ships, and its prediction, and details can be found in (Junger 1987), (Filcek 2006), (Zinchenko 1957), (Fischer et al. 1983) and many others.

Numerical Approaches

As discussed as a general subject in Smith (2011), there are several numerical methods which are often applied to structural response and noise propagation problems of this nature. Low frequency or low modal density problems are typical addressed using Finite Element Analysis (FEA), whereas higher frequency or higher modal density problems are addressed using Statistical Energy Analysis (SEA). Hybrid approaches are also under development for problems which fall in between the applicability of these two methods (Zoet 2013; Zoet et al. 2011).

A very simplified version of the SEA energy balance is presented below, from Smith (2011) and originally from Lyon & Jong (1995), which allows for the calculation of the frequency average vibration levels:

$$P = \omega m_0 S \langle v^2 \rangle \eta \tag{3. 36}$$

Where: P is the power input from the machinery

m_0 is the surface density of the structure to which it is attached

S is the surface area

$\langle v^2 \rangle$ is the space averaged mean square velocity

η is the damping loss factor

The space and frequency average mean square velocity is then:

$$\langle v^2 \rangle = \frac{P}{m_0 \omega \eta S} \tag{3. 37}$$

The radiated acoustic sound power can then be found from:

$$W_{rad} = \rho_0 c\sigma S\langle v^2\rangle \qquad (3.\ 38)$$

Where: W_{rad} *is the radiated acoustic sound power*

$\rho_0 c$ *is the characteristic impedance of air*

σ *is the radiation efficiency of the structure*

This approach is however only applicable to the simplest case of a point force applied orthogonally to a plate structure. Even this very simplified approach requires a substantial level of detail to be available regarding the exact nature of the machinery and the structure to which it is attached. The more complex the method then becomes for greater accuracy, the more complex and extensive the required model becomes, and hence the more in-depth detail is needed. These approaches, however accurate they could prove to be, are unlikely to be suitable for this particular application of an early stage design prediction model. This is also true for FEA approach, and for hybrid structural and modal analysis methods, hence these have also been deemed to be inappropriate for this work.

Empirical Approaches

It is relatively easy to predict the frequencies at which different items of machinery will emit tonal noise, as these are related to operational speeds of the machinery and associated harmonics, which can be easily found. The problem however lies in the prediction of the tonal amplitudes. Due to the complexity of the modelling and information required for the prediction of tonal amplitudes, these are considered outside the scope of this work.

The diagram below, taken from Fischer & Collier (2007) demonstrates an example of how the tonal frequencies in a machinery system can be calculated for the different components, in this case a diesel-electric system:

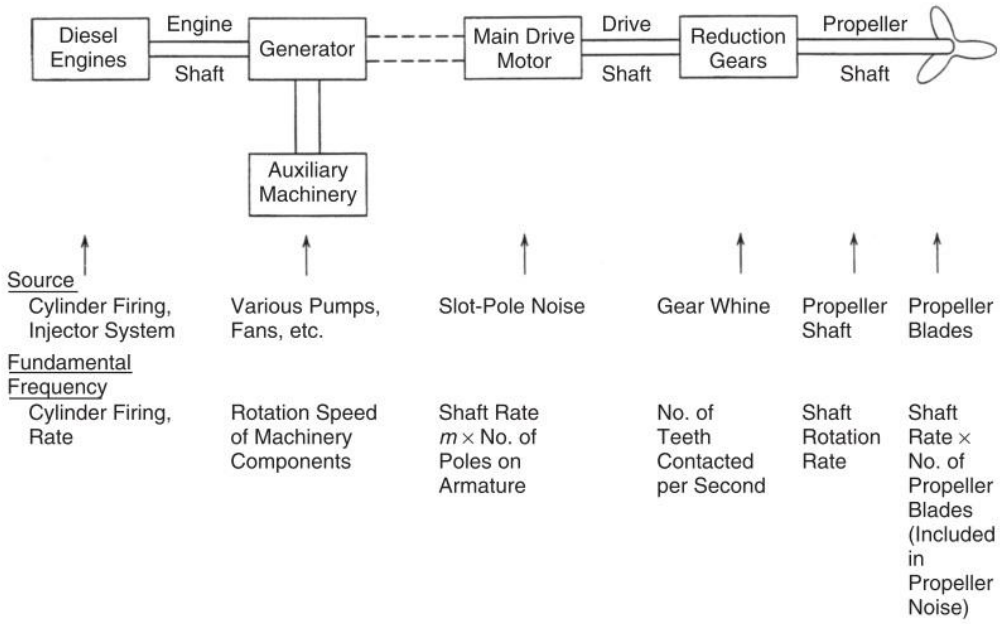


Figure 3.27 - Machinery Components and Noise Source Frequencies on a Diesel-Electric Ship (Urick 1983)

The disadvantage of the above calculations is that they give no indication of the sound pressure level, or tonal amplitude, which might be observed at these frequencies.

The formulae below, taken from Table 3 in Collier (1997), gives empirical approximations for the machinery vibration source levels, at source on board the vessel, for a range of typically installed machinery classes:

1) Diesel

$$-20\,log(w) + 20\,log(kW) + 30log\left(\frac{rpm}{rpm_0}\right) + 136 \qquad \textit{(3. 39)}$$

2) Reduction Gears

$$64 + 10log(kW) \qquad \textit{(3. 40)}$$

3) Generator

$$53 + 10\,log(kW) + 7log(rpm) \qquad \textit{(3. 41)}$$

4) Non-Hydraulic Pumps

$$65 + 10log(kW) \qquad \textit{(3. 42)}$$

5) Non-Hydraulic Pumps

$$63 + 10log(kW) \qquad \textit{(3. 43)}$$

Where: w is the gross weight in kg

kW is the rated power

rpm is the given rotational speed

rpm_0 is the rated rotational speed

In terms of airborne noise sources on the vessel interior, the following empirical formulae are suggested in Collier (1997), and are based on data presented by Fischer et al. (1983), which also discussed many other important aspects pertinent to this field. Is not clear on what data these empirical formulae have been based on, and what assumptions have been included.

1) Diesel Engines - Intake, Exhaust and Casing Radiation Baseline

$$L_{WB} = 58 + 10log(kW) \qquad \textit{(3. 44)}$$

2) Gas Turbines, Intermediate - Exhaust

$$L_{WB} = 74 + 10log(kW) \quad (3.45)$$

Where: L_{WB} *is the Sound Power Source Level, in dB re* 10^{-12} *W*

It can be seen that these predictions depend largely on rated power of the machinery. It is usual to assume a strong relationship between acoustic characteristics of machinery and its rated power. The approximations could also be representative of general machinery noise levels. However these formulae are based on data for older vessels than are seen in the current fleet and in new builds to which they might be applied. They also only give a prediction at source, so the problem of predicting corresponding propagation to the water would still apply.

The following formula, for airborne diesel engine broadband noise level estimation is taken from Ross (1976) and can also be used as a good indicator of the expected onboard machinery noise:

$$SPL = 91 + 10logj + 28lognD \quad (3.46)$$

Where: j is the number of cylinders

n is the rotational speed in rps

D is the cylinder diameter in metres

This formula requires more detailed specifics for a diesel engine and therefore is less suitable to early stage estimates. The results are also less suitable for use as general machinery noise estimation, as they would not make sense in the context of an alternative main propulsor type. The result is based on data published in (Zinchenko 1956; Zinchenko 1957), which was focused purely on marine diesel engines available at the time.

The frequencies of the main engine noise tonals and their harmonics can be calculated using the formulae below, taken from Gloza (2002):

Table 3.3 - Main Vibration Frequencies for a Diesel Engine

Vibration Frequencies	Cause of Vibrations
$f_{cfr} = \frac{k \times n_s}{2 \times 60}$	Cylinder firing rate
$f_c = \frac{k \times n_s}{60}$	Crankshaft
$f_v = \frac{k \times z_p \times n_s \times z_z}{m \times 60}$	Engine Valves
$f_{ps} = \frac{k \times z_p \times n_s}{60}$	Piston Slap
$f_{pr} = \frac{k \times b \times z_p \times n_s}{60}$	Piston Rings

Where: k is the number of the harmonic (a whole number)

n_s is the rotational speed of the engine in rpm

z_p is the number of pistons in the engine

z_z is the number of valves for one piston

m indicates if it is a two- or four-stroke engine

b is the number of piston rings for one piston

It should be noted that it can be difficult to translate on-board vibration or airborne noise values to its corresponding noise at an underwater receiver, given the very complex propagation path from each source, the interactions between different sources, and the secondary impacts from vibration transmission. Therefore, these formulae will not be used for any further work within this study, and are considered outside the scope of work. Only the tonal component formulae will be used.

3.7 Effects of Noise on Marine Wildlife

Contrary to some depictions of the ocean as "The Silent World", there is in fact a great deal of ambient noise ever present in the underwater environment. As discussed in Ross (1976), prior to the industrial revolution this consisted of a wide range of natural sounds, including wave and current noises, rain and wind, thermal effects, seismic activity and of course wildlife noises. The hundreds of different species of animals and fish use noise, both created and received, for a wide variety of purposes integral to their existence. However following the rise of shipping, sub-sea oil and gas exploration the pleasure industry and other marine-based activity, significant levels of man-made underwater noise are also being radiated. The level of this noise has been steadily rising since the start of the Industrial Revolution, however its' significance, and effects on the marine environment are not as widely considered. There is much discussion and research on the use of noise by many species of marine creatures and the effects on man-made noise on their lives and behaviours. However it is acknowledged that further research is required to gain a better understanding. This section does not intend to provide an exhaustive review of all the available literature, it aims only to give an overview of the challenges and recent knowledge with particular relation to ship radiated underwater noise. It is acknowledged that much has already been done to further understanding of use of noise by marine wildlife, and the impacts of anthropogenic noise on their lives, and most of this research has been publically published.

It is estimated that "there is no direct behavioural or physiological hearing data for almost 80 per cent of marine mammals" (IFAW 2008). One source of marine animal audiogram data currently in the public domain however is a summary of available information by Nedwell et al. (2004) at Subacoustech Ltd; however many of the tests summarised are based on data for only a few subjects of a species in a laboratory environment, and can therefore only serve as an indication of the animal's hearing ranges in relatively quiet surroundings with no prior hearing damage in the subjects. There is also increasing concern from conservation groups concerning hearing test, especially those which aims to find the limits above which temporary threshold shift for hearing occurs in different species, as these are currently unregulated and could potentially cause harm to the test subjects. There are now calls for regulated guidelines on hearing test procedures and methodologies to be published and monitored. As well as ensuring the test subjects are protected, such guidelines would also ensure all results and reports contained the same information, allowing for comparison and general use of the data. However the existing data are still valuable and are being used when addressing underwater radiated noise.

Another difficulty is that there is limited published data on behavioural reactions to noise of specific frequencies or intensities; typically the reaction is detailed however information relating to the nature of the noise causing the reaction is sometimes omitted. This is the case for both immediate and short-term responses. Records relating to possible longer term population effects are also limited at present, however current research may add to the body of available literature. A summary of known effects of noise on the behaviour of cetaceans can be seen in the paper by Nowacek et al. (2007), but as the author states, much of the available information on studies lacks comprehensive data on source and received levels of noise, frequencies, exposure times, and other vital information required for full review. Nevertheless, the data can still be used in the context in which it was originally recorded, which all assist in building a comprehensive picture. An important discussion on anthropogenic induced stress in marine mammals is also provided in Wright et al. (2007).

Figure 3.28 below gives the estimated audiograms for a range of different marine mammal species:

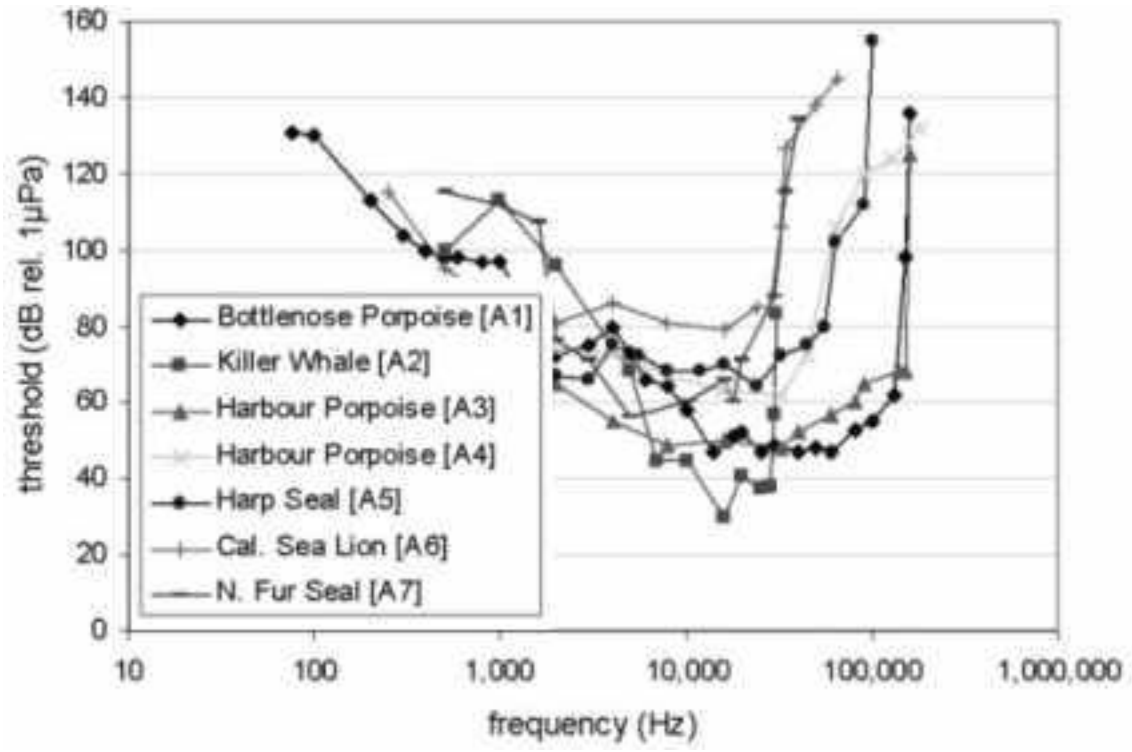


Figure 3.28 - Marine Mammal Audiograms (Pamguard 2010)

Most audiograms take the "U-shaped" form seen above, with peak sensitivity to frequencies approximately in the middle of the overall hearing range. Peak sensitivity means that at these frequencies the species can detect the sound at the lowest threshold above the background noise. It can be seen from these audiograms, marine mammal hearing covers an extremely wide range of frequencies, and therefore attempting to reduce the impact of underwater noise on all species would appear to be complicated, due to their differing requirements and sensitivities.

3.7.1 Cetaceans

The most comprehensive collection of data exists on noise with relation to Cetaceans; an order of marine mammals including whales, dolphins and porpoises. This order is split into two sub-groups: Mysticeti or Baleen whales, which include the blue and humpback whales, and Odontoceti or toothed whales, which include beaked whales, dolphins, porpoises and orca.

The exact uses of underwater noise by cetaceans, which can take the form of tonal sounds, clicks, pulses, knocks, whistles, booms and even full "songs", are still the subject of ongoing discussion, however some experimentation and observation has lead to generally accepted theories about their uses. According to IFAW (2008) (International Fund for Animal Welfare) in their paper on Ocean Noise Pollution,

"Marine mammals use sound to navigate and to detect predators and prey. It is essential for communication in order to attract mates, announce location and territory, to establish dominance and maintain group cohesion and social interaction".

The hearing ranges for marine mammals, especially cetaceans that have developed methods for communicating over many miles of ocean, are known to be much broader than those in land-based mammals (Ketten 2004). Naturally, they tend to vocalise at frequencies within their peak hearing sensitivity range, as this will ensure that their communication is as effective as possible. However, these vocalisations have been known to alter in frequency and also increase in intensity as a possible attempt by the animals to make themselves heard over the ever-increasing levels of ambient noise. This is due to a phenomenon known as "masking", where important biological sounds are effectively "hidden" by louder anthropogenic sounds which occur at similar frequencies. The changes to vocal behaviours are often only temporary, while a vessel is close by, but animal vocalisations have also been observed to have slowly increased in frequency over periods of several years to account for increasing ship traffic in certain habitats. This is known as the "Lombard Response" or "Lombard Effect", and has been observed in a number of marine mammal species, including Beluga Whales (*Delphinapterus leucas*) in the St Lawrence River, and Southern Resident Killer Whales (*Orcinus orca)* around Washington and Vancouver Island (Chapman 2007). These changes could potentially move the vocalisations outside the optimum ranges, compromising the efficiency of the communication and risking loss of key information transfer. Such seemingly small effects could have larger population-scale problems when they begin to affect feeding, mating or social cohesion behaviours.

The effects of underwater noise on cetaceans can be difficult to generalise, as these effects will vary depending on the nature of the sound, the frequencies present in the noise, the intensity or received level of the sounds, the duration for which the animal is exposed to the sound, and of course the species of the animal itself. Lower frequency, longer term shipping noises are likely to have a very different effect on animals to the mid- or high-frequency short impulses from airguns or sonar. The effect of the received level of sound will vary depending on the average ambient noise in the area, and also on the levels of sound typically used by the cetacean through natural behaviour. This would suggest that species living in quieter areas, such as previously un-chartered Polar Regions, with very little industrial activity could be much more sensitive to anthropogenic noise than those accustomed to areas of heavy shipping traffic, and therefore possibly habituated to some level of shipping noise. Different behaviours have also been observed depending on the season, with animals reacting differently depending on whether they are breeding or recently calved, migrating or feeding (Morton & Symonds 2002).

Figure 3.29 shows the potential impacts on marine animals at different respective distances from the source; for a known noise signature, location and species, specific values could then be added to this model. The model was suggested by Richardson et al. (1995).

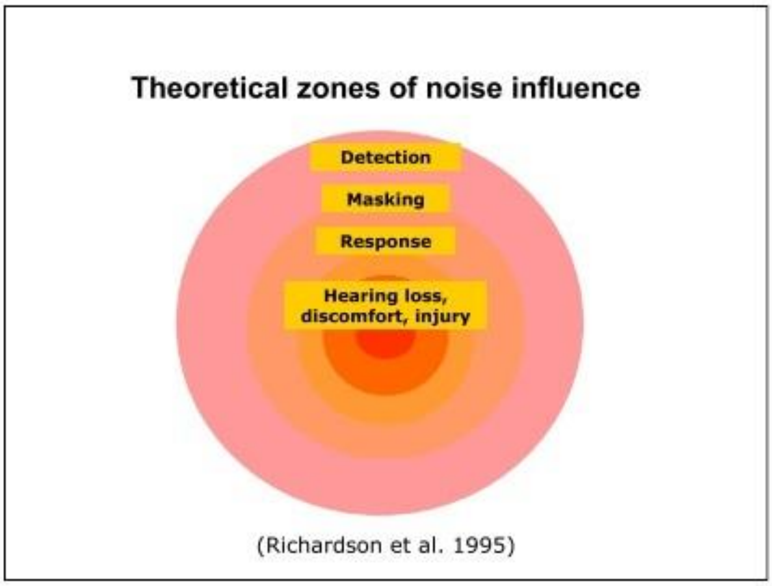


Figure 3.29 - Theoretical Zones of Noise Influence (OSPAR Commission 2009)

It is believed that there is a wide range of possible effects from noise, which could be responsible for the behaviours observed in the cases described above. In their report on Underwater Sound and Marine Life IACMST (2006) (Inter-Agency Committee on Marine Science and Technology) present a comprehensive summary of the possible effects of noise on marine life and marine mammals in particular those from Simmonds et al. (2004), some of which is presented below:

- **Physical** (**Non-Auditory)** - Damage to body tissue, induction of the "bends" (decompression sickness)

- **Physical (Auditory)** - Gross damage to ears, temporary or permanent hearing threshold shift (TTS and PTS)

- **Perceptual** - Masking of communication with co-specifics and other biologically important noises, interference with ability to acoustically interpret environment, adaptive shifting of vocalisations

- **Behavioural –** Interruption of normal behaviour both temporary and with decreased efficiency to more serious and permanent, displacement from area (short or long term), disruption of social bonds including mother-calf associations

- **Chronic/Stress** - Increased vulnerability to disease, increased potential for impacts from negative cumulative effects (e.g. chemical pollution combined with noise-induced stress), sensitisation to noise exacerbating other effects, habituation to noise causing animals to remain close to damaging noise sources

- **Indirect Effects** - Reduced availability of prey, increased vulnerability to predation or other hazards such as collisions with ships, entanglement in fishing gear and strandings etc., behavioural changes leading indirectly to physical damage, e.g. animals may be embayed and strand

A response severity table was also suggested based on data for wild and captive animal species, further highlighting the potential short-term impacts noise can have:

Table 3.4 - Scale of Severity Observed in Behavioural Responses of Wild and Captive Marine Mammals Exposed to Various Types of Anthropogenic Sound (André et al. 2009)

Response Score	Corresponding Behaviour (Individuals in the Wild)	Corresponding Behaviour (Individuals in Captivity)
0	• No response	• No response
1	• Short orientation response (visual orientation / research)	• No response
2	• Moderate or multiple orientation behaviours • Brief cessation / minor modification of vocal behaviour • Brief or minor change in respiration rate	• No negative response observed: may have appreciated sounds as some new object
3	• Prolonged orientation behaviour • Individual warning behaviour • Small changes in swimming speed, direction and/or diving but no fleeing from sound source • Moderate change in respiration rate • Cessation / lesser modification of vocal behaviour (<duration of source operation)	• Small changes in response to trained behaviours (e.g. delay in returning to initial position, intervals between longer tests)
4	• Moderate changes in swimming speed, direction, and/or in diving profile but no fleeing from sound source • Cessation / moderate modification of vocal behaviour (duration ≈ source operation time span)	• Moderate changes in response to trained behaviours (e.g. reticence to return to initial position intervals longer between tests)
5	• Consistent or prolonged changes in swimming speed, direction and/or diving profile but no fleeing from sound source • Moderate change in group distribution • Change in distance between animals and/or size of group (aggregate of separate) • Prolonged cessation / modification of vocal behaviour (duration>duration of source operation time)	• Severe and substantial changes in response to trained behaviours (e.g. splitting from position during test / experiment sessions)

6	• Moderate or less evasion of individuals and/or groups to sound source • Brief or small separation of mother from dependent young • Aggressive behaviour related to the exposure of the sound (e.g. tail / flipper slapping, opening and closing of mouth (making noise), abrupt changes in movement, formation of bubble clouds) • Cessation / modification of vocal behaviour • Visibly startled / frightened response • Brief cessation of reproductive behaviour	• Refusal to commence trained tasks
7	• Considerable or prolonged aggressive behaviour • Moderate separation between mothers and dependent young • Clear anti-predator response • Severe sustained evasion of sound source • Moderate reduction in reproductive behaviour	• Evasion from experimental situation or seeking refuge (≤ duration of the experiment) • Menacing behaviour or of attack towards the sound source
8	• Obvious aversion and/or progressive sensitization • Prolonged or severe separation between mothers and dependent young with disruption of acoustic regrouping mechanisms • Long term evasion from the area (>operation of the source) • Prolonged cessation of reproductive behaviour	• Total evasion from acoustic exposition area and refusal to carry out trained behaviours for over 24 hours
9	• General panic, fleeing, stampeding, attacking of congeners, or strandings • Evasive behaviour related to present of predators	• Total evasion from acoustic exposition area and refusal to carry out trained behaviours from more than 24 hours

It should be noted that some of the above will affect the individual, while others could have more profound effects on a population-wide scale. For a more detailed review of known effects of noise on marine animals, the reader is referred to the review by Weilgart (2007). Recently developed by the NRC (National Research Council), and published in a 2005 report, the Population Consequences of Acoustic Disturbance (PCAD) model attempts to predict the population-scale effects on anthropogenic noise, in order to assess and prevent these impacts before they occur on a serious scale. Figure 3.30 shows this theoretical model; the number of crosses at each stage represents the amount of data and information available on those variables to the authors at the time of publication. More recent work may well have advanced knowledge in these fields of research.

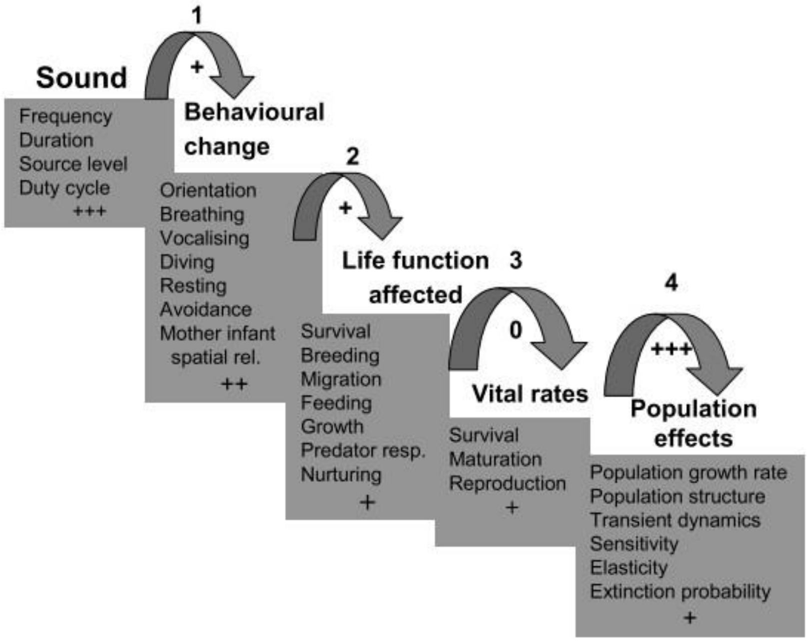


Figure 3.30 - The NRC PCAD Model (OSPAR Commission 2009)

As with most problems, a compromise has to be sought in order to find a solution which will reduce the effects of underwater noise without creating different issues. The effect on marine animals generally, but whales specifically, of a reduction in ship noise has to be considered. Unlike dolphins and other odontocetes, baleen whale use passive listening rather than active echolocation to locate prey and other objects such as ships within their environment. This can lead to a multitude of problems where the "masking" of critical noises by a high ambient noise level occurs. Each animal will have a critical ratio of hearing for different frequency bands. This value represents a dB value by which a sound level has to be higher than the ambient noise for that animal to hear it; this ratio stays constant regardless of the intensity of the sound. Consequently, at higher ambient noise levels, this critical value may never be reached by the important sound and hence it will not be detected. In particular, it is believed to be the cause of ship – whale collisions, as the low frequency noise of ships, particularly at the slower speeds, such as those required within a specified protection zone, are undetectable above the general environment noise, as discussed by Gerstein (2002). Ship noise, particularly from the main source of the ships propeller, can also be highly directional, with a large area directly in front the vessel having much lower sound levels due to shadowing effects from the ship's hull combined with shallow water absorption effects and surface attenuations. Whales will often seek refuge from the ship noise in this quieter area, resulting in often fatal ship strikes (Gerstein et al. 1999). This means that the consequences of any noise reduction also need to be carefully considered, and any associated problems in terms of different impact on the wildlife, or on the ship performance, have to be addressed accordingly.

3.7.2 Pinnipeds

Due to the tame nature of some of the species in this order, there is also a good body of data available with relation to Pinnipeds; the fin-footed sea mammals. This order is split into three sub-groups; namely the Odobenidae, which include Walruses, the Otariidae or eared seals, which include Sea Lions and Fur Seals, and Phocidaec or earless seals, which include the Harbour and Elephant Seal.

Many of the uses of noise, as well as the potential effects of man-made noise, described with particular reference to cetaceans (above) can also be equally well related to pinnipeds. Tests into the effects of broadband noise and tonal stimuli carried out by the Institute of Marine Sciences at Long Marine Laboratory, University of California, Santa Cruz on Harbour Seals (*Phoca vitulina*) found significant evidence of temporary threshold shift (TTS) occurrence, and good correlation between shift and recovery time, and the sound exposure level experienced by the subject (Kastak & Reichmuth 2007). Such data is extremely valuable in helping to understand animal behaviours in relation to noise, as well as providing solid evidence of the damage it can cause. These types of noise, in such areas, at such exposure levels can then be specifically avoided or at least reduced. These same subjects were also used along with subjects from two other pinnipeds species, the Californian sea lion (*Zalophus californianus*) and the northern elephant seal (*Mirounga angustirostris)*, in a separate experiment to establish critical ratio levels at different frequency levels, as these could give indication to the possible "masking" effects of anthropogenic noise on these species (Southall et al. 2000). Although these particular species were found to have comparatively low critical ratio values, this should not be taken as a sign that anthropogenic noise in innocuous to these animals; indeed this high processing capability is likely to be integral to their survival, and should not be compromised in any way.

3.7.3 Fish

With an estimated 31,500 species of fish living in the world's waterways, it is unrealistic to expect to have noise related data for even a small percentage of them. However, work by fisheries research vessels, as well as observation by fishing and trawling vessels in particular, have shed some light on the potential effects of noise on some of the more "commercial" species, as well as their use of noise. Unfortunately, it has been suggested that the very vessels least wishing to disturb fish, such as research and fishing vessels, could in fact be eliciting avoidance behaviours and inducing stress, which can lead to stunted growth (Mitson 1995). It has also been suggested that in fish with swim bladders, resonance responses can be induced at certain frequencies of underwater noise, which will have an increased effect on fish with larger swim bladders. This would suggest that the larger fish are most likely to avoid the vessel, causing undersize catches and incorrect biodiversity data collection. In some very specific circumstances, the occurrences of high sound level impulses such as seismic charges have been known to cause mortality in fish in the immediate area (McCauley & Kent 2008). Reduced catch rates of 50-80% and the presence of fewer fish near seismic surveys have been reported in species such as cod, haddock, rockfish, herring, and blue whiting in a number of studies worldwide (Weilgart 2006). In other species, their primary sensing mechanism of "noise" is through particle velocities in the water, which are generally at low frequencies and tend to have lower amplitudes.

Due to the fact that the peak sensitivity of fish hearing lies in the lower frequency ranges, where ship radiated noise typically occurs, it was assumed to be this noise that was eliciting the fish avoidance behaviours of either diving deeper, or swimming horizontally away from the vessel. However, recent vessel-vessel comparisons between traditional design research vessels and new ICES-approved designs have left the success of the noise reductions on minimising avoidance behaviour in some doubt. One experiment surveying Atlantic Herring (*Clupea harengus*) in Norwegian waters found that statistically the herring displayed a greater avoidance behaviour towards the quieter ICES-designed vessel, although this tended to occur once the transducer had passed by the fish (Ona et al. 2007). The ICES limits are discussed later. Another survey carried out in the East Bering Sea looking at Walleye Pollack (*Theragra chalcogramma*) stocks did not provide results where this clear difference was apparent , nevertheless again indicated that the fish displayed greater avoidance behaviour to the ICES vessel, once the transducer had passed by (De Robertis et al. 2008). Both of these experiments would suggest that the real stimuli for fish avoidance behaviour requires further research, however a benefit is still available from vessel quieting in terms of general environmental impact and also improved acoustic instrumentation performance. The latter experiment also highlighted the issue of maintaining a vessels acoustic performance; the vessel complied with the ICES limit when first built, but failed at certain frequencies when subsequently tested in the two following years of service.

3.7.4 Other Marine Organisms

There is less research data available in the literature regarding the effects of noise, or noise-related behaviours of other marine wildlife, as they are much less visible and many have limited commercial value. There is therefore nothing in the way of guidelines relating to these creatures as there is at present for fish. However, there are some known instances of extreme reactions to high source level underwater impulse noise. Snow crabs under seismic noise conditions showed bruised organs, stress, and smaller larvae, while even giant squid have seemingly mass stranded because of air gun noise, suffering massive internal injuries and badly damaged ears (Simmonds et al. 2004).

A well-known and fairly prominent source of ocean noise in shallow water regions is the “snapping “of millions of shrimp, caused by them clicking their claws together, although it has recently been discovered that the noise in fact originated from cavitation caused by the closing motion, rather than from the claws themselves (ScienceDaily 2000). The shrimp apparently uses its snap to stun its prey, defend its territory, and communicate with other shrimp. The resulting noise can have significant effects on the local ambient noise in coastal areas, and has also been known to “drown out” military sonar. The presence of this noise appears to be integral to the lives of the shrimp, and should therefore not be “masked” wherever possible.

Sea turtles are widespread across the global, inhabiting every world ocean except the Arctic, and are known to have hearing adapted specifically to low frequency sounds, similar to Mysticeti (Baleen Whales). This would suggest that they are likely to be affected by the continuous low-frequency noise created by shipping, and will be able to hear these noises at large distances from the original source, although there is no data on the nature of these effects in free-swimming turtles, and only one case of data for captive subjects (Lenhardt 1995).

It should be considered that underwater radiated noise from ships could also impact non-marine wildlife, such as sea-bird, penguins and polar bears.

3.8 Underwater Noise Regulations and Standards

There are currently no overarching regulations or standards for controlling the radiated underwater noise of commercial vessels. As sound propagation does not conform to any political or legal barriers, it is also very difficult for countries to impose underwater noise limits in territorial waters. Any regulation of this issue therefore needs to have international agreement, and to apply to all branches of the marine industry. Such agreement could still be a long way from fruition, especially given the current uncertainties associated with the potential impacts of different anthropogenic marine noise sources on the marine environment.

The IMO (International Maritime Organisation) recently released a set of guidelines providing advice and guidance to designers, ship owners and shipbuilders on reducing ship radiated noise (IMO 2014), however there is at this point no reference to the noise receptors within this report. This takes the form of document MEPC.1/Circular 833 - Guidelines for the Reduction of Underwater Noise from Commercial Shipping to Address Adverse Impact on Marine Life, and also provides non-mandatory guidelines which apply only to Commercial Shipping. It also gives a set of definitions, and a list of noise prediction techniques and related references.

This section will outline current suggested limits, some of which could be applicable to commercial vessels. The origins and intentions of these limits will be discussed. It will also outline current regulations for noise measurement and reporting procedures which could be applicable.

3.8.1 Ship Radiated Noise Limits

Fisheries research vessels, and any other vessels involved in research work are now required to be built to comply with the International Council for the Exploration of the Seas (ICES) recommendations for underwater radiated noise levels at a vessel speed of 11 knots; taken as the typical operational speed for a vessel conducting research work. The figure below presents the ICES recommended noise limits for fisheries research vessels:

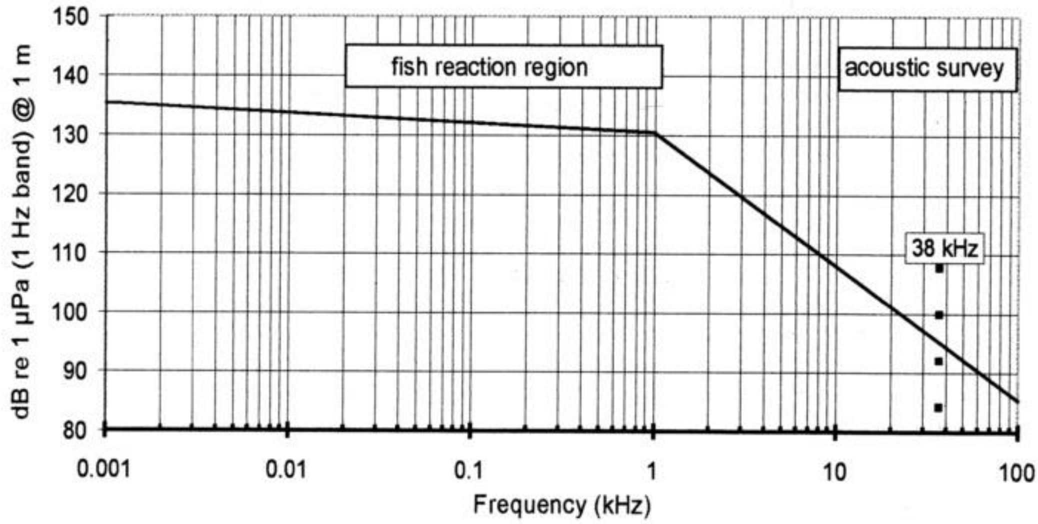


Figure 3.31 - ICES Fisheries Research Vessel Noise Limits (Mitson 1995)

Research into the above limits initially began as it was suggested that erroneous results for fish stocks were being obtained. The fish, it appeared, were being scared away by the research vessels themselves. The ICES carried out detailed investigation into ship underwater noise and fish hearing, the result of which are the recommended noise limits at 11 knots forward speed. The values are based on the hearing sensitivity of Atlantic Herring (*Clupea harengus)* and Atlantic Cod (*Gadus morhua*); these fish species are believed to have the highest hearing sensitivity and range. It is therefore assumed that if the noise level is now "acceptable" to these species, then the same can also be said for the majority of other fish species. The low range limits are set specifically to try and limit fish avoidance behaviours, while the higher frequency limits are set to try and maximise the performance of the acoustic instrumentation used during the surveys (Mitson 1995). There are several papers in the literature such as Bonney & Bahtiarian (2006), Hotaling et al. (2001) and Rolland & Clark (2010) which outline the measures adopted on fisheries research vessels in order for their acoustic performance to comply with the ICES limits, some of which may be transferrable to larger commercial vessels.

In January 2010, the classification society Det Norske Veritas (DNV) released a new Silent Vessel Notation Class. These non-obligatory requirements are split into 5 classes: Acoustic, Seismic, Fishery, Research and Environmental. The first four classes are aimed at improving the efficiency of the vessels in their designated work through noise control, whereas the final Environmental class is more geared towards giving merchant vessel owners a way of proving to the public and industry that they are trying to improve the "green" credentials of their vessels. The table below outlines the purpose and requirements of these five classes:

Table 3.5 - DNV Silent Class Notation (DNV 2010)

Name	**Refer to**	**Applies to**	**Frequency Range (Hz)**	**Source Level (dB re 1µPa at 1m, 1 Hz band)**
Silent Class Notation - Acoustic (A)	Rules for ships, January 2010, Pt. 6, Ch. 24, Sec. 2	Vessels using hydro-acoustic equipment	1,000-100,000	Light Survey (kHz) = 156-12logf Thruster condition (kHz) = 165-12logf
Silent Class Notation - Seismic (S)	Rules for ships, January 2010, Pt. 6, Ch. 24, Sec. 2	Vessels engaged in seismic research activities	3.15-315	168 in each 1/3 octave band 175 integrated over the frequency range
Silent Class Notation - Fishery (F)	Rules for ships, January 2010, Pt. 6, Ch. 24, Sec. 2	Vessels performing fishery activity	10-100,000	Light Search (Hz) = 162-6logf (10-100) / 138+6logf (100-1,000) / 156-13.2logf (1,000-100,000) Heavy towing (Hz) = 178-8logf (10-100) / 162 in each 1/3 octave band (100-1,000) / 162-15logf (1,000-100,000)
Silent Class Notation - Research (R)	Rules for ships, January 2010, Pt. 6, Ch. 24, Sec. 2	Vessels engaged in research or other noise critical operations	10-100,000	Research (Hz) = 171.8-22.5logf (10-25) / 128.7+8.3logf (25-1,000) / 153.6-12logf (1,000-100,000)
Silent Class Notation - Environmental (E)	Rules for ships, January 2010, Pt. 6, Ch. 24, Sec. 2	Any vessel wanting to demonstrate a controlled environmental noise emission	10-100,000	Quiet cruise (Hz / kHz) = 171-3logf (10-1,000) / 162-12logf (1,000-100,000) Transit (Hz / kHz) = 183-5logf(10-1,000) / 168-12logf (1,000-100,000)

Rather than being based on specific marine animal species hearing ability or on known environmental effects, these values are based on what DNV believe can be realistically achieved through good design philosophy, with current propeller and hull design, materials, and operational conditions. It could therefore be difficult to justify whether these limits are "sufficient" in combating the issue of underwater radiated noise impact; however it does mean that the values are not based on conjecture regarding the potential environmental effects of this noise, which as has been suggested above, is not always fully justifiable. Their introduction does however highlight the change in public perception and that of the scientific community, towards beginning to focus on the issue of underwater noise, and the need for its regulation and control. It should be noted that to date, the Classification Society has noted a significant demand for notations under the Acoustic, Seismic, Fisheries and Research classes, however the lack of economic drivers has meant that little interest has been expressed in the Environmental class.

It is believed that classification society Bureau Veritas (BV) are also intending to release non-compulsory guidelines for underwater noise, while Lloyd's Register (LR) and Germanischer Lloyd (GL) are offering technical assistance on the matter to customers who require it.

The main output of the FP7 Project SILENV, outlined in the Project Overview Chapter, is a set of "Green Label Limits". These are designed to suggest achievable limits for onboard, underwater and airborne noise for commercial vessels. Similar to the DNV limits underwater noise in transit and quiet conditions for commercial vessels have been specified, and are outlined below. These were developed following analysis of the DNV limits, and comparison of a range of ship source level data with these limits (SILENV Consortium 2012a).

Table 3.6 - SILENV Green Label Limits for Commercial Ship URN Levels (SILENV Consortium 2012a)

Limit	Frequency Range (Hz)	Source Level (dB re 1µPa at 1m, 1 Hz band)
SILENV Quiet Limit	10 - 100	157 - 10Log[f(Hz)/10]
SILENV Quiet Limit	100 - 1000	147 - 12Log[f(Hz)/100]
SILENV Quiet Limit	1000 - 10,000	135 - 23Log[f(Hz)/1000]
SILENV Transit Limit	10 - 100	167 - 10Log[f(Hz)/10]
SILENV Transit Limit	100 - 1000	157 - 12Log[f(Hz)/100]
SILENV Transit Limit	1000 - 10,000	145 - 23Log[f(Hz)/1000]

The figure below provides a comparison of the underwater noise limits specified by DNV and the SILENV Green Label proposal. It can be seen that the SILENV Limits are consistently more stringent than those proposed by DNV. As both have claimed to have defined their limits on what they feel is possible based on current technologies and knowledge, there appears to be a discrepancy in opinions even amongst practitioners and experts in the subject. It is felt that these are still somewhat arbitrary limits and clearer understanding of the reasons for these differences are required. The additional expense and training required to achieve even these limits is unlikely to be acceptable to commercial ship builders and operators while they remain optional, and there is a lack of evidence to suggest that the noise reduction is having a positive effect on the problem.

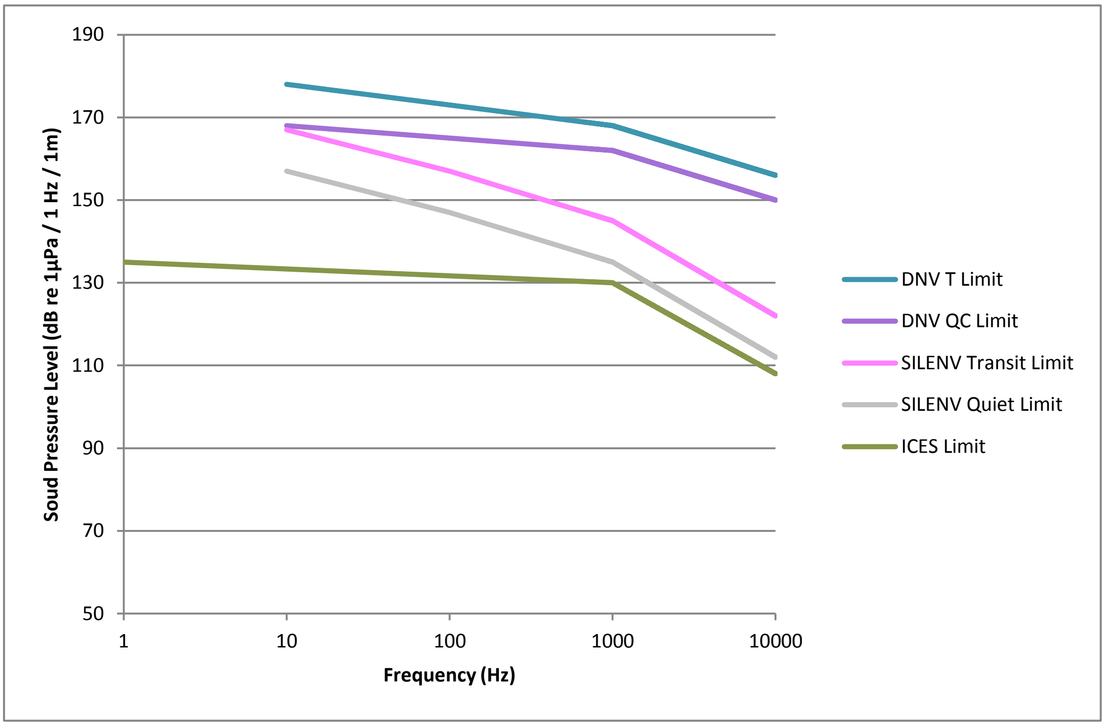


Figure 3.32 - Comparison of Underwater Noise Limits

In some areas, such as the Florida coastal waters in the US, speed limited zones have been introduced to try and reduce the impact of underwater radiated noise. The particular species of interest within this area is the West Indies Manatee (*Trichechus manatus*), and the aim has been to reduce speed of vessels to allow the creatures more time to move away from the vessels path and avoid collisions. In other areas such as near the Port of Boston, Massachusetts in the US, major shipping lanes have been relocated to reduce the acoustic impact on a marine sanctuary nearby. In other areas, it has been suggested that geographical or seasonal exclusion zones should be put into place, to avoid the ensonification of biologically significant areas such as feeding or breeding grounds, or migration routes, especially those of particularly sensitive or endangered species. For smaller areas where the impacted species are more easily identified and understood, such pragmatic limits could prove much more beneficial than general noise level limits for all vessels in all areas.

On a smaller scale, some comprehensive guidelines exist which cover how whale-watching vessels are required to operate in the vicinity of these creatures. These guidelines include a minimum distance of approach to whales, low speed zones at given distances, a requirement for turning off engines rather than idling while stationary, not following the creatures or interfering in any way with their movement, and maximum numbers of vessels allowed near the creatures at any given time. The specifics will vary depending on the location, but the aim remains the same. Although these guidelines cannot be directly applied to commercial vessels, there may be areas in which similar strategies can be used to reduce the impact of commercial vessels on marine animals.

The German Federal Maritime and Hydrographic Agency has recently released a noise limit which is to be adhered to during construction of offshore wind farm installations, of 160 dB re 1μPa at 750m, and requires full Environmental Impact Assessments (EIA) for any such planned work within its Exclusive Economic Zone (EEZ). Under the United Nations Convention of the Laws of the Sea (UNCLOS) Art. 211, member states have a duty to establish international rules and standards and flag states to adopt laws and regulations. The absence of any international regulations on noise do not change the duty to coastal and flag states to implement their own laws, and in fact gives states an option to introduce unilateral regulations in their own waters. The above is an example of this taking place.

