Raymond Fischer
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Noise and Vibration Control on Ships

Understanding and Cutting Through the Noise

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Raymond Fischer
(Retired)
Billerica, MA, USA

Leonid Boroditsky
(Retired)
Canton, MA, USA

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The ASA Press

ASA Press, which represents a collaboration between the Acoustical Society of America and Springer Nature, is dedicated to encouraging the publication of important new books as well as the distribution of classic titles in acoustics. These titles, published under a dual ASA Press/Springer imprint, are intended to reflect the full range of research in acoustics. ASA Press titles can include all types of books that Springer publishes, and may appear in any appropriate Springer book series.

The Acoustical Society of America

On 27 December 1928, a group of scientists and engineers met at Bell Telephone Laboratories in New York City to discuss organizing a Society dedicated to the field of acoustics. Plans developed rapidly, and the Acoustical Society of America (ASA) held its first meeting on 10–11 May 1929 with a charter membership of about 450. Today, ASA has a worldwide membership of about 7000.

The scope of this new Society incorporated a broad range of technical areas that continues to be reflected in ASA's present-day endeavors. Today, ASA serves the interests of its members and the acoustics community in all branches of acoustics, both theoretical and applied. To achieve this goal, ASA has established Technical Committees charged with keeping abreast of the developments and needs of membership in specialized fields, as well as identifying new ones as they develop.

The Technical Committees include acoustical oceanography, animal bioacoustics, architectural acoustics, biomedical acoustics, computational acoustics, engineering acoustics, musical acoustics, noise, physical acoustics, psychological and physiological acoustics, signal processing in acoustics, speech communication, structural acoustics and vibration, and underwater acoustics. This diversity is one of the Society's unique and strongest assets since it so strongly fosters and encourages cross-disciplinary learning, collaboration, and interactions.

ASA publications and meetings incorporate the diversity of these Technical Committees. In particular, publications play a major role in the society. The Journal of the Acoustical Society of America (JASA) includes contributed papers and patent reviews. JASA Express Letters (JASA-EL) and Proceedings of Meetings on Acoustics (POMA) are online, open-access publications, offering rapid publication. Acoustics Today, published quarterly, is a popular open-access magazine. Other key features of ASA's publishing program include books, reprints of classic acoustics texts, and videos. ASA's biannual meetings offer opportunities for attendees to share information, with strong support throughout the career continuum, from students to retirees. Meetings incorporate many opportunities for professional and social interactions, and attendees find the personal contacts a rewarding experience. These experiences result in building a robust network of fellow scientists and engineers, many of whom become lifelong friends and colleagues.

From the Society's inception, members recognized the importance of developing acoustical standards with a focus on terminology, measurement procedures, and criteria for determining the effects of noise and vibration. The ASA Standards Program serves as the Secretariat for four American National Standards Institute Committees and provides administrative support for several international standards committees.

Throughout its history to present day, ASA's strength resides in attracting the interest and commitment of scholars devoted to promoting the knowledge and practical applications of acoustics. The unselfish activity of these individuals in the development of the Society is largely responsible for ASA's growth and present stature.

Without a doubt, we would like to thank our wives for supporting and aiding in decisions and choices made over our careers. This often meant putting up with extensive overtime and time away from home. They found ways to keep the home front running and bringing up our families. Additionally, our wives were very encouraging and supportive while we were writing this book. We could not have done all this without their support.

Preface

> Even in the most beautiful music, there are some silences, which are there so we can witness the importance of silence. Silence is more important than ever, as life today is full of noise. We speak a lot about environmental pollution but not enough about noise pollution.
>
> ~ Andrea Bocelli

Quiet is proving to be a desirable goal with respect to both humans and animals. Creating a quiet ship is a complex process. Every ship type from small craft to large tankers can be engineered to be quieter than their previous types. How to achieve this given noisy, powerful machinery placed in the restricted space of a thin shell that accepts and radiates both airborne and underwater noise is the objective of this book. How to deal successfully with these contradictive requirements is laid out in terms of criteria, program planning, acoustic modeling, engineering of optimal treatments, and testing for compliance.

This book should prove useful to those involved in ship acoustics: designers, ship builders, ship owners/operators, marine engineers, and naval architects. They need a practical guide that should clarify the process of developing reasonable acoustic goals and the method of meeting them. Currently, there are no books in the technical literature that combine existing theoretical and practical experience in this field. This book is the first attempt to systematize this information. To the extent possible, the use of sophisticated mathematics was avoided; physical models are used to explain acoustical constructs.

The book contains multiple references that can be used to extend one's knowledge on a particular topic. With this approach, the reader should have the opportunity to solve acoustic problems from beginning to end. Specifically, how to meet acoustic criteria while designing optimal noise and vibration controls for a complex ship. Cutting through the noise can be simplified with a better understanding of the various facets.

Billerica, MA, USA — Raymond Fischer
Canton, MA, USA — Leonid Boroditsky

Acknowledgment

We would like to take the opportunity to acknowledge all the outstanding engineers and staff members who aided us in creating a quiet marine environment.

Contents

About the Authors

Raymond Fischer Fischer received his MS in Ocean Engineering from the University of Massachusetts, Amherst, in 1976. He was employed by Bolt, Beranek, and Newman (BBN) for 12 years and by Atlantic Applied Research for 5 years. In 1990, he started his own company, Noise Control Engineering, Inc. (NCE). He retired in 2016.

Fischer was extensively involved with the prediction of radiated noise, sonar self-noise, habitability noise, aural detectability, and structural vibrations on over 400 different vessels. He has also directed many shipboard noise and vibration measurements for demonstration of compliance with habitability criteria, for diagnostic testing and to verify noise modeling techniques. His areas of specialty include design, development, and noise control engineering to solve noise and vibration problems in small high-speed crafts, yachts, work boats, ice breakers, naval surface ships, and other nonconventional vessels. He was the Technical Leader for a US Navy Small Business Innovative Research (SBIR) program to provide more accurate shipboard airborne noise predictions. He coauthored the original and a Supplement to SNAME's Design Guide for Shipboard Airborne Noise Control, T&R 3-37. He has also contributed to the development of acoustic goals and several design guides for the US Navy and the Canadian Navy.

Fischer was the Principle Investigator for the Office of Naval Research's (ONR) Noise Induced Hearing Loss (NIHL) program. This multiyear effort started in 2010. This resulted in a new and verified acoustical engineering tool – DesignerNOISE®. New management and programmatic approaches for Navy acquisitions were also developed. Acoustic modeling tools were established for composite material constructions, underwater radiated noise, and return on investment.

Fischer was the project manager for the habitability and radiated noise control efforts on NOAA's Fisheries Research Vessel (under contract with VT Halter Marine), Fast Missile Craft (FMC), the Alaska Regional Research Vessel, and the development/optimization of spray-on damping treatments. Under his guidance, NCE conducted radiated noise trials on SEAFIGHTER (ONR's X-Craft), research vessels, seismic vessels, and commercial vessels.

He has 84 published papers on ship acoustics. He has over 400 technical reports or technical memos to his credit over his career.

Leonid Boroditsky Boroditsky graduated from Leningrad Shipbuilding Institute in 1959 with a master's in science degree. From 1959 to 1994, he was employed by the Acoustical Lab at the Central Research Institute of Shipbuilding Technology.

Boroditsky defended his PhD thesis about ship structure sound vibration insulation in 1966. His areas of special interest were icebreaker noise, flow noise, and propeller noise on fast craft. Boroditsky coauthored, with V. Spiridonov, a book entitled *Structureborne Noise Reduction in Ship Compartments* (1974).

Boroditsky joined NCE in 1996 and retired in 2021. He was a major contributor to the development of DesignerNOISE®, 3-D noise prediction software. He saw the implementation of this software into an ongoing design practice. His latest areas of interest were the acoustic characteristics of composite materials, aircraft carrier noise reduction, damping coating optimization, and on-deck noise control.

Boroditsky participated in numerous government and commercial projects to abate noise levels on ships. He performed multiple diagnostic tests for different kinds of vessels and developed treatments to mitigate mechanical and hydrodynamic noise.

Boroditsky has published papers in *Sudostroenie*, *Sound and Vibration*, *Noise Control Engineering Journal*, *American Acoustical Society*, *International Congress on Sound and Vibration*, *Inter Noise*, and *Naval Engineers Journal*.