3.8.2 Underwater Noise Measurement

The main standard currently available for the measurement of underwater noise is ANSI S12.64-2009 - Quantities and Procedures for Description and Measurement of Underwater Sound from Ships - Part 1: General Requirements. This was released in December 2009, and provides guidance on the methodologies and procedures which should be adopted for the measurement and reporting of underwater noise for surface vessels. A summary of the requirements of this standard can be seen in the table below:

Table 3.7 - Summary of ANSI Underwater Noise Measurement Requirements (The Acoustical Society of America 2009)

Grade	**A**	**B**	**C**
Grade Name	**Precision Method**	**Engineering Method**	**Survey Method**
Measurement Uncertainty	1.5 dB	3 dB	4 dB
Measurement Repeatability	+ / - 1 dB	+ / - 2 dB	+ / - 3 dB
Bandwidth	One-third Octave Band	One-third Octave Band	One-third Octave Band
Frequency Range (one-third octave bands)	10 to 50,000Hz	20 to 25,000Hz	50 to 10,000Hz
Narrowband Measurements	Required	Required	As Needed
Number of Hydrophones	Three	Three	One
Nominal Hydrophone Depth(s)	15^o, 30^o, 45^o angle	15^o, 30^o, 45^o angle	20^o + / - 5^o angle
Minimum Water Depth	Greater of 300m or 3x overall ship length	Greater of 150m or 1.5x overall ship length	Greater of 75m or 1x overall ship length
Minimum Distance at Closest Point of Approach (CPA)	Greater of 100m or 1x overall ship length	Greater of 100m or 1x overall ship length	Greater of 100m or 1x overall ship length
Acoustic Centre Location	Determined during testing	Halfway between the Engine Room and the Propeller	Halfway between the Engine Room and the Propeller
Minimum Number of Runs per Condition	6 Total 3 Port 3 Starboard	4 Total 2 Port 2 Starboard	4 Total, at least one starboard and one port
Recommended Weather / Sea Conditions	Wind Speed < = 20 knots	Wind Speed < = 20 knots	Wind Speed < = 20 knots
Auxiliary Measurements	Engine shaft speed, wind speed and direction, sound speed profile	Engine shaft speed, wind speed and direction	Engine shaft speed, and wind speed and direction

The International Organisation for Standardization (ISO) has adopted the above ANSI Standard. Many other measurement organisations will have their own procedures, developed to suit the nature of their usual work. For example, both DNV and SILENV outline recommended measurement, analysis and reporting procedures in relation to their noise limits. It should be noted that all of these procedures currently require the measurements to be carried out in deep water, and no corrections are proposed for measurement in shallow waters. Suitable deep water locations are not easy to come by within European waters, and travelling to find such conditions can increase the costs significantly. It is also noted that as such European-based vessels would hence be operating in shallow water for the majority of the time their acoustic performance should also be assessed in such conditions. The need to carry out the measurement of vessels in realistic operational conditions extends not only to large commercial tankers and bulk carriers, but also to more specialised vessels such as dredgers, which always operate in shallow coastal areas. Work is being carried out as part of the EC funded FP7 framework project "AQUO" (Achieving QUieter Oceans) to establish more suitable procedures to suit European water conditions. Despite this however, these is a gap in current practice regarding the measurement methods for vessel noise in shallow water conditions, and extending the procedures for tests carried out to more accurately reflect noise operations.

DNV are currently looking at the possibility of developing a standard for through-hull noise measurements in the near field to give an indication of far-field properties, chiefly as verification and trouble-shooting option. Initial data shows relatively good agreement at medium to high frequencies, but that such methods may be less suited to low frequencies. Others are looking at methods for using onboard noise to predict underwater noise, and to identify the onset of cavitation etc. The ISO is currently looking at the possibilities of creating a more "simple" standard for underwater noise measurement during sea trials that would be less technically demanding than the current standard, to reduce costs and encourage more shipbuilders to carry out such measurements. The details of this standard are not yet clear. It is too early to state whether such moves would be positive for the industry, but it does have the potential to make vessel noise measurements cheaper and easier for ship operators, and hence increase the likelihood of them being carried out.

Once the data has been collected, the standard then details the post-processing procedures for normalising for distances and multiple hydrophones, background noise and sensitivity. The data is then corrected to a nominal distance of 1m from the acoustic centre, as is common practice in most underwater noise applications. Finally, the standard provides a reporting layout which should be used for all underwater measurement data, for consistency of information and to allow for comparisons between data for different cases. This standard is typical of other similar guidelines, however they have been said to contain flaws for certain applications, specifically smaller, quieter vessels. Assumptions made in post-processing to account for corrections of the measured values back to source level are also unjustified within the ANSI standard.

Regulation 39 of the Marine Offshore Conservation Regulations of 2009 (UK Government 2009) states that 63Hz and 125Hz have been selected as the centre frequencies for analysis of background noise, as they relate to the most power from shipping noise, and these are also typically used for specific measurements of vessels. Work currently being carried out at LAB (Laboratori D'Aplicacions Bioacústiques) at Universitat Politècnica de Catalunya (UPC) regarding the modelling of underwater noise impacts on marine animals has also suggested adding a further three centre frequencies. These are 25Hz, 500Hz and 1000Hz, and are intended to be applied specifically to account for Baleen Whale, mid-frequency and high-frequency cetacean impacts respectively. More direct links between marine mammals and noise is positive, and more research should be carried out looking more specifically at relationships between noise and impact in these critical frequency regions.

Ambient noise measurement standards are also being proposed within the EU, and some national standards already exist in Germany and the Netherlands. It is again important to understand the environment in which the marine species are living in worldwide, in comparable terms, so that the potential increases in noise from vessel transits can be more realistically quantified.

Guidelines detailing the procedures to be followed during onboard noise and vibration are outlined in the IMO (International Maritime Organisation) Resolution A.0468(12) - Code on Noise Levels on Board Ships, adopted in 1981. These standards give the operational conditions, measurement locations, measurement procedures, and many other key parameters required to ensure that all measurements and data reporting practices maintain consistency and allow for comparison of data. For full details of the requirements of this resolution, the reader should refer to IMO (1981). Other national standards, such as those published by the MCA (Maritime Coastguard Agency) in the UK, are based on the requirements of this resolution, and most IACS (International Association of Classification Societies) Classification Societies will refer to this resolution for required measurement procedures. The current regulation on on-board noise is important to underwater noise. In some cases, most notably in the cruise ship industry, the significant improvements made to reduce onboard noise levels have had the additional benefit of significantly reducing underwater noise levels. However care should be taken that when reducing on-board noise, the problem is not simply moved underwater. It has been suggested that podded propulsion units, while beneficial for on-board noise levels, could increase comparable underwater level compared to a typical single propeller case, however there is little data available in the literature on this subject, and more research may be required. This is due to the presence of the propulsion motor in the water unit rather than onboard. It is felt that it could be beneficial for guidelines on the reduction of on-board noise to be released, to demonstrate best practice, to ensure that changes benefit the global rather than just the local levels.

As part of the SILENV Green Label proposal, procedures for the quantification of on-board, underwater and airborne noise were also suggested. These aim to overcome some of the shortcomings observed in the procedures discussed above, while remaining realistic and limiting costs (SILENV Consortium 2012a).

As well as using the correct measurement procedure, it is also important that the impact of different post-processing methods are well understood. For example, the way the data is divided up and averaged can make a significant difference to the results obtained. Using the arithmetic mean, the geometric mean or the median will give very different values, and hence it should be clearly stated how the post-processing has been carried out.

3.8.3 Marine Wildlife Standards and Frameworks

Other than the ICES ship noise requirements, discussed above, for fisheries research vessels based on fish hearing ability, there are no official regulations or standards to govern these concerns. Discussed below are a number of existing guidelines which can be related to marine animal conservation, and also some recent work on marine animal weighted measures of sound.

Existing Regulatory Frameworks

There are a number of legal instruments which have at least in part attempted to acknowledge and regulate anthropogenic impacts on the environment; and pertinent to this research, in particular the impacts of noise on marine wildlife, typically cetaceans. Some of these, as identified by Simmonds et al. (2004) and other sources, are listed below:

National

- The US Marine Mammal Protection Act (MMPA) of 1972, as amended in 2007
- The EU Marine Strategy Framework Directive (MSFD) (2008/56/EC) established in 2008
- 1997 Council Directive 97/11/EC, on the Assessment of the Effects of Certain Public and Private Projects on the Environment

- 1992 Council Directive 92/43/EEC, on the Conservation of Natural Habitats and Wild Flora and Fauna

Regional

- 1992 Agreement on the Conservation of Small Cetaceans of the Baltic and North Seas (ASCOBANS), extended in 2008 to include the North East Atlantic and Irish Seas
- 1996 Agreement on the Conservation of Cetaceans of the Black Sea, Mediterranean Sea and Contiguous Atlantic Area (ACCOBAMS)
- The Helsinki Commission (HELCOM), which is the governing body of the "Convention on the Protection of the Marine Environment of the Baltic Sea Area"
- 1979 Bern Convention on the Conservation of European Wildlife and Natural Habitats
- The Convention for the Protection of the Marine Environment of the North-East Atlantic , also known as the “OSPAR Convention”
- The Joint Nature Conservation Committee (JNCC) which is the statutory adviser to Government on UK and international nature conservation.

International

- 1979 Convention on the Conservation of Migratory Species of Wild Animals (also known as CMS or the Bonn Convention)
- The United Nations Convention on the Law of the Sea (UNCLOS) from 1982 and the United Nations Open-Ended Informal Consultative Process on Oceans and the Law of the Sea (UNICPOLOS) which has been operating since 2004
- International Maritime Organisation's (IMO) Resolution A.927(22) 2001"Guidelines for the Designation of Special Areas Under MARPOL 73/78 and Guidelines for the Identification and Designation of Particularly Sensitive Sea Areas"

- The International Union for the Conservation of Nature (IUCN) Resolution RESWCC3.068 which was the first to deal with underwater noise pollution problem at the global level
- The International Whaling Commission (IWC) Resolution 1998–6
- Work by many conservation groups, such as International Council for the Exploration of the Seas (ICES), the International Fund for Animal Welfare (IFAW), the Whale and Dolphin Conservation Society (WDCS), and the International Ocean Noise Coalition (IONC).

These regulations take a much more general approach to conservation of the environment, and concurrent parties are in agreement that their respective national laws should reflect this aim towards conservation; however the approach of each will depend on their interpretation of what is required. In general, the above regulations require the parties to ensure that prohibition against intentional taking or killing of any covered species or habitat is covered by national law, where "take" is typically defined to include harassment or activities which would result in the harassment of these species or habitats. None of the above instruments however specify how this should be achieved, and there limited scientific data to provide any means for justifying the efficacy of measures which are imposed. These instruments, or other methods, could in future be used to require designers, ship owners and indeed governments to take action on noise pollution in a way that they have not until now.

There are a number of publications which provide an extensive discussion on the regulation of underwater noise including (Scott 2004), (Dotinga & Elferink 2000), (André et al. 2009) and (McCarthy 2004).

Threshold Noise Values and Data

The US government body National Marine Fisheries Service (NMFS) has suggested threshold level for behavioural disturbance in Cetaceans and Pinnipeds. This value of 120 dB re 1Pa for continuous sounds was suggested by Southall et al. (2007). It has also suggested thresholds for TTS and PTS in Cetaceans of 224 / 230 dB re 1µPa, and in Pinnipeds of 212 / 218 dB re 1µPa.

A large body of data exists which details the behavioural and physical effects of many different types and levels of underwater noise on a range of marine animal species. The papers available in the public cover both data gained during experimental conditions, usually on captive species, and also open water observations during a range of marine industrial activities. This research has focussed only on those papers which record data on marine animal reactions to shipping noise, either actual or simulated, and those papers in which details of the nature of the sound and receiver distances have also been recorded. It was noted, however, that a large proportion of the cases presented failed to present detail of the noise, the propagation conditions, or the distances of the affected marine animals, meaning that the information was of limited use to this research. This also highlights the need for better guidelines and regulations for reporting such data, however it is encouraging that such studies have been done and continue to be carried out, given their great importance to future understanding.

3.9 Reducing Ship Radiated Underwater Noise

There are two general approaches to reducing ship radiated noise; by looking at either the cause or the effect. Focus can be either on reducing the noise and excitation vibrations created by the many sources, with most improvement gained from addressing the few more dominant sources, or can be addressed instead to reducing the overall levels of noise once they have been created, through insulation, isolation and damping. Ideally, both aspects should be given attention; however this can prove to be a very expensive exercise. Care should also be taken that the noise mitigation measures do not compromise the efficiency and main function of the vessel in question. It should be noted that addressing ship underwater radiated noise level may also have positive impacts on vessel efficiency, however this is not guaranteed and requires a compromise on propeller Blade Area Ratio (BAR) and other parameters (Hauerhof 2013).

Until noise reduction methods are shown to be beneficial to the environment, cost effective, efficient, and suitable for large-scale commercial applications, the issue of ship radiated underwater noise is likely to continue to be largely overlooked. As highlighted by Mr Francesco De Lorenzo of Fincantieri at the 2012 "Fish & Ships" Conference, ship designers, owners and operators are unlikely to act until formal regulations are put into place, with clear requirements and assessment methodologies for their vessel.

The International Maritime Organization (IMO) has recently published a circular outlining guidelines for designers to aid in awareness and reduction of commercial ship underwater radiated noise, which should be made available to all member governments for circulation (IMO 2013). This also mentions some of the measures discussed below.

The following sections will discuss the multitude of options available for reducing and mitigating ship noise from the many sources identified in previously. These will cover methods suitable both for new vessel designs, as well as retro-fit options for in-service vessels. Some attention will also be given to different available budgets, and to measures specifically intended for use on commercial vessels. The three sections will cover aspects of reduction through design, installations and equipment, and operations. Most of the attention will be directed towards reduction of underwater noise, however in some cases, the measures will also improve on-board noise levels, and these added benefits will also be highlighted.

3.9.1 Reduction through Design

The greatest improvements in the noise characteristics of a ship can be achieved when onboard and underwater acoustics become a contractual requirement which is addressed from the outset of a new design. It is much more complicated and costly, and in some cases simply impossible, to try and make the required design stages once the vessel is in service, or even when it reaches the latter stages of the build. Most of the noise reduction and mitigation measures discussed in this section are only applicable for new build vessels, however in a few cases, they can also be applied to existing vessels, and these will be highlighted as such.

Some types of vessel, namely Military, Research and Cruise Vessel have already made significant advances in the design of low-noise vessel. The ways in which these have been achieved vary depending on the size of the vessels, and the reasons for which low noise is an important quality, however there are measures taken from these vessel types which are discussed below, and which indicate what could be achieved on other commercial vessels where low noise is considered as a contractual requirement. This section will have three main sections, covering the design of the underwater aspects of the vessel, the structure of the vessel, and the on-board systems.

Propeller and Hullform Design

At higher operational speeds, the noise from cavitation and the propulsion system tends to dominate the underwater radiated noise signature of commercial vessels. For this reason, much of the design focus on the underwater components is directed at reducing cavitation and increasing the cavitation inception speed.

Looking first at the hullform and fluid inflow into the propeller, it is widely accepted that a more uniform inflow into the propeller can increase propulsive efficiency, and reduce the occurrence of conditions which leads to cavities forming. During the design of a new vessel, special attention can be paid to the hullform in the stern section, to ensure smoother and more uniform transitions in shape, for better inflow. Since the introduction of the IMO's EEDI (Energy Efficiency Design Index) in 2013, which benchmarks the energy efficiency of new ship designs against a reference line, aspects such as improving the energy efficiency through hullform optimisation will become increasingly important. In some cases changes to the hullform can also have the added benefit of reducing noise induced by the flow of fluid over the hull, and may also reduce hull resistance and hence improve fuel efficiency of the vessel. Such investigations into the stern flow field may illustrate the optimal location for the propeller in terms of inflow however other aspects of propeller performance should also be considered when the locations are selected. The impact on propeller inflow from other appendages such as struts, skegs, rudders and stabilizers should also be carefully considered, as well as their own cavitation properties. While there may not be significant cost implications in such hullform design developments for a new design, such changes are not a viable option for in-service vessels.

Now focussing on the propeller itself, a wealth of work on designing propellers for improved efficiency and reduced cavitation has been carried out, and is available in the public domain. Only a few key aspects will therefore be presented here. The interested reader should refer to other publications on the subject for comprehensive discussions and data; references used in this work include (Renilson Marine Consulting Pty Ltd 2009), (Vesting & Bensow 2011), (Kruger et al. 2012), (Spence et al. 2007) and (Pereira et al. 2004). Below are a few specific design changes which have the potential to reduce propeller and cavitation noise:

- Increase propeller blade area
- Change propeller blade skew
- Vary propeller blade section thicknesses
- Change the pitch-diameter ratio and radial distributions of this
- Unload the propeller blade tips
- "De-tune" the propeller away from resonant frequencies to prevent "singing", though thickness, material, blade area or mean pitch ratio or speed changes
- Reduce propeller loading with two rather than one propeller, a larger diameter and /or lower RPM
- Increase the number of propeller blades
- Increase CIS through blade design development
- Compare acoustic advantages of single-screw systems with smoother wake fields, and twin-screw systems with reduced tip speed and propeller loading
- Compare acoustic properties of inwards-turning and outwards-turning twin-screw systems
- Consider moving the propeller to a more optimal location for wake inflow characteristics
- Investigate potential improvements from use of composite propellers, and foul-release coatings and other treatments
- In more extreme cases, the designer may wish to consider alternative propulsors, such as waterjets for high-speed vessels (Kallman & Li 2001)

Changing propeller RPM can prove very effective in altering the noise and vibration properties directly associated with the propeller, however this measure should be applied with caution. As discussed by Zoet et al. (2012), it is imperative that the structural response of the ship is well understood before such changes are made, otherwise there is a danger that the new frequency could coincide with a natural frequency of the aft structure of the vessel and hence greatly increase vibration and noise.

Reductions in cavitation noise and the impacts of cavitation and unsteady flow on the stern part of the ship and appendages may also reduce the amount of structure borne noise which is propagated from outside the vessel to on-board locations. Some of the above design changes to the propeller may also be applicable to in-service vessels, as a new propeller can easily be fitted during a routine dry-docking, providing that supporting propulsion systems do not also require alterations. It should be noted that the presence of cavitation may also impact the more tonal propeller noises, so some attention should also be paid to these cases (National Research Council 2003). As shown above, ship noise signatures as well as cavitation performance and efficiency may well benefit from a short investigation into inflow into the propeller, and potentially minor design changes, such as the addition of a bottom plate.

Another important measure is in fact to use more accurate techniques to ensure that the delivered propeller matches the requirements of the design, which has been optimised so carefully. These techniques involve modern optical or laser technology. QinetiQ are currently looking at a possible improvement to the current ISO 484-2:1981 (Shipbuilding -- Ship screw propellers -- Manufacturing tolerances) S-Class standard for propeller manufacturing accuracy, as it has been demonstrated through analysis that industry is already capable of this.

Rudders are also prone to cavitation, especially when there are very turbulent wakes from propellers etc (Brehm et al. 2011). Another aspect which should be considered for cavitation and general noise problems is the design of the any transverse thruster propellers, such as bow thrusters as discussed by Noise Control Engineering (n.d.) and Fischer (2000). The conditions in which these are typically required to operate tend not to be conducive to low cavitation performance, for example in relation to the tunnel entrance shapes, need to reverse propeller direction, and propeller clearance in the tunnel.

One key issue of hull, propeller and rudder design is that they are typically designed for only one "design condition", which often corresponds to a full loaded service speed voyage in calm water. In reality, these conditions are very rarely met during normal service. It would seem prudent to investigate propeller performance and noise at a variety of conditions, which would more realistically reflect reality, for example modelling in a seaway, and investigating at a "slow steaming" speed or different loading condition. Of particular concern might be a ballast loading condition, as this significantly changes the conditions in which the propeller is operating. It is at these "off design" condition where the propeller is most likely to make the most noise and be least efficient, therefore such a step could also benefit fuel consumption and other areas.

Structural Design

Significant changes to the way in which sound and vibration from noise sources on-board the vessel are propagated through the structure and into the water, can be achieved through careful structural design or "tuning" of the vessel (Zoet et al. 2011). This is where resonant responses of the structure are eliminated, through identifying the key excitation frequencies of tonal machinery sources, and designing a structure whose natural frequency does not coincide with these frequency ranges. This can become a complex exercise for vessels requiring large numbers of installed potential excitation sources, and is only possible for new designs where this is addressed as a requirement from the outset. Work carried as part of the SILENV project has also suggested that transmission losses can be increased though the use of non-uniform stiffener spacing on structural panels, and potentially through the use of composite materials for some parts of the structure (SILENV Consortium

2012b). These measures may also benefit the design from reduced design if properly applied. Again, these are not changes that can be made to existing vessels, and their suitability for new designs is subject to other vessel criteria and constraints. It should also be noted that a sub-optimal structural design for acoustics may in fact increase noise and vibration propagation as was warned by Zoet et al. (2012), and could also increase structural weight.

Onboard Design

Most of the reductions made in on-board machinery and systems is through selections of installations and reduction treatments, which will be discussed in the next section. However there are several ways in which the designer may be able to reduce the underwater radiated noise; it should be noted that these measures are unlikely to have much impact in a general sense on the onboard noise levels. Where possible due to other requirements and restrictions, locating machinery spaces and installations which are likely to cause high levels of noise and vibration towards the centre of the vessel, above the waterline, away from the hull plating will increase transmission losses and hence reduce the amount of noise that is propagated to the water through direct and structure-borne paths. For topside equipment, and external machinery and systems, radiation of sound to the water can only be achieved at angles between 0 and 30 degrees without reflection. Therefore, location of these items where this is not possible will have an impact, albeit a small one, on underwater noise levels. However, as stated before, other requirements of the vessel operations should also be duly considered, and the relative advantages of each requirement properly assessed, to ensure the operation of the vessel is not compromised for a small gain in acoustic performance.

At later stages of the design, where the chosen machinery and systems noise might be particularly high and difficult to reduce directly, but their selection has been driven by external factors, it can be possible to reduce the radiated on-board noise through careful engine room layout and system placement within the available space, both of the compartment itself and its general placement within the whole vessel.

3.9.2 Reduction through Installations and Equipment

Reducing and mitigating noise through equipment and installations can be applied to both new design and to some extent on in-services vessel as well, and can tackle reduction in both the sources and the propagation of noise.

This section will discuss some of the low noise equipment, unconventional designs and propagation reduction measures available to the designer, which have the potential to have significant impacts on the underwater radiated noise spectra of ships and in particular commercial vessels. However, as noted in Plumb & Kendrick (1981), the energies involved in machinery creating noise and vibration are very low, only a matter of a few watts of sound power in comparison to the overall power, mean that it is much less likely to make economic sense to focus on the reduction of the sources of noise for machinery items.

On-board Machinery

At vessel speeds below the cavitation inception speed (CIS), machinery noise will typically dominate the underwater radiated noise profile of a vessel. They also typically have the majority of their acoustic power in the low frequency ranges, up to around 500Hz, which can propagate over large ocean distances with little attenuation.

Looking first at the key sources of noise, the main propulsors on a vessel will typically have the highest emitted noise levels as was found in the study by Jun & Dan (2003). Diesel engines and gearing systems, which are the prime movers in the majority of large commercial vessels are known to be the noisiest option, however quieter diesel-electric systems, or even steam turbines can become a viable substitute. This is not a change which is suitable for application to in-service vessels, however these options should be considered in new designs. Diesel-electric systems, when applied to new designs, tend to have inherently quieter electric motors with lower vibration levels, and the smaller diesel generators are easier to isolate, using the methods outlined later. Steam turbines are well known for being a quieter option where the design requirements allow for these, and they are particularly notable in not displaying the tonal frequency patterns in underwater noise spectra which are typically associated with diesel systems.

Other items of machinery known to radiate significant levels of noise and vibration will also likely have quieter alternatives, albeit at a potentially higher cost. Where quieter machinery variations are not a suitable option due to other criteria and requirements, the design should consider the specification of higher quality components, with high quality materials, machining and manufacturing processes and finishes, as these can lower noises such as gear meshing, "piston slap", and general vibration levels. Although the initial costs will be higher, they could also have the added benefit of requiring less maintenance, improved efficiency, and of having a longer life-span. These noise reductions have been tried and tested, and are known to reduce noise, however require higher initial expenditure. In general, rotating machinery with continuous loading is quieter than reciprocating machinery items under impulsive loads.

In order to address the effects of noise sources, and limit noise and vibration propagation through the ship, there are a couple of transmission paths to consider. The first structure-borne paths are the most efficient, and hence require the most attention in terms of underwater noise concerns. When considering on-board noise levels, the other paths will also become much more significant, especially airborne and in-pipe fluid-borne noise. A popular method for addressing this path in prime movers and other noisy and highly vibrating machinery such as auxiliaries and gear boxes is to use resilient mountings to isolate the machinery from direct contact with ship structure. These can also benefit levels of on-board noise, particularly in machinery compartments and adjacent spaces. These can be applied in single or double stages, are made of a variety of absorptive materials and designs, and can be either of a passive or active type. The study presented by Zheng et al. (2001) aims to investigate the impacts of resilient mountings on propagation. Unfortunately, very large slow speed diesel engines, which tend to have the highest levels of low frequency noise and vibrations, are much too larger and heavy to be used together with resilient mounting systems. These systems can also take up precious machinery room space, and will add some weight to the vessel. An example of a resilient mount system can be seen in the figure below:

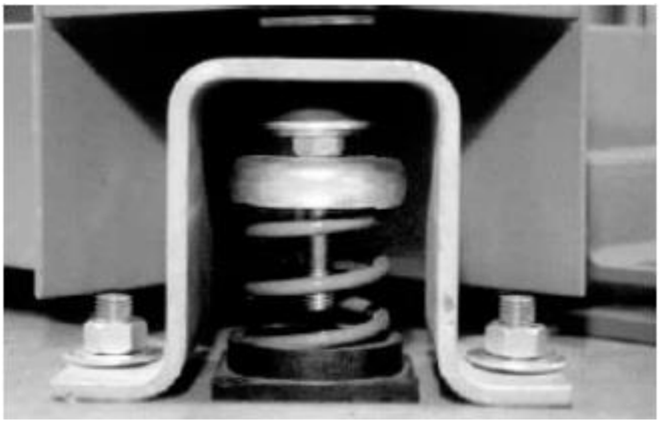


Figure 3.33 - Resilient Mounting for Machinery (Spence et al. 2007)

Table 5 in Collier (1997) gives estimated frequency band centre values for transmission losses achievable with different levels of machinery mounting, which is summarised in the points below:

- With hard or direct mounting, transmission losses of up to 13 dB can be achieved for lighter machinery items in the low frequency ranges
- With distributed isolation material, transmission losses of up to 15 dB can be achieved for lighter machinery items in the high frequency range
- With low frequency isolation mounting, transmission losses of up to 30dB can be achieved for most machinery items in the mid frequency range
- With two-stage mounting systems, transmission losses of up to 50 dB for most machinery items in the high frequency range

However as noted above, the more efficient systems will incur increased costs in terms of installation and materials, added weight, and loss of space. Seating design should be carried out very carefully, to ensure the best solution is achieved. Where this is appropriate, even the optimum connection points can be found, to coincide with vibration nodes or low response points, for minimal vibration transmission. Care should also be taken in considering the system as a whole, to avoid "noise shorts" where interaction occurs between resilient mountings and rigid connections and related issues. As noted by Plumb & Kendrick (1981), the energies involved in generating machinery noise and vibration tend to be very minimal compared to the overall power, therefore in some cases, more efficient reductions can be achieved from isolation rather than reduction systems.

Machinery manufacturers are also now making more active mounting systems available which are known as Active Noise Control (ANC) systems, where shakers of similar output but 180^{o} out of phase can significantly reduce the radiated noise and vibration. These systems also have advantages in lower weight and volume than a similarly performing passive system. In some cases, reductions of around 15dB at the mountings and 5dB at the hull can be achieved in the lower frequency ranges, where the main engine tonals are generally observed (Salm et al. 2013). These can be very effective where the system as a whole has been considered and its noise and vibration path properties taken into account.

More complete isolation of main propulsors can also be achieved by using electrical transmission systems which removes gearbox noise altogether, and removes continuous shaft lines.

In order to tackle the airborne and secondary structure-borne paths, some form of damping typically needs to be applied to surrounding bulkheads, deck-heads and other structure. The materials used for these purposes will be discussed later. However, other methods are also available, such as floating floor installations for more complete isolation. These can be very effective in reducing noise and vibration propagation, however will take up a significant amount of space, and also have a notable weight penalty (Noise Control Engineering n.d.).

Some items of machinery also require additional noise reduction to be applied, for instance silencers on exhaust stacks. Large marine engine and machinery companies, such as Wärtsilä, are now starting to offer "Low Noise Packages" which include coverings for engines, cylinder heads, turbochargers etc, to limit the amount of noise that can be transmitted to the surroundings. These can again be very effective solutions, however issues such as access for maintenance, and added weight and cost should always be considered with such measures. An example of the Wärtsilä Low Noise Package cylinder head cover is presented in the figure below:

Figure 3.34 - Wärtsilä Cylinder Head Cover (Aura et al. 2010)

Noise source mapping of the main engine, using either measurements from similar systems or data from the manufacturers can then highlight the relative noise sources within the main engine itself, and therefore the areas for focus and noise reduction can clearly be identified. Combustion noise is one of the sources of engine noise (Rakopoulos et al. 2011); this is known to be very sensitive to both the engine load but also the timing of the fuel injection. Engine load can be difficult to alter, however a small change in injection timing can change the Sound Pressure Level (SPL) of the engine, therefore at key frequencies, it could be used to reduce the noise level by several dB (Win et al. 2005).

Propellers and Outboard Equipment

Aside from the genetic propeller design variations discussed in the previous section, there are also a number of specific design for propellers which have been proposed, whose designers claim can reduce some types of cavitation or noise, or improved efficiency. These include:

- Forward-skew propellers, which are briefly discussed in Spence et al. (2007), and which are claimed to be less susceptible to changes in inflow and vessel speed . They need to be installed in a tunnel or nozzle, such as a Kurt nozzle, to reduce fouling and potential for physical damage to wildlife, however these can help to increase thrust
- Highly-skewed propellers, which are said to reduce blade tip excitation (Renilson Marine Consulting Pty Ltd 2009)
- Over-Lapping Propeller Systems (OLP), (Ebira et al. 2007) which it is claimed are capable of high propulsive performance in cases where there is significant wake away from the single screw propeller plane
- Contracted and Loaded Tip (CLT) propellers, which are claimed to reduce the occurrence of top vortices and hence allow for small propeller diameters, as the blades can be more highly loaded, if carefully designed. These are discussed by Renilson Marine Consulting Pty Ltd (2009), as well as other designs with blade-end plates
- Kappel Propellers, which are said to potentially reduce cavitation and increase efficiency (Renilson Marine Consulting Pty Ltd 2009)

It should be noted that many of the above suggestions are in the concept stage only, and most have not been fully tested for commercial applications. For most, it is also not known whether the claims of reduced cavitation, noise and improved efficiency are justified, and what impact this might have on underwater noise, therefore further study is required.

Below can be seen an example of the Overlapping Propeller System proposed by Kawasaki (Ebira et al. 2007):

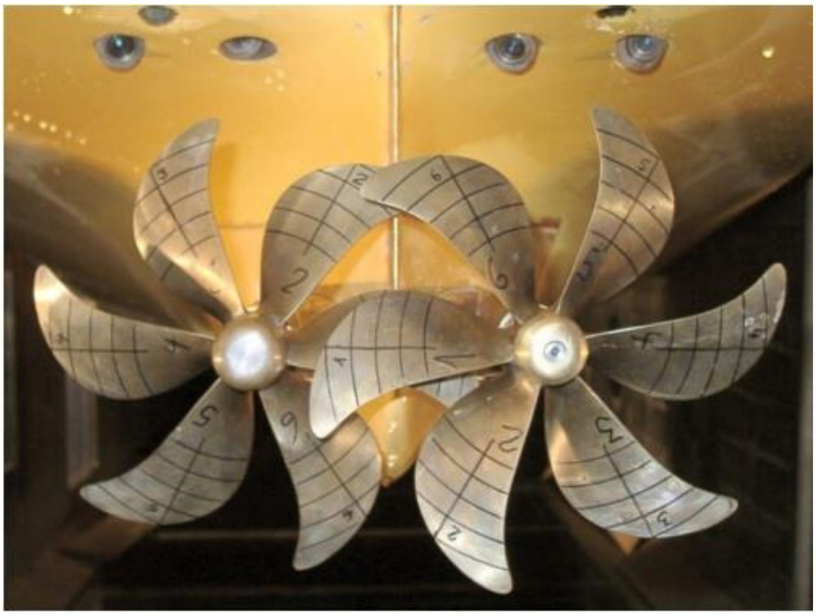


Figure 3.35 - Kawasaki Overlapping Propeller System (Ebira et al. 2007)

As well as the propeller blades, several alternative designs have also been proposed for the propeller hub, in an attempt to improve cavitation performance. These designs include propeller fins, propeller hub cap shape variations, boss cap fins and propeller cap turbine: (Renilson Marine Consulting Pty Ltd 2009), (Southall & Scholik-Schlomer 2007), (Druckenbrod et al. 2012). As above, further study is required to ascertain the ability of these designs to reduce underwater radiated noise from marine propellers. In some cases, such as for ice-strengthened propellers, additional considerations will be required in order to fully assess the suitability of the propellers.

As with machinery components, it is again suggested that use of high-quality components, and better casting, machining and finishing techniques will also improve performance and can reduce the need for maintenance, although at a higher initial cost.

As well as tackling the propellers themselves, there are also a range of equipment suggested for use in improving inflow into the propeller by Renilson Marine Consulting Pty Ltd (2009), which include:

- Vortex Generators
- Schneekluth Duct
- Mewis Duct
- Simplified Compensative Nozzles (SCN)
- Grothues Spoilers

If shown to be effective in improving inflow and reducing noise, many of these adaptations could be easily and relatively cheaply achieved, making them a good solution for in-service vessels. Masking systems which use the addition of bubble curtains can also be applied, and the additional of bubbles for the propeller itself can potentially be used to reduce on-board and higher-frequency noise.

Below is a figure of a wake equalizing duct applied to a vessel:

Figure 3.36 - Wake Equalizing Duct Installed on Vessel (Shipbuilding Tribune 2011)

Aside from conventional single- or twin-screw propulsions systems, comparatively little is known about the radiated noise of different propulsions systems such as podded propulsion, waterjets and Voith Schneider propulsion, and their relative suitability to commercial vessel applications. Further study is recommended in this area. Further research will also be required for establishing the general noise characteristics of the vessels more typically making use of these propulsion sources such as medium and high-speed craft.

Some have suggested that improvements in propeller-rudder interaction can improve noise characteristics, as rudders are also prone to cavitation, and designs such as the Costa Propulsion Bulb, twisted rudders and rudder fins have been proposed as relatively simple solutions to assist in this problem (Renilson et al. 2013).

In some marine applications, mostly offshore pile-driving activities, bubble curtains are used as a barrier between the noise source and the surrounding of the ocean, to try and limit the amount of noise which is propagated (Spence et al. 2007). It is possible that an adaptation of this technology, where a layer of bubbles is pumped round the outer hull of the vessel, could be adapted for ship noise applications. However, it is suggested by André et al. (2009) that such systems may not be effective in the frequency ranges of most importance. There are also many aspects to consider, such as efficiency, cost, and even how this would work is reality. Nevertheless further study should also be carried out for less conventional mitigation methods such as this.

Materials

There are a number of materials and material systems being developed for more efficient noise and vibration attenuation. These can take the form of both passive absorptive materials, and more active attenuation systems.

Looking first at passive materials, typical choices include composites such as M-P-M (an aluminium-plastic sandwich) and others, and details of these can be gained from manufacturers. These can be used in a variety of ways, including lining of bulkheads and deck-heads in machinery spaces and other noisy compartments, wrapping around piping, and as covers for high noise level machinery and installations. They do not impact in the noise source as such they simply aim to reduce the amount of noise and vibration it can propagate to the surrounding structure-borne and airborne paths. They can be a simple solution which requires no significant design and operational changes, however attention should be paid to the amount of space they make require, as well as the associated additional weight and cost.

Now focussing on the more active systems, systems such as the growing number of SMART materials, which have one or more properties that can be significantly changed in a controlled manner by external stimuli, such as stress, temperature, moisture, pH, electric or magnetic fields. It is suggested by Turkmen (2014) and (2012) that such materials could be developed as active resilient mounts, as their shape and other properties could be altered with the addition of a current or other external stimuli to suit the noise and vibration source in question. They could also be applied as damping material to the structure of machinery spaces and other noisy compartments, and their properties again varied to best suit the sources present at a given operational condition or speed. There would however be a significant cost in monetary, weight and space terms.

Sea-Connected Systems

Another potential source of underwater noise is from sea-connected systems, as these have a direct transmission path into the water (Urick 1983). Such systems, typically sea chests, and sea water cooling piping and pumps, cannot be avoided as they are necessary to the design and operation of typical commercial vessels. Therefore, steps must be taken to limit the amount of noise they can radiate. This is usually achieved through the use of flexible pipe connections, and through introducing dog-leg piping configurations where possible and appropriate. This can be achieved with little cost implications or complexity in both existing vessels and new designs, however impacts on overall noise levels are likely to be limited. Pulsation damper systems, or acoustical absorbers, either in-line or parallel, can be "tuned" to address a particular problematic frequency in such areas as well (Salm et al. 2013).

In some case, quieter machinery items such as low-noise pumps or impellers with some of the treatments discussed in the propeller section can be used, however the additional costs of these items should be weighed against the potential advantages in terms of overall noise characteristics.

3.9.3 Reduction through Operation

Reduction of on-board and underwater ship noise through operational changes can be achieved for both new designs and existing in-service vessels. These methods are discussed below, in relation to day-to-day on-board operations, and to changes over the voyage as a whole.

On-board Operations

The key to achieving reductions in underwater and on-board noise levels, and also potentially increasing fuel efficiency, is care and organization. Regular and thorough maintenance schedules for machinery can lead to small noise reductions, as the systems operate more smoothly. This reduces the incidence and amplitudes of tonals from sources such as gear meshing noises and "piston slap" in reciprocating machinery. Proper maintenance of damping materials to reduce wear is important, as well as ensuring work such as painting does not detrimentally affect its operation.

Continual condition monitoring of onboard and outboard systems is also an effective way of identifying when the noise and vibration of a particular noise source has increased or changed. There are a wide variety of systems available for such monitoring, however newer concepts such as those using fibre optic technology offer great potential. These lightweight systems can deliver a large volume of data for a very small unit size. If designed into the systems and propellers from the outset, their use could outweigh any additional manufacturing costs, as well as ensuring the units were suitably protected. This is particularly important for fitting to a propeller, as the glass equipment may not be able to withstand the pressures of being fitted to the blades, however if fitted inside the blade, they require very little removal of the parent material. Monitoring systems can also be applied to vessels to track crack propagation in structural elements. The appearance and increase in these could alter the vibration and propagation characteristics of the structure, so should again be taken into account. Where acoustic-based systems are used for crack or machinery condition monitoring, they can also provide data on through-life ship operational profiles and dynamic loading.

Crew should also be fully trained and briefed in the workings of the noise mitigation equipment, to ensure a stray spanner etc does not affect their performance. Crew training and awareness can also assist in achieving improvements; it was discussed by Banks et al. (2009) that this could be achieved for fuel efficient operation, so should also be applicable to noise considerations. For the hull and propeller, regular dry-docking, monitoring and repair of propeller blade damage (SILENV Consortium 2012a), and the use of suitable anti-fouling and foul release coatings can reduce flow noise and increase efficiency. There may be increased costs associated with these methods however they are likely to be outweighed by the reductions in fuel usage and major repair costs.

A good knowledge of the design conditions of the vessel and how these can be approximated in reality can also lead to reduced noise. Where a vessel is fitted with a Controllable Pitch Propeller (CPP), establishing the optimal combination of pitch and RPM for different vessel operating and loading conditions, and operating at these, can assist in the aim towards lower radiated noise (Spence et al. 2007). The same could also be applied to stabilizers etc. Thrusters used during manoeuvring exercises are also known to be significant sources of noise, and so finding ways in which their use can be limited will also make a difference.

Voyage Changes

Any major changes made to how voyages are conducted have to be carefully considered and weighed up to ensure that they don't have a detrimental effect on customer requirements, economic viability of the voyage, ship efficiency and emissions, and a number of other aspects. One suggested solution for reducing noise is simply to reduce ship speed. This will decrease both the underwater and propulsion noise, and also the speed-dependant machinery noise. However, this can have a number of disadvantages, in terms of longer voyage time requiring increased loaded consumables, less economic transport efficiency with fewer voyages per year, and operation of the vessel at off-design conditions, among others. However, even a modest reduction in ship speed can improve the acoustic characteristics of the vessel, and where vessel commonly spend significant amounts of time waiting outside ports, this could become a viable option. It could also become an option to only reduce vessel speed in particularly sensitive areas, such as those identified below.

Another possible change is to vary typical routes to avoid particular areas, such as those of biological importance to sensitive marine animal species: migration routes, feeding grounds and breeding areas. This could be a simple solution in cases where the alternative route is not significantly further than the original. Changes of this kind are already being enforced by local and national authorities in some areas, such as the area around the Port of Boston on the east coast of the USA, where major shipping lanes have been moved to avoid transiting through the Gerry E. Studds Stellwagen Banks National Marine Sanctuary, and to reduce the general ensonification of this area (Ecklund 2008). It is likely that in the near future, more requirements of this kind will be enforced on ship operators through a variety of legal instruments. Alterations to voyages for other reasons, for example as a consequence of weather routing, to avoid particularly challenging sea conditions for improved fuel efficiency, may also improve acoustic performance. The reduced propeller loading will reduce both propulsions and propulsor machinery noise. The same could also be said of route alterations to avoid areas of surface sea ice. In the coming years, systems for avoiding excessive ensonification of particular ocean regions though monitoring of vessel transits in the area may come into play, possibly making use the ship AIS (Automatic Identification System) tracking. Vessels may be forced to change their routes according to requirements outside their influence.

Very little study has been carried out into the small reductions or spectral changes which might be achievable through changes in operational conditions such as trim and draft. If these were found to have an influence, even in a small way, they could be useful tools for operators to apply where conditions and operations would allow.

3.10 Research Gaps

This chapter has provided an overview of current research and knowledge on ship radiated underwater noise and related subjects. The diversity of the topics covered has highlighted how extensive the research has been, and should continue to be, in order to fully address underwater noise. It has also demonstrated how underwater noise should not be considered as an issue in isolation. The chapter has identified commonly used methods and areas of promising ongoing research, as well as some gaps in knowledge which are yet to be investigated.

The majority of these gaps in the knowledge lie in the availability of a range of prediction models for different purposes and users. There tend to be a lot of research into empirical models which can provide good initial approximations for ship noise sources and propagation, and into very complex numerical methods which have been identified as being complex and computationally demanding.

There appears to be a need for more research into proposing, modifying or automating methods that can be used by designers and operators, which will provide a reasonable level of accuracy without the significant computational demands. It is felt that the availability of these approaches would make it easier to take ship and shipping noise into account and may encourage this, while regulations are not yet available.

Another notable gap in the knowledge is in the understanding of the long- and short-term impacts of ship and shipping noise on marine wildlife. While significant work has already been done in this area, there is still much to address. It is felt that this research could benefit from being carried out alongside studies into noise reduction and regulation, as a greater understanding on both sides would help in creating eventual guidelines and regulations which may be proven to be beneficial. Further measurement, gathering and publication of data on both marine wildlife and ships in relation to sound and noise would be of great help as availability is limited in the public domain.

The range of noise reduction and mitigation measures discussed is significant, however as identified in the paper, many are in their infancy in terms of design, application and even understanding of the efficacy. Further research into these and other measures, the applications they might be suitable for, costs and other implications, and their effectiveness would be extremely beneficial for members of the maritime industry interested in making changes. It is also important in the current shipping industry that these measures are not detrimental to other aspects already being regulated such as emissions and ballast water.

A final factor to note is an apparent lack of awareness of the issues and potential implications of ship and shipping noise in the wider maritime community. This can easily be addressed through education and involvement. Greater understanding and appreciation could lead to more involvement by designers and owners, as well as researchers, in addressing this issue and taking steps in their own work to make a difference. As stated in the chapter, if considered from the very beginning of a design, improved acoustic performance does not have to be a burden on the designer and owner.

3.11 Chapter Summary

This chapter has reviewed the literature currently available on ship radiated underwater noise, as well as related and relevant topics. It has also identified some important gaps in current research and state of the art, some of which will be addressed within the scope of this project.

The next chapter will overview the approaches to be adopted within this project, in order to achieve the outlined aims and objectives.

Chapter 4 - Approaches Adopted

4.1 Chapter Overview

This chapter will discuss the various approaches to be adopted in different parts of this study. The critical review which was carried out is introduced (§4.2) initially. The approach for propeller and hydrodynamic noise prediction and propagation is then outlined (§4.3). Further, the approach taken for the prediction of machinery noise is discussed (§4.4). The methods used for validation and implementation are then overviewed (§4.5), followed by a brief introduction to the assessment approach for impact on marine wildlife in the study (§4.6). The consequent discussions are outlined (§4.7), and finally the software used is introduced (§4.8). A flow chart presenting an overview of the main stages to be discussed later is shown below:

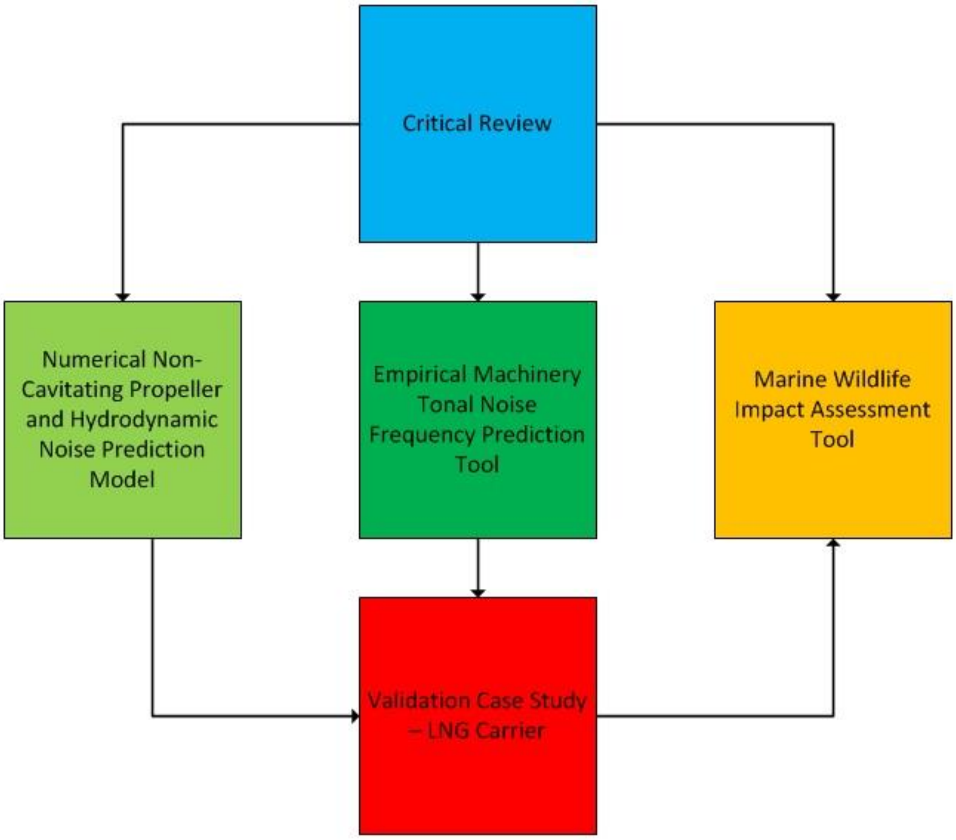


Figure 4.1 - Thesis Overview Flow Chart

4.2 Critical Review

An in-depth and wide-ranging critical review was carried out as a starting point for this study. The scope included the many areas of research relating to commercial ship radiated underwater noise, namely: typical ship noise sources and signatures, ambient underwater noise, noise prediction and propagation modelling approaches, effects of underwater noise on marine wildlife, noise regulation and finally reduction options.