Chapter 1
Introduction

If your goal is to achieve a quiet vessel, be it a ship, a small boat or anything in between, this generates the questions of 'how quiet' and 'for whom'? Are crew and/or marine life to be considered? Are noise and vibration both to be studied? What is the operating profile of the vessel, and how much time, effort and money will be invested to accomplish a quiet vessel? In the long run, designing, building and operating a quiet vessel will have significant impacts. These include weight, scheduling, space and cost. Other considerations are the impacts on the level and complexity of the design effort. The only way to minimize these sometimes 'adverse' impacts is to ensure that valid and implementable controls are utilized. These controls will need to be optimized with respect to the acoustic goals, vessel operation, and environment being protected. This means that this control process needs to be integrated into the overall ship design.

Ships have many intensive noise and vibration sources in close proximity to compartments that are noise sensitive – berthing, work and watch stations, and other manned positions. Source-path-receiver modeling is often used to predict this noise environment. Depending on the accuracy of the modeling used, one can determine and identify the optimal controls. Controls consist primarily of additional materials, noise and/or vibration treatments, and/or operational changes. In addition to the material/treatment costs, scheduling, testing and labor become factors affecting decisions on optimal procedures to meet goals or criteria. One assumption here is that the acoustic goals are appropriate and achievable.

The authors believe that necessary and sufficient noise and vibration control on ships should be based on detailed acoustical studies conducted during the design stage. The most successful projects show that acoustical design should be incorporated starting from the Concept or Preliminary Design stage. Fixing noise problems after Builder Sea Trials usually cost an order of magnitude more than if those treatments and material were considered during a proper design process. Unfortunately, consideration of acoustic goals is omitted or ignored in many designs due to their "usual" impact on space, weight and cost of treatments or just plain lack of interest.

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The core part of any acoustical design is based on realistic noise and vibration predictions: airborne and structureborne induced noise level calculations. These calculations must reflect the actual ship construction and systems. This book is an attempt to cover these issues in a comprehensible and concise manner. The noise prediction process includes knowledge of three parts: acoustic[1] source description, acoustic path analysis, and receiver compartment or location acoustic characteristics.

There is an enormous variety of acoustic sources on ships, more so than for most land-based factories. The most well-known are mechanical sources: main and auxiliary engines, gearboxes, pumps, compressors, fans, and other equipment basically located in machinery rooms. A noise source may be located not only inside but also externally to the ship. Aircraft landing and taking off may be a source of intensive on-deck noise, but they also influence the noise inside a ship. Main engine intake and exhaust systems very often generate high noise levels at on-deck stations and, sometimes, at inner compartments such as the Pilothouse.

Extensive ventilation systems may create excessive noise levels very distant from the machinery room. Ventilation noise is a combination of aerodynamic and mechanical noise. Practice shows that at some distance from engine rooms, ventilation (air conditioning) may be the main contributor to overall noise levels in compartments.

Propellers, thrusters, water jets and other propulsors are another category of noisy sources. Underwater radiated noise is basically connected with excessive noise levels from propulsors when they cavitate. Icebreaking and waves and inflow interactions with the hull are very important for noise in the bow compartments of some ships. The preponderance of these sources are considered in this book.

The acoustic path analysis requires information concerning hull structure, joiner panels, insulation, coating structures, machinery location, and locations of interest relative to critical sources. Noise path analysis includes analytical consideration of sound waves spreading through air and structure and analysis of a structure's ability to insulate, absorb, and radiate sound. Considering and analyzing this information can be a labor-consuming segment of the noise prediction process. The process is complicated by the strong frequency dependance of the acoustic sources, path effects, and receiver characteristics. For airborne noise, for instance, there are usually nine octave bands to be considered from 31.5 Hz to 8000 Hz, the typical range of human hearing. In addition, there are frequency weightings such as A-weighting for human hearing or vibration frequency weightings to account for vibrations affecting the human body along different axes. For underwater radiated noise, the frequency span can be much lower and higher – 1 Hz to 50,000 or 100,000 Hz.

Notwithstanding the sophistication of the noise prediction process for a ship's environment, one major principle must be formulated. That is, the separation of airborne noise from structureborne noise. This terminology basically reflects the different media related to the spreading of sound. Airborne noise is transmitted through the air. Structureborne noise is vibration spreading through ship structural

[1]The term "acoustic" as used herein generally includes both noise and vibration.

components with eventual noise radiation from this structure into air in the compartment of interest. Each of these components needs to be analyzed with different methods and are processed differently. These methods for airborne and structureborne noise calculation are discussed in detail in this book.

When excessive noise levels are expected and the relative contributions of structureborne noise and airborne noise have been determined, noise and vibration control approaches can be developed. This would include consideration of operating conditions such as speed and machinery line-ups, administrative controls such as allowed exposure time, and application of physical treatments. The treatments may insulate noise or vibration. They may absorb noise and vibration or may block it. In each case, it is important to qualify and quantify the effectiveness of each of the recommended treatments versus frequency. The treatments may have "global" (whole ship) or "local" (one structure or compartment) effectiveness; they may reduce only airborne or structureborne noise or both. For example, resilient mounts under machinery equally reduce the structureborne noise of this source everywhere in a ship; therefore, this treatment has a "global" effect. A joiner panel may reduce radiation from a bulkhead in one particular room where this joiner panel is installed. This is an example of a "local" effect. Damping tiles, depending on location and area, may have a combination of local and global effects. The effectiveness of salient noise and vibration treatments will be considered in this book individually and in combination. Other factors, such as the frequency range over which the treatment is effective, its weight and space impact, and that it meets marine regulatory requirements, need to be factored into this process.

The authors worked as acoustical consultants for over 40 years each. This book reflects their combined experience in marine acoustical consulting and the corporate experience of the consulting company Noise Control Engineering (NCE) established in the USA in 1991. Other papers and reference material by the authors are noted as needed. Prominent works by others in the field are also included.

The authors believe that it is not reasonable to provide the basics of acoustics, vibration theory, and wave spreading in elastic media, as these are widely published (Beranek 1971; Beranek and Ver 1992; Junger and Feit 1986). It is also assumed that the reader has basic knowledge regarding acoustics: frequency, decibel, sound pressure, acceleration, resonance, etc. A simplified engineering approach sometimes prevails over extensive and complicated scientific considerations. If an empirical approach leads to the goal faster than an analytical approach, the authors will use the first with a reference to any limiting assumptions.

The objective of this book is to educate ship owners/operators, naval architects/ marine engineers, ship builders, and engineering students on how to understand and address relevant factors involved in designing, building, delivering and operating a 'quiet' vessel. A vessel herein is defined as any mechanical system operating in an enclosed volume in a marine environment – fixed, floating, or submerged. 'Quiet' in this case addresses both the noise and vibration of a vessel and how it influences the operating environment both inside and outside the vessel.

A comprehensive discussion of the terms used in the field is given in several basic texts on acoustics or noise control (Fischer et al. 1983; Fischer 2020; Fischer and Boroditsky, 2001; Beranek 1971; Beranek and Ver 1992; Crocker 2007; Pierce 2019; Kinsler and Frey 1962; Fahy and Thompson 2015; Loeser 1999; Junger, 1986; Bies and Hansen 1988; Boroditsky and Spiridonov 1974; Fischer and Bahtiarian, 2017a, 2017b; Humes, 2005; Nikiforov, 2005; Plunt, 1980; Ross, 1983; Southall 2019; Urick, 1983; Discovery of Sound in the Sea - www.dosits.org; and Junger and Feit 1986). These references provide a discussion of relevant acoustic terminology and general principles that are needed as a basis for understanding the acoustic environment in and around a ship. This includes a discussion of the primary units for both noise and vibration – decibels, dB, and decibel math. The concepts of airborne noise and structureborne noise as important components are introduced.