This was done to gain a good grounding of the subject and the key areas of research. The findings then informed the selection of the prediction approaches which were applied and developed within the study. It also determined the information used for assessment of the impact of ship noise on marine wildlife, and of a vessel's underwater radiated noise characteristics in general.

4.3 Propeller and Hydrodynamic Noise Prediction

Using the findings from the critical review outlined above, a numerical approach was found to be the most appropriate and suitable for the prediction of the propeller and hydrodynamic noise. The approach used an Unsteady Reynolds-Averaged Navier-Stokes (URANS) solver within commercially available Computational Fluid Dynamics (CFD) software to predict the distribution of pressure around the hull, propeller(s) and rudder(s) of the vessel. A built-in noise propagation approach based on Farassat Formulation 1A of the Ffowcs-Williams Hawkings equation then predicts the vessel noise spectra at a given receiver location for a specified time period and frequency range.

This approach was selected for several reasons. Firstly, the approach requires the user to apply only simple hull and propeller outline geometry, which is thought to be easily accessible within the early design stages. Where the hull geometry is not available but the propeller characteristics wake velocities are, these could be used as inputs for the domain inlet, and simulation could then be carried out for the propeller operating in this approximated field. Furthermore, only simple information such as vessel and propeller speeds is required in order to carry out the simulations. A similar approach utilising the Ffowcs-Williams Hawkings approach has previously been applied by researchers for similar problems (Ianniello 2012), and has been found to be successful. The particular application of the approach in a coordinated manner within only one dedicated software programme is felt to be advantageous as it may be more representative of what would be available to a designer, and minimises the need for expert development. If applied in a suitable way, the approach offers a compromise between higher accuracy and more detailed results than an empirical approach could provide, whilst retaining the option to reduce computational complexity, demand, cost and time through simulation variations.

4.4 Machinery Noise Prediction

Due to the complexities and large amounts of information involved in fully predicting the machinery noise contributions to underwater noise, it was decided that the predictions for the machinery noise aspect of this study would be limited to tonal frequencies only. This is thought to be reasonable, as it is known that in general machinery noise contributes only narrow tonal peaks to ship underwater noise spectra, and would therefore have limited impact on the broadband level as a whole. There are a number of empirical formulae available in the literature for frequency prediction, which have been be used to calculate the first four harmonics for key machinery items, namely: the main propulsor, reduction gears, auxiliary equipment, generator(s), propeller shaft and propeller blades. This has been implemented in a machinery noise tonal prediction tool using simple data inputs and formulae, which also displays predicted frequencies over broadband ship noise spectra.

4.5 Validation and Testing

In order to validate and test the noise prediction, a case study of an LNG carrier has been used. The noise prediction approaches have been directly applied to the geometry of the vessel in order to assess their accuracy, applicability and suitability. Predicted underwater noise spectra results for the vessel, using several simulations variations, are presented and discussed. For validation purposes, they are compared with field measurement data.

4.6 Impact Assessment

Several approaches have been used for assessing the potential impact of a ship radiated underwater noise on the marine environment and wildlife. Firstly, current non-mandatory noise limits and thresholds from various sources are compared with either measured or predicted ship noise spectra, for a general indication of performance and possible areas for concern, and could also be applied in the future in the case of mandatory rules-based regulation of underwater noise. Secondly, an extensive database of marine wildlife has been collated, containing information on habitat by area, conservation status, hearing and vocalisation ranges, and examples of reactions to ship underwater noise sources. This database can be filtered for a specific area and highlighted by conservation status to assist in identifying the most at-risk species in an intended operational area. Thirdly, the database can also be highlighted using a frequency range of interest, and the hearing and vocalisation ranges of the species, to identify those most likely to be affected.

4.7 Discussions

Following the development and application of the approaches outlined above, they have been discussed in detail, with the main areas of focus being the successes of the approaches as tools for designers to use in early design stages, areas for further development within the approaches, research gaps and topics for further study.

4.8 Software

The CFD software used in this research was the commercially available StarCCM+ Versions 6.04.014 and 7.02.011 from CD-Adapco, released in 2011 / 2012. These were run in parallel mode on the University of Strathclyde's High Performance

Computer (HPC) facility to enable more complex and demanding simulations to be solved in a much shorter time.

The main body of the impact assessment and data analysis tool developed for machinery noise prediction and impact assessment was created in Microsoft Office's Excel 2007. This was chosen as the most appropriate medium as this software is widely used and therefore widely available. Most designers who are likely to use the model should be familiar to a suitable degree with the program. However the methodologies used within the tool are easily transferrable to other programs.

4.9 Chapter Summary

This chapter has introduced the various approaches which have been adopted for the prediction and assessment of ship radiated underwater noise in this study.

The next chapter will discuss in detail the numerical model for the prediction of propeller and hydrodynamic noise which has been developed. It will also outline the empirical approaches used for machinery noise prediction, and the data analysis tool which has been developed.

Chapter 5 - Numerical Modelling of Ship Radiated Noise

5.1 Chapter Overview

This chapter will discuss the details for the numerical modelling of propeller and hydrodynamic noise modelling developed in this study. The acoustic modelling approach will be outlined (§5.2), followed by details of the CFD simulation set-up requirements (§5.3). The ideal case simulation will then be discussed along with details of the geometry and data requirements (§5.4). Following this, the likely problems will be identified and discussed (§5.5). The chapter will also discuss the modelling of machinery noise, and the empirical approaches adopted in this study within a data analysis tool (§5.6), and for machinery tonal prediction (§5.7).

5.2 Acoustic Modelling

As outlined in the previous chapter, the acoustic modelling which forms the core of this study will be based on an application of the Ffowcs-Williams Hawkings equation, through Farassat Formulation 1A. The sections below will provide details of the model and governing equations, followed by particulars of the application of the model in numerical methods generally, and in the CFD Software StarCCM+ in particular.

5.2.1 Acoustic Model Details and Governing Equations

Noise in fluids can be identified as small pressure fluctuations or changes in the flow field, caused by, in this instance, the flow of water past the hull and the rotation of the propeller within the fluid. Computational aero- or hydro-acoustic models use the equations of motion for the fluid to solve these changes in the flow field into acoustic pressure fluctuations. The approach applied in this instance is one based on the F-WH equation, and is detailed below.

Ffowcs-Williams Hawkings Equation

As was introduced in the Literature Review Chapter, the fundamental Ffowcs-Williams Hawkings (F-WH) equation presented below was proposed by Ffowcs-Williams & Hawkings (1969). The Ffowcs-Williams Hawkings equation uses generalized functions to extend the application of Lighthill's Acoustic Analogy to the aerodynamic noise generated by rotating bodies. The general equation rearranges the conservation of momentum and continuity into an inhomogeneous wave equation.

$$\overline{\Box}^2 p' = \frac{\bar{\partial}}{\partial t}[\rho_0 v_n \delta(f)] - \frac{\bar{\partial}}{\partial x_i}[l_i \delta(f)] + \frac{\bar{\partial}^2}{\partial x_i \partial x_j}\left[T_{ij} H(f)\right] \tag{5. 1}$$

Where: $\overline{\Box}^2$ *is the D'Alembert, or wave operator:*

$$\Box^2 = \left(\frac{1}{c^2}\right)\left(\frac{\partial^2}{\partial t^2}\right) - \nabla^2$$

p' *is the acoustic pressure in the undisturbed medium, in this case* $p' = p - p_0 = c^2 \rho'$

c *is the sound speed in a quiescent medium*

ρ_0 *is the density of the quiescent medium, or fluid static density*

v_n *is the local normal velocity of the source surface*

$\delta(f)$ *is the Dirac delta function*

T_{ij} *is the Lighthill stress tensor:*

$$T_{ij} = \rho u_i u_j + P_{ij} - c^2 \rho' \delta_{ij}$$

$H(f)$ *is the Heaviside Function*

As the Ffowcs-Williams Hawkings equation is valid in the whole 3D space, it is common to use the Green's Function of the wave equation, defined below, to turn the equation into an integral form, allowing it to be solved numerically. When the integration surface coincides with the body being considered, in this case the hull and propeller, the three terms on the RHS of the equation are then the thickness, loading and quadrupole source terms respectively. The thickness term relates to noise arising from fluid being displaced by the body movement and the loading term relates to the noise arising from the unsteady motion of force distributed over the body. The quadrupole term relates to noise arising within the fluid from non-linear sources such as turbulence and cavitation. Most of the Analogy is now linear, with all non-linearity being collected in the Lighthill Stress Tensor.

Green's Function is defined as:

$$G(\boldsymbol{x}, \boldsymbol{y}, t, \tau) = \frac{\delta(g)}{4\pi r} \; for - \infty < \tau \leq t$$

$$G(x, y, t, \tau) = 0 \; for \; \tau > t$$

Where: $g = \tau - t + \frac{r}{c_0}$

$r = |\boldsymbol{x}(t) - \boldsymbol{y}(\tau)|$

*(**x**, t) are the observer space-time variables*

*(**y**, τ) are the source space-time variables*

Farassat Formulation 1A

In his 1975 Technical Report for NASA Farassat (1975) proposed a method for evaluating the Ffowcs-Williams Hawkings equation through a time domain formulation that can predict the noise of an arbitrary shaped object in motion, without the need for numerical differentiation of the observer time. The formulae presented below are those implemented within the StarCCM+ software:

$$p'(\boldsymbol{x},t) = p_T'(\boldsymbol{x},t) + p_L'(\boldsymbol{x},t) \tag{5. 2}$$

$$p_T'(\boldsymbol{x},t) = \frac{1}{4\pi}\left(\int_{(f=0)}\left[\frac{\rho_{0(\dot{U}_n+U_{\dot{n}})}}{r(1-M_r)^2}\right]_{ret} dS + \int_{(f=0)}\left[\frac{\rho_0 U_n\left[r\dot{M}_r + c(M_r - M^2)\right]}{r^2(1-M_r)^3}\right]_{ret} dS\right) \tag{5. 3}$$

$$p_L'(\boldsymbol{x},t) = \frac{1}{4\pi}\left(\frac{1}{c}\int_{(f=0)}\left[\frac{\dot{L}_r}{r(1-M_r)^2}\right]_{ret} dS + \int_{(f=0)}\left[\frac{(L_r - L_M)}{r^2(1-M_r)^2}\right]_{ret} + \frac{1}{c}.\int_{(f=0)}\left[\frac{L_r\left[r\dot{M}_r + c(M_r - M^2)\right]}{r^2(1-M_r)^3}\right]_{ret} dS\right) \tag{5. 4}$$

Where: $M_i = \frac{v_i}{c}$

$U_i = v_i + \left(\frac{\rho}{\rho_0}\right)\left(\frac{u_i}{v_i}\right)$

$L_i = P_{ij}n_i + \left(\frac{\rho}{u_i}\right)\left(\frac{u_n}{v_n}\right)$

and the other variables are defined above.

Now, the numerical integration is mainly dependant on kinematic and geometric variables, with the exception of hydrodynamic loading on the body. This information can be provided by the near-field incompressible CFD prediction of the flow field around the body. It should be noted that this formulation considers only the thickness and loading terms. As the propellers for a vessel will be operating at sub-sonic speeds, the quadrupole sources are typically considered negligible, and are hence not included. This however has been disputed in work carried out at INSEAN (Ianniello 2012).

In order to account for the quadrupole sources, the use of the hull and propeller as the integral surface can be replaced by an enclosing permeable surface which is located within the flow, at a distance from the hull which would then also capture the in-flow noise sources. This approach is known as the porous formulation, and avoids the need for volume integration to take into account the non-linear sources. Both approaches have been investigated in this study.

5.2.2 Application within CFD

In numerical applications, Farassat Formulation 1A, neglecting the quadrupole sources, is most typically implemented. The aim of the F-WH solver is to predict the small scale acoustic pressure variations at the specified receiver locations. Providing that the input data from the near-field simulation is of high quality and accuracy, the noise prediction results will also be accurate. For this reason, LES (Large Eddy Simulation) or even DES (Detached Eddy Simulation) is often suggested as the most appropriate selection for the initial solver as these approaches are more capable of producing good details of the flow field aft of the propeller and in other key areas, however in order to reduce complexity, a RANS solver is used in this study. This can be a suitable compromise when coupled with suitable mesh resolution and time-step size. Acoustics are inherently transient problems, and therefore unsteady simulations are the most appropriate approach. The problem is typically approached in two steps. The first step computes the time-accurate flow and pressure field data on and around the hull and propeller surfaces, which are the emission surfaces for the F-WH solver. The second step is to compute the temporal variation of acoustic pressure at the receiver locations using the F-WH Solver.

As the noise is predicted in the time-domain, the initial acoustic results will take the form of pressure time histories at each of the receivers. Once the simulation has completed, a Fast Fourier Transform (FFT) is carried out on the data at each receiver to give the underwater radiated noise spectra for the selected time-frame. The purpose of a Fast Fourier Transform is to take the time-domain acoustic data collected during the simulation, and transform it to the frequency-domain to produce spectral data which can then be analysed. The FFT analysis used in this study applies a Hanning window which is where the Hann Function, a discrete probability mass function, is used as a window function, as is often done in signal processing, to select a subset of a series of samples to perform a Fourier Transform. In this case, the Hanning Window is a medium-to-high resolution window function and part of the "raised cosine" family of functions due to its use of cosine as seen below:

$$\omega(n) = 0.5\left(1 - cos\left(\frac{2\pi n}{N-1}\right)\right) \qquad (5.\ 5)$$

This analysis uses 4 analysis blocks with an overlap factor of 0.5, to smooth the results between blocks. These results are then compared against the field measurement data discussed in Chapter 6 (§6.3) to assess the accuracy of the simulation results, both in terms of general spectral shape and predicted Sound Pressure Levels (SPL).

5.2.3 Application within StarCCM+

An acoustic modelling module is already built in to StarCCM+, and is based on the Ffowcs-Williams Hawkings (F-WH) method and using Farassat Formulation 1A, which was introduced above. The unsteady simulation approach of the F-WH is stated in the software user guide to be the preferred approach for mid- to far-field noise prediction, and is hence the appropriate choice for this work.

The user is required to specify the uniform speed of sound and far-field fluid density values. The user then specifies the emitting surfaces, which may be the vessel's hull and propeller(s), or a permeable surface located within the fluid. The user also specifies the location of the receivers in terms of their distance and depth, and the emitting surfaces they are related to. The software does not require for these receivers to be located within the meshed region of the computational domain, allowing for predictions in the far field to be carried out.

In this software, the solver uses a "forward in time" approach, whereby it calculates when the currently generated sound from the emitting surfaces will be received at the receiver locations. The total sound pressure at the receiver is then made up of the contributions from the different emitting surface, with the spectra at the receivers then being the summation of the different contributions over a given time duration.

5.3 CFD Simulation of Non-Cavitating Propeller and Hydrodynamic Noise

The following sections will outline the details of the CFD Simulation approach developed for the purposes of non-cavitating propeller and hydrodynamic noise prediction.

5.3.1 Propeller Modelling

It is imperative that the propeller is appropriately represented within any ship underwater noise prediction simulation; it has been highlighted previously as a key source of underwater noise, both in non-cavitating and cavitating conditions. The following relates to non-cavitating conditions only; the subject of modelling cavitating is covered later, as this was not specifically carried out as part of this study.

Whilst for simulations dealing more generally with flow around a hull a momentum source or an actuator disk approach may be a suitable compromise, in the case of acoustics, the propeller geometry and an aspect of rotation are indispensable. There are two suitable options in this case: A moving frame of reference approach, or a rotating mesh with sliding interfaces. In both cases, it is required that the computation domain is split into two regions, one rotating region enclosing the propeller, and one stationary region which includes the hull, rudder and computational domain.

The moving frame of reference approach assigns a frame of reference to a static mesh region incorporating accurate propeller geometry, and the frame of reference itself moves in a purely rotational motion with respect to the global co-ordinate system. The simulation will calculate the variations in flow and pressure distributions at each time-step according to the relative location of the rotating frame of reference. A source term is added to the momentum equation, to include the added Coriolis forces which occur due to the rotation. In StarCCM+, the axis of rotation, axis origin and direction, and rate of rotation are defined by the user. This provides a steady-state approximation to a transient problem, with time-averaged results, and hence is the less accurate approach.

In the rotating mesh approach, a purely rotational motion is applied to the entire rotating region, meaning that the propeller geometry and associated mesh physically rotates. This results in a transient calculation, which will provide time-accurate results; the closest representation of reality of the two methods presented here. The interface between the rotating and stationary mesh components is known as a sliding interface, and it is important that the interface is properly defined, to allow flow properties and calculation variables to be properly propagated through the simulation domain. In StarCCM+, the axis of rotation, axis origin and axis direction, and rate of rotation are again all defined by the user. This is the preferred method and will be applied in the first instance.

5.3.2 Free Surface

In this study, in order to maintain a simulation which is faithful to reality, the free surface is fully modelled as a water-air interface, rather than by using a symmetry plane boundary condition, or neglected entirely. As calm water conditions are likely to present the worst case scenario for noise propagation into the far field, a flat free surface is simulated. This is done using a Volume of Fluid (VoF) flat wave approach with two Eulerian phases; in this case air and water, for which the physical properties are defined. In this approach, it is assumed that the two phases share the same velocity, pressure and temperature fields. Hence the same equations for momentum, mass and energy transport are solved as for a single-phase flow, using an equivalent fluid whose properties are defined by the physical properties and volume fractions of the constituent phases. The transport equation for volume fractions is then:

$$\frac{d}{dt}\int_V \alpha_i \, dV + \int_S \alpha_i \left(\boldsymbol{v} - \boldsymbol{v}_g\right) . d\boldsymbol{a} = \int_V s_{\alpha_i} dV \qquad (5.\ 6)$$

Where: α_i *is the phase volume fraction*

s_{α_i} *is the source or sink in the* i^{th} *phase*

The water level is initially uniform across the computational domain at a level defined by the user, with volume fractions of air and water being 1 above and below the water line respectively, with only the band of cells at the specified as partial volume fractions each of air and water. Ideally, this air-water interface should be one line of cells wide. The speed of the wave past the hull represents the speed of the vessel through water and air, and is again specified by the user. The surface then develops over the simulation, adjusting to mimic real-life interaction with the hull surface, creating a bow wave and wake features. The interaction of the air and water is dictated within the software by a Multiphase Mixture model. Gravity is applied to the simulation domain to ensure that the free surface acts in a realistic manner. Numerical damping can also be applied to the VoF wave however this is a purely numerical tool and has no physical meaning, and hence should be applied with caution.

The accuracy of the free surface is heavily dictated by the mesh. Refinement is required throughout the domain in the free surface region, with additional refinement required on the hull surface, to ensure that is it properly resolved, and to avoid large areas of diffusion by ensuring a sharp interface. Ideally, the mesh should also be aligned with the undisturbed free surface. Additional refinement should also be added aft of the vessel if the wake pattern is of interest. Steps should be taken to minimise reflection by the domain boundaries of the very small waves that are generated back into the domain.

It should be noted that any potential impacts of a sea bed, particularly in shallow water conditions are not considered within this work. This could however present an area for further investigation.

5.3.3 Porous Formulation

As discussed by Ianniello et al. (2010a), in the Literature Review Chapter, and above, the quadrupole noise sources present within the fluid may not be negligible in hydroacoustics as they are in aeroacoustic applications. Quadrupole noise sources arise from the unsteady shear stress and vorticity content of a highly turbulent flow domain such as that observed in the wake of the hull and propeller. The most appropriate method for achieving this in CFD simulation is to use the Porous Formulation (Farassat 2007). In this approach, a permeable source surface, which encloses the vessel, is placed within the flow and is used as the radiating surface for the acoustic model. The surface integrals are calculated on this surface, meaning that the contributions of all the flow properties and non-linearities contained within the porous surface are accounted for, without the need for volume integration. This surface will then allow for the monopole and dipole sources from the hull and propeller surfaces to be captured, as well as the quadrupole sources within the flow, such as those arising from turbulence. This approach has been shown to be a robust approach and effective approach in work carried out by INSEAN (Ianniello, Muscari, & Mascio, 2010). The F-WH equation for the properties of the porous data surface can be seen below:

$$\square^2 c^2 \rho' \equiv \square^2 p' = \frac{\partial}{\partial t}[\rho_0 U_n]\delta(f) - \frac{\partial}{\partial x_i}[L_i \delta(f)] + \frac{\partial^2}{\partial x_i \partial x_j}\left[T_{ij} H(f)\right] \tag{5. 7}$$

Where:

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

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

and the other variables are as defined in above and in Appendix A.

Within this approach, this permeable surface takes the form of a half-cylinder which stops at the free surface and which encloses the entirety of the vessel up to the waterline, and part of the flow downstream of the propeller.

In StarCCM+, an internal interface boundary is created using a cylindrical surface to create the permeable surface, within the stationary region, and within the Ffowcs-Williams Hawkings model, this surface is selected by the user as the emitting surface to be monitored by the receivers using a simple drop-down menu.

5.3.4 Boundary Conditions

The table below gives an outline of the boundary types assigned to each of the computational domain boundaries within the model:

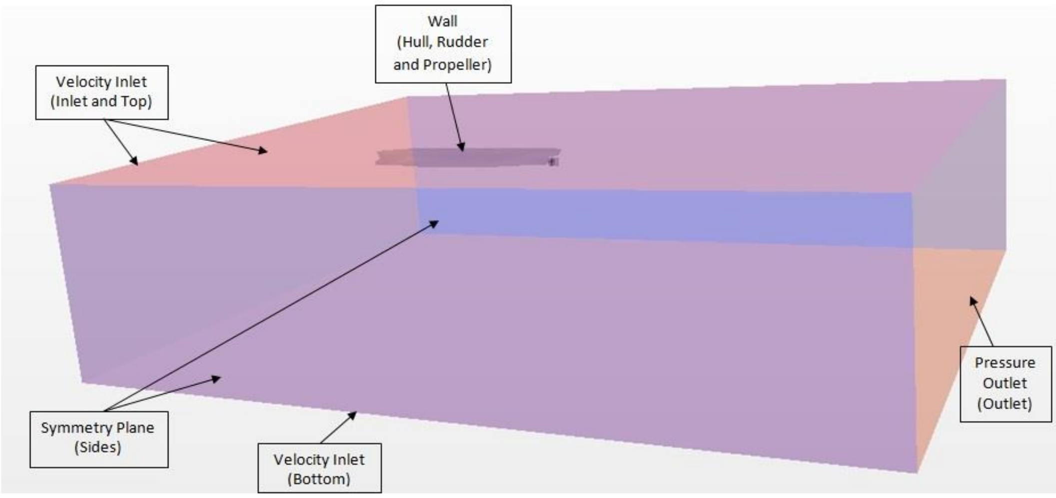


Figure 5.1 - Boundary Conditions

The velocity at the inlets, pressure at the outlet, and volume fractions of water and air at the boundaries are all set to field functions dictated by the VoF flat wave which represents the free surface. Damping of the wave can also be applied directly at the boundaries, but again should be applied with caution. The other parameters are linked to turbulence modelling and are again left at default values. The symmetry plane boundaries have no specific settings applied to them. The hull, rudder and propeller have to be defined as wall boundaries, and the no-slip condition has been applied meaning that the fluid "sticks" to the wall giving rise to a velocity gradient from stationary at the wall to the free stream velocity at some point away from the wall boundary.

For velocity inlet boundaries, the inlet face velocity is specified directly, as is the static temperature. The mass flow rate can be calculated using the known velocity as shown in the equation below: a Neumann Condition is applied for pressure correction, and mass flux corrections are zero.

$$\dot{m}^*{}_f = \rho_f\left(\boldsymbol{a}.\boldsymbol{v}^*{}_f - G_f\right) \qquad (5.\ 8)$$

Where: $\dot{m}^*{}_f$ *is the face mass flow rate*

$\boldsymbol{a}$ *is the face area vector*

$\boldsymbol{v}^{*}{}_{f}$ is the known boundary velocity

G_f is the grid flux

For wall boundaries with no-slip walls, the velocity is set to the specified value. The pressure is calculated as discussed above and temperature for an inviscid solution is calculated from adjacent cells.

For pressure outlet boundaries, the boundary face velocity is extrapolated from the interior and where the Segregated Flow model is being applied, the normal component of the velocity for inflow can be used to stabilise the solution. The pressure correction is not zero as was the case above, and is calculated using the equation below, and a mass flux correction is required, which is dependent on whether the flow is subsonic or supersonic. For inflow, the boundary pressure is dependent on the boundary inflow velocity, to discourage backflow. For subsonic outflow, the boundary pressure is specified. For inflow, the specified static temperature is used, whilst for outflow, the boundary face temperature is extrapolated from the adjacent cell.

$$\dot{m}^{*}{}_{f} = \rho_f\left(\boldsymbol{v}_f . \boldsymbol{a} - G_f\right) - \gamma_f \qquad (5.\ 9)$$

Where: $\boldsymbol{v}_f$ is the boundary velocity

γ_f is the Rhie-Chow-type dissipation

For symmetry plane boundaries, the shear stress is zero and the velocity, pressure and temperature are calculated from adjacent cells.

5.3.5 Turbulence Modelling

The k-Epsilon turbulence model formulations were developed separately by Jones & Launder (1972), and Launder & Sharma (1974). These models require the solution of two extra transport equations, compared to the simpler Spalart-Allmaras model, to calculate turbulent kinetic energy (k) and turbulent dissipation (ε), hence the name. This group of turbulence models represents a compromise between robustness and reliability, computational cost, and accuracy of results. They are best suited to industrial-type applications which include complex flow and regions of recirculation.

The two basic transport equations for the Realizable k-Epsilon Turbulence model are as defined below:

$$\begin{aligned} \frac{d}{dt}\int_V \rho k dV + \int_A \rho k(\boldsymbol{v}-\boldsymbol{v}_g).d\boldsymbol{a} \\ = \int_A \left(\mu + \frac{\mu_t}{\sigma_k}\right) \nabla k.d\boldsymbol{a} \\ + \int_{\boldsymbol{V}} \left[G_k + G_b - \rho\big((\varepsilon-\varepsilon_0)+\gamma_M\big) + S_k\right] dV \end{aligned} \qquad (5.\ 10)$$

$$\begin{aligned} \frac{d}{dt}\int_V \rho \varepsilon dV + \int_A \rho \varepsilon(\boldsymbol{v}-\boldsymbol{v}_g).d\boldsymbol{a} \\ = \int_A \left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right) \nabla \varepsilon.d\boldsymbol{a} \\ + \int_{\boldsymbol{V}} \left[C_{\varepsilon 1} S_\varepsilon + \frac{\varepsilon}{k}(C_{\varepsilon 1} C_{\varepsilon 3} G_b) - \frac{\varepsilon}{k+\sqrt{v\varepsilon}} C_{\varepsilon 2}\rho(\varepsilon-\varepsilon_0) \right. \\ \left. + S_\varepsilon \right] dV \end{aligned} \qquad (5.\ 11)$$

Where: k is the turbulent kinetic energy

ε is the turbulence dissipation rate

$\boldsymbol{v}$ is the velocity

$\boldsymbol{v}_g$ is the grid velocity

μ_t *is the turbulent viscosity*

σ_k *and* σ_ε *are the turbulent Schmidt numbers*

G_k *is the turbulent production*

G_b *is the turbulent production due to buoyancy*

γ_M *is the dilitation dissipation*

S_k *and* S_ε *are strain rate parameters*

$C_{\varepsilon 1}$ *,* $C_{\varepsilon 2}$ *and* $C_{\varepsilon 3}$ *are coefficients*

The Realizable version of the basic model contains a different equation for the turbulent dissipation rate, ε, and has the coefficient C_μ which is used to compute the turbulent viscosity, defined in terms of mean flow and turbulence properties rather than assumed to be constant as is done in the standard model. The Realizable model is said by the software developers to be "substantially better than the standard k-Epsilon model for many applications". The two-layer approach allows the viscous sub-layer to be resolved in fine meshes. The computation is divided into two layers, with the layer adjacent to walls solving ε and μ_t in terms of wall distance. These values are blended smoothly with those calculated away from the wall. The turbulent kinetic energy is solved in the entire flow as in the standard approach.

The k-Epsilon model in general, and the Realizable Two-Layer k-Epsilon model was therefore deemed suitable for this work as it is a good model for drag and hydrodynamic prediction, and with the surface noise sources being generally considered, this would seem appropriate. The Realizable aspect includes a modified transport equation for ε, as well as calculating coefficients based on actual values rather than assuming a constant value, hence allowing for the physics of turbulence to be realistically modelled. The Two-Layer aspects means that this model can also be used with fine meshes which resolve the viscous sub-layer. This was hence felt to be well suited to the application for which it was being used. The k-Omega turbulence models which are a similar type of two-equation model, and in particular the SST model, would be a suitable alternative as they are generally thought to be more suited to problems involving a propeller. This would particularly be the case if quadruple sources and cavitation were being considered.

As all of the coefficients and general settings for this model have been optimised for accuracy and reliability within the CFD software, settings associated directly with these models were left as default. For all models within StarCCM+, All Y+ Wall Treatment is suggested. The segregated Volume K-Epsilon Turbulence Solver uses the Gauss-Seidel relaxation scheme, with no acceleration.

For further details on turbulence modelling, the different models available, and derivation of these models, refer to Ferziger & Períc, (2002).

5.3.6 Regional Conditions

The computational domain, as discussed previously, is split into two regions; the stationary and the rotating region. The rotating region is cylindrical, and encloses the propeller blade and hub geometries. The rest of the domain, including the hull and rudder(s) geometry comprises the stationary region. The two regions are linked using three in-place interfaces, located at the forward and aft circular faces of the rotating region cylinder, with the third corresponding to the enclosing cylinder ring.

The only pertinent variables within these sections are the specification of the motion for each of the two regions with reference to the global frame of reference for the simulation. The rotating region can hence be set to use a rotating mesh approach, where its motion is set to match the required rotation relating to the stationary global coordinate system, or a moving frame of reference approach, where the motion is set to be stationary with a rotating coordinate system. The stationary region remains unchanged.

It is possible for multiple rotating reference frames to be set up simultaneously, meaning that multiple propellers could be modelled using this method. It is also possible to include translational, as well as rotational motions for both the hull and propeller, either separately or coupled together, which again provides further scope for future opportunities.

5.3.7 General Conditions

The model applies a RANS (Reynolds Averaged Navier Stokes) solver. Although it has been suggested that LES or even DES may be more appropriate for this nature of study, these approaches are significantly more costly and time-consuming to apply. A designer is unlikely to have access to the power required to develop and run such simulations in the required time-scales for early stage design.

Temporal Modelling

Although it is possible within most CFD software to carry out steady state acoustic simulations, this is not recommended, as the results are time-averaged only and are hence prone to omitting important details regarding both flow and acoustic characteristics. In this study an unsteady simulation is used.

Time-step size for the simulation of noise is generally dictated by the frequency range which is to be modelled. As the majority of the acoustic power for commercial vessels is located in the lower frequency ranges, a modelling range of 0-500Hz has been selected. Ideally the range should be 0-1000Hz however for this case the full scale results for the case study vessel showed that the majority of the sound power was located in this lower range. The lower range would also allow for a slightly higher time-step, hence reducing computation time. For the time period over which the acoustic results are being generated therefore, a time-step of no larger than 0.001s is required. Overall simulation time is determined by convergence speed and requirements, and propeller rotation speed to ensure that sufficient full rotations have been performed during acoustic simulation, and hence is assessed on a case by case basis.

Flow Modelling

A Segregated Flow model is used, in which the three components of velocity and the pressure component are solved in an uncoupled manner, and instead the continuity and conservation of momentum equations are solved using a Predictor-Corrector approach. The equation below shows the Navier-Stokes equations for continuity and momentum in continuous integral form as applied in the segregated solver:

$$\frac{d}{dt}\int_{V} \rho\chi dV + \oint_{A} \rho(\boldsymbol{v}-\boldsymbol{v}_g).d\boldsymbol{a} = \int_{\boldsymbol{V}} S_u dV \qquad (5.\ 12)$$

$$\begin{aligned}\frac{d}{dt}\int_{V} \boldsymbol{\rho\chi v dV} + \oint_{A} \rho\boldsymbol{v}\otimes(\boldsymbol{v}-\boldsymbol{v}_g).d\boldsymbol{a} \\ = -\oint_{A} \rho\boldsymbol{I}.da \qquad (5.\ 13) \\ + \oint_{A} \boldsymbol{T}.d\boldsymbol{a} + \int_{V} (\boldsymbol{f}_r + \boldsymbol{f}_g + \boldsymbol{f}_p + \boldsymbol{f}_u + \boldsymbol{f}_\omega)\, dV\end{aligned}$$

Where: χ is the porosity

$\otimes$ is a tensor product

$\boldsymbol{I}$ is the identity matrix

$\boldsymbol{T}$ is the viscous stress tensor

$\boldsymbol{f}_r$ is the body force due to rotation

$\boldsymbol{f}_g$ is the body force due to gravity

$\boldsymbol{f}_p$ is the porous media body force

$\boldsymbol{f}_u$ is the user-defined body force

$\boldsymbol{f}_\omega$ is the vorticity confinement body force

Solution is achieved using the SIMPLE (Semi-Implicit Method for Pressure-Linked Equation) algorithm to link the pressure and the velocity field, in order to satisfy the continuity equation. Second-order convection has been chosen, meaning that a second-order up-winding scheme will be used by the solver. Details of the steps taken by the SIMPLE solver are outlined by CD-Adapco (2012).

A Segregated Flow model was selected as the most suitable, as it arises from solving mainly constant-density problems where the fluid is considered incompressible or mildly compressible, and this simulation is assuming a constant fluid density throughout the domain.

Within the segregated flow model, an Algebraic Multi-Grid (AMG) Linear Solver is used to run two sub solvers, for velocity and pressure, as required for the SIMPLE Algorithm. For the velocity solver, the Gauss-Seidel iterative relaxation scheme is used with no acceleration, while for the pressure solver, this relaxation scheme is used together with Conjugate Gradient iterative acceleration method. The segregated Volume of Fluid (VoF) solver uses the Gauss-Seidel relaxation scheme, with no acceleration. These solvers were used as the solver within the software has been optimised for this combination of approaches, and other options are not easily available. Use of the SIMPLE approach is satisfactory in this case as the time-steps used are small, and to ease convergence issues which can occur AGM solvers have been applied, assisted by the implicit nature of the solver.

Solvers

Within StarCCM+, the majority of variable gradients are reconstructed using the Green-Gauss Cell Based method, except for the pressure, which uses the weighted least squares cell based approach. For general details of these approaches, please refer to Ferziger & Peric (2002).

Initial Conditions

The initial conditions for the simulation are largely dictated by the free surface:

- Pressure: Hydrostatic pressure of flat wave

- Velocity: Velocity of flat wave
- Volume Fraction of Water (H_20): Volume fraction of heavy fluid in flat wave
- Volume Fraction of Air: Volume fraction of light fluid in flat wave

The turbulence model related initial conditions are not changed from the default settings, as these have been optimised by the software developers.

Solution Monitors

Ideally, buoyancy (lift) and resistance (drag) forces on the hull, and thrust and torque for the propeller should be monitored throughout the simulations. Where available, these parameters should be compared against model or full scale data for the vessel, as these will be ideal for validation of the physics of the simulation.

Validation of CFD Approach

The software as a whole has been developing using an extensive library of test cases to validate the build-in solvers, and hence it can be assumed that in general StarCCM+ has been validated.

The approach used within the Ffowcs-Williams Hawkings (FWH) solver has been validated, and within the StarCCM+ User Guide, a test case by Caraeni et al. (2011), of CDNA and CD-Adapco, has been cited. It should be noted that this case is based on an aeronautical test case using a DES model rather than a RANS approach for the hydrodynamic field.

The turbulence model is based on the validated approach outlined in work carried out by NASA (Shih et al. 1994).

5.3.8 Meshing and Discretization

As with all CFD applications, the mesh is the key to a successful simulation. Judicious selection of cell sizes, distributions and areas of refinement will ensure that all relevant flow features and details are captured. Again, a balance is required between accuracy and computational time hence some compromise is required, especially in a case such as this one where simulation time is required to be short. Whilst areas of mesh refinement around the surface for accurate capture of the boundary layer, around the free surface height for simulation of that, and in other areas of complex flow, areas of coarser mesh are appropriate in areas which provide few details of interest to the noise investigation.

5.4 Requirements for Ideal Case Numerical Model

This section will outline the ideal case for the CFD simulation coupled with the F-WH Solver used to predict commercial ship noise, especially in early design stages, which has been found to be the most accurate approach and to constitute the most suitable compromise.

5.4.1 Geometry and Data

In order for the provided geometry to usable for the CFD simulation, it should ideally be an outline model comprising of a limited number of connected surface, which accurately reflects the as-build vessel form. Gaps in the surface or areas of intersecting surfaces place significant levels of demand in terms of pre-processing on the user. In some cases, exact hull or propeller data is not made available, which can also be extremely problematic.

In terms for required information, the key vessel and propeller geometrical particulars, vessel speeds and corresponding propeller rotation rates and vessel trim conditions are required, to ensure the simulation developed is an accurate reflection of reality.

5.4.2 Propeller Modelling

As discussed above, the most accurate and appropriate form of propeller modelling is to use a rotating mesh approach with sliding interfaces for data transfer between the rotating and stationary regions. This is recommended for all such studies of this nature.

5.4.3 Free Surface

It was also discussed above how the free surface can impact on the noise levels predicted, and therefore it is felt that it is important to use a transient VoF approach for representing the free surface throughout the simulation.

5.4.4 Cavitation Condition

The majority of this study and subsequent model development has been conducted considering a non-cavitating condition for the propeller. The simulation of cavitation is a very complex problem, and would add significantly to the demands of the CFD simulations, making them much less suitable for use in early design stages. For this reason, it is thought to be appropriate to consider only non-cavitating conditions simulations, providing that the user is aware of the errors this would introduce to the results, especially at higher speed where cavitation noise becomes dominant. These levels of error are discussed in a later chapter. However, if facilities and expertise are available to enable the inclusion of cavitation phenomena, they should be used.

5.5 Modelling Concerns for Numerical Modelling

In cases where an exact geometrical model is not available, it is even more important that resistance / thrust / torque and other performance data is made available, so that the model applied can be appropriately validated. This can often occur in cases where the vessel design is not the vessel owner, and for confidentiality reason, the full vessel details cannot be released, which is problematic. In some cases the required data is also not available, and in these cases it is important that the error is well understood.

As has been stated previously, the free surface capture is very dependent on the mesh and so problems may arise in relation to the surface, which particularly becomes a problems where the propeller appear to experience ventilation, rendering results unusable. That is to say, the free surface interacts with the hull and eventually the water phase starts to surround the propeller, rather than water. This then leads to incorrect results. A similar problem might also occur in a ballast draft condition when the propeller is partly emerging. In this case, a different free surface approximation may be required. Drag force prediction can also be affected.

5.6 Data Analysis Tool

In order to carry out more details data analysis, a methodology has been developed within this study for this purpose. Although the model for this has been developed using Microsoft Excel and Visual Basic macros in this instance, the general methods could be applied in other data analysis software as well. The general purpose of this model is to assist the user in the following ways:

- Obtaining useful data for use within their own CFD software, such as estimation of speed of sound based on input values for temperature, salinity, depth and density
- Predicting the machinery noise tonal frequencies, using input data for the different machinery items
- Providing visual comparisons of numerically calculated data with empirically estimated data, and with predicted machinery noise tonals for different installed items, at different speeds
- Providing visual comparison of the numerically calculated spectra against recommended underwater noise limits
- Providing visual comparison of the numerically calculated spectra against suggested threshold values for biological impact
- Providing information which could be used in assessing the potential impact of the noise on marine wildlife, in the form of a filterable marine wildlife database

The sections below will provide more details of the approaches.

5.6.1 Data Inputs

The data which can be input into the model, and the ways in which the data will be used, is outlined below.

General Particulars

The key Naval Architecture general particulars of the vessel in question can be input, and values will be used principally in empirical estimations of the vessel spectra (SPL) using the Speed-based formula from (Ross 1976) presented in Chapter 3 (§3.6.1). The first (equation 5.14 below), speed-based and third (equation 5.15 below), propeller-based equations have been found to most closely predict the actual ship, and therefore an average of the predictions from these two formulae is found and used to provide the most accurate indication on broadband source level at a user specified frequency. It should be noted that all the empirical approaches discussed provide an estimate for the source level noise rather than the receiver level results which are obtained numerically and through measurements. Correcting results to a different distance (source or receiver) introduces additional errors.

$$SL_1 = 112 + 50log\left(\frac{V}{10\ knots}\right) + 15log\Delta \qquad (5.14)$$

$$SL_3 = 175 + 60log\left(\frac{V_t}{25\ m/s}\right) + 10log\left(\frac{B}{4}\right) \qquad (5.15)$$

Where: SL is the source level above 100Hz in dB re1μPa at a reference distance of 1 yard

V is the ship speed in knots

$10\ knots$ is a reference speed

Δ is the displacement tonnage

25 m/s is the reference tip speed

B is the number of blades

V_t is the propeller tip velocity,

$$V_t = \pi n D$$

Where: n is the rotational speed in rps

D is the propeller diameter in m

The limitations of applicability of these formulae are also much more suited to modern commercial vessels. The results from the second, displacement-based Ross model (equation (3.18) in Chapter 3), along with the Wales & Heitmeyer (2002) and Junger (1987) models, are also presented for information only. Please refer to these papers for details of the equations.

The variables required are all basic vessel characteristics, and are typically defined very early in the design stages of a new vessel, and should therefore be easily available. The full list of required general particular values is given below:

- Length Overall (m)
- Maximum Beam (m)
- Operational Draft (m)
- Maximum Depth (m)
- Service Speed (knots)
- Displacement (tonnes)

Machinery Installations and Propeller

The input also requires some basic details regarding the machinery and propeller. The machinery data focuses mainly on the vessels main propulsor. Details are also required for the propeller.

The machinery data will be used within in empirical formulae which will predict the likely onboard source level noise contributions from diesel engines and generators (equations (3.44) and (3.45)), and the diesel engine onboard broadband noise estimation (equation (3.46)). These formulae were presented previously in Chapter 3 (§3.6.5). It will also be used to predict the machinery noise tonal frequencies, which are outlined above and in Chapter 6. The list of required input for machinery is provided below:

- Main propulsor rated power (kW)
- Main propulsor rated rpm
- Main propulsor first speed rpm
- Main propulsor second speed rpm
- Number of engine cylinders
- Cylinder diameter (m)
- Number of generators
- Generator rated power (kW)
- Generator rotation rate (rpm)
- Propeller shaft rotation rate first speed (rpm)
- Propeller shaft rotation rate second speed (rpm)
- Auxiliaries rotation rate (rpm)
- Reduction gear teeth contacted per second, first speed
- Reduction gear teeth contacted per second, second speed
- Propeller diameter (m)
- Number of blades
- Propeller rpm first speed
- Propeller rpm second speed

As this analysis approach is designed for use in early design stages, where exact values for the new design are not known, estimates can be used.

The propeller data will be used in two different ways. Firstly, the values will be used to predict the likely ship source level in the cavitating condition using an empirical formula (equation 3.35 in Chapter 3). Secondly, it will be used within the input itself in the Ross propeller-based broadband ship noise spectra prediction formula, as discussed above. The full list of required input for the propeller is provided below:

- Number of propellers
- Propeller diameter (m)
- Number of propeller blades

As above, where elements of this information are not readily available at the given design stage, assumptions or substitution can be made based on similar or previous designs.

Operational Profiles

The definition of the typical or most critical operational profiles of the vessel in question is important. These profiles will require information, or closest assumptions, for the following data:

- First speed (knots) to be modelled, which will also be converted to m/s
- Second speed (knots) to be modelled, which will also be converted to m/s
- Main propulsor first speed rpm, which will also be converted to rps
- Main propulsor second speed rpm, which will also be converted to rps
- Propeller first speed rpm, from which blade tip rotation speed will be calculated
- Propeller second speed rpm, from which blade tip rotation speed will be calculated

The required receiver location is also a required input, as these distances are used later when carrying out approximate corrections of receiver sound pressure levels to source levels (SL).

Operational Area

The intended operational area for the vessel will be selected from a drop down list of pre-defined regions. The reason for using a pre-defined list is that it matches the habitat areas used in the wildlife data used in the impact assessment, as discussed later. This database will be fully updatable. The list of initially defined regions is provided in full below:

Table 5.1- Operational Area Options

Location
Atlantic Ocean
Pacific Ocean
Indian Ocean
Southern Ocean
Baltic Sea
North Sea
Arctic Circle
Antarctic
Mediterranean Sea
Africa Coastal
Europe Coastal
Asia Coastal
Oceania Coastal
North America Coastal
South America Coastal
Caribbean

There is a requirement for the Ffowcs-Williams Hawkings equation used for the numerical modelling for an average speed of sound in water value for the chosen operational area to be specified. As such data is unlikely to be easily available for each region, it will be calculated empirically. These formulae require operational area water data, specifically:

- Average water temperature (^oC)
- Average water salinity (parts per thousand)
- Water depth of interest (m)
- Average water density (kg/m^3)

If these values are not known, links are provided to the online World Ocean Atlas (NOAA 2009) which contain global data. Three formulae are then used to predict the speed of sound in water: the Leroy method (equation (3.12), Medwin method (equation (3.14)) and Mackenzie method (equation (3.15)). All three approaches have been previously introduced in Chapter 3 (§3.5.2). An average of these three formulae is the calculated and can be used in numerical modelling. Indications will also be provided regarding whether or not the specified water data lies within the validity limits of the above formulae.

5.6.2 Visual Comparison of Results

Different graphs will be used for the visual comparison; two graphs for each of the selected vessel speeds.

The first graph will demonstrate the numerically calculated results over a user specified frequency range, and they will be able to choose whether to display the original receiver levels, or an approximation of the levels at source, using a cylindrical spreading approximation. Although this geometrical propagation estimation is not very accurate, it is felt that in this case, where the results are intended as indicative only, it is a suitable approximation. The averaged Ross model empirical prediction result, from the speed-based and propeller-based formulae, is used for generating predicted spectral data over a required frequency range for use in impact assessment purposes where no numerically calculated data is available. This will only be presented as an estimated source level value. There is also an option to display the estimated four tonal frequencies for each of the machinery items listed previously. It should be noted that the model will only permit the tonal frequencies for one machinery item to be displayed at a time, to avoid confusion.