References

Beranek, L. (ed.): Noise and Vibration Control. McGraw-Hill Book Co., New York (1971)
Beranek, L., Ver, I. (eds.): Noise and Vibration Control Engineering, Principles and Applications. John Wiley & Sons, Inc, New York (1992)
Bies, D., Hansen, C.: Engineering Noise Control. UNWIN HYMAN, Boston (1988)
Boroditsky, L., Spiridonov, V.: Structureborne Noise Control in Ship Compartments. Sudostroenie, Leningrad (1974) (in Russian)
Crocker, M. (ed.): Handbook of Noise and Vibration Control. Wiley, NY (2007)
Discovery of Sound in the Sea – http://www.dosits.org
Fahy, F., Thompson, D.: Fundamentals of Sound and Vibration, 2nd edn. CRC Press, Boca Raton (2015)
Fischer, R.: Sound and noise control. In: *The Marine Engineering Series*, 4th edn. The Society of Naval Architects and Marine Engineers (SNAME), Alexandria VA (2020)
Fischer, R., Bahtiarian, M.: Chapter 38: Ship noise control. In: Encyclopedia of Maritime and Offshore Engineering. Wiley (2017a)
Fischer, R., Bahtiarian, M.: Ship noise control. In: Finch, D. (ed.) Encyclopedia of Maritime and Offshore Engineering. Wiley (2017b). Online ISBN: 9781118476406
Fischer, R., Boroditsky, L.: Technical and Research (T&R) Bulletin 3–37 (Supplement), Supplement to the Design Guide for Shipboard Airborne Noise Control. The Society of Naval Architects and Marine Engineers (SNAME), Alexandria VA (2001)
Fischer, R., Collier, R.: Noise prediction and prevention on ships. In: Crocker, M. (ed.) Handbook of Noise and Vibration Control. Wiley, New York (2007)
Fischer, R., Nelson, D., Burroughs, C.: Technical and Research (T&R) Bulletin 3–37, Design Guide for Shipboard Airborne Noise Control. The Society of Naval Architects and Marine Engineers (SNAME), Alexandria VA (1983)
Humes, L., Joellenbeck, L., Durch, J. (eds.): Noise and Military Service: Implications for Hearing Loss and Tinnitus. National Academies Press (2005) http://www.nap.edu/catalog/11443.html
Junger, M., Feit, D.: Sound, Structures, and their Interactions, 2nd edn. The MIT Press, Cambridge, MA (1986)
Kinsler, L., Frey, A.: Fundamentals of Acoustics, 2nd edn. Wiley, NY (1962)
Loeser, H.: Fundamentals of Ship Acoustics, Acoustical Phenomena in and Around Ship Hulls. The Society of Naval Architects and Marine Engineers (SNAME), New Jersey (1999)

Nikiforov, S.: Acoustic Design of Naval Structures, D. Feit (English translation), Office of Naval Research, ADA 463069 (2005)

Pierce, A.: Acoustics. In: An Introduction to its Physical Principles and Applications, originally published in 1981, Reprinted 1989 in Physics Today, **34**(12) (2019).

Plunt, J.: Methods for Predicting Noise Levels in Ships. Chalmers University of Technology, Goteborg, Sweden (1980)

Ross, D.: Mechanics of Underwater Noise, 3rd edn. Peninsula Publishing, Los Altos. CA (1983)

Southall, B.L., Finnean, J., et al.: Marine mammal noise exposure criteria: updated scientific recommendations for residual hearing effects. Aquat. Mamm. **45**(2), 125–232 (2019)

Urick, R.: Principles of Underwater Sound, 3rd edn. Peninsula Publishing, Los Altos. CA (1983)

Chapter 2
Basic Acoustic Principles – Noise and Vibration

This chapter discusses the basic acoustic parameters used throughout this book. This includes information on the speed of sound and vibration in various media, relationships between acoustic power and sound pressure, sound propagation, direct v. reverberant fields, and transmissibility of sound through ship structures. Other parameters are the radiation efficiency of air-backed and water-backed plates and structure/fluid interaction. Finally, noise and vibration units are defined. Unless otherwise noted, the mks system is utilized throughout this text.

Basic parameters used in acoustics are covered quite well by Beranek (1971), Thumann and Miller (1990) and Engineering Acoustics from Wikibooks.[1] Some basic concepts, especially applicable to ships, are as follows:

- Some of the sound impinging on a bulkhead, deck or deckhead will be reflected, transmitted, or absorbed. The controlling factors are the Young's modulus, density, and thickness of the solid material and the amount of absorption on the source/receiver sides.
- The noise in each ship compartment is affected by the amount of internal noise due to any sources in that compartment, generally HVAC, and the amount of noise transmitted from the adjacent compartment(s).
- The best way to minimize noise transmitted throughout a ship is to isolate the vibration source from the structure before it can excite the compartment boundaries/foundations (as structureborne noise).
- The best way to reduce airborne transmitted noise is to vibration isolate joiner work and/or add mass and decoupling to the solid surfaces. Limp mass on a resilient underlayment such as fiberglass will provide this "decoupling."
- Any finite-sized structure on the ship has a resonant frequency – whether it is equipment, structure, piping or HVAC systems – which, if excited, may become a sound short or way for acoustic energy to be easily transmitted and likely cause a noise and/or vibration problem.

[1] https://en.wikibooks.org/wiki/Engineering_Acoustics

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- Ships, no matter how well designed, can become noisy due to sound shorts, poor maintenance, incorrect installation of treatments, and the fact that acoustic energy, such as heat, will take the shortest and path of least resistance to any compartment of concern.

Unfortunately, and this is the bane of acoustical consultants specializing in shipboard noise, the typical treatments discussed above:

- Add weight
- Take away space
- May be costly in terms of implementation and maintenance.

2.1 Decibel Addition/Subtraction and Units for Noise and Vibration

The decibel units used in acoustics are not linear. Going from 60 dB to 120 dB, such as going from a temperature of 10 °C to 20 °C, is not twice as loud or twice as hot. One rule of thumb to remember is that an increase of 5 dB in sound is usually *perceived* as being twice as loud. Thus, going from 60 to 65 dB is considered by most as twice as loud. However, some people are more sensitive to noise (and vibration), and a change of even 2 dB can be noticeable and problematic.

A majority of the calculations carried out in this book are simple arithmetic operations, except in regard to adding and subtracting decibel values. In this case, it is necessary to 'combine' the values logarithmically. When two or more sound pressure levels, vibration levels, or power levels are combined, the following steps need to be taken:

- Take the antilogs of each dB value
- Arithmetically sum these antilogs
- Determine the log value of this sum

For example, the combination of three acoustic sources of 90, 95, and 96 dB re 20 μPa (or 2×10^{-5}N/m^2) would be:

$$Combination = 10\ \mathrm{Log}\left(\sum\left(10^{90/10} + 10^{95/10} + 10^{96/10}\right)\right) = 99\ \mathrm{dB\ re\ 20\ \mu Pa} \quad (2.1)$$

It is slightly more complicated when trying to determine the contribution of only one source when all that is measured is the combined level of two sources and the level with just one of the sources secured. Then, the math is as follows. For the case where the combined level from two sources is, L_{comb} is 90 dB, and the measured level with just one source L_{p1} is 86 dB. Then, the unknown source L_{p2} is

$$L_{p2} = 10\ \mathrm{Log}\left(\sum\left(10^{L_{comb}/10} - 10^{L_{p1}/10}\right)\right) = 88\ \mathrm{dB\ re\ 20\ \mu Pa} \quad (2.2)$$

When pressure levels, vibration levels, or power levels are added to any type of transfer function,[2] the values are treated arithmetically. For example, if a source level of 90 dB re 20 μPa is attenuated by spreading losses by 6 dB, the received level is 90–6 = 84 dB re 20 μPa.

All values in decibels should be rounded up to the nearest whole number. Fractions of a dB do not amount to anything of value. Along the same vein, values within +/−2 dB can be considered almost equivalent and sometimes are with certain criteria.

2.2 Units for Noise and Vibration

One should note that the primary quantities of interest are sound pressure (p), acoustic power (W), and vibration – as either velocity (v) or acceleration (a).[3] One should always bear in mind that sensors and instrumentation that measure these parameters always use squared quantities for pressure and velocity; hence, '20 Log' is used in the equations. Power is not reported as a squared quantity – hence 10 Log is used in the power level equations.

Therefore, when computing a dB value for a physical quantity, the following equations apply:

Pressure, p, measured in N/m^2 or Pa, and using a standard reference of $p_o = 20$ μPa (or 20×10^{-6} Pa) is converted to a sound level value in decibels as:

$$L_p \text{ dB re } 20\ \mu\text{Pa} = 20 \text{ Log}\left({}^{p}\!/_{20 \times 10^{-6}} \right) \tag{2.3}$$

This book uses acceleration whenever possible. Thus, a given value of acceleration, a (m/sec^2), can be converted into an equivalent level dB re 1 μG (1 G = 10 m/sec^2 and 1 microG[4] is 10×10^{-6}m/sec^2):

$$L_a \text{ dB re } 1\ \mu\text{G} = 20 \text{ Log}\left({}^{a}\!/_{10 \times 10^{-6}} \right) \tag{2.4}$$

Using the relationships between rms acceleration, a(m/sec^2), velocity, v(m/sec); and displacement, d(m); at frequency f (Hz):

$$a = 2\pi f \nu \text{ and } \nu = 2\pi f d \tag{2.5}$$

One can develop the following equations to convert to velocity in dB re 1×10^{-8} m/sec or displacement in dB re 1×10^{-11} m:

[2]Transfer functions relate an input quantity to an output quantity.