The second graph will again present the numerically calculated results, this time only using the estimated source level values. The user will then have the option of comparing these results against threshold and limits, which are discussed in Chapter 7.

5.7 Machinery Noise Prediction

As discussed in the Literature Review Chapter, machinery noise and other onboard sources are typically prevalent below the Cavitation Inception Speed (CIS) of the propulsion system, and the spectra of this noise typically take the form of distinct tonal peaks at specific frequencies relating to system rotation or operational frequency, and its harmonics.

In almost every commercial vessel, the main engine(s) will be the most dominant source of machinery. This is especially the case for large, 2-stroke slow-speed diesel engines which cannot be easily isolated from the ship structure. After this, some of the other typical machinery noise emitters were identified in the Literature Review Chapter. These include auxiliary engines, gear boxes, generators, turbines, boilers and electric motors. As can be seen even from this short list, the sources will vary greatly depending on the type of propulsion system a vessel has, the type of vessel, and it's current operation. An understanding of how a vessel will be used, and the systems that will be installed on board should give the designer an indication of likely noise sources which could receive greater focus, with the potential for detailed modelling at later stages of the design.

There are some issues relating to the prediction of machinery and onboard noise, and in particular their contribution to underwater radiated noise spectra of commercial vessels which need to be taken into consideration. These are briefly outlined below:

- The nature of the onboard machinery sources is entirely dependent on the exact details of the installed systems and conditions under which they are operating. It is unlikely that at the early design stages for which this numerical underwater radiated noise prediction model is intended such in-depth details would be known. The final specifications for installed machinery are likely to be finalised at a later stage in the design, therefore a noise prediction based on assumptions only would be most suitable.

- Where concurrent underwater and onboard noise and vibration measurements are available, as well as data regarding the operational frequencies of some of the machinery sources, it can be possible to predict which source is responsible for which observed noise peaks. However even this is not easy and interaction effects can add extra complexity. Where a vessel is still in the design stages, even the prediction of tonal frequencies is only an approximation based on averaged operating system data and conditions.
- The onboard noise sources combine and travel through the ship structure to be radiated out from the hull. In order to accurately model and predict all the many transmission paths and losses for these sounds, the internal ship structural arrangements, materials and any damping equipment used need to be well understood. As with the above point, it is highly unlikely that such complex details would be known and finalised at an early stage in the design, making informed assumptions much more appropriate. The timescales and computational complexities involved in carrying out a full direct Finite Element Method (FEM) / Boundary Element Method (BEM) / Statistical Energy Analysis (SEA) analysis, or a hybrid method analysis, of the noise and vibration propagation would also be unsuitable for this intended purpose of this model. These approaches are discussed briefly in the Critical Review Section, and details specifically relating to the focus LNG carrier can be found in (Zoet et al. 2011).

It is most important to note that the key noise sources for each commercial vessel will vary, depending on a number of variables, and this cannot be easily predicted in great detail, however some knowledge on the nature and use of a vessel will give some reasonable indications of areas which may require greater focus.

5.7.1 Operational Influences

There are many aspects of the operation of the vessel which could impact on the noise and vibration characteristics of the vessels installed machinery systems.

The first of these is ship speed. Although there is a speed dependence of machinery noise in relation to the main engines, some machinery items such as the generators will run to supply hotel load requirements, which will be independent of speed.

The variations in hotel load requirements will also therefore have an impact on the machinery noise contributions from generators etc, however as these variations are very difficult to predict, especially at an early stage, they will be disregarded. It should also be noted that generally, such machinery items only make a very minor contribution to ship underwater noise hence this omission of noise variations will not be detrimental to the results.

The loading on the propeller will also vary the noise emitting from the main engines, which could also be assumed to have some speed dependence, although there are other factors which could also influence propeller loading. The cargo or ballast loading of the ship could also have an impact on the propeller loading and emitted noise observed. Another significant contributor to propeller loading variations is weather conditions and sea state. The more severe they are, the greater the loading on the propeller and therefore the main engine is likely to be, as it will need to do more work to continue propelling the vessel forward at the same speed.

However at the early design stages, it would not be practical to carry out underwater noise contribution predictions for the machinery to cover this extensive range of different operational conditions.

5.7.2 Tonal Frequency Prediction

As discussed above, within the data analysis tool developed as part of this work, prediction of tonal frequencies for a number of typical machinery noise sources will be carried out using empirical approaches. This will be done at both the selected vessel speeds, providing that the required information is inputted for both speeds. The tool also provides the option to present the tonal frequencies for each machinery item in turn overlaid on the underwater noise spectra. This could be particularly helpful in a case where actual field measurement data is available, for source identification. Where predicted underwater noise spectra are being used, the indications of tonals may be helpful when addressing potential impact of the noise, as discussed later. The first order tonal frequencies will be predicted as presented in the table below. Second, third and fourth order tonal will then be presented simply as multiples of the first order value.

Table 5.2 - Typical Machinery Noise Source Tonal Frequency Prediction

Machinery Item	Formula	Information Required	Assumptions
Main Propulsor	f (Hz) = Engine RPS	Engine RPM	Rotation is the only source of noise. Does not account for cylinder firing related noise, piston slap etc.
Reduction Gears (Gear whine)	f (Hz) = Reduction gear teeth contacted per second	Number of teeth, rotation rate	
Auxiliary Engine	f (Hz) = Auxiliary engine RPS	Auxiliary engine RPM	Speed independent
Generator	f (Hz) = Generator RPS	Generator RPM	Rotation is the only source of noise, speed-independent
Propeller Shaft	f (Hz) = Propeller shaft RPS	Propeller shaft RPM	
Propeller Blades (Blade rate)	f (Hz) = Propeller shaft RPS x number of propeller blades	Propeller shaft RPM, number of propeller blades	Does not account for any interaction or cavitation effects

It can be seen that although the formulae are simple, some of the required information may be difficult to access at early stages of a design, when details of the systems to be installed are not well known. In these cases, as the predictions are providing an indication only, estimated values could still be used. Where data from field measurements is being analysed, actual values should be recorded alongside the noise and vibration measurements to ensure more accurate assessment of likely sources.

It should be noted that this information may also be useful to in-service vessels. Monitoring of machinery tonal peak frequencies may assist in identification of problems or maintenance issues when the frequency shifts.

5.7 Chapter Summary

This chapter has outlined the numerical model which has been developed for the prediction of non-cavitation propeller and hydrodynamic noise within this study. It has also discussed the typical key machinery noise sources for commercial vessels, and what may cause these to vary. It has presented the problems associated with accurate prediction of these sources, especially in relation to their tonal peak noise amplitude. Empirical approaches for the prediction of tonal frequencies were presented.

The next chapter will present an application of the numerical and empirical models discussed in this chapter to an LNG Carrier in a validation case.

Chapter 6 - Application of Modelling to an LNG Carrier and Validation with Full Scale Measurements

6.1 Chapter Overview

This chapter will introduce the vessel being used in the case study for the validation of the numerical model and empirical models (§6.2). The field measurements carried out to obtain underwater radiated noise data will be outlined (§6.3), followed by a demonstration of how the ideal case numerical model has been applied to this vessel (§6.4). This will be followed by results (§6.5) and an outline of the problems experienced (§6.6), and a grid sensitivity study (§6.7). There will then be a discussion of the available variations in modelling from the ideal case (§6.8), and a presentation of the results obtained (§6.9). Some preliminary results for cavitation prediction are then presented (§6.10). Finally, some empirical modelling results for machinery contribution will be presented (§6.11).

6.2 Vessel Particulars

The focus of this case study is the Shell G-Class LNG carrier "Gallina", which can be seen in the figure below. This vessel has been used as it is powered by steam turbines and hence the majority of the underwater radiated noise is that arising from the propeller and hull which is the main focus of this study. There is also a good availability of full scale measurement data and general information for this vessel.

Figure 6.1 - LNG Carrier Used in Field Measurements

The vessels general principles are presented in the table below. As is typical for LNG Carriers of this size, the vessel is powered by steam turbines, which drive a single screw propeller arrangement. The vessel also has bow and stern thrusters to aid manoeuvring however as this study focuses on predicting radiated noise during normal transit operations, the thrusters and their associated noise properties will be henceforth neglected.

Table 6.1 - LNG Carrier Ship Particulars

Ship General Particulars	
Name	Gallina
IMO Number	9236626
Year of Construction	2003
Class	LNG
Shipyard	Mitsubishi Heavy Industries
Length Overall, LOA	289m
Maximum Waterline Length, LWL	266.9m
Maximum Waterline Beam, BWL	46m
Displacement	84491 tonnes
Maximum Speed, V_{max}	19.84 knots
Draft during Trials	Aft = 9.37m
	Forward = 9.37m
Propeller	
Maximum Power	21569 kW
Maximum Propeller Revolutions	81 RPM
Number of Propeller Blades	4
Propeller Blade Pitch	Fixed
Propeller Rotation Direction	Clockwise

Steam Turbines	
Combined Maximum Power	21569 kW
Steam Turbine Revolutions	High Pressure Turbine = 5804 RPM
	Low Pressure Turbine = 3966 RPM

6.3 Field Measurements

This data was gathered as part of the European Commission Funded FP7 Framework Program Project "SILENV" (Ships Oriented Innovative Solutions to Reduce Noise and Vibrations, Project number 234182, FP7-SST-2008-RTD-1). The author was not involved in the measurements, however was involved in subsequent related activities

6.3.1 Obtaining the Field Underwater Radiated Noise Data

In 2010, field measurements were carried out on an LNG Carrier, which included both underwater radiated noise at 2 speeds, and also corresponding onboard noise and vibration measurement to enable a greater understanding of the key noise sources in different conditions.

The field data was measured in calm, shallow water (approx. 45m depth) in the area east of Singapore (latitude: 001°25’N / longitude: 105°21’E), with a hydrophone suspended at 30m depth. This area was selected for its relatively low levels of background noise, as well as the calm water conditions. The figure below shows the measurement area:

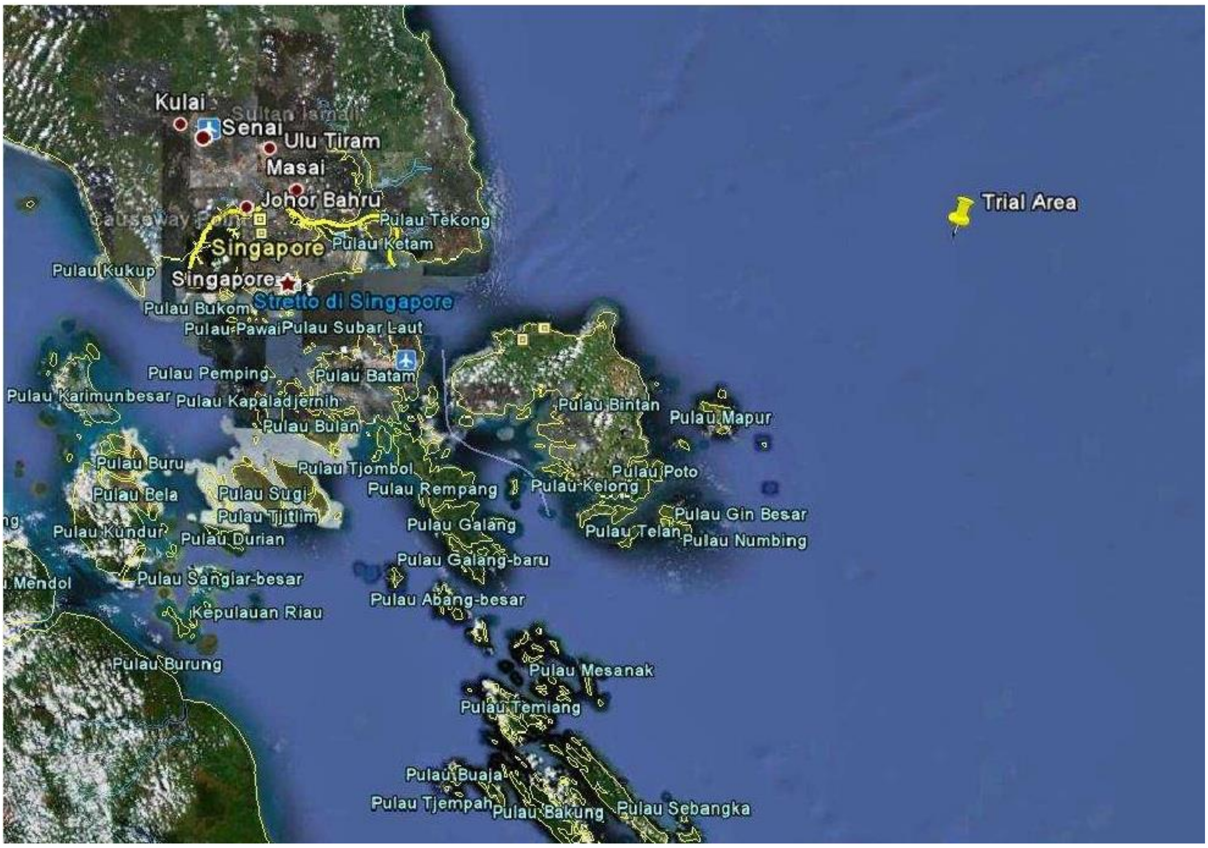


Figure 6.2 - Field Measurement Area for LNG Carrier

Measurements were taken with the LNG Carrier at anchor with machinery systems running, and at 9 knots and 19 knots forward speed, at a trial condition draft of 9.37m. The LNG Carrier passed by the hydrophone at a closest point of approach for each run as given as shown in Table 6.2 below. These distances were calculated using GPS latitude and longitude data for both the vessel and hydrophone buoy during each run; the closest distance was then calculated using the Haversine formula (Wikipedia n.d.). This formula is widely used in navigation, as it gives the greatest-circle distance between two points on a sphere based on their latitude and longitude, using the equation shown below:

$$d = 2r\,arcsin\left(\sqrt{sin\left(\frac{\varphi_2-\varphi_1}{2}\right)^2 + cos(\varphi_1)\,cos(\varphi_2)\,sin\left(\frac{\psi_2-\psi_1}{2}\right)^2}\right) \tag{6. 1}$$

Where: d is the spherical distance between the two points, in metres

r is the radius of the sphere

φ_1, φ_2 *are the latitudes of point 1 and 2 respectively*

ψ_1, ψ_2 *are the longitudes of point 1 and 2 respectively*

Table 6.2 - Closest Point of Approach Distances for LNG Carrier

Run Number and Details	Closest Point of Approach of Ship to Hydrophone (m)
Run 001 - 19 knots to Port	48.17 m
Run 002 - 19 knots to Starboard	125.20 m
Run 003 - 9 knots to Port	88.43 m
Run 004 - 9 knots to Starboard	136 m
Run 005 - Ship Still	35.70 m

Noise was measured for both the port and starboard sides on alternate runs, to ensure that any asymmetry in sources mainly arising from machinery installations, were captured. This data is available as both as-measured data from the hydrophone location and also data corrected to source level. However, as the data was only corrected to source level using spherical spreading laws (Transmission Loss = 20logR, where R is distance from source to receiver), with no account made of refraction, reflection or surface interaction effects, the as-measured data at the closest point of approach will be used for comparison. This will require that each set of field data has a corresponding simulation data set, obtained with a receiver in the same position as the closest point of approach of the LNG vessel on each field run.

The technical specifications of the hydrophone used for these measurements can be seen below:

Table 6.3 - Hydrophone Technical Specifications

Hydrophone Manufacturer	Co.l.mar., Italy
Identification	GP0280 - port 107
Type	Pre-Amplified Spherical Omnidirectional Hydrophone
Frequency Band	5 - 90,000 Hz
Attenuation at Low Frequencies	6dB/octave with -3dB at 15 Hz
Sensitivity	-164dB re 1 V/μPa
Directivity	Spherical - Omnidirectional
Maximum Operational Depth	1500m
Gain in Band at 5kHz	30dB
Input Impedance	10 mega Ohms
Power	11.5 - 30 Volts
Maximum Signal Output	5.5 Volts peak-to-peak
Absorption	15mA @ 12 Volts
Unit Length	230 mm
Unit Width	33.7 mm
Unit Weight	400 grams

6.3.2 Field Measurement Results

Results were obtained for ambient noise, ship still, 9 knots from the port and starboard sides, and 19 knots from the port and starboard sides, over the frequency range 0-20,000 Hz. All the field measurement underwater radiated noise results for the LNG Carrier are presented in Appendix B. The figure below compares the source level results for all the different measurements. Receiver level results in this case should not be directly compared as the receivers are all located at different distances from the vessel. The source level results are the measured results corrected to 1m from the source. The receiver levels are the values measured directly at the receiver location, with no corrections.

It can be seen that the results at 9 knots and those with the ship at zero speed are similar in level. The ambient noise level is also significantly lower than the measured SPL's, meaning that it has little influence on the measured results.

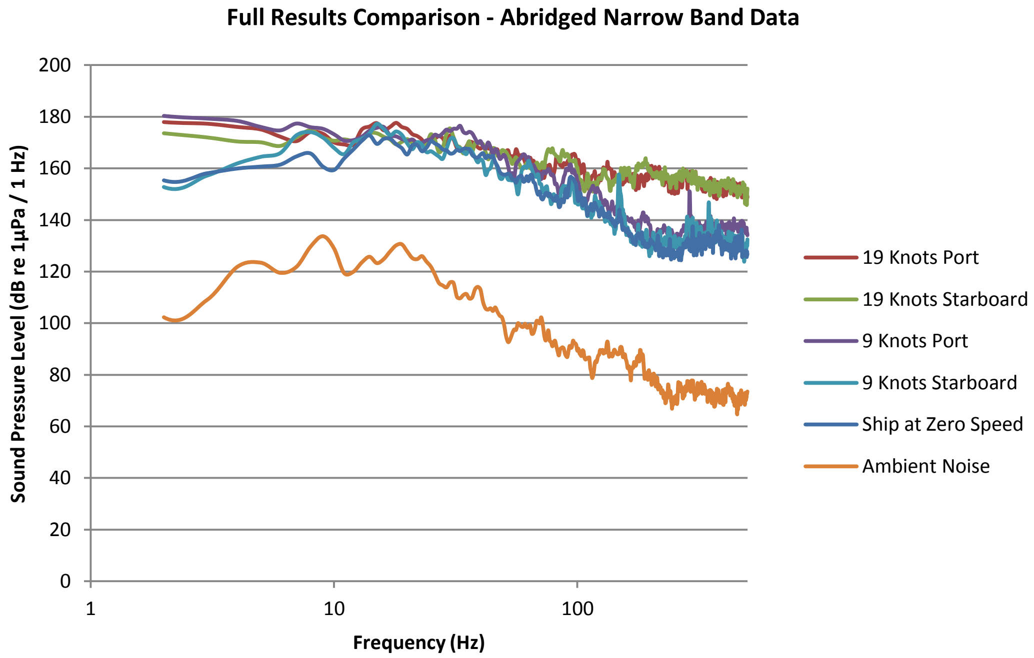


Figure 6.3 - Comparison of Field Measurement Data, Source Level

As can be seen from the different graphs presented in Appendix B, a significant portion of the ship noise is in the lower frequency ranges, up to around 500 Hz. This results in an average sound pressure level for the abridged data of around 20dB re 1 µPa higher than for the full data range. It is also known that higher frequency sounds propagate over much shorter distances, and are much more prone to attenuation from surface and sea bed interaction. For this reason, this 0-500 Hz range will be the focus throughout the simulation and data re-creation exercise, as discussed in later sections, for this vessel. In general, a range of 0-1000Hz would be more appropriate

6.3.3 Water Properties

Information on the water properties were also recorded during measurements, so that the conditions could be recreated as accurately as possible. Data on temperature vs. water depth, and speed of sound vs. water depth was recorded, and is shown below. An average of this data will be used to provide the reference water temperature, and speed of sound to be used in the numerical modelling. The speed of sound data was calculated using the 9-term Mackenzie Formula introduced in the Chapter 3 (§3.5.2).

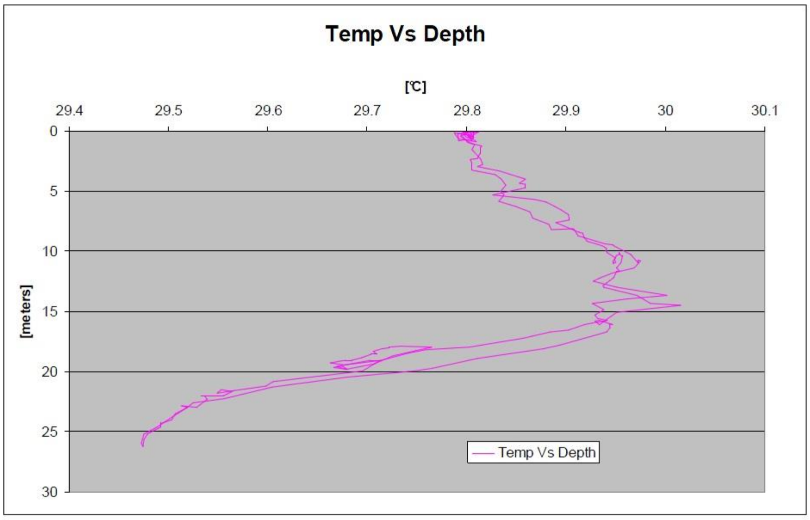


Figure 6.4 - Temperature Vs Depth During Trials

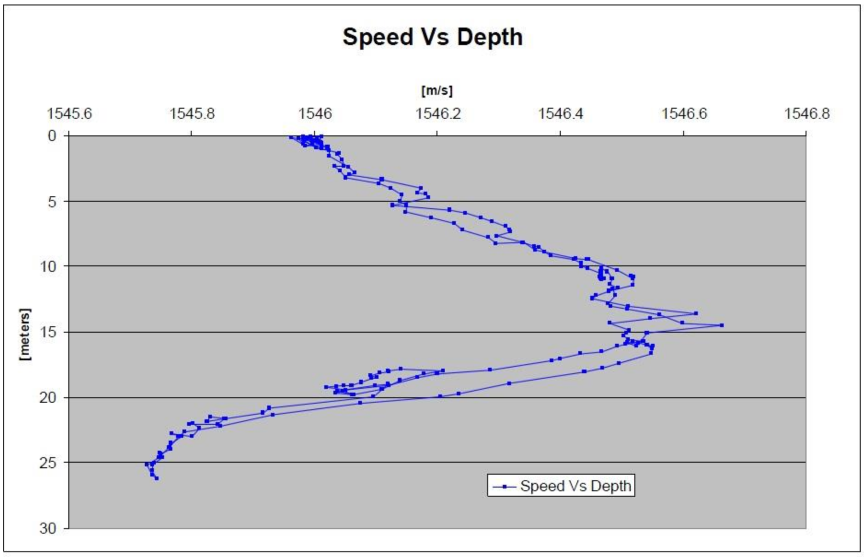


Figure 6.5 - Speed of Sound Vs Depth During Trials

6.3.4 Ambient Noise

As discussed above, the measurements area was chosen for its lower lever of background ambient noise however no measurement area will ever be "silent". The measured levels of ambient noise are presented in Figure 6.3 above.

In all cases where field measurement results are presented, the ambient noise contribution has been removed from the as measured results. To remove these values, the dB values have to be converted back to a linear scale, then the subtraction is carried out, and then the results is converted back to the logarithmic scale. This has been done to try and remove one source of error when comparing measured and predicted results, so that all results presented are purely due to the acoustic characteristics of the ship in question.

6.4 Application of the Model

The following section will outline how the ideal case numerical model outlined in Chapter 5 was applied to the validation case of the LNG carrier. The results achieved will then be presented in the next section, followed by a section discussing the problems which arose. Details of the work and results presented below can also be found in (Kellett et al. 2013).

6.4.1 Acoustic Modelling

In the case of the LNG Carrier a sound speed value of 1546.2 m/s was used; an average value taken from the sound speed profile measured during field trials. The corresponding far-field density was set to 1025 kg/m^3. Operating temperature was also specified using an average value from the temperature-depth profile found during measurements. Both the sound speed profile and the temperature vs. depth plot from the trials are presented above.

Receivers were located to coincide with the closest-point-of-approach hydrophone locations for each run, resulting in a port and starboard receiver in each simulation. Their locations also depended on the speed used. Time history data for the pressure signal at these receiver locations is recorded. It has been noted from (Hallander et al. 2012) and work carried out at INSEAN that results "obtained from upstream locations seems to have better agreement with measured data than downstream locations", and this was be taken into consideration within this work. The receivers were placed level with the vessels amidships. A summary of the receiver locations used are presented below:

Table 6.4 - Receiver Locations for LNG Carrier Simulations

Simulation	X Location	Y Location	Z Location
19 Knots Port	145m aft from bow	48.17m to port	30m deep
19 Knots Starboard	145m aft from bow	125.2m to starboard	30m deep
9 Knots Port	145m aft from bow	88.43m to port	30m deep
9 Knots Starboard	145m aft from bow	136m to starboard	30m deep

The sound source surfaces are also specified. In the ideal case, using the rotating mesh approach only, these were identified as the hull, rudder and propeller.

6.4.2 Propeller Modelling

As specified in the ideal case, the rotating mesh approach with sliding interface was applied. For both 19 and 9 knot simulations, the rotation axis was set to correspond to the propeller centre, with RPM of 79 and 50 respectively. For each time-step, the mesh will be rotated by the appropriate amount, and the model will recalculate the interfaces before carrying out iterations to find a solution.

6.4.3 Free Surface

As specified in the ideal case, outline in the previous chapter, the free surface was defined using a Volume of Fluid (VoF) flat wave, which was allowed to continually develop throughout the simulation. It was set to an initial height of 9.37m to match the vessel draft during the field measurements. For the 19 knots simulation the speed is set to 9.7736 m/s, whilst the 9 knots simulation is set to 4.6296 m/s. The respective densities of the two fluids were then specified; in this case 1.2 kg/m^3 for air and 1025 kg/m^3 for water. The figure below illustrates how the free surface is represented in this model at initialisation; it should be noted that the free surface will adjust to a natural position on the hull throughout the simulation.

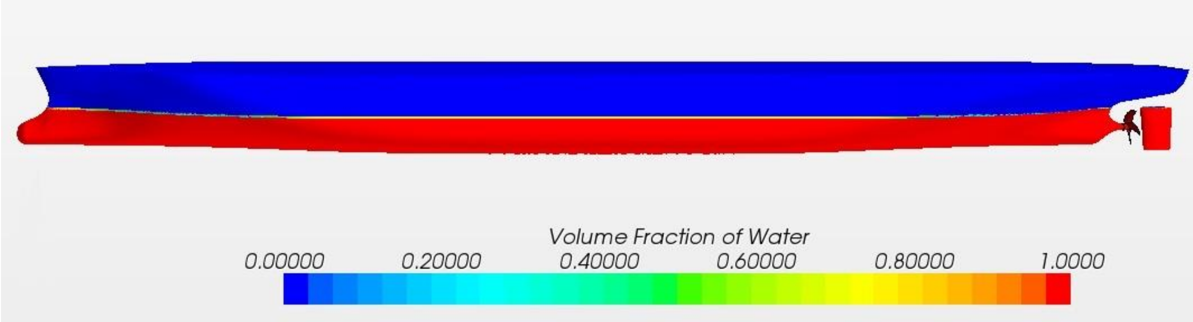


Figure 6.6 - Free Surface Representation

6.4.4 Boundary Conditions

The boundary conditions were defined as required in the numerical model discussed in Chapter 5. It should be noted that contrary to the condition for the inlet boundaries, where the volume fractions are related directly to the volume fractions of the light (air) and heavy (water) fluids of the VoF flat wave, for the pressure outlet boundary, the volume fractions are related to the resulting air and water fractions calculated from the developing free surface.

6.4.5 Turbulence Modelling

The Realizable Two-Layer k-Epsilon model was applied as required. The wall surfaces were treated as "smooth" for this simulation and no roughness correction has been applied, which may slightly affect the predicted forces.

6.4.6 Regional Conditions

The propeller was entirely enclosed in a cylindrical region. It was also crucial to ensure that problems would not arise with any parts of the hull aft being enclosed in the rotating region due to the rotating motion which will be applied, hence a small gap was left between the hull and the cylinder just forward of the propeller.

6.4.7 General Conditions

Temporal Modelling

For reasons discussed previously, the prediction of noise in the frequency range from 0 - 500 Hz was the focus. In order to ensure that results up to this value are obtained, with a sufficient level of accuracy, a time-step size of 0.001 seconds is used throughout the acoustic simulation stages. However it was also necessary to run an initial phase of the simulation with a larger time-step of 0.005 seconds to allow the flow and free surface to establish before the acoustic modelling commenced. The simulations were run for up to 30s in real time, with 20s for the initial phase and then a further 10s for acoustic data gathering. This equates to around 13 full propeller revolutions at 19 knots and 8.5 at 9 knots during the acoustic data gathering phase. It is accepted that this time is a compromise between accuracy of results and CPU time. A longer real time simulation run would be desirable to ensure better convergence of results however the small time steps used would make this very computationally demanding, expensive and time consuming.

In the simulation, the initial phase was carried out using a 1st order implicit unsteady solver, and then the 2nd order implicit unsteady solver is used for the acoustic simulation phase. This approach allows the simulation to settle without divergence problems, and then use the more accurate solver in the data gathering phase. In each case, the free surface and flow field was checked to ensure it was properly established during the initial phase.

Flow Modelling

The segregated flow solver was applied as required in the numerical model discussed in Chapter 5.

Initial Conditions

The initial conditions were set as discussed in the ideal numerical model presented in Chapter 5 (§5.4).

Monitors

Three monitors are set up for each simulation, to act as a check for the models realistically representing reality, and also to act as a comparison between modelling variations, to ensure the results can justifiably be compared. It is assumed that good agreement between these values between different model variations at the same speed suggests that the acoustic results can be justifiably compared.

The lift monitor measures the pressure and shear force in the positive z-direction on the hull and rudder re-meshed surface representations, i.e. the buoyancy force. The drag monitor measures the pressure and shear force in the negative x-direction on the hull and rudder re-meshed surface representations, i.e. the resistance force. The thrust monitor measures the pressure and shear force in the positive x-direction on the propeller blades and hub. A report is run once the simulation completes to establish the finishing values for each monitor, which is assumed to have converged to a representative value.

Validation

In order to validate the acoustic results, the field data for resistance is being used. The aim is to achieve good agreement between the field and simulated resistance forces at 19 knots, as some model scale resistance data is available for this condition. This value was estimated at full speed. The weight of the vessel is also known, and is speed independent, and therefore can be used to compare against predicted buoyancy data. A summary of the expected buoyancy and resistance forces is presented below:

Table 6.5 - Field Measurement Estimated Buoyancy and Resistance Forces

Field Measurement Estimated Resistance (kN)	$1.4\text{x}10^{3}$
Field Measurement Estimated Buoyancy (kN)	$8.288\text{x}10^{5}$

Buoyancy and resistance values for each simulation will be monitored throughout running, and the final value when the simulations ends will be recorded. This will be done so that the values from each simulations can be compared; good agreement between the values will be used to indicate that acoustic results are also comparable as they are modelling the same phenomena but using different methods.

A good agreement between predicted acoustic results and measured field data will be taken into account for validation purposes, as well as a comparison with the levels of accuracy quoted in other work of a similar nature, such as (Hallander et al. 2012).

It should be noted that propeller performance characteristics data (thrust / torque etc) were not available for either of the vessels and hence this approach could not be used for validating the simulation, as was suggested in the ideal case.

6.4.9 Mesh and Discretization

The LNG Carrier is enclosed within a rectangular domain of size 1000 x 800 x 250 m around the hull, rudder and propeller geometry. The distance from the inlet to the vessel is 200m; slightly under 1 hull-length. The stationary region, which includes all elements except the propeller and enclosing cylinder, as seen below, has a target cell size of 4m for the volume mesh. The hull and rudder surfaces however have been separately meshed to give a much finer grid, with additional refinement at the free-surface height of 9.37m, as seen below. These surfaces have a minimum cell size of 0.04m, with a target size again of 4m.

Figure 6.7 - Hull and Rudder Surface Mesh for the LNG Carrier

The cylinder surrounding the propeller geometry is kept small and is as shown below. The volume mesh in the rotating region has a target size of 0.5m. The surfaces have a minimum cell size of 0.01m and a target size of 0.5m.

Figure 6.8 - Propeller and Surrounding Cylinder Surface Mesh for LNG Carrier

Mesh generation was carried out using the automatic meshing tools in StarCCM+, resulting in a computation mesh of approximately 3 million cells in total with 20,000 in the rotating region and the remaining 2.98 million in the stationary region; a typical medium density grid. It is acknowledged that this number of cells is not really suitably sufficient for a full scale simulation of this magnitude. The results achieved may therefore suffer some loss of accuracy. The majority of the mesh in both regions is comprised of the "Trimmer" type mesh, which is predominantly a structured hexahedral mesh. Prismatic cells are applied using the "Prism Layer Mesher", in order to improve the capture of the flow gradients at the surfaces. 5 layers of prismatic cells have been used, with a stretch of 1.5, and a total thickness of 0.4m and 0.1m in the stationary and rotating regions respectively.

The simulation mesh has areas of progressively refined mesh size in the area immediately around the hull and propeller up to the inflow and outlet boundaries, as well as in the wake region, to ensure the complex flow properties are captured. These were in the form of progressively coarser rectangular grids around the hull at the waterline, and aft of the propeller. Cone-shaped areas of refinement were also added fore and aft of the propeller, to assist in capturing inflow and wake properties here more accurately, as these are likely to have a large influence on the resulting acoustic pressure. In order to improve this mesh, the refinement at the domain walls should be removed, and additional refinement should be added in the Kelvin Wake region.

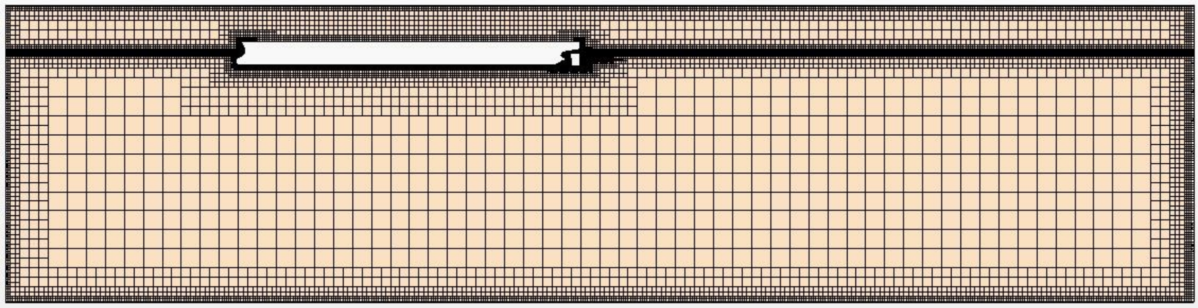


Figure 6.9 - Mesh Refinement Across Domain

In-place interfaces were then added at the propeller fore and aft ends, and around the circumference of the enclosing cylinder around the propeller. These are required to transfer flow variables between the stationary and rotating regions.

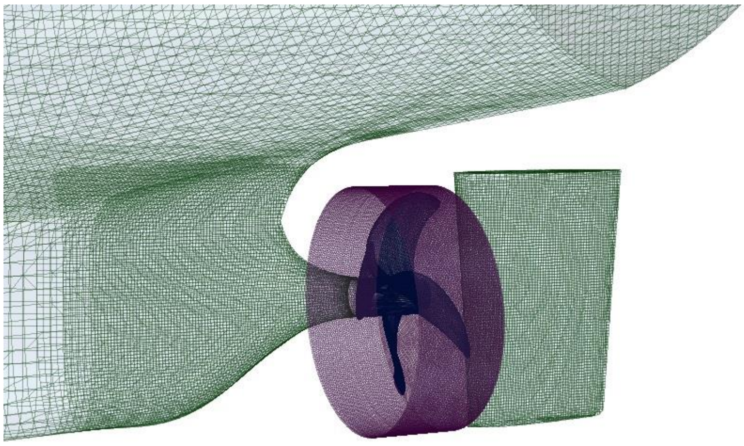


Figure 6.10 - Aft section showing stationary and rotating parts, and interfaces for LNG Carrier

6.4.10 Sources of Error

In applying modelling methodologies to simulating a real-life scenario, it is critical that the user is aware of the many sources of error which arise. The main sources have been summarised in the table in Appendix D, divided into the main areas of the work, and where applicable a comment has also been made on how they have been eliminated or addressed.

The Reynolds Number for this simulation is in the region of 2.8×10^9 at 19 knots and 1.08×10^9 at 9 knots which is very high for CFD simulation and hence the simulation of this vessel in full scale may in fact increase the inaccuracies, despite avoiding errors associated with scaling which would be present with simulation at model scale. It is proposed therefore that future work should be carried out at a scale of around 1/10 in order to solve the flow more accurately. The results would then be converted to full scale using the International Towing Tank Conference (1987) approach.

6.5 Results Achieved - Ideal Case

The section below will present the results achieved when the ideal case numerical model, as discussion previously and above, is applied to the LNG Carrier case, for both 9 and 19 knots. Full prediction results can be found in Appendix C. The results presented have all been corrected from receiver level to source level results using the spherical spreading law and the receiver distances quoted in Table 6.4 above.

6.5.1 Simulations for 19 Knots

The prediction results for the 19 knots simulation at the port receiver compared to the field measurement results can be seen below:

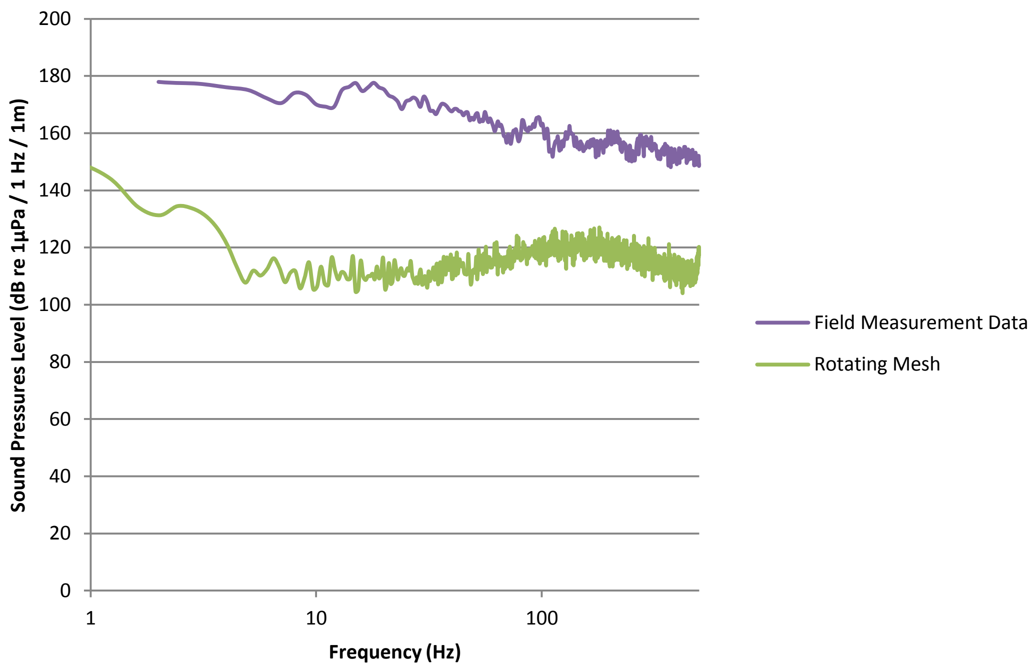


Figure 6.11 - Comparison of Predicted and Measured Results at 19 knots, Port Receiver

It can be seen that there is a discrepancy in results of approximately 40dB above 50 Hz. As the vessel operating at 19 knots is likely to be experiencing cavitation, some of the gap has arisen from cavitation not being included in the simulation. In field measurements, an increase of approximately 20dB was observed between 9 and 19 knots, which is attributed to cavitation conditions. However no cavitation observations were carried out during the field measurements. Assuming a similar increase from cavitation for the predicted results, there still remains a 20dB discrepancy which is significant. This is thought to be due to the many inaccuracies which arise from the simulation of a very complex real-life scenario using a compromised approach, along with the insufficient cells which have been used in the simulation mesh. Future applications of this approach would require a much more refined initial mesh. The sources of error are outlined more fully in Appendix D.

It was observed that the buoyancy force monitors for the above simulations tended towards a value of $6x10^5$ kN, which although not in agreement with the field measurement value of $8.288x10^5$ kN, provides a figure of suitable magnitude. It is felt that longer run times would cause the buoyancy force value to settle to an accurate figure eventually if the simulation had not been stopped and use of a much more refined mesh would also correct these significant discrepancies. The resistance force was found to be between $1.1x10^4$ kN and $1.3x10^4$ kN. It is suggested that better mesh refinement around the hull and wake could be considered if the increased mesh size and subsequent increase run time could be accommodated, as this would significantly improve the prediction achieved. The Wall Y+ values observed are up to 30,000 for the hull and slightly higher for the propeller and rudder. The Wall Y+ is expected to be high for high Reynolds Number simulations however it could be reduced to more manageable levels in this case by using an improved mesh.

The figure below shows the streamlines of velocity around the hull in the rotating mesh simulation which appears realistic:

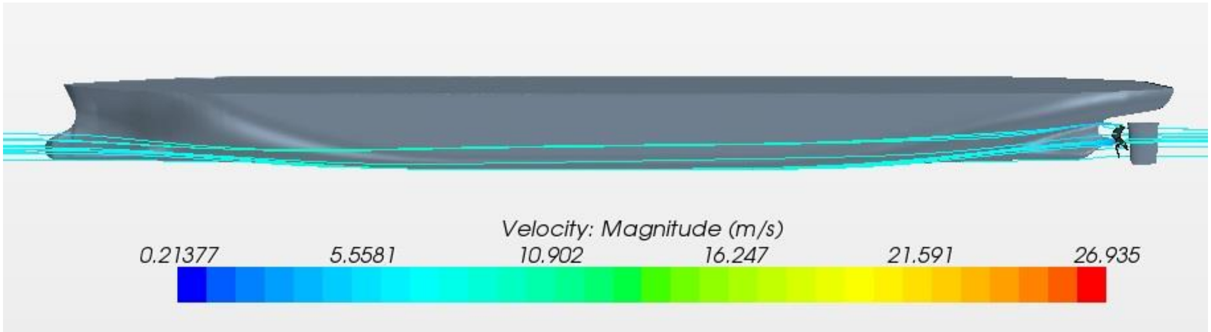


Figure 6.12 - Streamlines of Velocity for LNG Carrier at 19 knots

6.5.2 Simulations for 9 Knots

The simulation was also run at 9 knots. Firstly results for the port receiver are presented below:

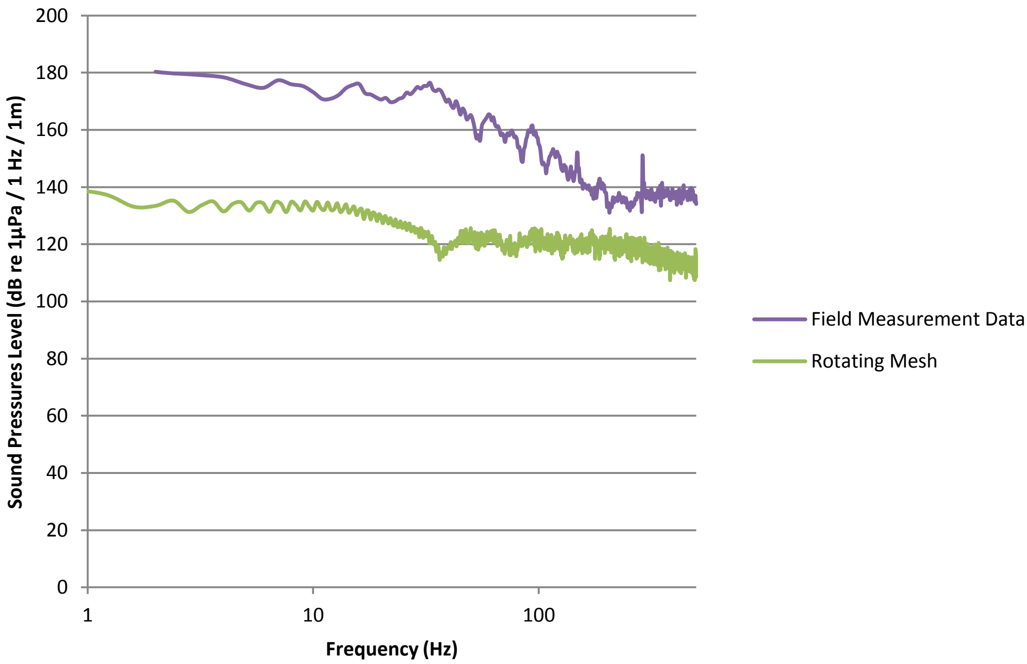


Figure 6.13- Comparison of Predicted and Measured Results at 9 knots, Port Receiver

As can be seen in the figure, at 9 knots, the simulation results are much closer to the field measurement data than at 19 knots. This is due to the fact that at 9 knots, it is though that there is little cavitation present as it has been estimated from model test data that the CIS for the vessel is in the region of 8 knots. As the simulation is only modelling the propeller in non-cavitating condition at this stage, the inaccuracy in the prediction is therefore significantly less at 9 knots. The improvement in prediction accuracy is in fact in the range of 20dB above 150 Hz, which coincides with the observed increase in sound pressure level between 9 and 19 knots in the field measurements, which was attributed to the influence of cavitation noise. The overall error in prediction in sound pressure level is generally around 20dB above 50Hz at 9 knots, which again is still significant.

It can be seen in the results for the starboard receiver, shown below, that the prediction error in this case is generally less than the 20dB observed at the port receiver, above 100 Hz. It is felt that this may be due to the increased distance of the starboard receiver from the noise source in this instance. The FWH method is intended as a far-field noise propagation approach, and therefore it is likely that the accuracy of the prediction would improve at larger receiver distances.

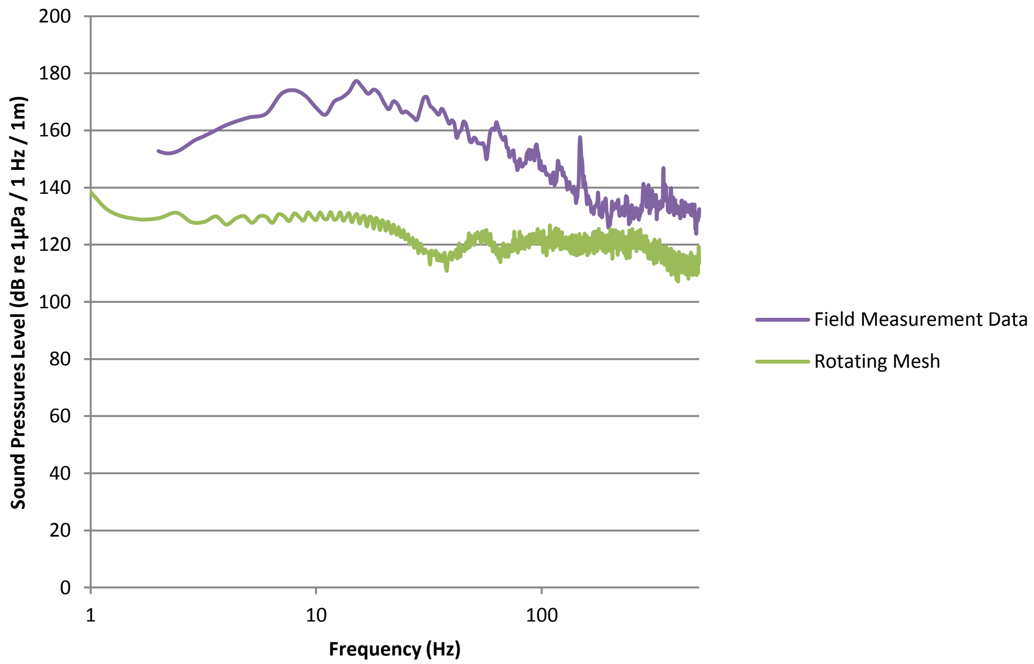


Figure 6.14 - Comparison of Predicted and Measured Results at 9 knots, Starboard Receiver

6.6 Problems Experienced

A number of problems arose from the application of the ideal numerical model to the LNG Carrier case, which could also be representative of those which could be experienced by a designer using the developer approach.

Firstly, some problems arose with the free surface, where it was observed to experience some small reflections from the domain boundaries, and also to cause large amplitude waves on the vessel hull. This was due to insufficient mesh refinement about the free surface despite increases. As further refinements would have made the resulting mesh very large and hence time consuming to run in this case, some damping was applied to the flat wave at the domain boundaries instead, which solved the problem. Again this is not an ideal solution, and in future cases, a more appropriate mesh would be preferable.

Some initial unsteadiness was also observed in the simulations which meant that turbulence under-relaxation factors had to be reduced in early stages of the simulation to allow it to converge on a more suitable result before increase them back to original values for the latter stages of the simulation. It is generally good practice to built up a simulation gradually and therefore this appears to be a suitable solution. Other alternatives might be to consider initially freezing the free surface, or starting with a moving reference frame approach and then switching to a rotating mesh approach later.

6.7 Grid Sensitivity Study

A grid sensitivity study was carried to test the mesh dependency of the achieved results for the 19 knot simulation. A new coarser mesh of 1.29 million cells used, which was approximately half the number of cells of the original 2.98 million cells. This was mostly achieved using a larger base size, to which all the other mesh parameters relate, meaning the new mesh could be generated quickly, and ensured a similar distribution of cells and refinement was achieved compared to the original mesh. The results are presented below.

It can be seen that there is very good agreement between the acoustic prediction results, although the coarser mesh gives slightly different spectra shape predictions at low frequencies. It was observed that the predicted buoyancy, resistance and thrust forces were all higher than for the original mesh. The buoyancy force was closer to the field measurement value however the resistance value was further from the required value, which is likely to be due to the coarser mesh. The higher thrust force prediction is also unlikely to be accurate, although comparison values are not available as discussed previously.

A second study was carried out with a finer mesh of approximately 6.5 million cells. From the results of this are again presented below. It can be seen that these results are slightly more accurate than those from the coarser meshes above 50 Hz, but in general they provide good agreement. Given the additional run time required for the larger mesh, it may not be required for the purposes of an early stage design assessment of noise characteristics. The buoyancy force and resistance forces were again broadly in the required region.

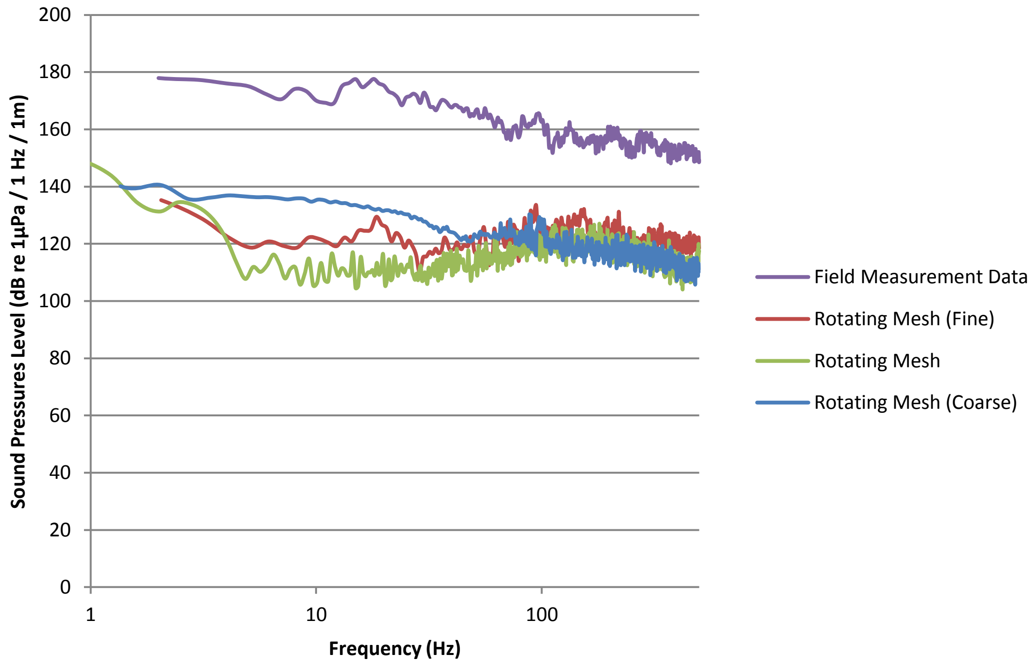


Figure 6.15 - Comparison of Results with Finer and Coarse Mesh at 19 knots, Port Receiver

6.8 Variations in Modelling

6.8.1 Propeller Modelling

For comparison purposes, the model was also run using a moving frame of reference approach, and with a static propeller geometry which had no rotation associated with it at all. These approaches, whilst less accurate also require less computational power and time and hence may be preferable to a designer with limited time available for prediction.

6.8.2 Free Surface

A comparison was carried out with and without a free surface, using a moving reference frame propeller approach to speed up the run time for this short investigation.

It should be noted that the omission of the free surface will have a noticeable impact on the buoyancy and resistance forces observed, as the vessel appears immersed fully in water. A comparison with the field measurement values for buoyancy and resistance is therefore not suitable in this instance.

6.8.3 Porous Formulation

Within this approach, this permeable surface takes the form of a half-cylinder which stops at the free surface and which encloses the entirety of the vessel up to the waterline, and part of the flow downstream of the propeller. Current knowledge on the subject appears undecided on the optimum location of this surface therefore several different half-cylinder radii values were compared. Radii of 40m and 50m were both tested with a moving frame of reference approach, and additionally to 40m and 50m radii, cylinders of 30m and 60m were tested with a rotating mesh approach. The reasons for these different tests and the findings of this comparison are discussed later. The final location and dimensions of the half-cylinder selected based on the results achieved are as seen in the table below, and also in Figure 6.16:

Table 6.6 - Permeable Source Surface Dimensions Relative to Hull in Model

Radius	40m
Length	400m
X forward relative to hull	-20m (20m in front of the bow)
X aft relative to hull	380m (approx. 100m aft of stern)
Z Top relative to hull	9.37m (at free surface)
Z Bottom relative to hull	-30.63m (approx. hydrophone depth)

Figure 6.16 - Permeable Source Surface in Relation to Hull Model

In StarCCM+, an internal interface boundary is created using the cylindrical surface to create the permeable surface, within the stationary region, and within the Ffowcs-Williams Hawkings model, this surface is selected as the radiating surface to be monitored by the receivers, rather than the hull and propeller surfaces as before.

6.9 Results Achieved - Variations

The following section will present the results achieved from the application of the modelling variations to the ideal case presented above.

6.9.1 Propeller Modelling

As was expected the rotating mesh approach produced the most accurate results, as this approach is seen as the most complex and the closest to what is occurring in reality. The moving frame of reference results are generally around 20dB lower again in terms of SPL and the static geometry approach is lower still. The static geometry approach is hence considered unsuitable and will not be used in any further simulations. The moving frame of reference approach could be considered when used together with other modelling variations discussed later, as the much shorter run time and lower computational demands of this approach are an advantage it has over the rotating mesh approach. The figure below compares the three propeller modelling approaches with the field measurement data:

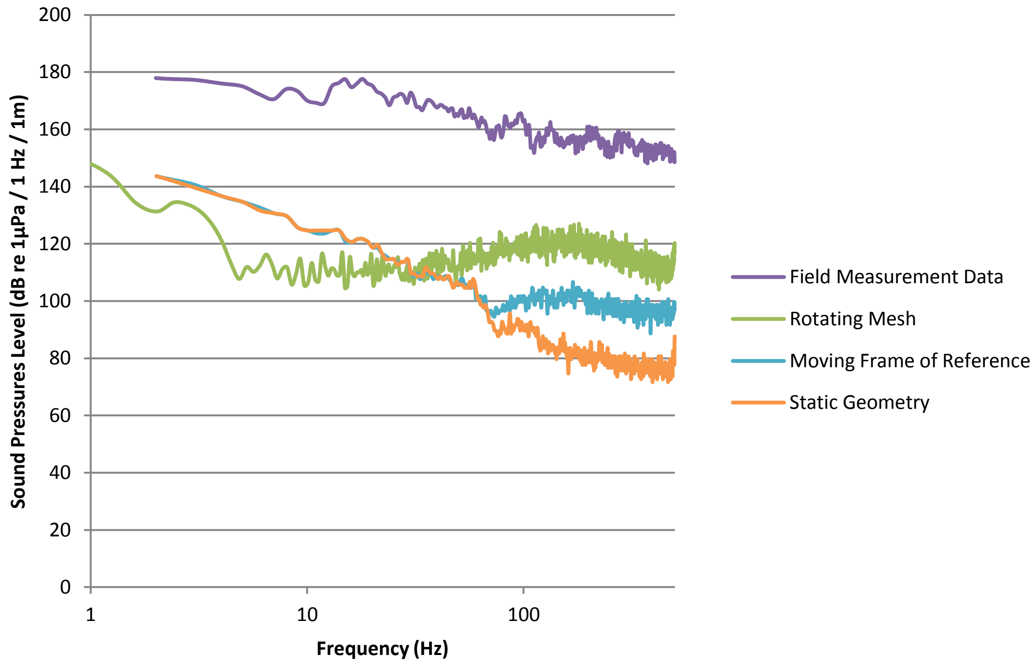


Figure 6.17 - Comparison of Propeller Representation Methods at 19 knots, Port Receiver

A similar comparison was also carried out for the simulation at 9 knots, and the results are presented below. It can be seen that the results for both approaches is as above, with the rotating mesh approach being approximately 20dB closer to the field measurement data than the moving frame of reference approach. However as was observed previously, the predicted results at 9 knots are generally closer to the measurement data, especially above 50 Hz, due to the very low levels of cavitation noise at this lower speed making a much lower contribution to the spectra than at 19 knots.

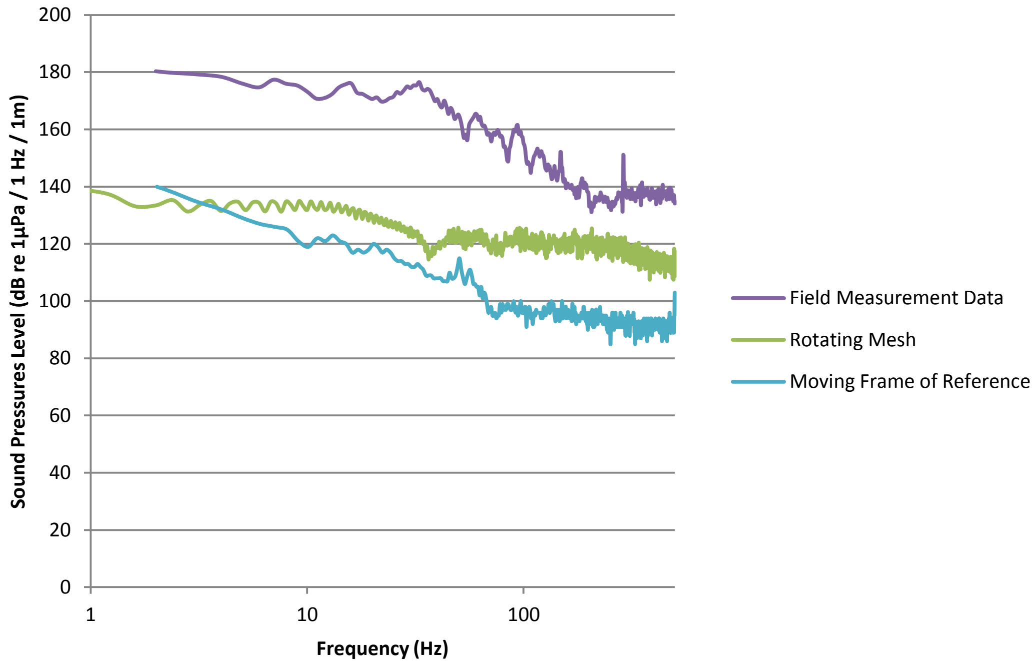


Figure 6.18 - Comparison of Propeller Representation Methods at 9 knots, Port Receiver

6.9.2 Free Surface

It can be seen from the results presented below that at the lower frequencies, up to around 200Hz, the model with no free surface predicts the sound pressure level to be higher than the simulation which includes the surface. However, it can also be observed from the results that above 200Hz, the results with and without a free surface present show good agreement, suggesting little impact from the free surface at these frequencies. This is useful to note as running a simulation with no free surface will speed up computation time and reduce complexity, and as can be seen, there may be little penalty in prediction accuracy in comparison to the other approach discussed here.

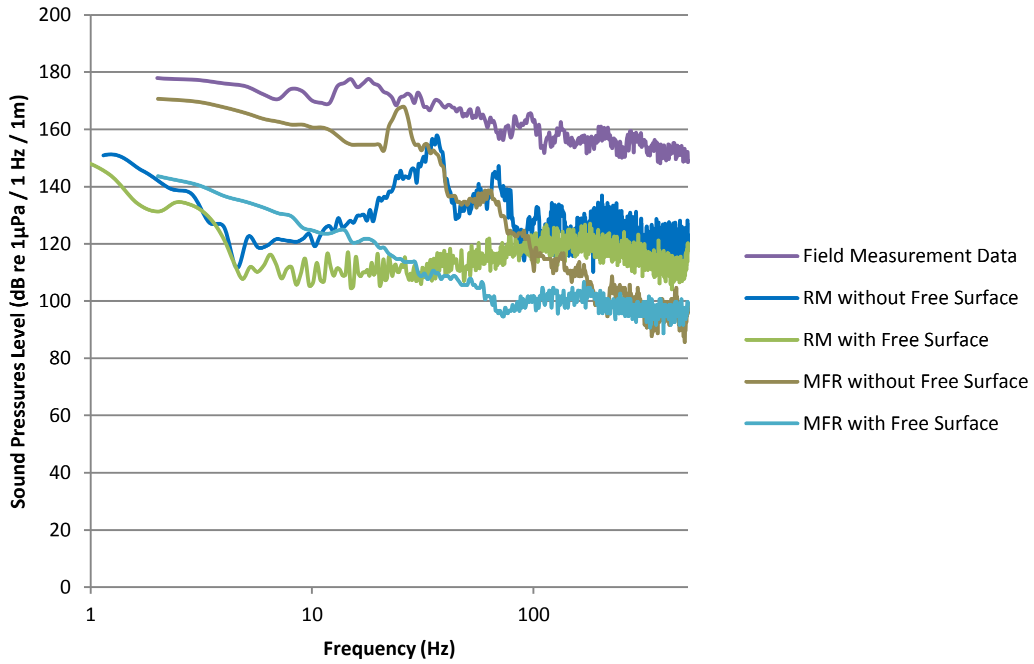


Figure 6.19 - Comparison of Spectra with and without Free Surface at 19 knots, Port Receiver

6.9.3 Porous Formulation

Comparing the results achieved with a 40m, 50m and without a permeable source surface, using a moving reference frame propeller approach, it can be seen that the inclusion of a free surface has a significant impact on improving the accuracy of the results obtained with an improvement of around 20dB, as was suggested. For this particular case, it appears that the permeable surface with 40m radius generally produces a more realistic spectral shape than the surface at 50m. The reasons for this are not clear and further research into the optimal permeable surface locations for different cases is required. It should be noted that the reasons for the apparent peak at around 350Hz with the 40m surface are also not known.

In this case, the resistance values were found to be between 1.22×10^4 kN and 1.23×10^4 kN, which are again too high but with a downward trend. Meanwhile the buoyancy values were between 7.67×10^5 kN and 7.69×10^4 kN at the time the simulations were stopped, again showing reasonable agreement.

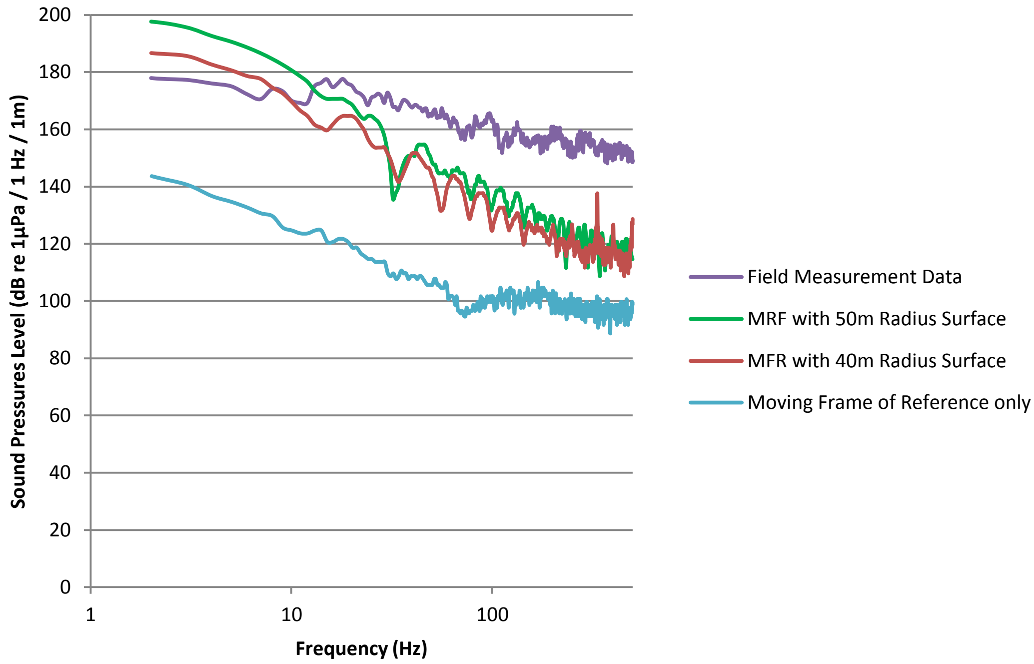


Figure 6.20 - Comparison of Moving Frame of Reference with Different Permeable Surfaces at 19 knots, Port Receiver

When a comparison, below, is made between the results achieved with a moving reference frame propeller representation with a 40m permeable source surface, and a rotating mesh propeller approach, it can be seen that there is very good agreement between the predicted sound pressure levels in both cases. However the spectral shape of the rotating mesh approach is in general more realistic, and does not include an arbitrary tonal peak.

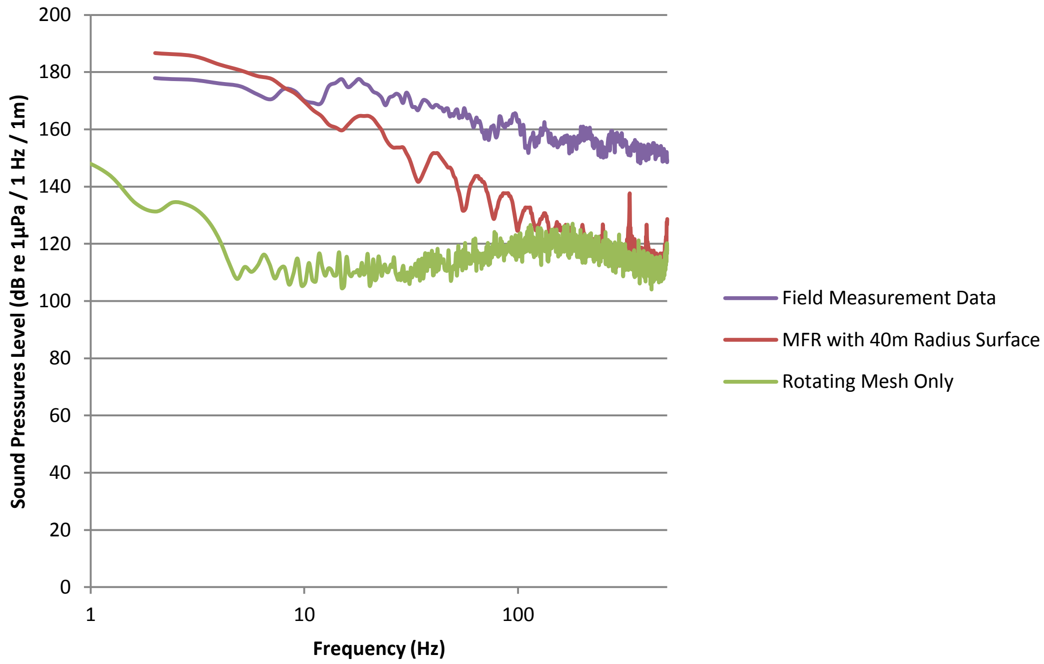


Figure 6.21 - Comparison of Rotating Mesh and Moving Frame of Reference with Permeable Surface at 19 knots, Port Receiver

As it was observed from the figure above that both variations gave very similar results in terms of sound pressure level, it was postulated that a combined rotating mesh propeller approach with a permeable source surface could produce more accurate results and therefore this was tested, as presented below. A variety of different permeable source surface radii were tested and the results are presented below. Permeable surfaces with radii between 30m and 60m were investigated.

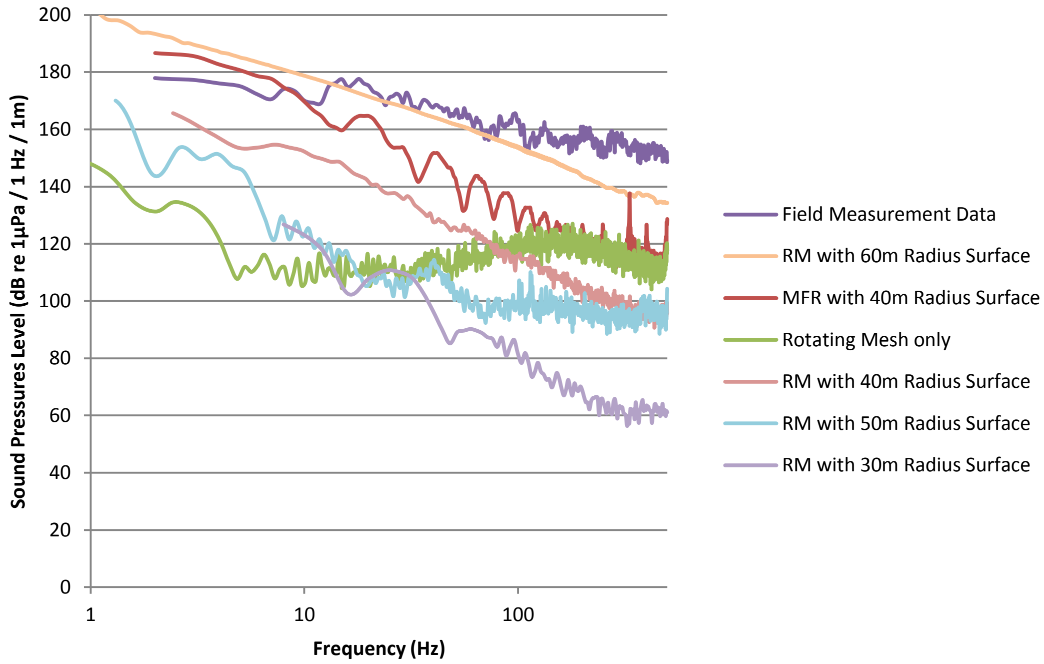


Figure 6.22 - Comparison of Rotating Mesh with Different Permeable Surfaces at 19 knots, Port Receiver

The results observed above generally support the earlier observations of the most reliable results being achieved with a 40m radius cylinder surface however the combination of the moving frame of reference and permeable surface still appears to be more suitable than a combination with a rotating mesh approach.

The most interesting results can be seen with a 60m radius. The dimensions of this surface meant that the port receiver was within the surface, while the starboard one was outside, as is usually the case. It can be seen in the figure below, showing the starboard receiver results for the same simulations as above, that this had a significant effect on both the predicted sound pressure levels and the spectral shape. Both the resistance and buoyancy values for the rotating mesh simulations with permeable surfaces were found to be extremely inaccurate in comparison to the field measurement data. This would suggest that the simulations were not providing an accurate model of reality. A general look at the results presented also suggests significant uncertainty in the application of this approach which would hence require additional investigation.

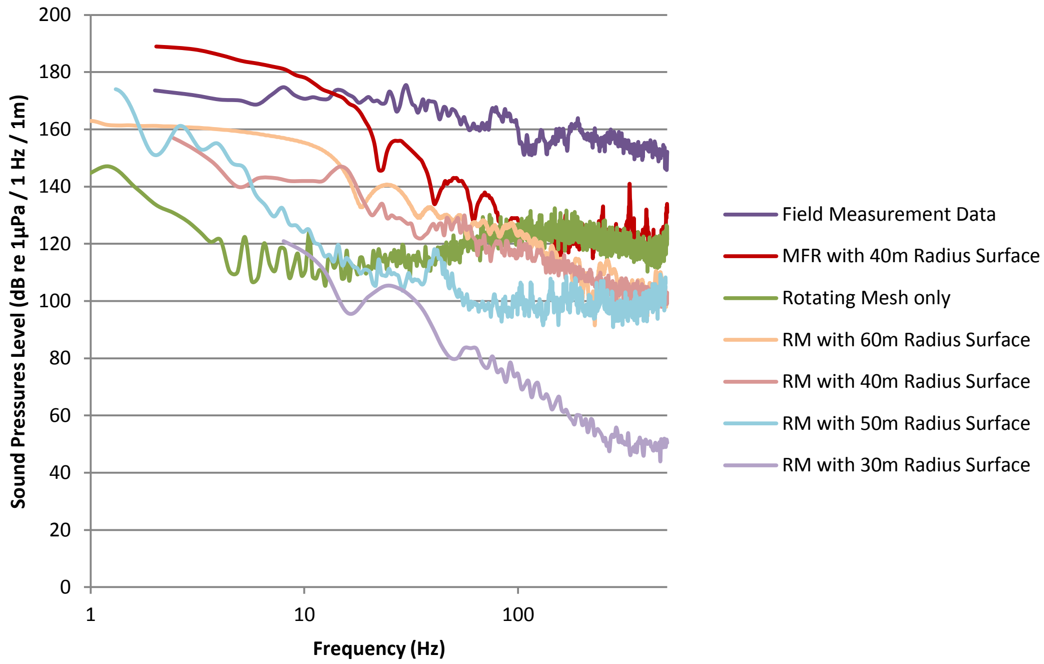


Figure 6.23 - Comparison of Rotating Mesh with Different Permeable Surfaces at 19 knots, Starboard Receiver

It can be observed from the above that the results achievable with the current simulation set-up and mesh, with a rotating mesh propeller representation and a permeable source surface, are not as accurate or reliable as those achieved by either a moving frame of reference propeller approach with a permeable source surface, or a rotating mesh approach alone. Great variation is observed in the predicted results for the different permeable surface radii with no apparent pattern. It is felt that this is more a product of the inaccuracies present in the current full scale simulation rather than a true reflection of the capabilities of a combined rotating mesh propeller approach with a permeable source surface. Both approaches are likely to perform much better where much more accurate simulation into and aft of the propeller can be achieved. This would require a much more detailed mesh which would need to be fully optimised to capture complex flow details especially in the vicinity of the propeller, and possibly also use a more accurate hydrodynamic approach than the current URANS method. In this case where the propeller wake field may play a more significant role, an LES or DES approach would be better able to capture these details. It is felt that in the case of this model, the inherent inaccuracies which arise from a full scale simulation of a real-life situation, and the intention of the model to be used in early stage design, means that the required improvements to enable this approach to function to its full potential are not merited.

6.10 Cavitation Simulation

This section will outline the preliminary work which was carried out to investigate the possibilities of using the built-in cavitation solver in StarCCM+ for the LNG Carrier to see whether this might be a viable option for cavitation noise prediction. The multiphase interaction optional model uses two Eulerian phases for the water and water vapour which will interact as described by the cavitation model, where water is the primary phase and water vapour is the secondary phase. The saturation pressure, seed density and seed diameter are prescribed by the user. In this case, the default values setting of seed density of 1.0×10^{12} per m^3 and seed diameter of 1.0×10^{-6} m have been used. A full discussion of the approach used within the software can be found in the StarCCM+ User Guide.

It is acknowledged that a full analysis and development of cavitation condition modelling and prediction would entail a study in its own right. For this reason, in this study predictions were carried out in open water conditions only. In these simulations the propeller was enclosed in 50 x 40 x 30 m domain, with a finer grid than previously. A time-step size of 0.0001 seconds was used, with a first-order unsteady implicit solution, and the simulation was run for a total of 0.5 s. The reason for the short run-times is due to the very high computational complexity and therefore simulation time. A maximum of 10 inner iterations was set for each time-step to allow the solution to converge more easily. The free surface is modelled as it was previously, as its proximity is very important to the generation of cavitation.
The simulations were run at both 19 and 9 knots as before, in order to enable a comparison. It is anticipated that at 9 knots, the level of cavitation will be very low, as this is at a speed close to CIS whereas at 19 knots, it is expected that the

propeller is in full cavitating condition. The results will not be a true reproduction of the actual conditions as the wake into the propeller from the vessel hull stern shape has not been recreated. The aim is simply to investigate whether such a simulation could be run with relative ease, to give results which could be realistic. It should be noted that no cavitation visualisations are available from field measurements for the LNG Carrier, therefore assessment of the results can only be done through comparisons between the predictions at the two speeds.

6.11.1 Cavitating Noise Prediction Results

The results for the cavitation noise propeller open water simulations at both 19 and 9 knots are presented below. Several different aspects of the simulation results will be presented and compared, namely pressure distributions, cavitation visualisations and acoustic results.

The figures below show the absolute pressure distribution on the propeller. The results could feasibly be realistic, as the areas of higher pressure are in suitable regions of the blades. It can be seen that the high pressure areas appear in similar locations on the blades however the magnitudes are lower at 9 knots than can be observed at 19 knots.

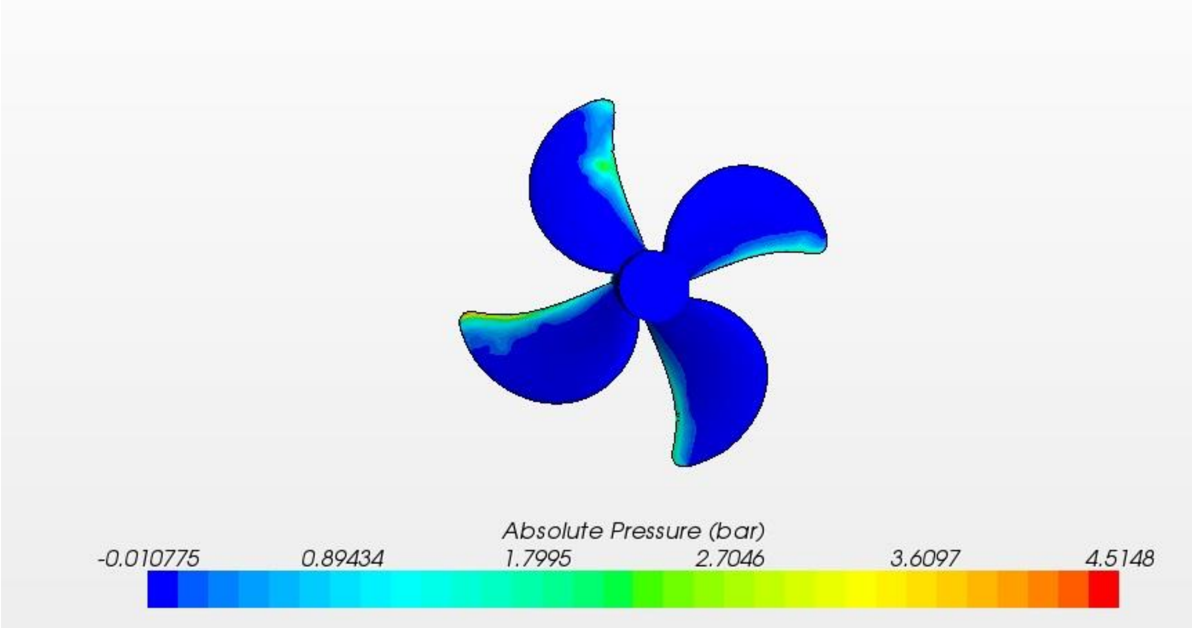


Figure 6.24 - Pressure Distribution on the Pressure Side of the Propeller at 19 knots

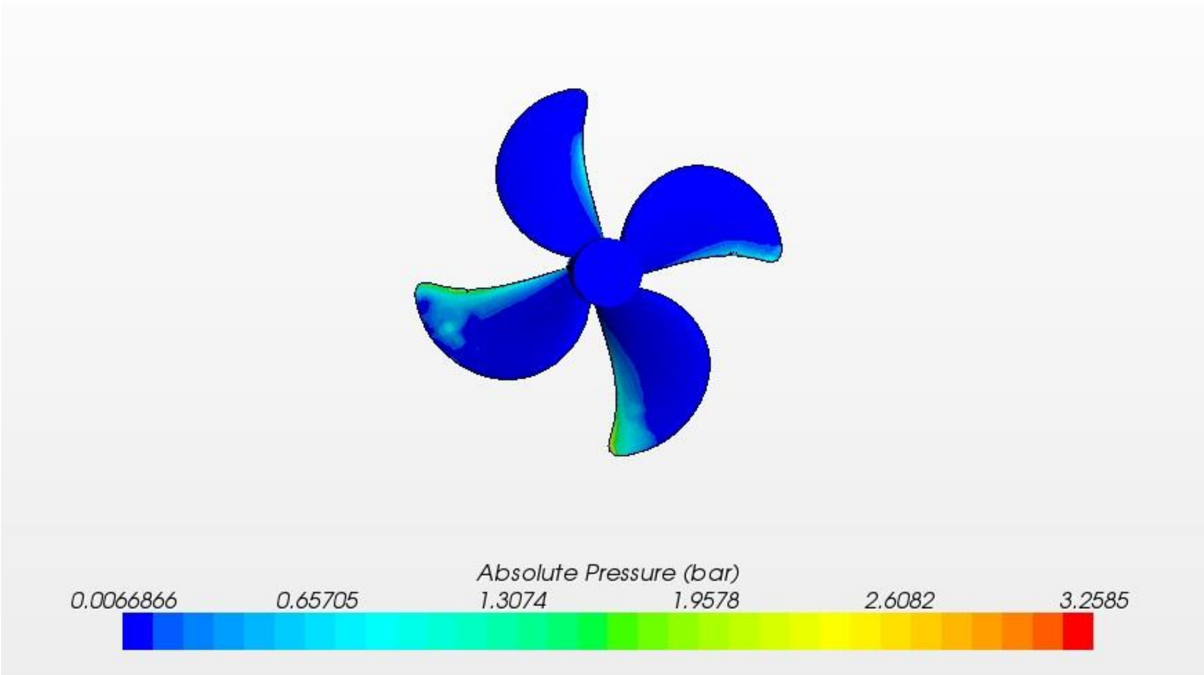


Figure 6.25 - Pressure Distribution on the Pressure Side of the Propeller at 9 knots

The figure below shows the predicted acoustic results for both the 19 and 9 knot simulations. The receivers were both placed in the same locations as were used above for the 19 knots non-cavitation simulations. It can be seen that the results for both speeds appear to indicate very similar acoustic levels, which is contradictory to the observations in field measurements where the addition of cavitation at 19 knots accounted for an increase in SPL of around 20 dB. Similar results were also observed in work on the same vessel carried out by SSPA (Hallander et al. 2012).

This observation together with the inconclusive visualisations would suggest that there are more significant errors to address in the simulations. Given the time demands and complexity of even these simplified cases, it would appear that such predictions are not practical for early stage design URN assessments in cavitating condition. It may be more suitable to propose a broadband increase factor or simple empirical formulae to predict the increase to the non-cavitating results when in a cavitation condition; either partially or fully cavitating. A great deal of further study is required in this area.

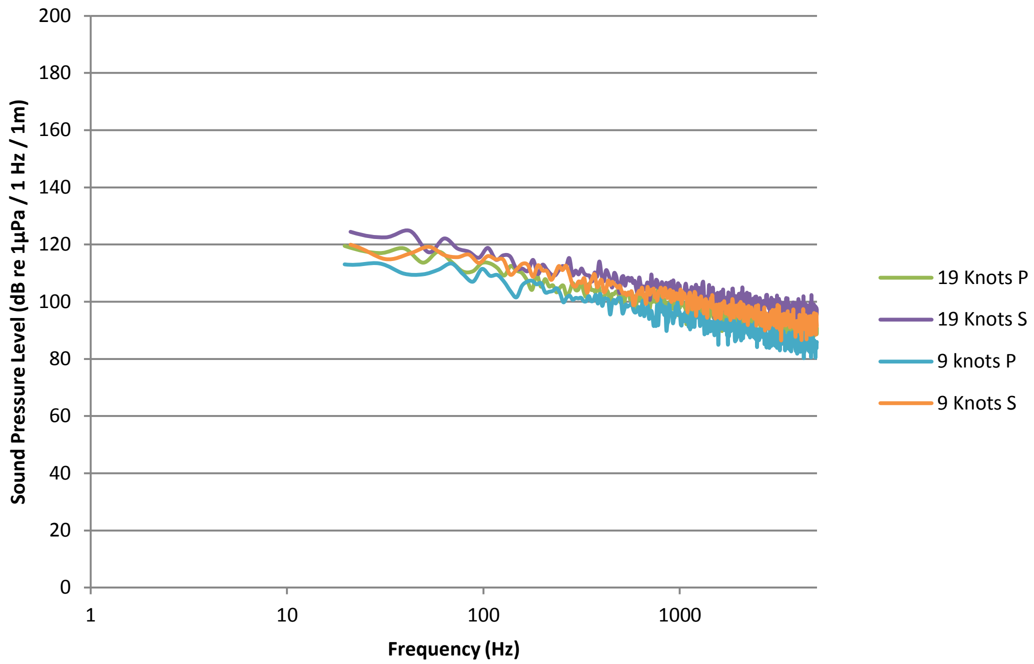


Figure 6.26 - Comparison of Acoustic Prediction Results for Open Water Propeller in Cavitation Condition

6.11 Machinery Noise

The following section will present the results for the analysis of the LNG Carrier machinery noise sources and tonal predictions, along with the empirical spectra predictions achieved using the data analysis tool.

6.11.1 LNG Carrier Machinery Noise Sources

As discussed in the previously, alongside the underwater noise measurements carried out on the focus LNG Carrier, concurrent onboard measured were carried, measuring both the noise and vibration levels, close to key machinery items as well as in a wide variety of internal spaces throughout the vessel. This work and its findings is fully discussed in (Zoet et al. 2010a).

These measurements which were also carried out at 19 and 9 knots identified some key onboard noise sources, as well as suggesting how different sources propagate through the hull structure. Some indication is also given regarding how the radiation from the different sources may be affected by different vessel speeds and operational conditions. The main sources will be briefly discussed below, at the two main speeds under investigation. It is hoped that this succinct review will reinforce and justify the choice of key onboard noise sources identified above, which will form the basis of the predictions carried out.

In terms of general overall vibration levels, measured at suitable locations for these machinery items using an accelerometer, the most dominant sources appeared to be:

- The Propeller (measured on adjacent inboard hull plating fields)
- The Main Steam Turbines (measured at the turbine feet and foundations)
- The Gearbox (measured at the foundations)
- The Auxiliary Sets (measured at the foundations)
- The Feed Water Pumps (measured at the foundations)

The sections below will look more specifically at which sources can apparently be observed from the tonal noises present in the underwater noise spectra at 19 and 9 knots speed. The results when the ship was stationary will also be considered briefly.

It should be noted that although significant amounts of noise onboard can be attributed to equipment such as the Heating, Ventilation and Air Conditioning (HVAC) system, these sounds do not appear to propagate into the water, as the noise is lost in transmission though the ship, and so they will not be considered within this work.

The observations highlighted below are more thoroughly discussed in (Zoet et al. 2010b). It also demonstrates the difficulties associated in accurately dissecting underwater ship noise spectra into its constituent parts.

19 Knots

As stated previously, the main propulsion for this vessel is delivered by two steam turbines rather than diesel engines which are much more commonly used onboard large commercial vessels. Steam turbines are well known for being significantly quieter than the more conventional diesel engines, and this is reflected in the observations made. In general, the majority of the tonal peaks observed were deemed to be linked to the propeller blade rate frequency and harmonics. Above 2000Hz, the pattern of the spectra also appears to show links with the steam turbine rotation frequencies.

Some correlation can also be seen between tonal peak frequencies in the lower frequency range of the spectrum, and propeller / turbine shaft rotation frequencies.

9 Knots

At this speed, as at 19 knots, the majority of the tonal peaks are linked to the propeller blade rate frequency. Between 100 and 4000Hz, distinctive peaks associated with the steam turbine rotation frequencies could also be observed, however above 4000Hz, this association appeared more loosely as an influence on the general spectral pattern. It is assumed that at this lower speed, cavitation and turbulence is less dominant and does not mask these other sources so significantly.

As well as these more dominant sources, other sources that could be responsible for tonal peaks observed in the lower frequency section of the spectrum include the propeller shaft and turbine shaft rotation, and gear meshing at the 1st and 2nd stages. However it should be noted that it is very difficult to accurately assign different tonal peaks to different sources, even where such a depth of measurement data, both underwater and simultaneously onboard, is available.

Ship at Zero Speed

When the ship is stationary, the propeller will not be turning, and there will be negligible flow noise, so in theory, the remaining acoustic contribution should be mostly attributed to the onboard machinery noise. Figure 6.27 below shows the ship still source level and receiver level (as measured) spectra. The ambient noise contribution has been removed from the results. It should be noted that when the propeller is not operating, and therefore not loaded, the loading on the main engine will also be different from normal operating conditions, and hence the acoustic characteristics in the ship still condition also differ slightly from those which are likely at 9 and 19 knots.

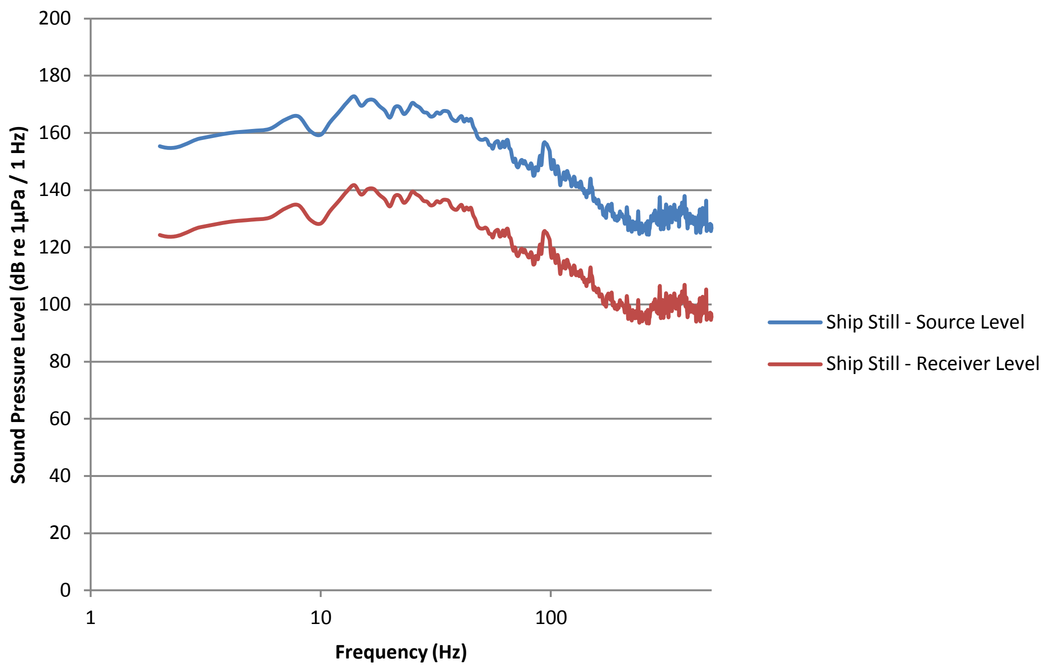


Figure 6.27 - Ship at Zero Speed Underwater Noise Source and Receiver Level Spectra for LNG Carrier

Although, as discussed, the differences in loading will vary these acoustic properties, this data can still give an indication of an approximate level of machinery noise contribution to the underwater source level spectra of the vessel. The lack of noticeable tonal components is again highlighted for the particular case of this LNG carrier: it would be expected that for a diesel engine installation, there would still be tonal peaks clearly visible at key frequencies.

6.11.2 Machinery Tonal Prediction

As discussed previously, in the particular case of the LNG Carrier, the steam turbine main propulsion system means that the main propulsor is not a dominant source of machinery noise tonals, and in fact its contribution to the underwater radiated noise spectrum is almost negligible. At 19 knots in particular, the underwater noise spectrum is dominated by cavitation noise and few tonal peaks can be observed. There are only a few significant tonal peaks at 19 as can be seen in the figure below, and based on the predicted tonal frequencies for the main machinery equipment, may be attributable to the indicated sources, based on visual comparison:

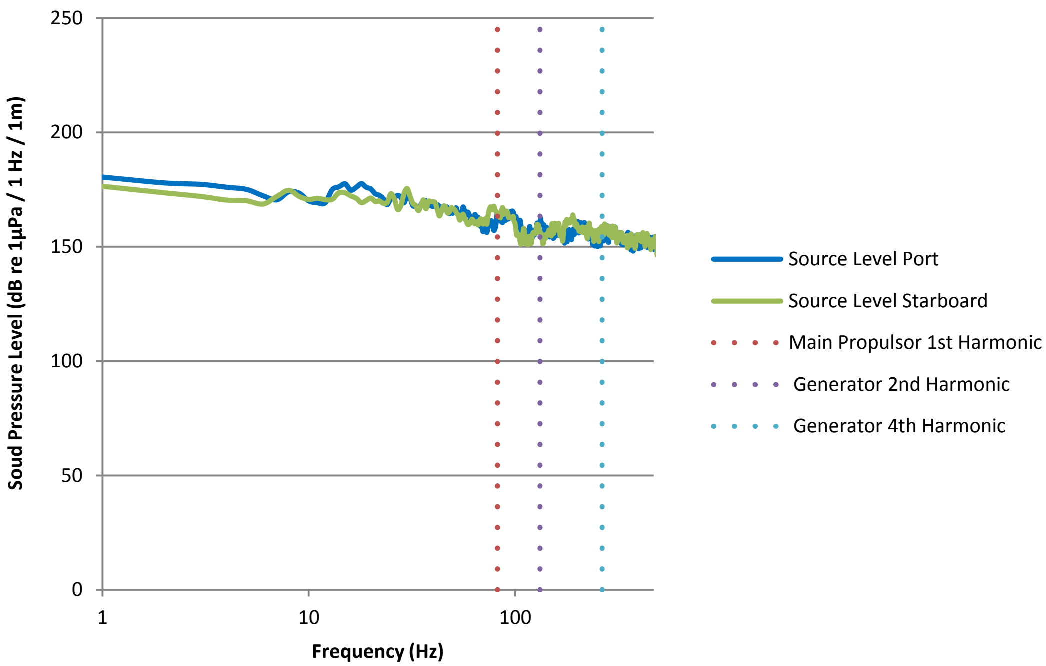


Figure 6.28 - Potential Machinery Tonal Sources for LNG Carrier at 19 knots

At 9 knots, there are some more distinct tonal peaks visible, and some of these may be attributable to onboard machinery sources as shown in the figure below. However there are also peaks which appear on only one side of the vessel, and so not appear to correspond with the main machinery tonal frequencies. They may in fact be related to the measurement activities and be an incidental noise, but as only one run was carried out for each set of data, this cannot be clarified. The analysis tool suggests the following potential sources at this lower speed, again based on visual comparison:

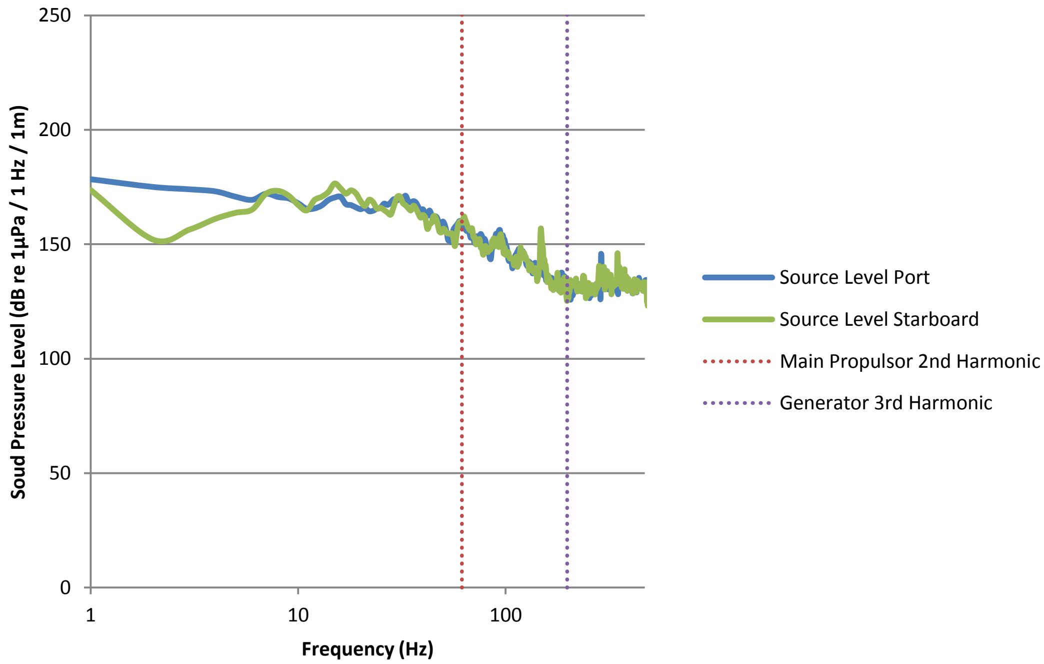


Figure 6.29 - Potential Machinery Tonal Sources for LNG Carrier at 9 knots

6.11.3 Empirical Spectra Prediction

A comparison of the empirical spectra prediction approaches which were tested and the measured LNG Carrier results is presented below for both 19 and 9 knots. At 19 knots, the results of the two Ross approaches are almost identical, hence why the second line does not appear to be visible. It should be noted that the LNG Carrier values presented below are source level spectra, i.e. the levels corrected to 1m from the source, as the empirical prediction also provides a source level estimate, and hence they appear higher than other displayed values, as they have been corrected from the as measured data gathered at receivers.

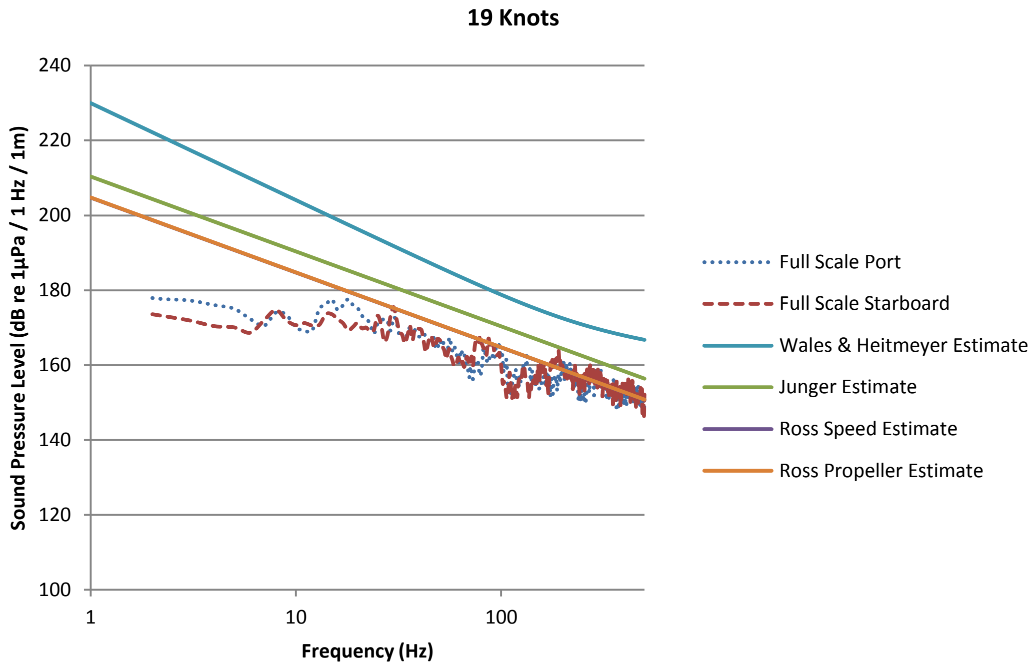


Figure 6.30 - Comparison of Field Measurement Results and Estimates at 19 knots

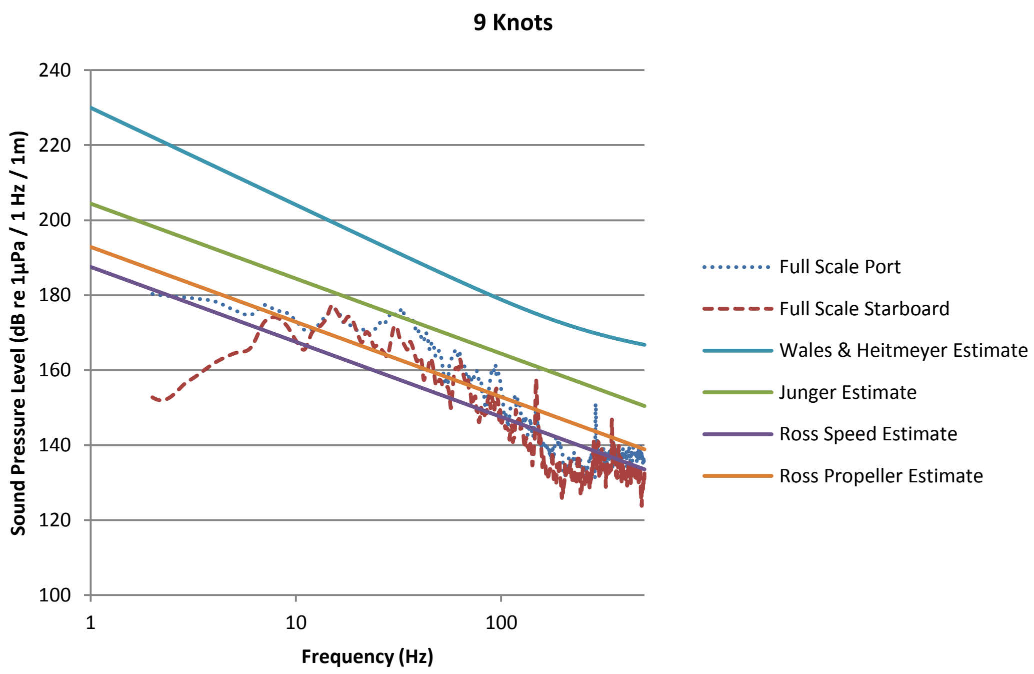


Figure 6.31 - Comparison of Field Measurement Results and Estimates at 9 knots

6.12 Chapter Summary

This chapter has presented details of the numerical modelling prediction results which were achieved for the LNG Carrier. Focus was on the prediction of non-cavitation propeller and hydrodynamic noise however some investigation of cavitation noise was also presented. A discussion on the identified main noise sources on board the LNG Carrier based on onboard noise measurements was then presented. Finally, some results arising from use of the tonal frequency prediction approach were presented. Likely sources of observed tonal peaks in the measured spectra for the LNG Carrier were identified.

The next chapter will present a discussion of potential impacts of URN on marine wildlife, with a case study to indicate how data compiled into the data analysis tool may be used in impact assessment.

Chapter 7 - Marine Wildlife Impact Assessment

7.1 Chapter Overview

This chapter will discuss the data which has been compiled in a database for a large range of marine wildlife species and which is intended to be used in assessing the potential impact of ship radiated underwater noise on these species (§7.2). It will then outline the different ways in which the impact assessment could be conducted (§7.3) before providing a case study of a LNG Carrier by way of an impact assessment example using the data compiled (§7.4).

7.2 Marine Wildlife Species Data

The aim of this chapter is to provide a form of assessment for whether or not the predicted radiated underwater noise of the commercial vessel in question is "acceptable" or not to the environment in which it is intended to operate. As the key subjects affected by these anthropogenic noise sources are the marine wildlife species residing in the specified operational area of the vessel, this assessment will focus on the potential impacts the measured or calculated ship noise may have.

As discussed at length in Chapter 3, it is difficult to specify a particular "acceptable" or "unacceptable" noise with relation to marine wildlife. Instead, species are affected differently by variations in noise levels, frequencies and noise source proximities. These depend on the noise levels to which they are habituated; their hearing range, their vocalizations and use of noise in everyday life, and can also vary for different times of the day or year, for example if it coincides with a mating or migration period. Therefore, it would be unsuitable to assess the noise impact as a blanket effect for all species by imposing a single threshold limit. Instead, it's likely impact on all relevant species for the selected operational area will be assessed individually.

The aim of this work is not to provide mitigation measures to address any issues identified in the predicted or measured ship spectra. The potential problems will simply be highlighted, and it will be left to the designer to determine the most suitable techniques for addressing it.

The sections below will discuss what data relating to the marine wildlife species has been used to assess these potential impacts and where this has been sourced from. They will also present the methods by which this assessment will be carried out, and any specific regulations which may need to be included. The results of an assessment for the LNG Carrier which has formed the focus of this study will be provided. Finally, some information on how this section of the model operates will be given.

As discussed previously, it is also likely that in the near future, restrictions and requirements will be put in place, either as general noise levels, or for the protection of specifically vulnerable areas, and specifically Marine Protected Areas (MPA's). Such approaches have previously been employed by the IMO and other international bodies in the regulation of other pollution sources such as emissions and ballast water. There will therefore be a need in the future for the predicted noise to be assessed with respect to these limits and requirements as well.

There are several ways in which the impacts of the predicted underwater noise spectra could be assessed, as listed below:

- Habitat and potential habituation to anthropogenic noise, using prior knowledge of different global locations
- Conservation status and hence assumed vulnerability of the species
- Known or predicted hearing range of species, and where known, key sensitive frequencies
- Potential use of noise by species during everyday life, and its assumed biological significance
- Known vocalisation range and key frequencies, where applicable
- Recorded observed reactions of the species to known levels and frequencies of underwater noise

In order to allow information relevant to the above points to be easily accessed, a database has been created which within the data analysis tool discussed previously in Chapter 5. This records the following data for a range of key wildlife species, including Cetaceans, Pinnipeds, Fish and a few other important species:

- Species - Common Name
- Species - Latin Name
- Species Sub-Order
- Species Order
- Species Type
- Number of Sub-Species, where applicable
- Habitats by Region
- Official Conservation Status
- Known or Predicted Hearing Range (Hz), using a lower and upper limit
- Critical Ratio (dB re 1µPa)
- Does the Species Vocalise?
- Full Vocalisation Range (Hz), using a lower and upper limit
- Average Vocalisation Sound Level (dB re 1µPa)
- Vocalisation Details

- Known or Observed Reactions to Noise
- Comments on the data provided

The data contained within the database could be periodically updated as new or more accurate data becomes available. It should be noted that there are also many marine wildlife species not currently listed in the database. This is mainly due to a lack of available and competent data for these species, for a variety of reasons. Some species such as the baleen whales are very large and elusive and therefore difficult to study, while other non-commercial species of fish etc also tend to be less studied. As more data becomes available, or is sourced, more species could be added, to ensure a comprehensive database is available for impact assessment.

Although current data availability does not allow for it, another useful item of information would be the typical swim or dive depth of the species, as this will also have an influence on the sound that they receive. For instance, those species more typically found near the surface will tend to receive much more noise from shipping than those which spend most of their time much deeper. It should be noted that this is not a hard and fast rule, as deep water channels can increase the propagation of noise at depth, by channelling them for great distances with little attenuation. This was discussed in Chapter 3 (§3.5.1). The methods for obtaining the data currently contained in the data analysis tool database, and the strength of it has been discussed in Chapter 3, and the references used can be found in the References section.

A good source of data on which species of marine wildlife are likely to be found in the territorial waters of EU member states, as well as additional information on their population size and conservation status, is the online database created by the European Topic Centre on Biological Diversity (2001) (EIONET). This information was collated to comply with the requirements of Article 17 of the Habitats Directive on all EU member states (Council of the European Communities 1992). All Member

States are requested by the Habitats Directive to monitor habitat types and species considered to be of Community interest. Article 17 of the Directive requires that every 6 years Member States prepare reports to be sent to the European Commission on their implementation of the Directive. The Article 17 report for the period 2001-2006 for the first time included assessments on the conservation status of the habitat types and species of Community interest. Article 11 of the Marine Strategy Framework Directive (European Parliament and Council 2008) requires that "Introduction of energy, including underwater noise, is at levels that do not adversely affect the marine environment" as a measure of Good Environmental Status (GES) to be demonstrated by each member state by 2020. Governments will also therefore be required to gather evidence and data to support their reporting with relation to this requirement, and such data, if publically available, will again be an invaluable addition to the current body of information available.

7.2.1 Habitats and Vulnerability

The vulnerability and previous habituation to underwater noise will greatly affect how a species in a particular habitat is likely to be impacted by ship radiated noise. The same level of noise would cause significantly more harm to those species resident in polar waters and other relatively unindustrialised areas, who may have had little or no previous expose to anthropogenic noise, than to those which reside permanently near shipping lanes and ports and may have lived their whole lives in "noisy" conditions. The data analysis tool allows the user to automatically filter the marine wildlife database, to show only those species listed as residing in the selected Operational Area location. The selected operational area for the ship is important not only to select the potentially affected species, but also to provide some suggestion of their likely vulnerability. This kind of information, while easily available in published marine biology papers and from data on typical ambient noise levels, is difficult to record in database format. This is because it would not be possible to specify the vulnerability of a species for several different habitats where a species is wide-ranging or migratory. Therefore the user will be required to use their own judgement in assessing the vulnerability of species in a given habitat. For example, if the operational area of interest for the vessel is known to be relatively unindustrialised, the user should be aware that the information provided for a species may not be suitably conservative.

The Conservation Status, by risk of extinction, of each species within the database has also been recorded, based on the International Union for Conservation of Nature (IUCN) Red List of Threatened Species (IUCN 2001) and other sources. The different categories and their meanings, as used by the IUCN, are listed below:

- **Extinct**
  - *Extinct (EX)* - A taxon is Extinct when there is no reasonable doubt that the last individual has died.
  - *Extinct in the Wild (EW)* - A taxon is Extinct in the Wild when it is known only to survive in cultivation, in captivity or as a naturalized population (or populations) well outside the past range.

- **Threatened**
  - *Critically Endangered (CE)* - A taxon is Critically Endangered when the best available evidence indicates that it meets any of the criteria A to E for Critically Endangered and it is therefore considered to be facing an extremely high risk of extinction in the wild. For these criteria, please see the above reference.
  - *Endangered (EN)* - A taxon is Endangered when the best available evidence indicates that it meets any of the criteria A to E for Endangered (see Section V), and it is therefore considered to be facing a very high risk of extinction in the wild. For these criteria, please see the above reference.
  - *Vulnerable (VU)* - A taxon is Vulnerable when the best available evidence indicates that it meets any of the criteria A to E for Vulnerable (see Section V), and it is therefore considered to be facing a high risk of extinction in the wild. For these criteria, please see the above reference.

- **At Lower Risk**
  - ♦ *Near Threatened (NT)* - A taxon is Near Threatened when it has been evaluated against the criteria but does not qualify for Critically Endangered, Endangered or Vulnerable now, but is close to qualifying for or is likely to qualify for a threatened category in the near future.
  - ♦ *Least Concern (LC)* - A taxon is Least Concern when it has been evaluated against the criteria and does not qualify for Critically Endangered, Endangered, Vulnerable or Near Threatened. Widespread and abundant taxa are included in this category.

- **Other**
  - ♦ *Data Deficient (DD)* - A taxon is Data Deficient when there is inadequate information to make a direct, or indirect, assessment of its risk of extinction based on its distribution and/or population status. A taxon in this category may be well studied, and its biology well known, but appropriate data on abundance and/or distribution are lacking. Data Deficient is therefore not a category of threat.
  - ♦ *Not Evaluated (NE)* - A taxon is Not Evaluated when it is has not yet been evaluated against the criteria.

As these categories are internationally recognised, and many regulations and directives worldwide already aim to address issues to improve circumstances for all categories of threat to ensure no further damage is done to these species, it is possible that future underwater noise regulations will be based on these categories. This data has therefore been provided both to indicate the severity of having a negative impact on different species in the area, and for use in conjunction with future regulations. It has been assumed here that those species within the "Threatened" category are specifically considered regarding their potential response to ship radiated noise. However, as has been discussed previously, there are some species in particular cetaceans who may be a high risk of impact, but for whom insufficient data is available to specify a conservation status. These species should not be disregarded during impact assessments. A level of user discretion is again required. The model provides an option to highlight the filtered database, based on the different Conservation Status categories. It should be noted that data for species which are considered to be "Extinct" or "Extinct in the Wild" are not included in the database.

7.2.2 Vocalisation and Hearing Range

Recording data on a species' hearing and, where appropriate, their vocalisation frequency ranges, is very important as it can give a good indication of whether or not the species is likely to be affected by a given ship underwater radiated noise spectra, in the frequency range under consideration. However this information especially on species audiograms, can be very difficult to obtain, as was discussed in Chapter 3 (§3.7). In order to obtain an accurate species audiogram, a range of data for a relatively large number of individuals from a species needs to be obtained, using both behavioural testing and Auditory Brainstem Response (ABR) approaches. Otherwise, the results can be skewed by outside influences such as background noise in the testing area, whether or not the individual grew up in captivity, and any previous hearing damage or habituation to noise.

Both the hearing and vocalisation range data will be used within the data analysis tool to indicate whether or not the species is likely to be affected by the predicted noise spectra. Where the focus frequency range of the predicted spectra lies within either the hearing or vocalisation ranges, or both, it will be assumed that some level of impact, whether neutral or negative, will occur. The data analysis tool allows the user to highlight the species in the database for which the hearing and/or vocalisation ranges coincide with the frequency range of the ship noise spectra under consideration.

The frequencies, both broadband and of tonal peaks, and their associated level could be used to predict whether the species will experience masking effects of biologically important sounds, such as communication from other members of the species, noise from predators or prey, or any noise that may help in navigation and positioning. The vocalisation range and average sound pressure level will also be used to indicate whether or not the species is likely to demonstrate avoidance behaviours. The average vocalisation level will give some indication of the level of noise which is deemed normal or tolerable, and therefore comparison with this will assist in providing an "acceptable" and "unacceptable" sound pressure level range (dB re 1µPa) for that species. Where the critical ratio was available for a species, this can be used to define the level at which the species will hear a given noise, and will hence improve the impact prediction.

This data has been collected from a wide range of published literature, which details both experimentally obtained data, and data implied from observed reactions and other known information such as recorded vocalisation. Where hearing range data was not available for a given species, it was assumed that this would correspond with the vocalisation range, where this is known. Where neither range was available, the values are simply listed as "Unknown" as it would not be suitable to make assumptions which could lead to a species being wrongly disregarded.

7.2.3 Observations and Reactions

Where data on reactions, either from experiment or observation, of species to known underwater noise sources is available, this could be used to suggest where similar reactions may be expected for some predicted noise spectra. Some observations are presented in papers such as (Simmonds et al. 2004) and (Weilgart 2007).

It was noted during the research for this data that some shortcomings exist in the information recorded and published in this field. While most publications and papers provide detailed observations of the reactions of a wide variety of species to all manner of underwater anthropogenic noise sources, many of these provide limited or no information regarding the source frequency(s), sound pressure level, and distance from the animal (Nowacek et al. 2007). The majority also provide no details of the water and weather conditions, and a few fail to state the location at which the observation has been made. Where this information has not been recorded, the associated reaction and observations have been recorded however are unlikely to be of significant use within this work, as it can be difficult to make any sensible assumptions about what they may indicate about a species. This highlights a wider need for regulation of the methods by which data is recorded during observations and experiments, and the way in which it should be presented, for wider inter-disciplinary use. This was discussed previously in Chapter 3.

7.3 Assessing Underwater Noise Impact

It was noted in the Literature Review Chapter that existing suggested noise threshold limits are more typically based on what is felt is possible to be achieved by existing technology, or in relation to a very specific species. These limits tend to be in the form of a maximum allowable sound pressure level (dB re 1μPa) across the whole frequency range of ship noise. However, it is known that not all species will be affected by all frequencies, at these specified levels. It would appear more prudent to address only those high levels which are at frequencies known to be harmful to a specific species, meaning that excessive penalties are less likely to be imposed on the vessel in trying to address the acoustic characteristics. Altering underwater noise spectra can be very expensive, and even unrealistic in some areas, therefore it would appear to make economic sense to address only those areas which can be justifiably assumed to cause harm. Also, as most of the suggested limits are based on perceived technological capability rather than in relation to marine wildlife species, the limits used are not in fact guaranteed to address the problem fully.

In some instances, more detailed impact assessment methodologies have been designed, chiefly by practitioners in the marine biology field, to account for the potential impacts of anthropogenic noise on marine wildlife. In (Wright 2009), this is done by including noise as an additional stress driver amongst other stressors such as pollution and reductions in prey, in cumulative terms both for individuals and for populations. Meanwhile, in Clark et al. (2009), masking impacts in particular are investigated, using a model which aims to predict the reduction in an animal's effective communication space in response to the presence of a ship operating in the area. Furthermore, in Halpern et al. (2008), mapping of cumulative impacts from different causes have been mapped to help identify at-risk regions. The data, both in the shape of predicted ship underwater radiated noise spectra and of species present in the operational area, together with their hearing and vocalisation characteristics, provided by this model could be very suitable for use in such models for more details impact analysis. However it should be noted that they have not been developed from an Engineering standpoint and therefore some of the requirements may be very challenging for a designer to achieve, or may be deemed excessive given the costs versus the benefits.

Conversely, where rules and regulations for ship underwater radiated noise exist, either currently or in the future, they will have to be complied with. Designers tend to find such prescriptive requirements easier to deal with, as the vessel is either compliant or it isn't and the eventual goal is well defined. However such approaches can be restrictive for the designer, and can curb innovation and technological advancement into new areas and previously untried approaches.

The following sections will discuss how an impact assessment could be carried out using the data discussed in the section above. This could be based on either predicted biological impacts, on recommended noise limits for vessels, or on a more goals-based approach. The author feels that a goals-based approach is particularly well suited to this problem, and could provide a very interesting opportunity to the marine industry. There are still significant unknowns on both the sides of engineering and marine biology, and striving for solutions which require multi-disciplinary collaboration and innovation could benefit and further the knowledge on both sides. All three approaches will be covered briefly in the following sections.

A statistically-based impact assessment, whereby the probable percentage of species or members of a particular species affected could be predicted, could also be an appropriate means of demonstrating impact from a given vessel. This approach is however not taken here as the data used and currently available is not sufficiently detailed to allow the required analysis to be carried out.

7.3.1 Biologically-Based Impact Assessment

As stated above, one basis for impact assessment can be the biological implications of ship underwater radiated noise. The sections below will briefly outline how the data presented in the data analysis model could be used for these purposes.

Masking Effects

Masking occurs when a source of noise at a similar frequency and higher sound pressure level "hides" another sound, which may be of importance to the survival of the species in question. It may either mask the important noise completely, or significantly reduce the range over which it can be heard, both of which can have a negative impact on the affected species. Although the longer term and population implications are not well understood, it could be assumed that reduced social cohesion, loss of navigational information and predator / prey detection could lead to isolation and reduction in feeding efficiency.

For the purposes of the impact assessment, the user will need to know whether or not masking is likely, and where this is the case, how severe this is likely to be. This would give an indication of how much either the sound pressure level at the problematic frequencies may need to be reduced, or whether these frequencies should be avoided entirely if possible through speed changes and structural tuning. Using the option to highlight the database for hearing and vocalisation frequencies, the user could easily identify the species which may need to be considered. The known reactions information may also be able to indicate assumed occurrences of masking for a particular species.

Avoidance Effects

Avoidance behaviours are where the marine animals avoid a given area, either by relocating to a different region for a period of time or taking a different route if the area lies on a known migration route. Behaviours such as swimming away from a noise source at high speed is also considered to be avoidance. The timescales associated with this avoidance will vary depending on the species and characteristics of the noise source in question. In some cases, this avoidance will simply mean that they maintain a greater distance from the noise source, whereas in others they will vacate the area entirely while the noise is active, and in rare cases, they may avoid the area entirely for weeks or even months, despite cessation of the noise source. The potential impacts which this avoidance behaviour can have on individuals and populations are discussed in the Literature Review, but it can be assumed that avoidance of an area which may be important for feeding, mating, breeding or migration to different habitats could again have longer-term impacts on a population group.

Given the complexity of the many factors affecting the behaviour of marine wildlife in different situations, it would not be possible to reliably state how they will behave in reaction to the predicted ship radiated underwater noise spectra. However, where observations and experimental data exist for a given species which demonstrate avoidance behaviour to given noise properties, and these could be used as an indicator for similar behaviour occurring. No prediction could be made for the timescales and severity of the avoidance behaviour, or the likely short and long term effects on individuals or populations using solely the information provided in the data analysis tool database.

Other Potential Impacts

As discussed in the Literature Review Chapter, there are a number of other impacts known to occur as a result of ship radiated underwater noise. These behavioural changes include variations in diving durations and depths, which have the potential to affect both breathing patterns and feeding efficiency. Other effects such as different swimming, vocalization and group dynamic behaviours have also been observed and may need to be considered. Where observations of such impacts for given species have been recorded, this has been presented in the Known Reactions data. This could be used to suggest the possibility of similar behaviours to comparable sound inputs.

To provide a more general overview, the user will be given the option to compare the spectral sound pressure levels to the US National Marine Fisheries Service (NMFS) suggested threshold level for behavioural disturbance in Cetaceans and Pinnipeds. This value of 120 dB re 1Pa for continuous sounds was suggested in (Southall et al. 2007). This limit arose from detailed review of the literature by an expert panel on the subject however it should be used as an indication only and does not necessarily mean that no impacts will occur for noise levels which are slightly under the limit. This can be done by selecting the appropriate tick box in the Wildlife Impact section of the data analysis tool. The suggested limit will then be displayed on a results graph.

In some more extreme cases, in sensitive species or at high sound pressure levels, the occurrence of Temporary Threshold Shift (TTS) or Permanent Threshold Shift (PTS) of hearing sensitivity has been noted. Where observation or experimental data on TTS and PTS for the species in question has been recorded, this will be presented in the database. For a general overview, there user will again be given the option to use the NMFS threshold sound pressure level limits, (Southall et al. 2007), which are presented in the table below. This will again be presented as an additional limit on the results graphs, activated using the appropriate tick box.

Table 7.1 - Proposed TTS and PTS Limits from NMFS

Proposed Limit	Limit for
224 dB re 1µPa	Temporary Threshold Shift (TTS) in Cetaceans
212 dB re 1µPa	Temporary Threshold Shift (TTS) in Pinnipeds
230 dB re 1µPa	Permanent Threshold Shift (PTS) in Cetaceans
218 dB re 1µPa	Permanent Threshold Shift (PTS) in Pinnipeds

It should be noted that in reality, it is not just the peak sound pressure level which should be considered for impact, but also the energy content associated with it. For example, a dolphin may emit an echolocation click at 219 dB re 1µPa, however it will only contain around 0.02 Joules of energy, whereas a transponder emits at around 194 dB re 1µPa with 0.8 Joules associated with it. In terms of shipping noise, prolonged exposure to lower noise levels is also likely to be the cause of any eventual hearing damage, rather than a shorter period of much higher noise pressure. For more detailed impact assessments, exposure time to the noise source, or combination of sources should also be account for, however this will not be dealt with in this work. These should of course also be accounted for in relation to the ambient noise levels already experienced.

7.3.2 Rules-Based Impact Assessment

As discussed previously, there is a high prospect of mandatory rules and regulations concerning the radiation of underwater noise of commercial ships being developed and imposed in the future. As these are published and come into action, the requirements which relate to noise levels or particular species can be added to the model. The performance of the vessel under investigation in compliance with these rules could then be included in the assessment. Any requirements or regulations relating to speed limits, or avoidance of certain areas, could be highlighted in future versions of the model.

The data analysis tool will also give the user an option to compare their predicted ship noise spectra against existing suggested noise limits, such as the Green Label limits proposed in the SILENV Project (2012a), the optional DNV (Det Norske Veritas) Silent Vessel Notation Environmental (E) category (DNV 2010), and the ICES (International Council for the Exploration of the Seas) mandatory limits for Fisheries Research Vessels (Mitson 1995). Full details of these limits and their use are provided in Chapter 3. This can be carried out by selecting the appropriate tick box in the Wildlife Impact section, and the limits will be displayed as additional lines in the results graphs, as for the biological limits. These limits will give the user an indication of how their vessel noise output compares to vessels specifically designed for quiet operation. While the ICES limit in particular is not designed to be applied to commercial vessels, it was designed with impacts on fish as a basis and can therefore give an indication of what might be required for a vessel aiming for very low impact. It is also important to understand that non-compliance may not be a problem, as occasional low levels of startle response and localised avoidance in some species may be found to have negligible long-term impact on the species and therefore designing a vessel to avoid this may be uneconomical. As more suggested threshold limits, non-mandatory for commercial vessels, are published, these could also be added as options for inclusion in the assessment.

7.3.3 Goals-Based Impact Assessment

Goals-based designs and standards are a relatively new idea however they are already seeing application in ship structural design regulations, and in relation to ship safety. It is defined by the International Maritime Organisation (IMO) (Hoppe 2001) as below:

"Goal-based regulation" does not specify the means of achieving compliance but set goals that allow alternative ways of achieving compliance. For instance "People shall be prevented from falling over a cliff" is goal-based. In prescriptive regulation

the specific means of achieving compliance is mandated, e.g. "You shall install a 1 metre high rail at the edge of the cliff"

Given the complexity of the problem regarding ship radiated noise, and anthropogenic noise in general, and the many potential impacts it could have on a wide variety of targets, the author feels that this problem is remarkably well suited to governance by goals-based standards. The main goal in this case is to minimise wherever possible the impact of the ships radiated noise on the marine environment, and the marine wildlife which inhabits it.

This approach may be more time-consuming for designers, who will have to justify how their design is suitable by proving compliance rather than installing a required part or achieving a required value in a formula. However, rules-based standards tend to only be updated in reaction to a negative occurrence rather constantly seeking a better solution. They can also lead to complacency amongst designers, and as discussed before, many of the threshold limits currently in existence for underwater noise are not based on species reactions to noise and have not necessarily been proven to address the problem. It should however be noted that based on discussion with shipbuilders and ship owners, they would prefer to be required to comply with specific regulations, as this gives them an idea of what the design is aiming for. It is also simpler to justify expenditure to reach a known goal. Nevertheless, the author feels that the use of goals-based approaches is the most appropriate. In this way, the designer would be able to demonstrate reasonable noise mitigation measures given the likely impacts. Application of specific rules may lead to unsuitable or unnecessary expenditure on mitigation measures for noise levels which would not in fact be particularly problematic for that particular vessel in its operational area.

In order to provide the data analysis tool user with some form of justification for goals-based standards for underwater noise, it is envisaged that the data contained within the marine wildlife database, and the assessment options available will ensure that the model output is suitable for this purpose. The user should be able to demonstrate through either a series of assessments spanning the design development stages, or using the final assessment and some discussion, that all possible steps have been taken to assess the predicted spectra against the marine wildlife species likely to be affected and wherever possible, potential negative impacts have been minimised or eliminated entirely.

Although this could be potentially more time-consuming for the designer, and may require greater levels of communication with the regulating body, the saving which should result, from addressing only those areas which require changing, installing equipment and sound-proofing methods only where required, and altering operational conditions from optimum only where shown to be necessary, should far outweigh this.

The EU's Marine Strategy Framework Directive (European Parliament and Council 2008) aims to more effectively protect marine waters within the EU, with a specific aim being to "achieve good environmental status of the EU's marine waters by 2020". Noise is specifically mentioned as an undesirable anthropogenic input into the marine environment. Indicators such as the requirement for the noise not to be at a level which adversely affects the marine environment are stated, however specific limits are not proposed. This would therefore require goals-based indications of compliance and work towards achievement of a good environmental status, which all member states are required to demonstrate. Impact assessments such as those which can be carried out using the information which this model can provide could be used.

A proposed framework for goals-based impact assessment is presented below. It is not appropriate to propose specific details which might suggest that impact will be limited, due to the complexities involved. The species are so varied, with variations even between individuals of the population, and in addition to these variations are also those associated with operational area, and those arising from the available data on different species. Therefore proposing specific requirements would be unsuitable, especially in the scope of a goals-based approach, where the measures taken should not be dictated.

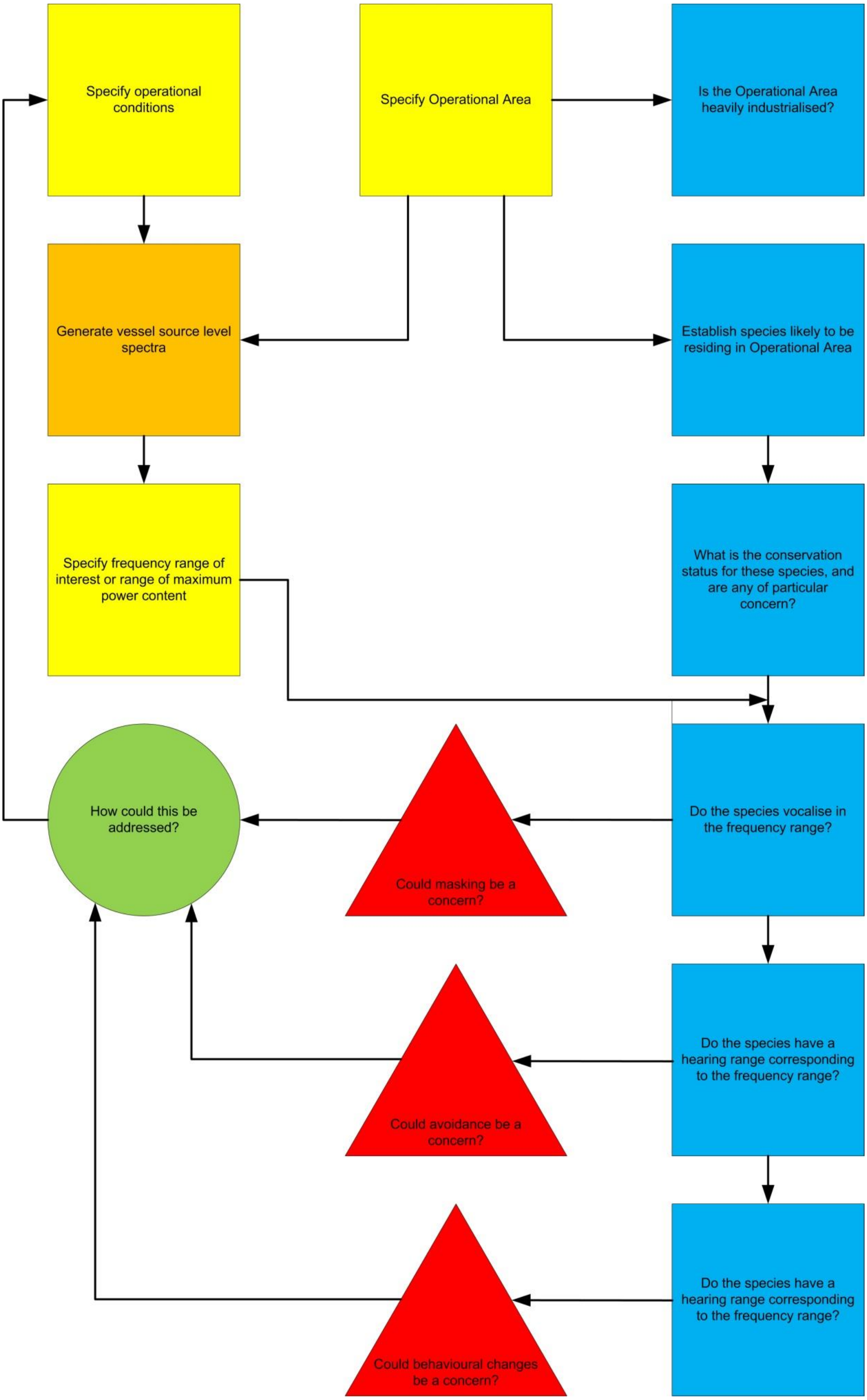


Figure 7.1 - Proposed Goals-Based Assessment Framework

7.4 LNG Carrier Case Study

The following sections will use a case study of the LNG carrier as an example of an impact assessment which could be carried out using an underwater radiated noise spectrum along with the information provided by the data analysis model.

7.4.1 Operational Area and Marine Wildlife Species

The actual operational area of the focus LNG Carrier is not known, therefore the trials area, off the East coast of Singapore, in the China Sea will be used by way of an example. Filtering the marine wildlife database for "Asia Coastal" gives a list of 41 different species who are recorded as inhabiting the Asia Coastal area, and which therefore could potentially be impacted by the vessel. As the area "Asia Coastal" covers a vast area, there may be species listed who would not be found in this particular area, so the user should apply some judgement with regards to the list.

When filtered based on Conservation Status, the following can be seen:

- Data Deficient = 13 species
- Least Concern = 14 species
- Near Threatened = 1 species
- Vulnerable = 8 species
- Endangered = 5 species

Those species listed as "Near Threatened", "Vulnerable" and "Endangered" should be treated with particular concern. It can also be seen however that a significant proportion of the species are listed as "Data Deficient", highlighting the need for further research to be carried out on marine wildlife, as even basic data is often unavailable. As mentioned earlier, it may not be suitable to disregard those species with no data available.

When filtering the Marine Wildlife Database on hearing and vocalisation ranges, with a focus frequency range of 1-500Hz, 26 species are highlighted. Of these, 8 species are also listed as "Near Threatened", "Vulnerable" or "Endangered", and include Cetacean, Pinniped and fish species. These are presented in the table below:

Table 7.2 - Species Potentially At Risk from LNG Carrier Noise

Species	Conservation Status	Hearing Range Lower Limit (Hz)	Hearing Range Upper Limit (Hz)	Vocalisation Lower Limit (Hz)	Vocalisation Upper Limit (Hz)	Known / Observed Reactions
Bowhead Whale	Endangered	20	5,000	20	5,000	Avoidance and behavioural changes 90-115dB
Common Carp	Vulnerable	50	3,000	N/A	N/A	None published
Great White Shark	Vulnerable	10	800	N/A	N/A	None published
Indo-Pacific Humpback Dolphin	Near Threatened	50	175,000	1200	16,000	Behavioural changes
Northern Fur Seal	Vulnerable	500	40,000	200	400,000	Avoidance
North Pacific Right Whale	Endangered	Unknown	4,000	Unknown	500	Avoidance at 148dB and behavioural changes
Ribbon Seal	Vulnerable	Unknown	Unknown	100	7,100	None Published
Sperm Whale	Vulnerable	100	20,000	100	20,000	Avoidance and behavioural changes

7.4.2 Results

A comparison of LNG Underwater Noise levels against both biologically-based and rules-based limits will be presented at 19 and 9 knots vessel speed. The LNG Carrier results presented here use the field measurement data, as in this case it is available, and hence will provide a more accurate assessment. The results are also presented as the sound pressure level at source rather than at receiver, as in general the proposed limits and thresholds are specified for source level.

19 Knots

The figure below shows how the LNG carrier underwater noise, estimated at source, compares to various underwater radiated noise limits:

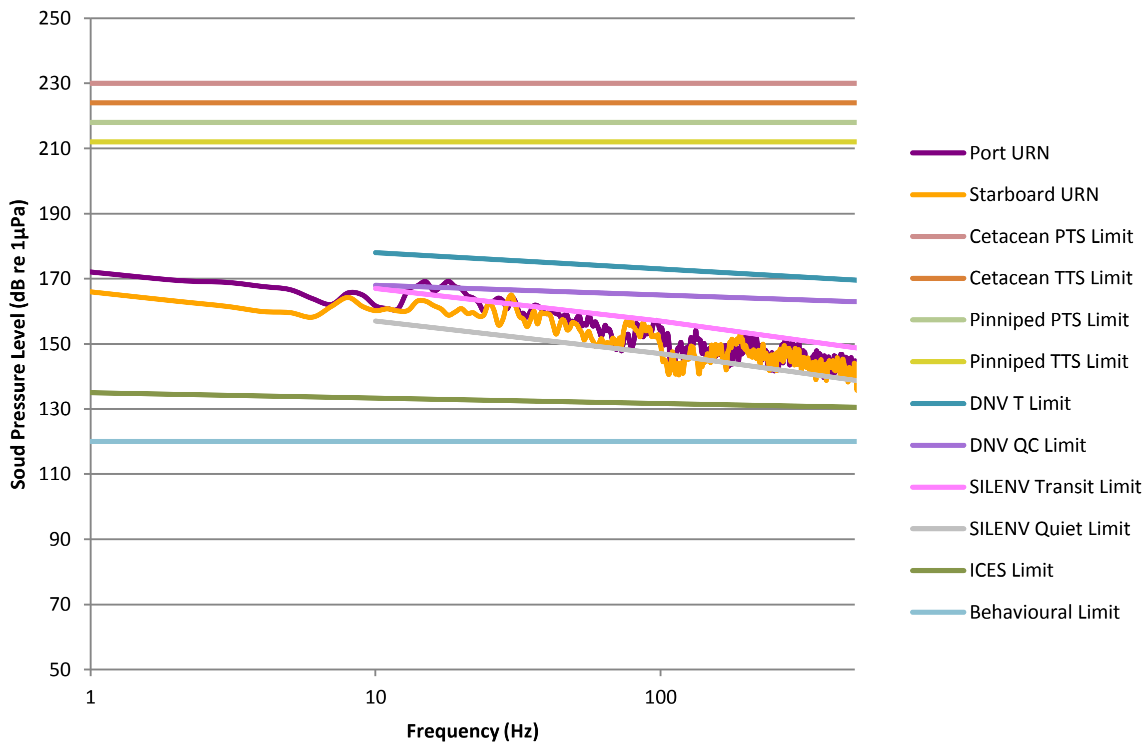


Figure 7.2 - Comparison of Source Level Results for LNG Carrier and Limits at 19 knots

It can be seen that except for a few tonal peaks at around 20Hz, the vessel complies with the DNV limits, and also generally complies with the SILENV transit limit. It does not comply with the quiet cruise limit, however as this is a limit set for vessels travelling at 11 knots rather than full speed this is not of great concern. The vessel does not comply with the ICES limit, but again, this is intended for vessels travelling at 11 knots, and is also more specific to fisheries research vessels, therefore this is not of concern.

It can also be seen that from a biological point of view, the vessel underwater radiated noise levels are significantly below limits for any physical hearing damage. They are however noticeably above the limit for behavioural impacts, which might include avoidance. This implies that the vessel, travelling at 19 knots is very likely to have an impact on the behaviour of any species sensitive to the frequencies involved. As this assessment has only been carried out for the frequency range 0-500Hz, no conclusions can be drawn for species affected by higher frequencies.

As an example, some assumptions will be made for the species identified as inhabiting the area in question, or near threatened or higher conservation status, and with hearing and/or vocalisation rages corresponding to the 0-500Hz range. Referring to the data in Table 7.2, a Bowhead Whale, Northern Fur Seal, North Pacific Right Whale, Ribbon Seal and Sperm Whale are all likely to experience masking of vocalisations and receipt of sounds, as well as to exhibit avoidance and behavioural changes. An Indo-Pacific Humpback Dolphin is likely to experience behavioural changes, and may also encounter reduced capacity to hear important sounds. Impacts on a Common Carp or a Great White Shark are not clear as there is insufficient data available on these species. It should be noted that the LNG Carrier noise levels are estimated source levels, and therefore the impacts will decrease with range.

9 Knots

The figure below again presents a comparison, this time for a vessel speed of 9 knots:

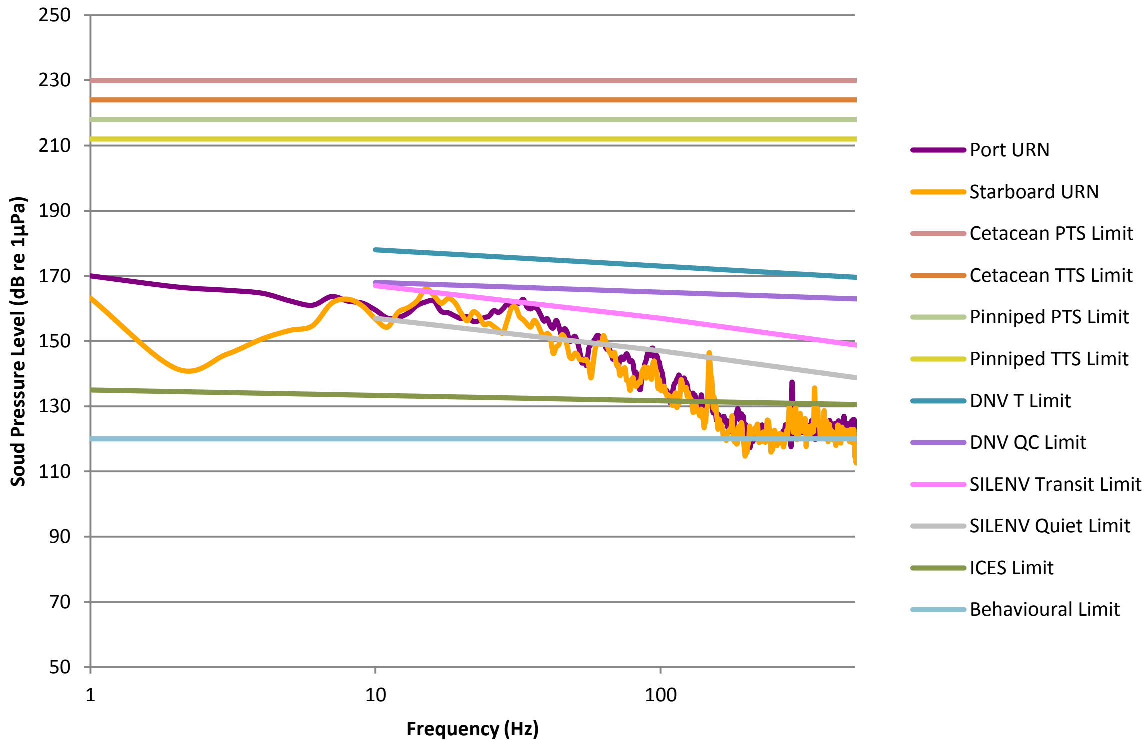


Figure 7.3 - Comparison of Source Level Results for LNG Carrier and Limits at 9 knots

The results at this lower vessel speed with reduced cavitation mean that it is also partially compliant with the SILENV quiet cruise limit. However, this does raise some concern regarding the low frequency characteristics of the vessel's spectra up to around 50Hz, as this limit is intended for vessels travelling at 11 knots. As the vessel's spectra are above the limit at 9 knots, it can safely be assumed that this will also be the case at 11 knots. It is not generally compliant with the ICES limit, however as this is intended for fisheries research vessels rather than commercial ships, this is less of a concern.

With regards to biological limits, the possibility of physical hearing damage in species is considered negligible. There is potential however, particularly in the lower frequencies, for avoidance and behavioural changes, and also possibly for masking to occur in some species which are particularly sensitive to low frequencies. Bowhead Whales, Common Carp, Great White Sharks and Indo-Pacific Humpback Dolphins are all known to be capable of hearing sounds at these frequencies. The distances involved will tend to be shorter than those over which the vessel travelling at 19 knots may have an impact.

7.4.3 Discussion

It can be seen from the above results and observations that at present, the limited availability of data on many species in relation to noise makes impact assessment an exercise of inference and assumption. Although work is ongoing in the field of marine biology to gather more suitable data, there is a long way to go, and developing a comprehensive understanding of this field will take time. It is felt however that using the data and approach presented above, areas of potential concern with regards to the acoustic spectra of a vessel can be identified. Comparison against suggested noise limits give an indication of how the vessel performs in a general sense, whilst comparison against biological limits and the database of values can highlight more specific frequency ranges which may prove problematic. This can also help to highlight particularly vulnerable species, for which operational consideration may also be appropriate, such as avoiding areas in which they are known to reside. As the tool also allows the user to compare receiver level values, a clearer picture of potential impact at a given distance from the vessel can also be assessed. Where actual measurement or predicted vessel spectra are not available, an empirical estimate can be used instead, and providing that the designer is aware of the inaccuracies associated with this estimate, the results and comparison can still be useful.

7.4.4 Addressing Areas of Potential Concern

From the results presented above, it can be seen that in the frequency range 0-500Hz, the area of greatest concern from a rules-based point of view is from 1-50Hz at both 19 and 9 knots. This would be the case regardless of the intended operational area for the vessel. This information would give the designer an indication that this may need to be addressed, and during the design stages, there would be several options for doing so. Some of these are discussed in the Literature Review Chapter.

From a biological point of view, the potential areas of concern regarding the vessels acoustic characteristics are much more dependent on the intended operational area, and therefore the resident species which may be impacted. Where a vessel has a known and limited operational area, the specific needs of the resident species can be taken into account. Where a vessel has multiple intended areas of operation, or where it is designed for global operation, a more general approach will be required. It may be the case that operational changes such as speed reductions or re-routing could be much more suitable than specific design tweaks, as there may be many different frequency range with the potential to have a negative impact. The main point to note however is that such considerations and decisions are demonstrable using the tools and approach provided here. This could form the basis for a goal-based assessment.

7.5 Chapter Summary

This chapter has presented the data available in the data analysis tool developed within this work for different types of assessment of commercial ship underwater noise characteristics. It has also presented an approach for using the data in an impact assessment of the LNG carrier operating in a particular operational area.

The next chapter will present a discussion on the finding of this study, along with some suggestions for areas of future research.

Chapter 8 - Discussion and Future Research

8.1 Chapter Overview

This chapter will present a summary of the main findings of the study discussed in this thesis (§8.2) along with a demonstration of how the research aims and objectives have been achieved. A discussion on the suitability of the approach proposed is then presented (§8.3). It will then highlight the contributions to knowledge which are the outcomes of the study (§8.4). Finally it will present recommendations for relevant areas of future research which are related to the study presented (§8.5).

8.2 Thesis Summary

Concern over the potential impact of ship underwater radiated noise on marine wildlife has grown in recent years and is now one of the key topics being investigated for potential future guidelines by the International Maritime Organisation (IMO). Except for specific classes such as Naval, Cruise or Fisheries Research Vessels, radiated noise characteristics are rarely considered during the design stages, and also tend not to be evaluated during ship trials. There would appear to be a need to bring this issue to the attention of the wider marine industry, to raise awareness of the potential impacts, and hence the need for considering them. This may give rise to more ship owners, operators and designers considering the underwater noise properties of their vessels. This in turn would mean that there is a greater need for URN prediction and assessment methods which can be applied by those concerned. This study has presented approaches which address this need, and has also identified flexibility in the approaches to suit the different needs of different users.

8.2.1 Achievement of Research Aims and Objectives

It was previously stated that research aims and objectives are as follows:

- *To review the available literature on ship radiated underwater noise sources, modelling, impact, regulation and other relevant areas, with particular reference to how they may be applied in this work*

The Literature Review in Chapter 3 addressed this by presenting a wide ranging overview of the various related topics, discussing; ship noise sources and signatures, ambient noise, underwater noise propagation, modelling techniques, effects on marine wildlife, regulations and reduction measures.

- *To develop a numerical prediction methodology for propeller noise, using field measurement underwater measurement data for validation*

As has been discussed throughout the study, there appears to be a need to be able to approximate the ship underwater radiated noise (URN) characteristics during the early design stages, as it has been shown that by considering acoustic performance as a contractual requirement from the very beginning can significantly reduce any associated costs in achieving suitable levels. Empirical and theoretical approaches for the prediction of various aspects of ship radiated noise, discussed in Chapter 3, have been used since WWII however more recent advances in computational power have lead to numerical methods increasing in popularity. These methods have the advantage of being able to providing more accurate results which are ship-specific, rather than being based on curve-fitting formulae for a small sample of vessels. However these approaches can be time consuming and expensive, with these increasing as the model is further refined. Therefore this study proposes a compromised approach which provides an indication of the acoustic characteristics of the vessel, using ship data which would be available at these early stages, for non-cavitating condition prediction. This was discussed in detail in Chapters 5 and 6. The results have been shown to be less accurate than would be ideal, and this has been attributed to several causes. Firstly the high Reynolds Number of the simulation being run at full scale has made the solution of flow characteristics difficult. Insufficient cells and a mesh which is not fully refined as required have also contributed to the errors observed. Other sources of error are discussed in Appendix D. A discussion on the suitability, advantages and drawbacks of the approach are presented in the next section.

- *To gain a better understanding of the importance of cavitation noise in relation to underwater noise*

A better understanding of the importance of cavitation has been gained through two paths. Firstly, research was carried out through the Critical Review into the phenomenon of cavitation, its prediction and its reduction. Secondly, some simplified simulation work was carried out on an open water propeller to investigate whether this could be used to predict cavitation performance. The results were found to be unsatisfactory and served to highlight the complexity of cavitation for marine propellers.

- *To develop a methodology for the prediction of machinery noise, using field measurement onboard measurement data for validation*

Chapters 5 and 6 also address the subject of machinery noise on board ships, before looking more specifically into its prediction. It is identified here that full and accurate prediction of the machinery SPL would form a study in itself therefore a simplified approach is taken whereby only the likely frequency of the tonal peak is predicted using simple formulae. The developed model uses user-input ship and machinery data or estimates to predict tonal frequencies for various equipment items, and presents these against the predicted spectra for peak identification purposes. This approach is deemed suitable for the likely usage of this approach, and was shown to highlight potential machinery noise sources using this tonal peak identification method.

- *To test the performance of the prediction methodologies for a commercial vessel, to establish the capabilities and limitations of the approaches*

In order to test the capabilities and limitations of the approach, it was used to predict the URN spectra for a case study commercial vessels; a large LNG Carrier. Plausible and useful results were achieved, providing that the error can be minimised in future. However this requires further study and comparison with a large number of additional commercial vessels to fully assess whether the performance of the method is acceptable. The limitations of the approach are discussed separately in the section below.

- *To develop a means of assessing the potential impact of the ship radiated underwater noise on marine wildlife*

The model also contains a detailed database of marine wildlife species, their conservation status, typical habitat by set regions, hearing and vocalisation range, and any published observations of their reaction to underwater noise sources. This can then be filtered by region and highlighted by conservation status or hearing and vocalisation range for a given ship noise frequency range. The purpose of this is to enable designers to take steps to assess the potential impacts of their vessels noise on any relevant species. The predicted or measured ship URN spectra can also be compared to a range of existing noise limits and guideline thresholds. A case study was then presented in Chapter 7 which used the model to assess the potential impact of a vessel. As was anticipated from the literature, masking, avoidance and behavioural changes were identified as being the most likely impacts on the at risk species recognised in the selected operational area.

8.3 Suitability of Underwater Radiated Noise Prediction Approach

As has been highlighted in Chapter 6, although the approach provided useable results for the vessel studied, there was a significant gap of at least 45-50 dB above 50 Hz between predicted and measured results when applying the moving frame of reference only approach. It was highlighted throughout the study that this approach is an over-simplification of reality, and would provide inferior results to a rotating mesh approach. As was shown for the LNG carrier, the rotating mesh approach would reduce the discrepancy to around 20 - 25dB above 150 Hz and improve prediction of the spectral shape. It is clear here that while the results achieved from the more accurate rotating mesh approach, with continually updating free surface is useful as both an indicator of likely broadband level, with a known under-prediction of around 20-25dB, and general spectral shape, those from the others approaches are unlikely to have much relevance to designers and ship owners. The errors observed are simply too great, especially given the set-up and run time demands of the numerical approach, and hence represent the limitations of the numerical approach. Even the results achieved using the ideal case approach is a significant under-prediction. That these errors arose during this study suggest that it is also likely that similar issues could arise for other practitioners, and therefore it should be clear whether or not the numerical approach should be used in different cases. It should now be tested whether or not the approach is limited by different types and sizes of commercial vessel, and whether the ideal case is again the most appropriate. In terms of the free surface, it may be possible to freeze or even neglect it as the results showed good agreement with and without the free surface above 200Hz. Given the added demands of creating the surface and re-meshing which arise when using a permeable source surface approach it is suggested that this is not suitable for such an application, when similar results can be achieved using only the rotating mesh approach.

In terms of computational penalty, the simulation time can be very variable, as it is dependent on the mesh and solvers activated, and the time-step size used. The simulation times depended on whether use was made of a high performance computer facility operating the software in parallel, and the number of cores being used. It also depends on the simulation time required for convergence, the number of inner iterations used and the different time-step sizes selected during the simulation. There is also a variation depending on whether a moving frame of reference of reference, permeable source surface, free surface and rotating mesh is being used. Due to the complexity of variations outlined above it is difficult to give a clear indication of relative times arising purely from the propeller representation method applied.

8.4 Contributions to Knowledge

Further to the limitations identified above, the disadvantages of the proposed approach are that it is reliant on the use of Computational Fluid Dynamics (CFD) software, and therefore places several requirements on the designers including availability and familiarity with such software, access to suitable computational power, and having software with the required capabilities for noise prediction and propagation. As part of this study, an Excel-based model was also developed, and outlined in Chapter 5 and 6. This model provides predicted overall and also spectral Sound Pressure Levels (SPL) using empirical approaches, for use where numerical or measured data may not be available. As can be seen in Chapter 6, these empirical formulae could provide a closer estimate to the measured values than the simplified numerical approaches, and therefore if for whatever reason the rotating mesh approach with detailed free surface cannot be applied, the empirical estimates would be a more suitable alternative. The benefits of the numerical approach over the empirical estimate when the rotating mesh approach can be used are the additional details and data which can also be gained, such as details about the flow around the vessel and operation in specific conditions.

The most appropriate way to present the contributions this study has made to current knowledge and the current state-of-the-art is by way of an answer to the research questions which was posed:

"Can the underwater radiated noise of a commercial ship be predicted and assessed using information available during the early design stages of a new build?"

The short answer to this question is yes. This study has presented an approach which allows a designer to gain a good estimate of the URN spectra of a vessel using only hull and propeller geometry, and approximate operational speeds and corresponding propeller rpm, all of which are likely to be available during early stages of a new design. Whilst similar work has previously been carried out in a few cases, it has been in model scale rather than the full scale applied in this study, although it has been noted in this study that use of a full scale simulation may have added to the eventual error in results. It has also typically been from a research rather than industry-based perspective and therefore the focus has been on greater accuracy rather than wider usability and faster simulation. During development of the approach, a study was also conducted into the variations in the CFD model which could be applied, and the impact these might have on the achieved results. These variations and findings are published in (Kellett et al. 2013). These variations could be useful for designers who have access to only limited time or computational power and are therefore prepared to sacrifice some accuracy in results in order to be able to generate them more easily. To the author's knowledge, such a study has not been previously made available to the public domain.

The impact assessment model which has been developed is then the means by which the URN of the commercial vessel can be assessed. This study takes a more global view of the subject of ship underwater noise, where others have focussed on very specific aspects of the issue. For this reason, both a noise prediction methodology, and an impact assessment model have been developed. It is important that once designers have established the acoustics characteristics of the vessel, they have some means of assessing what the impact of this vessel might be, either for contractual requirements or later, for the purpose of complying with regulations. The model developed and outlined in Chapter 5 (§5.6) presents some of the options for assessing likely impact, using literature on marine wildlife habitat regions, conservation status, hearing and vocalisation ranges, and observed reactions to known underwater noise sources. There are currently no specific guideline limits for "acceptable" or "unacceptable" noise levels, and it seems unlikely that these could be developed to satisfactorily protect the huge variety of species involved. Goals-based methods could therefore be the most suitable approach to addressing impact, and the data analysis tool is very suitable for use in such an approach. The model also allows the user to compare their ships spectra to different threshold and limits, which could be used should specific limits become the method of regulation.

It should be noted that it is the approaches in the data analysis tool in terms of the way the data can be presented and utilised which present the contribution; this was not intended to be a programming exercise and therefore the model itself is a fairly basic Excel-based spreadsheet which uses simple Visual Basic macros for some automation. The approaches could be translated into a more suitable program.

8.5 Recommendations for Future Research

As identified above, there were limitations to the existing approach which should be addressed. Due to the time pressures and focussed scope of the work conducted, there are also relevant areas which could not be investigated here. These will be briefly outlined below as suggested areas for future work in this specific field.

8.5.1 Improvements to the Early Stage Design Noise Prediction Tool

Firstly the accuracy of the applied approach should be greatly improved. This would require the application of the recommendations by International Towing Tank Conference (2011) on CFD studies for different approaches. The number of cells used in this study is also insufficient for a full scale vessel simulation, and should be increased in accordance with the guidelines. Additional refinement should also be added to better solve the boundary layer flows, wake, free surface and flow separation. Thirdly, the frequency range for sound prediction should be extended to the 0 - 1000Hz range to gain a better understanding of the non-cavitating propeller acoustic properties. Finally, detailed uncertainty analysis should be carried out in line with the various ITTC recommendations, especially those outlined in (International Towing Tank Conference 1999).

Alternatively, it is worth considering simulation at model scale rather than full scale. The very high Reynolds Numbers associated with these simulations have made it difficult to solve the flow characteristics. Simulation at approximately 1/10 scale would assist in overcoming this problem. The results would then be scaled back using the ITTC 1987 recommendations.

The noise prediction tool developed as an output of this work can provide a good deal of information to the designer, however there are several ways in which it could be extended. One of these is to include more details relating to marine protected areas in relation to the intended operational area of the vessel. As more guidelines, limits and eventually formal regulations for underwater radiated noise are published, these will also need to be included in the Microsoft Excel tool. The same applies to the addition of marine wildlife species and their relevant data to the database.

In terms of user-friendliness, improvements could be made to the user interface of the Microsoft Excel tool, or it could be transferred to a more technical base such as Matlab. Greater automation between sections, and if suitable, between the tool and the CFD programme used could also prove beneficial.

8.5.2 Development of a More Advanced Prediction Tool

The tool which has been developed is very much aimed at use in the early stages of a new ship design process however there are some ways in which it could be altered to create a much more advanced tool for later stages and more detailed analysis. The first way would be to include the prediction of noise and its propagation in realistic seaway conditions, as this would affect hydrodynamic noise, propeller loading and resulting noise, and also the range of frequencies propagated to larger distances. A growing area of research is also the propagation of noise in ice conditions, such as for vessels operating in polar waters, therefore such considerations could also be included. The influence of the seabed on noise propagation in shallower waters is also an area for potential inclusion. Another valuable extension could be to improve the prediction of the noise contributions from machinery installations and equipment, perhaps by coupling the general tool with a Finite Element Analysis (FEA) programme as well. This would allow for not only the separate frequencies of the tonals arising from the different installations, but also some ideas of the associated amplitudes of sound pressure level, and the interactions from the sound sources. If this could be developed to a suitably accurate level, it could also be used to investigate the impacts of onboard noise reduction and mitigation measures on underwater radiated noise levels and spectra as well.

As observed in Chapter 5, a thorough study of cavitation simulation approaches for acoustic prediction, especially in timescales suitable use in for early stage design but also for later stages by designers is required. This should explore both options of using built-in solvers such as the one in StarCCM+, as well as suitable alternatives. Some study should also be carried out to establish more up-to-date empirical or simplified methods that would not carry the same time and cost penalties.

The approaches used in the CFD aspect of the modelling have been chosen to fit the compromise of results accuracy and simulation run-time. For the purposes of a more advanced prediction tool, some more complex approaches could be more suitable. The hydrodynamic modelling in the current CFD is done using an Unsteady Reynolds-Averaged Navier-Stokes (URANS) approach, but as has already been discussed in the Chapter 5, more complex approaches such as Large Eddy Simulation (LES) or Detached Eddy Simulation (DES) could also be applied. These approaches are more capable of capturing the finer details of the pressure field around the vessels hull and propeller, which will improve the prediction of noise being emitted, especially for the impacts of wake fields on the propeller noise.

The current solver for propagation prediction used in the CFD simulations is based on the Ffowcs-Williams Hawkings (FWH) equation assumes that the water is entirely homogenous in terms of temperature, pressure, salinity and speed of sound. This assumption, while suitable over shorter distances, may not be as appropriate for receivers at longer distances from the source, and hence some modifications could be required.

Looking to wider applications than for use by designers only, the model could be expanded to suit use by either ship operators or even onboard officers. This could be adapted for specific vessels and used for predicting the potential impacts of a vessel at its current operational conditions and operational area. It could perhaps also include information of marine protected areas or important biological locations to avoid, or operational changes that could be made to decrease impact.

8.5.3 Applications for the Prediction Tool

There are several constructive ways in which the developed prediction tool could be applied for a more detailed investigation of different underwater radiated noise spectra for commercial vessels. For example, a study into the comparative noise characteristics of different propulsors and propulsion systems would allow for more informed decision-making during a new design. A similar investigation into variation in hullform, and into appendage designs and configurations could also prove valuable. Carrying out a study into the comparative underwater radiated noise of a vessel at different operational conditions such as trim and draft variation, as well as propeller loading would be extremely useful information for both designers and operators, as potentially cheaper and easier methods for noise reduction, or for changing the spectral properties.

More generally, a study into the suitability of goals-based underwater noise requirements as a form of government would help to answer important questions which are being asked in the field at present. This could also assess the applicability of models such as the early stage noise prediction tool developed in this work to these goals-based approaches.

8.6 Chapter Summary

In this chapter, a summary of the thesis has been presented along with a discussion of how the objectives of the research have been fulfilled. A discussion on the suitability and limitations of the proposed approach is also presented. An outline of the contribution to knowledge made by this study has been discussed, and finally, recommendations on future research have been made.

The next chapter will present some conclusions which have been drawn from this study.

Chapter 9 - Conclusions

9.1 Chapter Overview

This chapter will present some final remarks on the purpose, finding and contributions of this study.

9.2 Concluding Remarks

Increasing attention focused on the underwater radiated noise (URN) from marine industrial activity and its potential for negative impacts on marine wildlife has lead to it being brought to the attention of the IMO. Their particular concern has related to the acoustic properties of commercial shipping, and has lead to the recent release of a Circular of Guidelines relating to this (IMO 2014). This increase in attention has highlighted a need for suitable prediction and assessment approaches for URN of commercial ships during design stages, as once the vessel has been launched it can be very difficult and expensive to make any significant changes.

Regarding the prediction of underwater noise in the early design stages, this study confirmed that the numerical approach combining a URANS CFD approach with the F-WH solver is suitable. It was also shown through development, validation and implementation of the approach to two different vessels that suitably accurate and useable results can only be achieved by applying a rotating mesh approach with a sliding interface, and a continually developing free surface. However it was also found that results of similar magnitude could be achieved through the use of a moving frame of reference approach in combination with the porous formulation calculated using a permeable source surface of suitable radius. However it was found that establishing the optimum permeable surface radius could be problematic. Results achieved with a permeable source surface and rotating mesh approach are less accurate than was hypothesised, and this approach is therefore not considered appropriate. Less accurate variations, such as those neglecting the free surface or applying a moving frame of reference only, provide results which significantly underestimate the acoustic spectra of a vessel.

The lack of inclusion of the contribution of cavitation noise was observable in the achieved results, and hence indicates that the approach, whilst adequate, requires further improvements. At speeds corresponding with non- or low-cavitation conditions, an underestimation of approximately 20dB was achieved. This increased to 40dB for speeds corresponding to heavy cavitation conditions however it is thought that the additional 20dB is the contribution from cavitation noise. The reasons for this under-prediction have been addressed in Chapter 6 and in the Discussion, and sources of error are outlined in Appendix D.

The need for suitably high-quality geometry and operational data has been highlighted, as these ensure a good simulation can be conducted, in a reasonable timescale. Validation of the simulation is also critical for confidence in the results.

The approaches demonstrated for assessing the impact of vessel noise on marine wildlife were all successfully applied to the LNG Carrier case, and were all shown to be suitable for use by designers and ship owners. It also demonstrated that data available publically in the literature is suitable for use in impact assessment.

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Appendix A - Ffowcs-Williams Hawkings Equation and the Farassat Formulation

The Ffowcs-Williams Hawkings (FWH) equation, for the prediction of noise propagation, is introduced in Chapter 3, along with the Farassat Formulations of this equation, which have been applied in this work. The full equations, in their common form, are presented below. For details of how these equations arose and how they are derived, please refer to (Ffowcs-Williams & Hawkings 1969), (Farassat 1975), (Farassat 2007), (Farassat & Brentner 1998) and (Farassat & Brentner 1988).

Formulations 1 and 1A for both surface integral terms (thickness and loading) and Formulations Q1 and Q1A for volume integral (quadrupole) terms, again as introduced in Chapter 3, are given below. Formulations 1 and Q1 were the originally derived formulations, which have an observer-time derivative which is found numerically. Formulations 1A and Q1A were then derived from the original Formulations, and as the observer-time derivative is taken analytically, and hence they can speed up computation and improve accuracy.

Formulation 1 and Q1

Formulation 1 for Thickness and Loading Terms

$$4\pi p'(\boldsymbol{x},t) = 4\pi \left(p'_T(\boldsymbol{x},t) + p'_L(\boldsymbol{x},t)\right)$$
$$= \frac{\partial}{\partial t}\int_{f=0}\left[\frac{\rho_0 v_n}{r(1-M_r)} + \frac{p\cos\theta}{cr(1-M_r)}\right]_{ret} dS + \int_{f=0}\left[\frac{p\cos\theta}{r^2(1-M_r)}\right]_{ret} dS \qquad (A.\ 1)$$

Where: p'_T *is the acoustic thickness pressure*

p'_L *is the acoustic loading pressure*

$(\boldsymbol{x},t)$ *are the observer space-time variables*

M_r *is the Mach number of a fixed point on the blade,* $M_r = \boldsymbol{M}.\hat{\boldsymbol{r}}$

$[\,]_{ret}$ *denotes evaluation at the retarded time, i.e. sound speed is not infinite*

$r = |x-y|$

θ *is the local angle between normal to the surface and radiation direction at emission time*

dS *is for intergration over the surface in question*

Formulation Q1 for the Quadrupole Term

$$4\pi p'_Q(\boldsymbol{x},t) = \frac{1}{c}\frac{\partial^2}{\partial t^2}\int_{F>0}\frac{T_{rr}}{r}d\Omega d_\tau + \frac{\partial}{\partial t}\int_{F>0}\frac{3T_{rr}-T_{ii}}{r^2}d\Omega d_\tau + c\int_{F>0}\frac{3T_{rr}-T_{ii}}{r^3}d\Omega d_\tau \qquad (A.\ 2)$$

Where: p'_Q *is the acoustic pressure due to quadrupoles*

$T_{rr} = T_{ij}\hat{r}_i\hat{r}_j$ *where* T_{ij} *is the Lighthill Stress Tensor and* $\hat{r}_i$ *and* $\hat{r}_j$ *are unit radiation vectors*

$d\Omega$ *is an element of the surface area of the sphere* $r = c(t-\tau)$

d_τ *is integration at source time*

T_{ii} *is a summation result from the derivation*

Formulation 1A and Q1A

Formulation 1A for the Thickness Term

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

Where: $\dot{v}_n$ is the source time derivative of v_n

$\dot{M}_i$ is a result from differentiation of the Mach number at a fixed point at the source time

M is the Mach number

$\hat{r}_i$ is the component of unit radiation vector

Formulation 1A for the Loading Term

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

Where: n_i is a direction cosine

$\dot{p} = \frac{\partial p\,(\eta,\tau)}{\partial \tau}$, where τ is the source time

p is the unsteady blade surface gauge pressure

Formulation Q1A for the Quadrupole Term

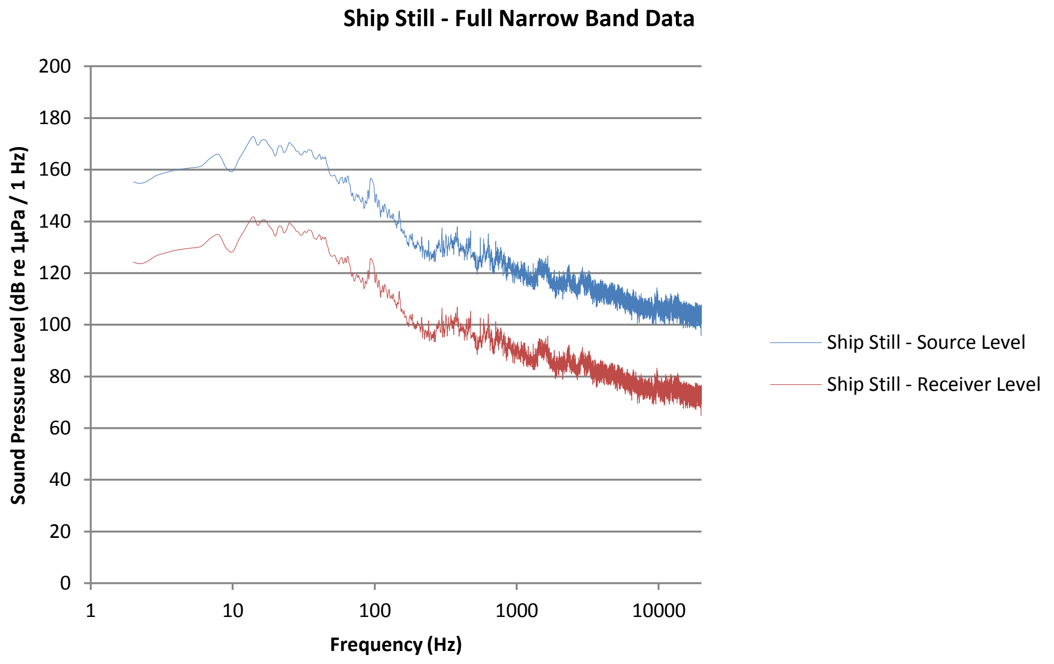
$$4\pi p'_{Q}(\boldsymbol{x},t) = \frac{1}{c}\frac{\partial^2}{\partial t^2}\int_{f=0}\left[\frac{Q_{rr}}{r|1-M_r|}\right]_{ret} dS + \frac{1}{c}\frac{\partial}{\partial t}\int_{f=0}\left[\frac{3Q_{rr}-Q_{ii}}{r^2|1-M_r|}\right]_{ret} dS + \int_{f=0}\left[\frac{3Q_{rr}-Q_{ii}}{r^3|1-M_r|}\right]_{ret} dS \qquad (A.\ 5)$$

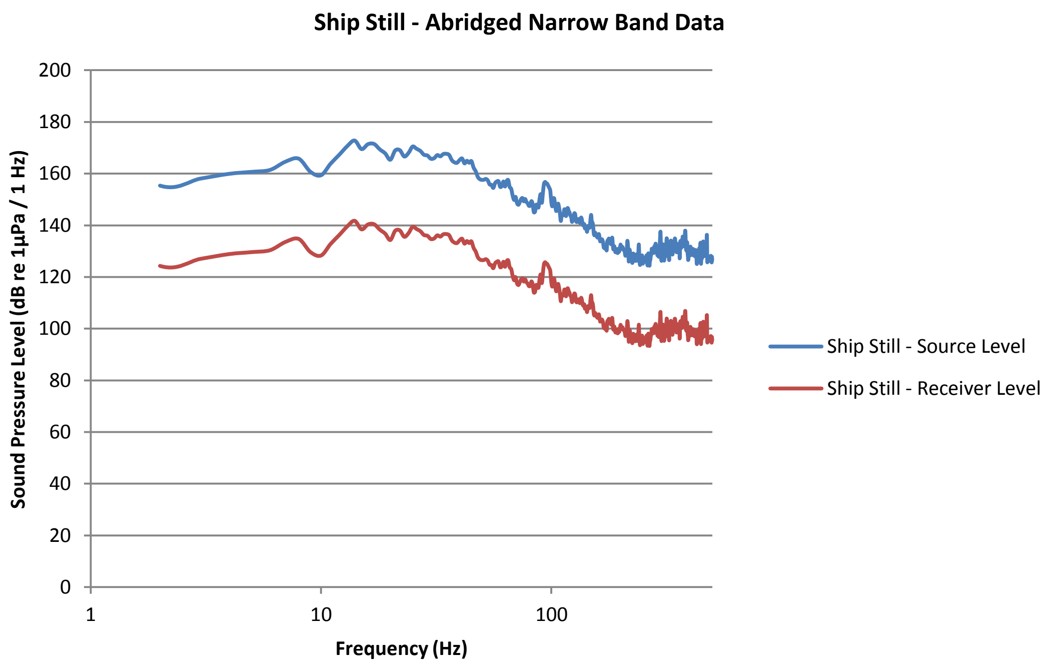
Where: Q_{rr} and Q_{ii} are related to the quadrupole source strength tensor in a similar fashion to T_{rr} and T_{ii}

Appendix B - LNG Carrier Field Measurement Results

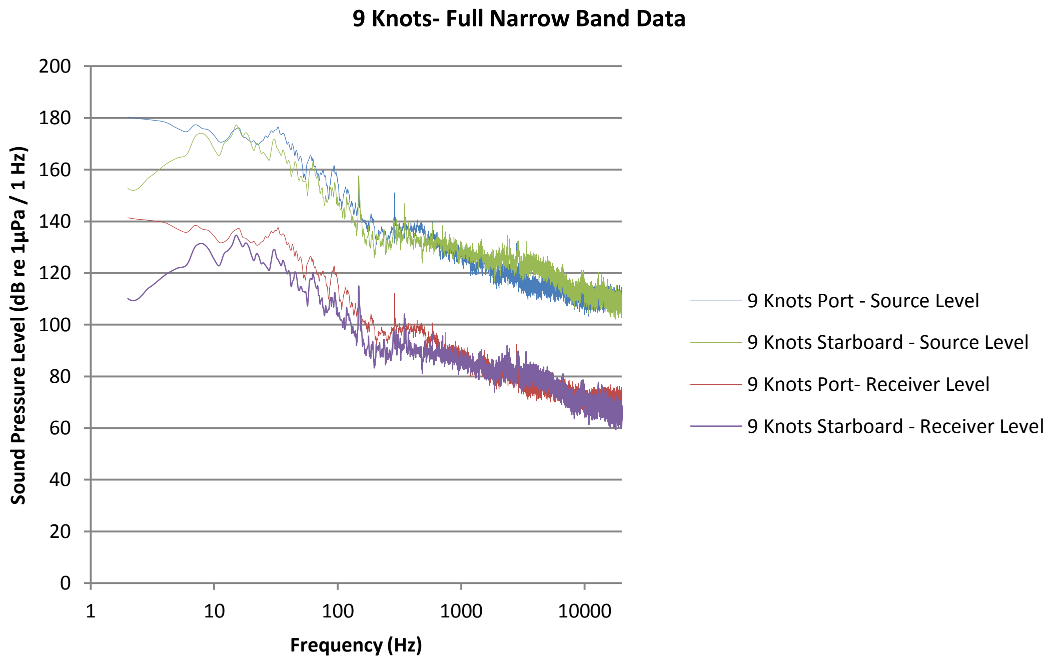
The field measurements carried out for the LNG Carrier were introduced in Chapter 6. The results for the ship at zero speed, 9 knots, 19 knots and ambient noise are presented below, for source and as-measured levels and for port and starboard sides where appropriate. The source level values are those corrected back to 1m from the source, while the as-measured or receiver levels are those directly measured at the hydrophones. Spectra are presented first for the full 0-20,000 Hz range, and then abridged to the 0-500 Hz range.

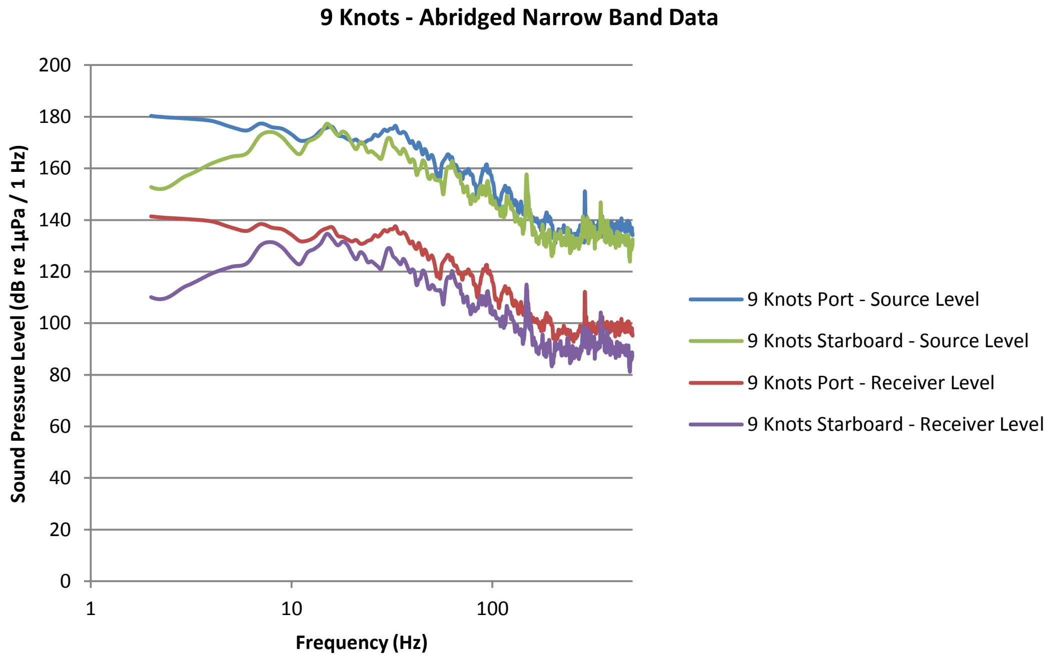
Results with Ship at Zero Speed



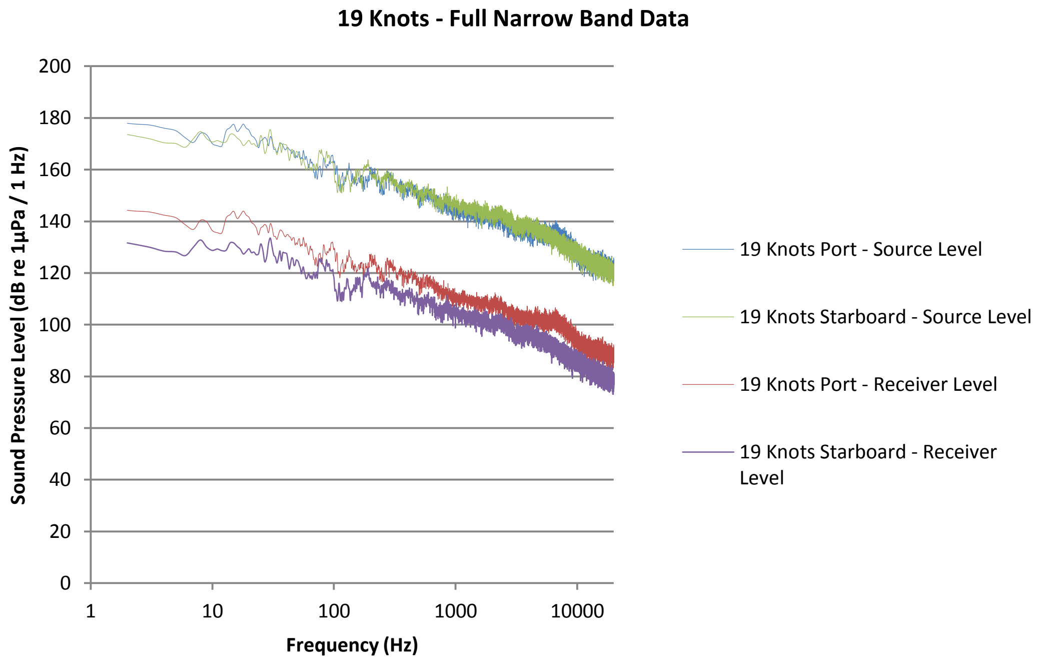


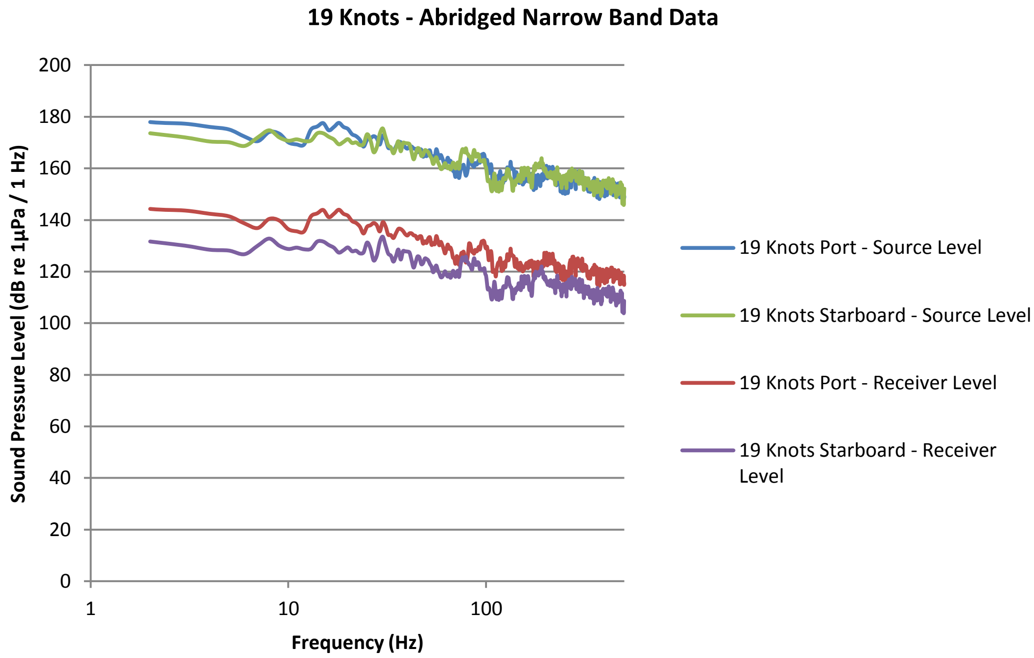
Results at 9 Knots



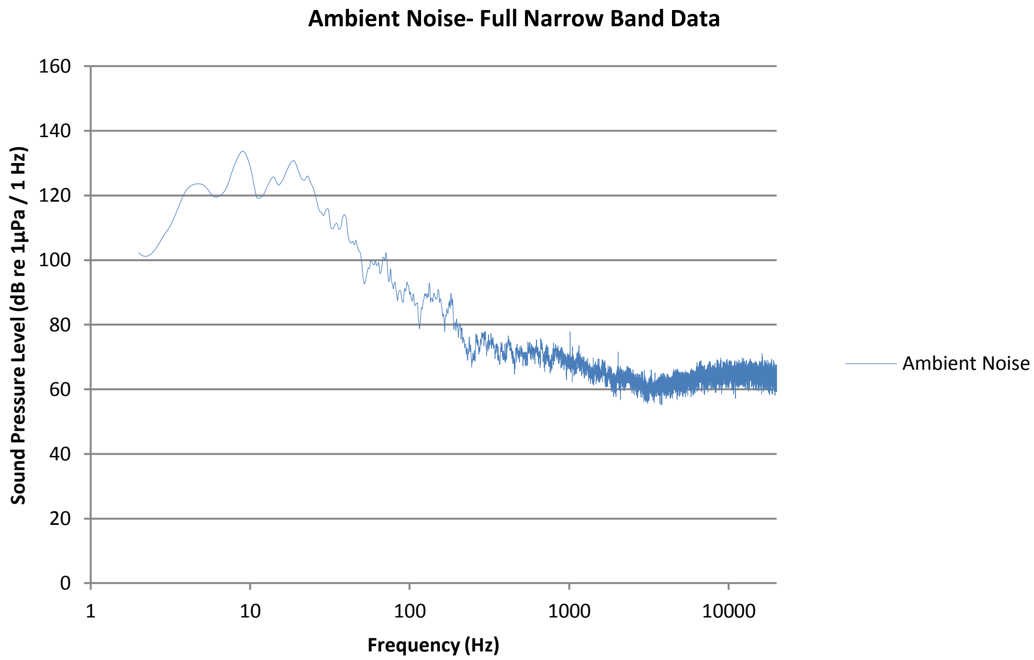


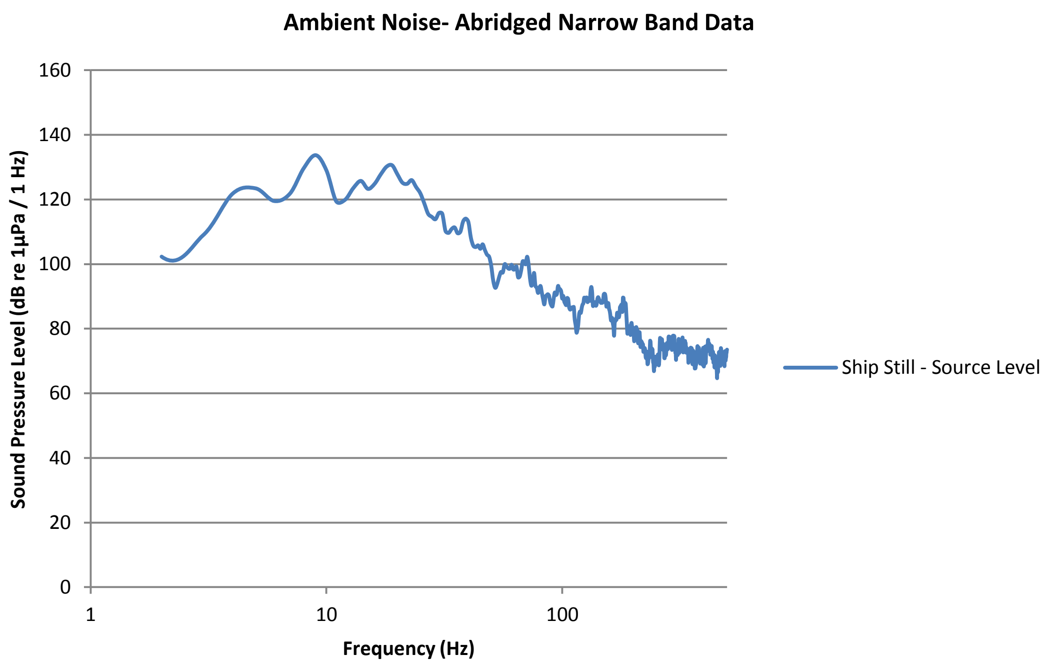
Results at 19 Knots



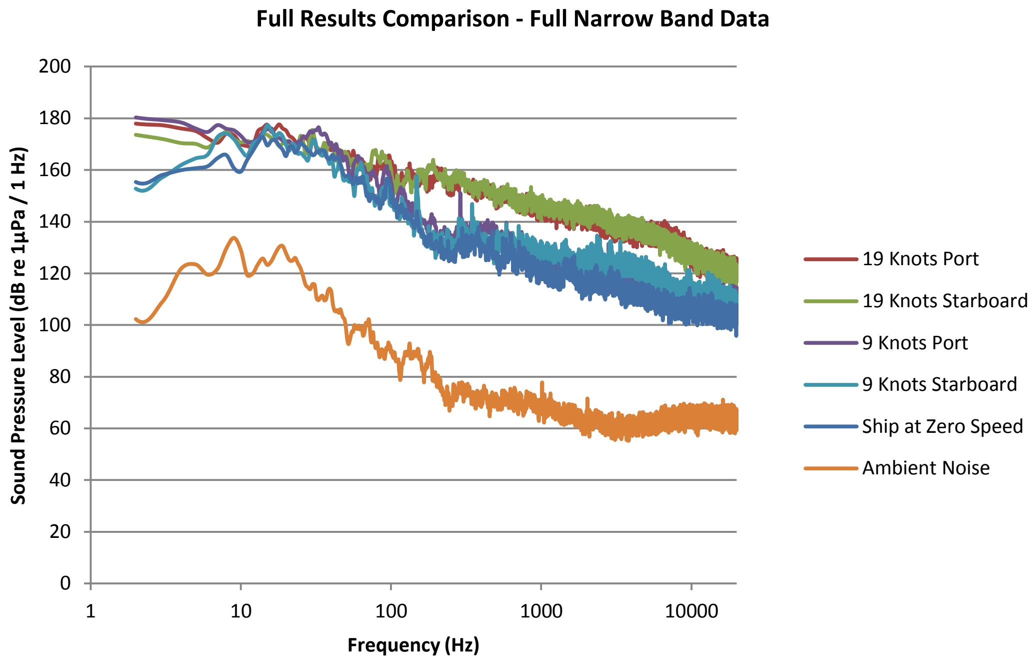


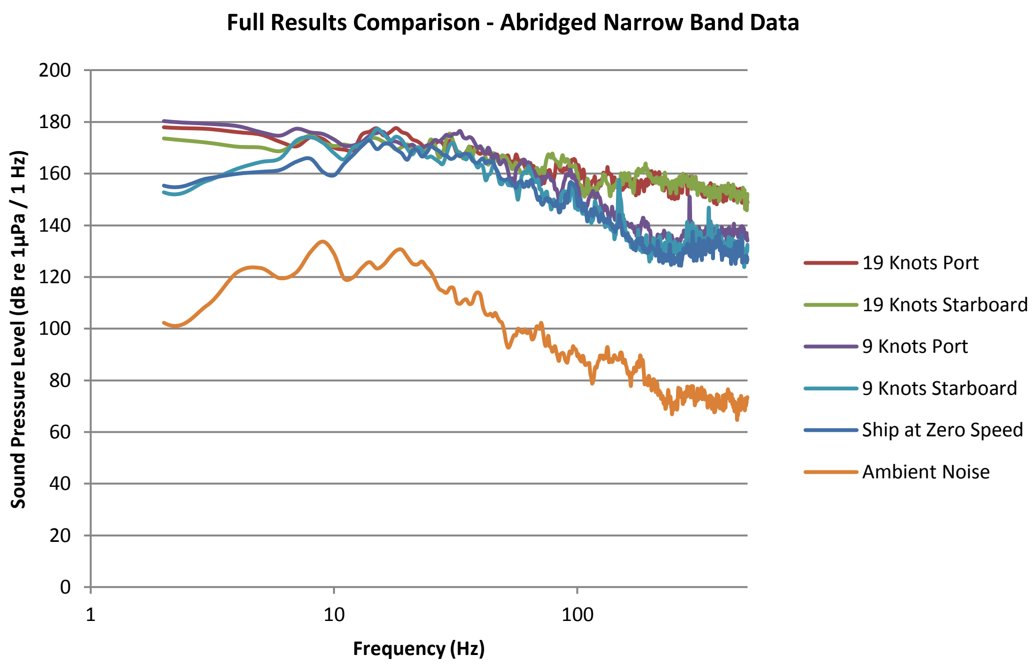
Ambient Noise





Full Comparison





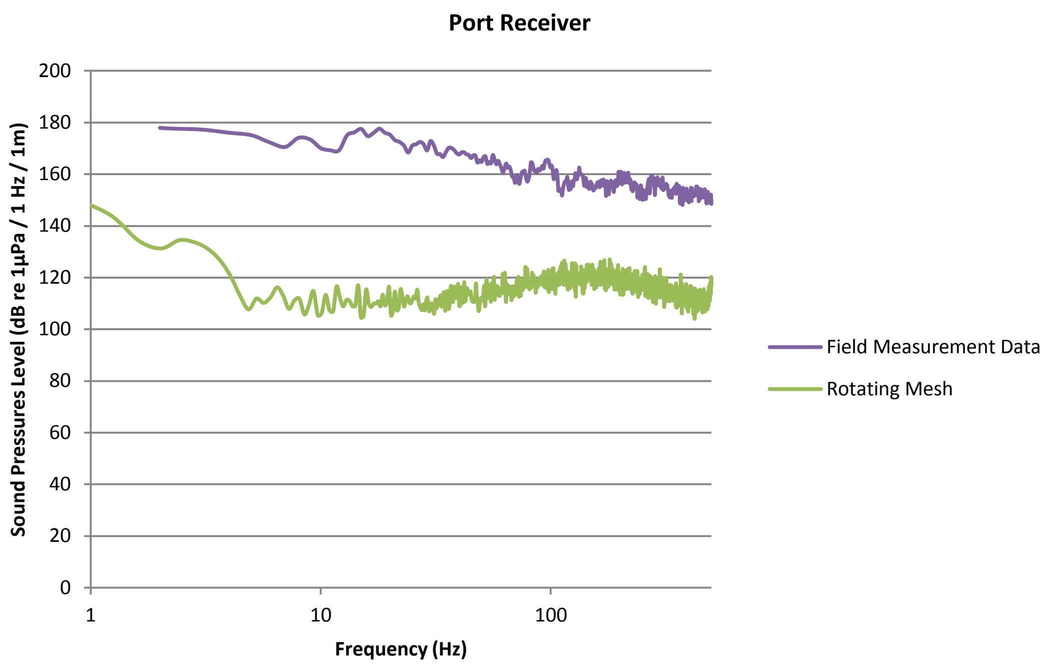
Full Results Comparison - Abridged Narrow Band Data
Sound Pressure Level (dB re 1µPa / 1 Hz)
200
180
160
140
120
100
80
60
40
20
0
1
10
100
Frequency (Hz)
19 Knots Port
19 Knots Starboard
9 Knots Port
9 Knots Starboard
Ship at Zero Speed
Ambient Noise

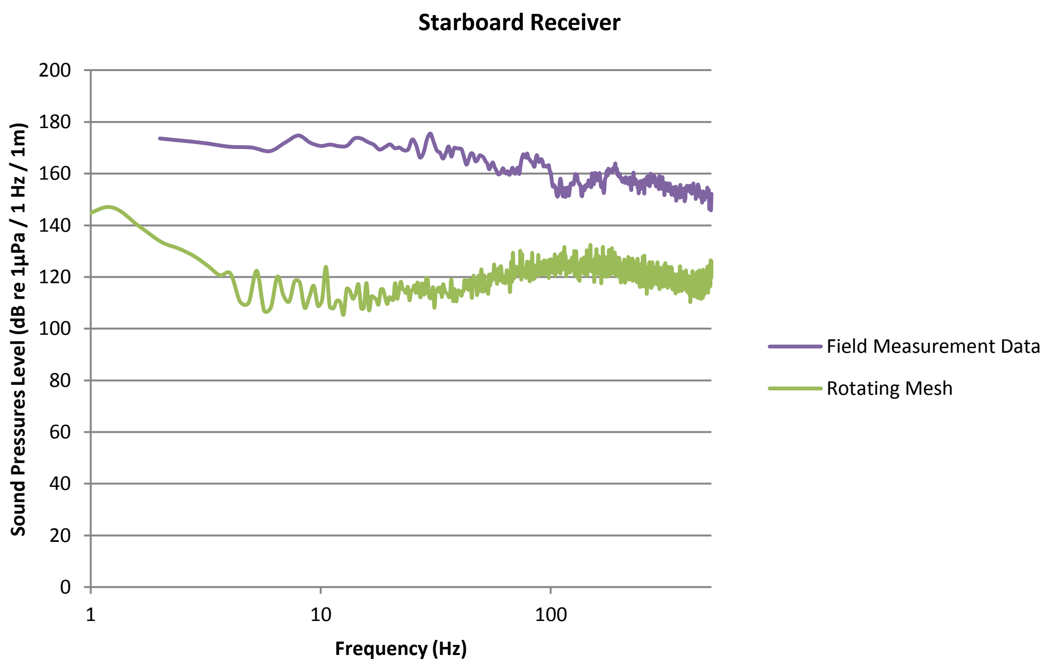
Appendix C - LNG Carrier Simulation Results

The LNG carrier simulation results were introduced and discussed in Chapter 6. The results for the LNG Carrier are presented in full below, with the data for both the port and starboard side receivers shown.

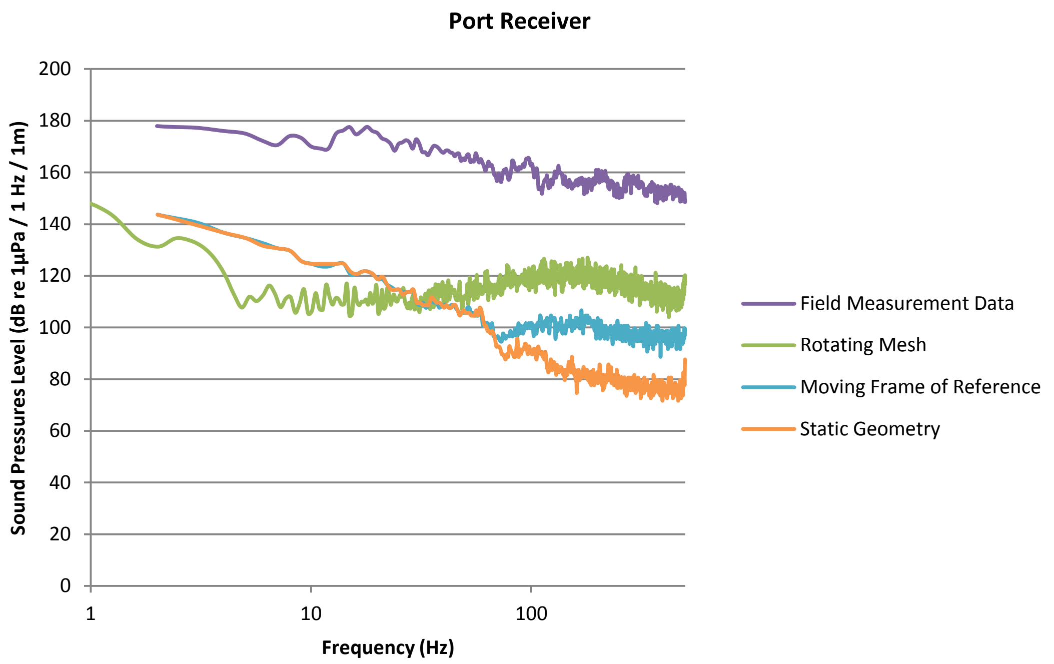
19 Knots

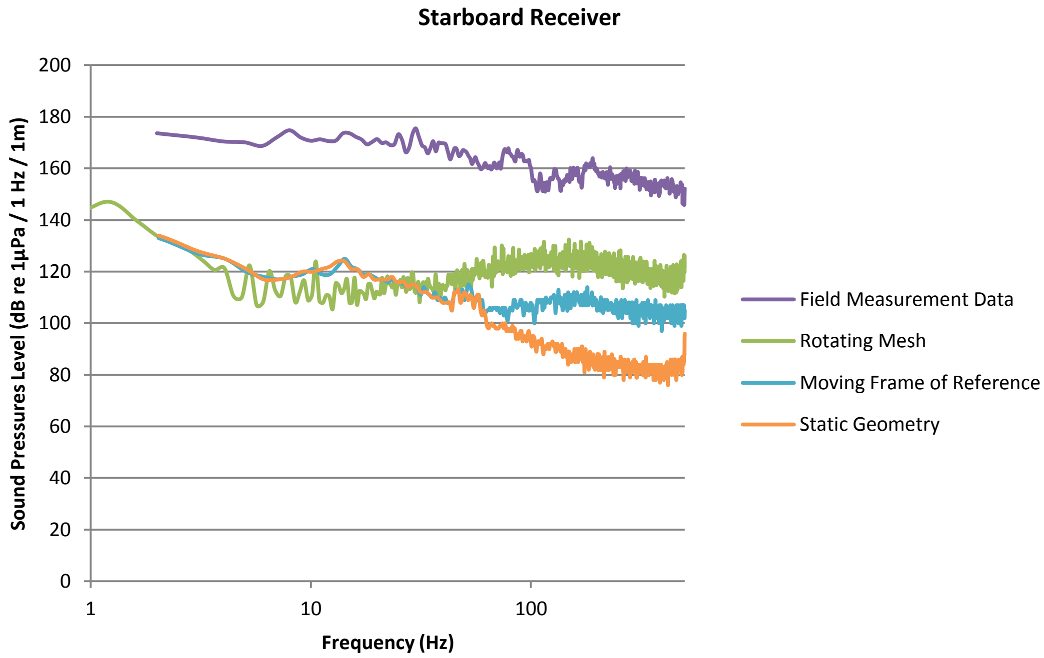
Ideal Case



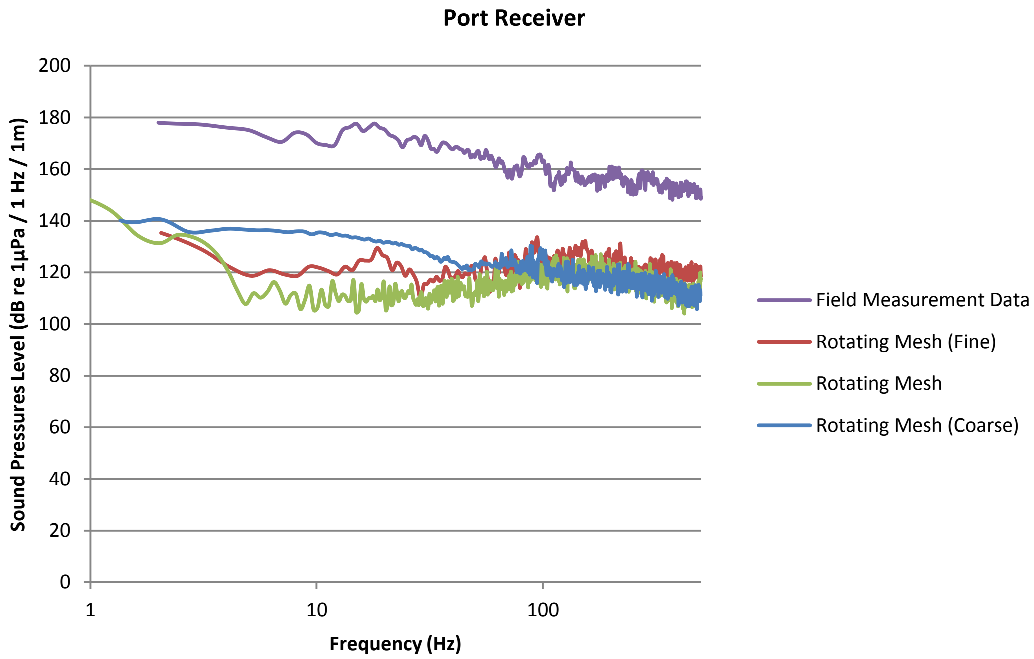


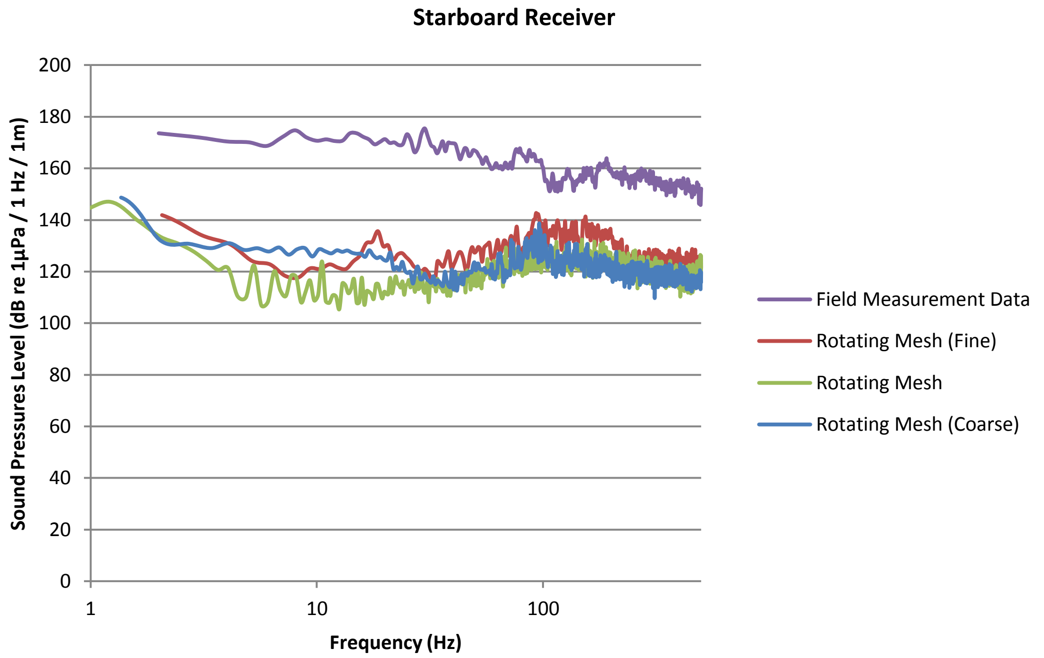
Propeller Representation Approaches



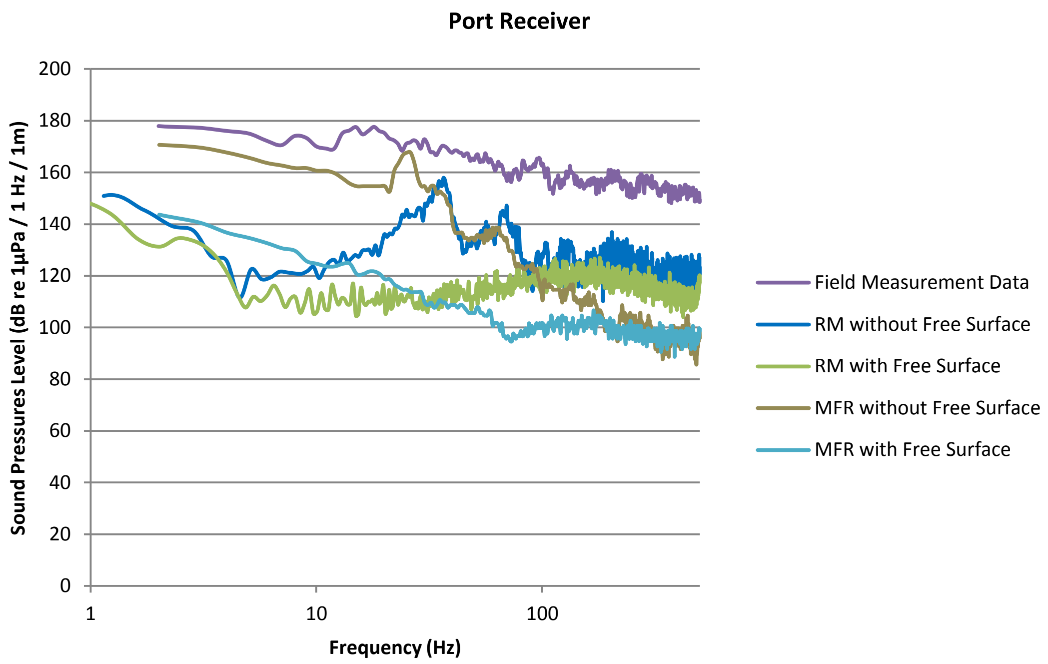


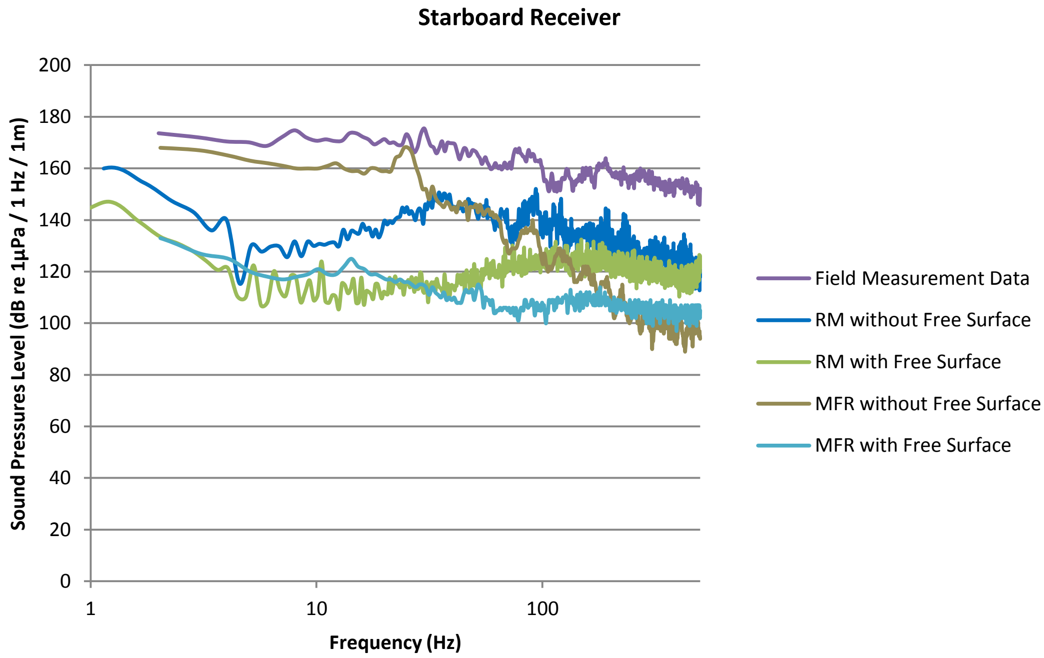
Grid Sensitivity Study



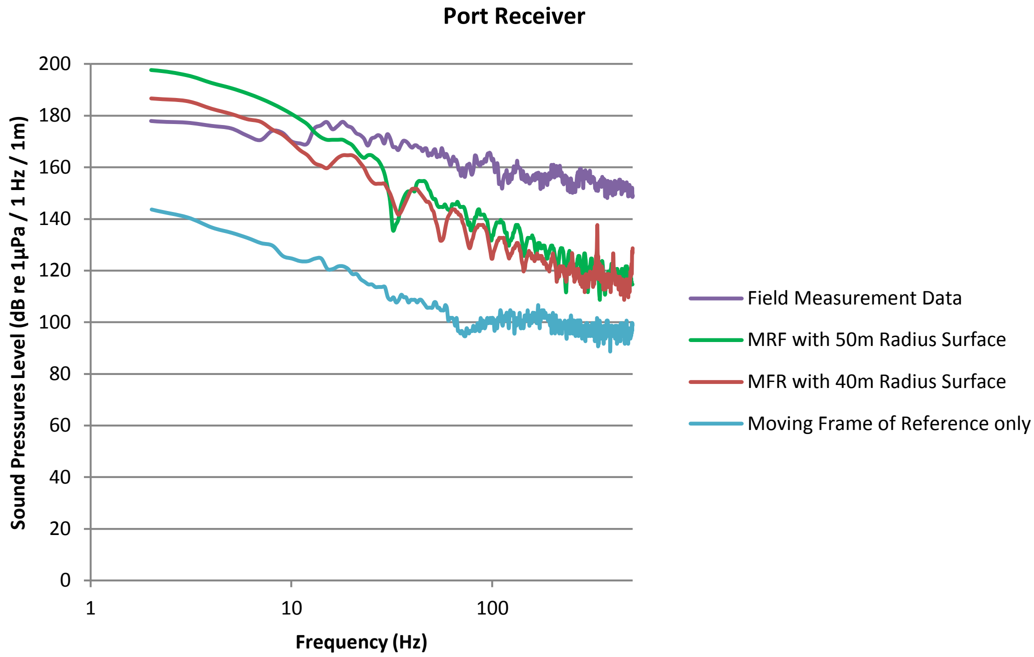


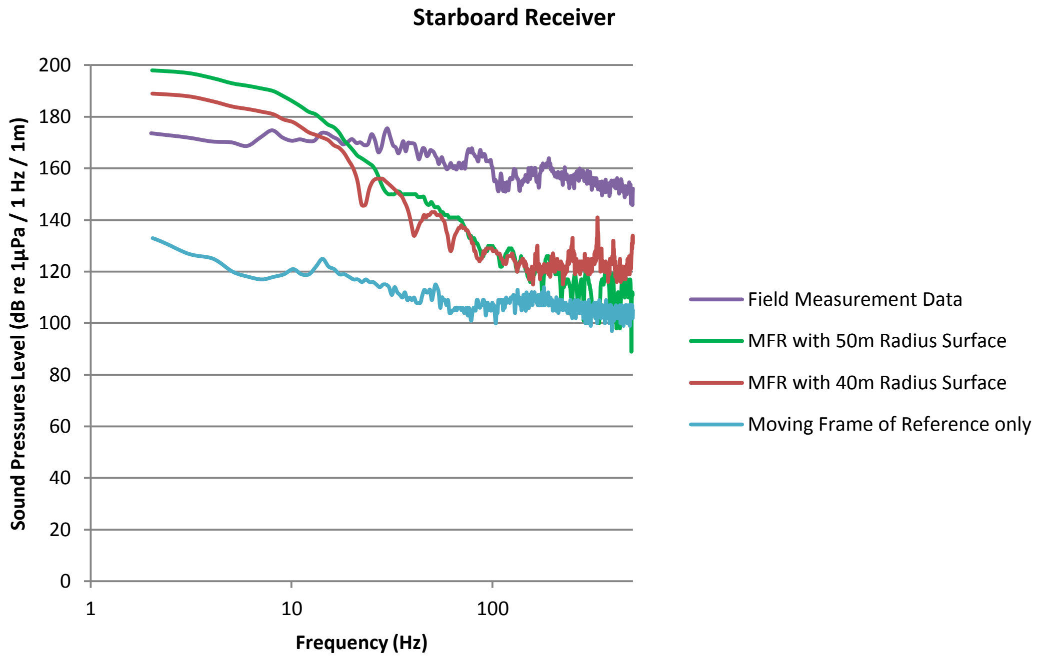
Free Surface



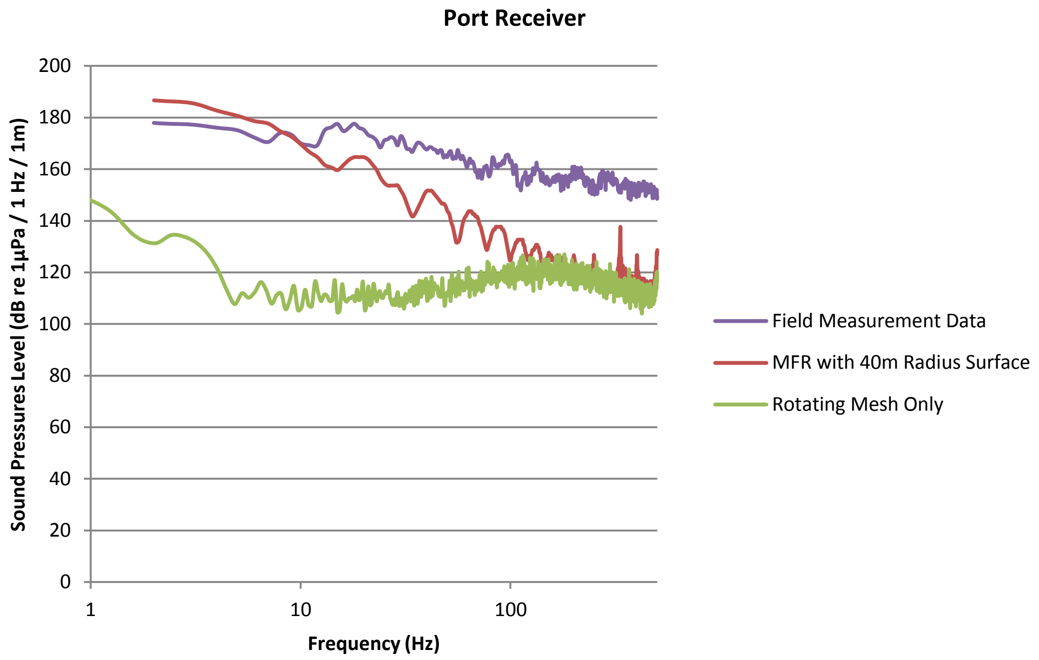


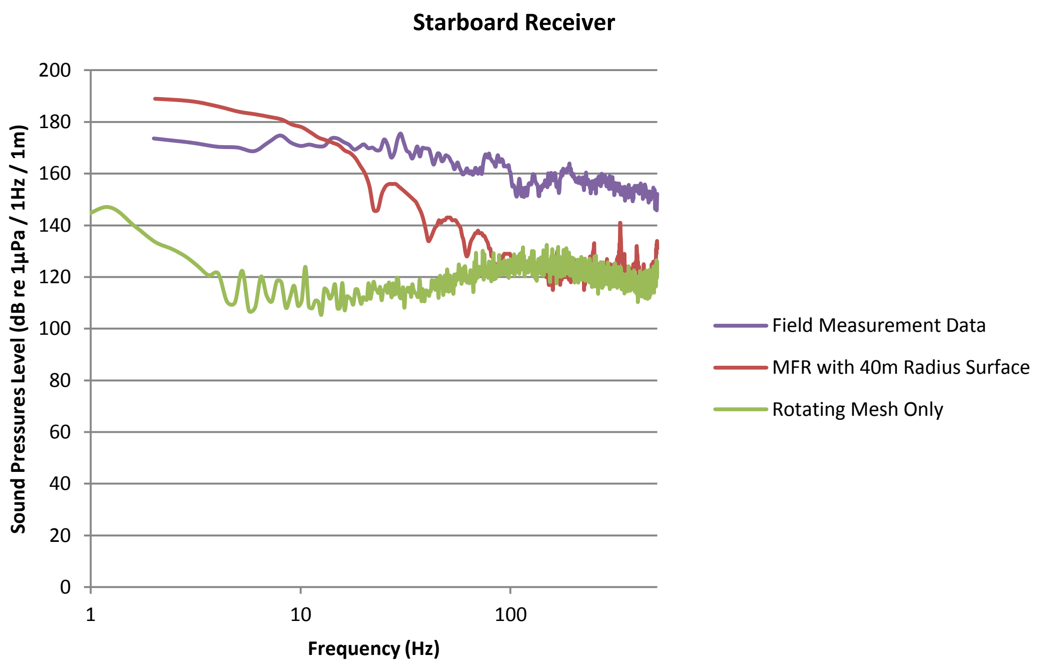
Moving Frame of Reference and Permeable Source Surface



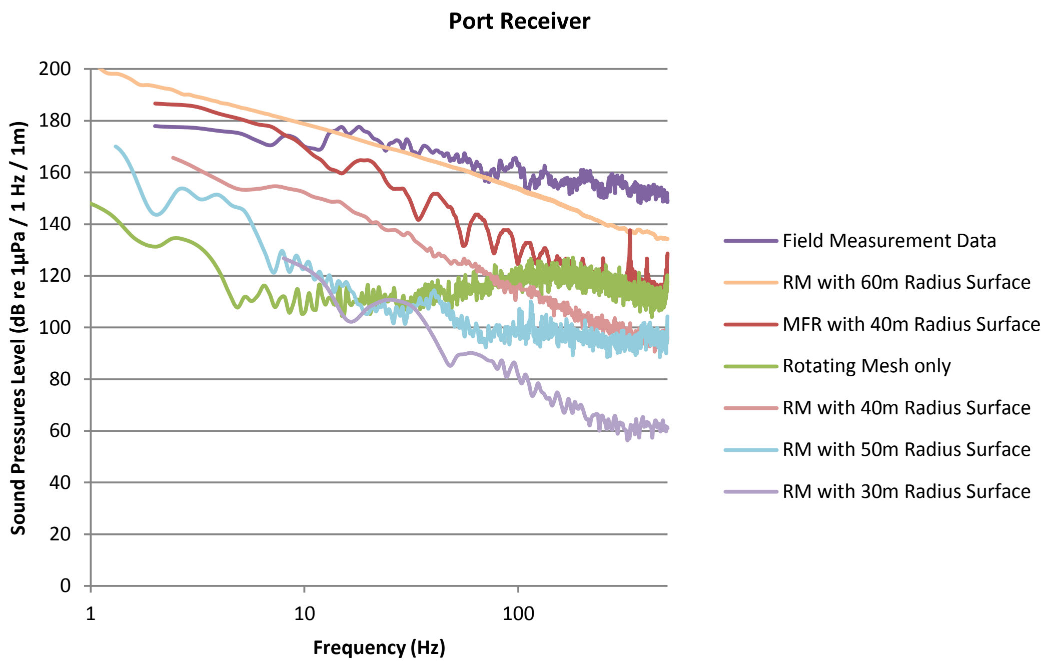


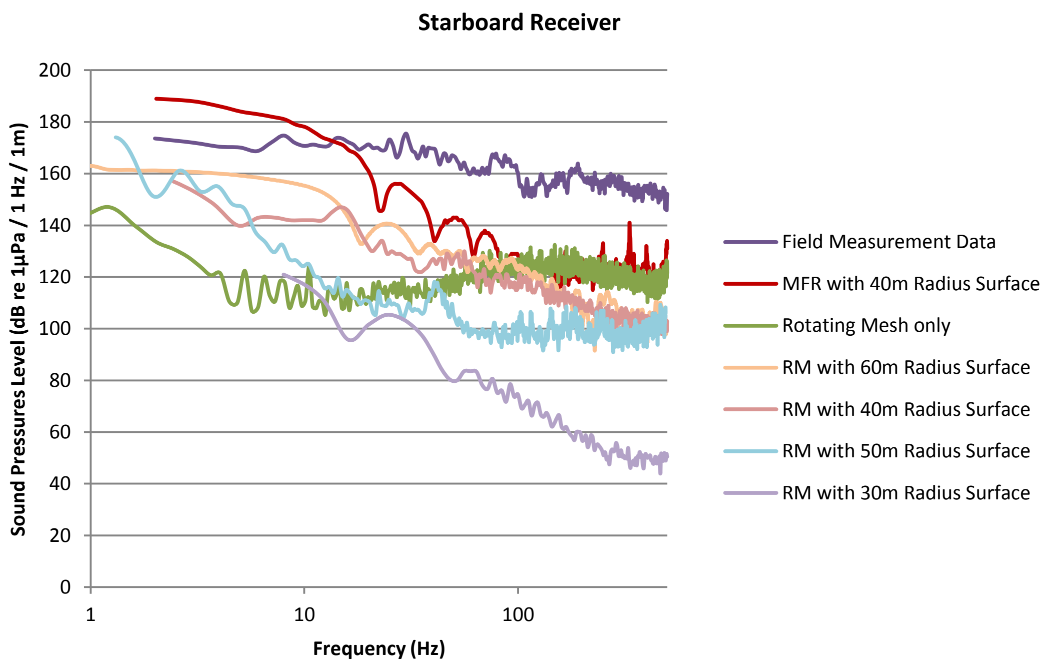
Rotating Mesh and Moving Frame of Reference with Permeable Source Surface





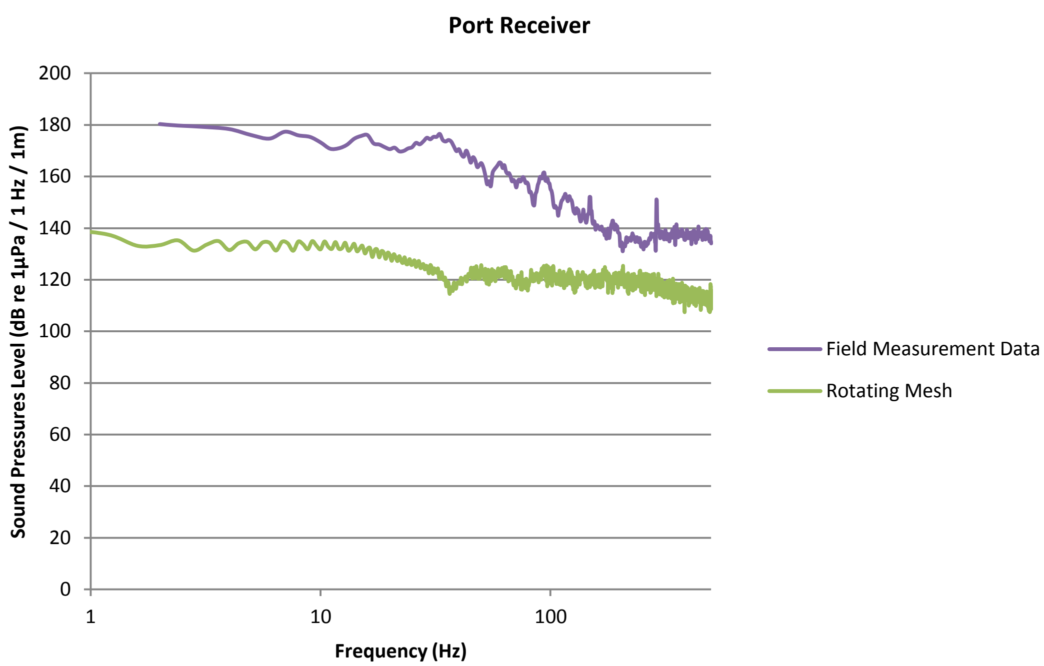
Rotating Mesh and Permeable Source Surface

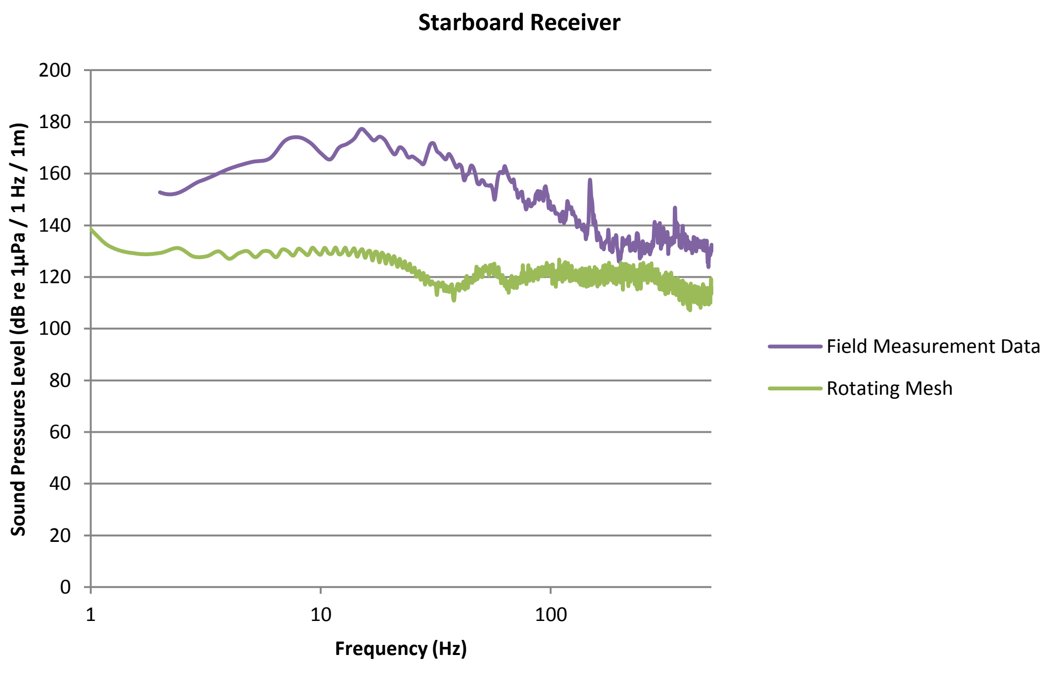




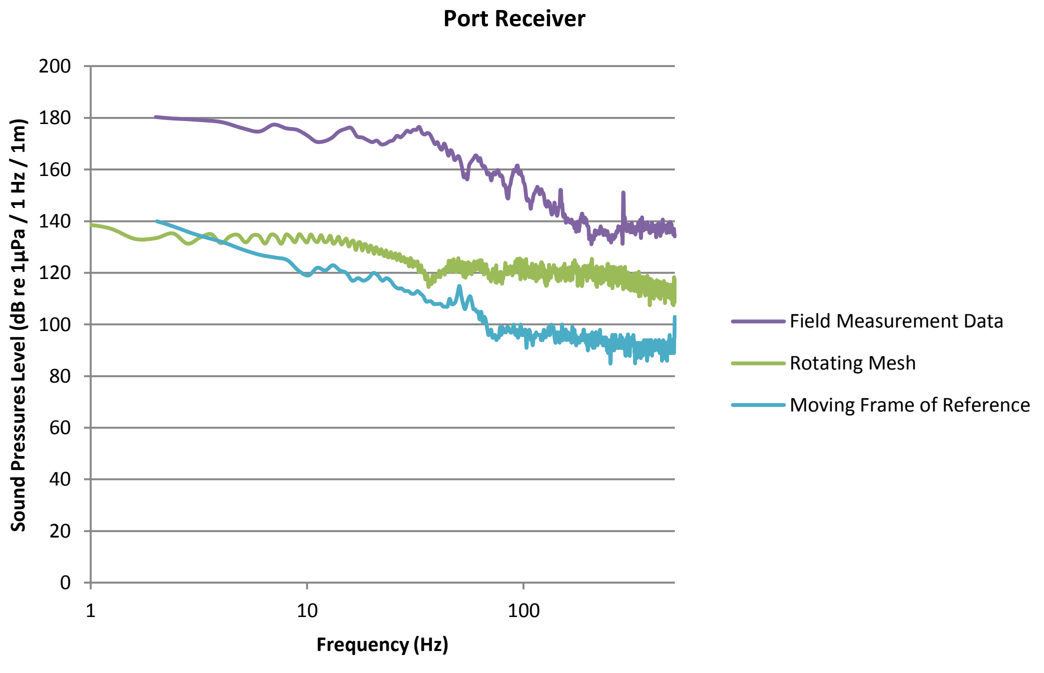
9 Knots

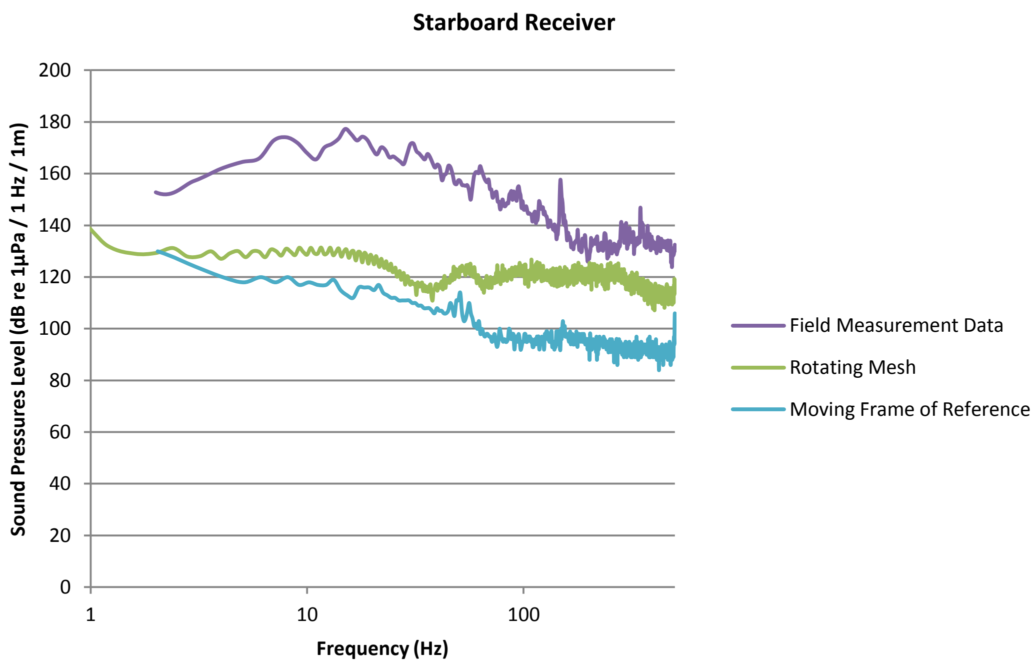
Ideal Case





Propeller Representation Approaches





Starboard Receiver
Sound Pressures Level (dB re 1μPa / 1 Hz / 1m)
200
180
160
140
120
100
80
60
40
20
0
1
10
100
Frequency (Hz)
Field Measurement Data
Rotating Mesh
Moving Frame of Reference

Appendix D - Sources of Error

The table below outlines the main sources of error which are associated with the study, and where applicable, details of how they have been addressed.

Section	Source of Error	Comment
Field Measurements	Inaccurate measurement of CPA distances, due to GPS error and hydrophone drift in current	If these errors are likely to be large, several sets of receivers may be required in the numerical model
	Measurement errors due to equipment used	The equipment should be calibrated prior to measurements
	Vessel speed and propeller RPM may not be constant throughout the measurement time frame	In cases where variations are small, the assumption of constancy will be suitable. Where there are large variations, either the run should be re-measured, or variations should be made in the numerical modelling
	Measurements were carried out in a comparatively shallow water environment	The complex interactions between sound waves and the sea bed will not be captured numerically
Field Measurement Results	Ambient noise contribution is included in the measurement results	Ambient noise also measured and contribution have been removed from field measurement results
	Results provided were corrected to source level using simplified spreading laws	Results were reverted back to "as measured" results for the CPA distance, and these were used for comparison
	Exact details of method used for averaging and post-processing of raw data are not known	In future, measurement standards may specify an approach to adhere to, to ensure consistency
Numerical Modelling	There may be discrepancies between a geometrical hull and propeller model, and the actual as-built vessel	It may not be possible to entirely eliminate this source of error
	Use of a full scale simulation leads to high Reynolds Numbers, which	Use of a model scale simulation may be more appropriate

	can cause difficulty in solving the flow characteristics	
	Use of a URANS approach introduces a significant level of approximation in solving the simulation in order to reduce computational demand	The user should be aware of the implications of different modelling choices. If there is concern, a comparative simulation could be carried out using LES or even DES
	There is a high level of mesh dependence in the numerical modelling results	Ensure consistency of meshing approaches, cell distribution and refinement. Grid sensitivity studies are indispensable
	The results are affected by time-step size and overall simulation run time	Ensure consistency in use of time-step sizes, always using the same size for acoustic predictions where possible. Also ensure overall run time allows suitable number of full propeller revolutions to occur, and for suitable simulation convergence
	F-WH approach uses averaged speed of sound and water density value	The method may not be suitable in its current form for very far field noise predictions
	The application of a flat wave for free surface representation will not be a fully accurate representation of operation conditions	The calm water condition represents the worst case scenario as any surface wave will increase sound attenuation resulting in lower propagated sound pressure levels, especially at higher frequencies
	The selection of models and solvers in the ideal case discussed is suitable for the software used only, as these combinations have been optimised in this case	Users of the numerical modelling approach will required a good level of CFD expertise
Empirical Modelling	The machinery tonal empirical models represent results in an ideal case where machinery is working exactly as expected	The tonal frequency prediction results are used as an indication only

	The empirical spectra estimations provide source level results using past ship data	The empirical estimates should be used as a preliminary indication only, and the user should be aware of errors arising from correcting results from source to receiver or vice versa
Impact Assessment	The database available for marine wildlife species is not exhaustive	The database should be updated regularly and should be used as a guide to the species likely to be affected, rather than a definitive list
	The hearing and vocalisation range data for species may be based on measurements or trials of only a few members of a species	The ranges should again be used a guide only, and updated whenever possible. It will be sufficient to highlight which species are likely to be affected by low, medium and high frequency noise
	The habitat areas of species may vary, or the operational area of a vessel may vary or be unknown	Impact assessment may need to be carried out for various operational scenarios, or using the database as a whole without filtering for area
	The published literature on observations of the reactions of different species to underwater noise can lack detail, or be limited to a few members of a species in one given population	The information provided in the database can not take into account all the variations within a species, but can still provide an indication of a possible reaction which should be considered