[3]Unless otherwise noted the metric system is used throughout.

[4]A microG is technically 9.8×10^{-6}m/sec^2; the G value for convenience is rounded to 10.

$$L_\nu \text{ dB re } 1 \times 10^{-8} \frac{\text{m}}{\text{sec}} = L_a \text{ dB re } 1\ \mu\text{G} + 44 - 20 \text{ Log}(f) \tag{2.6}$$

and

$$L_d \text{ dB re } 1 \times 10^{-11}\text{m} = L_a \text{ dB re } 1\ \mu\text{G} + 88 - 40 \text{ Log}(f) \tag{2.7}$$

In an historical note, the space/time averaged square-velocity on a surface of a structure is proportional to the total energy stored in a resonant system. This is proportional to the radiated sound power. In this case, the reference velocity can be chosen so that the velocity level numerically equals the sound pressure level on the surface of the structure radiating into air. Thus:

$$L_p = 20 \text{ Log } {p}/{p_0} = L_\nu = 20 \text{ Log } {\nu}/{\nu_0} \tag{2.8}$$

and

$$\nu = {p}/{(\rho_0\ c_0)} \tag{2.9}$$

where $p_o = 20$ μPa, $\rho_0 c_0 \sim 400$ mks rayls, and rayls – the sound specific impedance – is defined as the ratio between the sound pressure and the particle velocity it produces.

In this case, $\nu_0 = {p_0}/{\rho_0\ c_0} = 5 \times 10^{-8}$ m/sec rather than the 1×10^{-8} m/sec reference currently utilized. This implies that using the standard v_0 value of 1×10^{-8} m/sec would over predict the equivalent sound pressure by a factor of 14 dB.

Along this same line, the sound power, W (watts), radiated by a plate of surface area, S (m^2), is related to the velocity, v (m/sec), of the plate as:

$$W = \langle \nu^2 \rangle\ \rho_0\ c_0 S\ \sigma_{rad} \tag{2.10}$$

and

$$\nu = {p}/{\rho_0\ c_0} \tag{2.11}$$

where the brackets ⟨ ⟩ denote the root mean square value.

Then, the radiated power, L_w dB re 1×10^{-12} W, is

$$L_w \text{ dB re } 1 \times 10^{-12} \text{ W} = L_\nu \text{ dB re } 1 \times 10^{-8} \text{ m/s} + 10 \text{ Log}(S) + 10 \text{ Log}(\sigma_{rad}) - 14 \tag{2.12}$$

where σ_{rad} is the radiation efficiency of the ship plating (see also Sect. 2.7). At higher frequencies, typically above 2000 Hz, the radiation efficiency term goes to zero, and the radiated power is directly related to the sound pressure, as modified by the plate area.

The sound pressure squared, p^2, at a distance, r, m, from a source is equal to:

$$p^2 = (\rho_0\, c_0)^2\, Q\, \langle v^2 \rangle\, S\, \left(\sigma_{rad}/4\pi r^2\right) \tag{2.13}$$

and substituting acceleration for velocity yields

$$p^2 = (\rho_0\, c_0)^2\, Q\, \left(\langle a^2 \rangle/(2\pi f)^2\right)\, S\, \left(\sigma_{rad}/\left(4\pi r^2\right)\right) \tag{2.14}$$

Thus,

$$L_p \text{ dB re } 20\ \mu\text{Pa} = 19 + 10\,\text{Log}(Q) + L_a \text{ dB re } 1\ \mu\text{G} + 10\,\text{Log}(S) + 10\,\text{Log}(\sigma_{rad}) - 20\,\text{Log}(f) - 20\,\text{Log}\, r \tag{2.15}$$

where Q is the directivity factor and r is the distance source to the receiver.

In the hemi-spherical acoustic nearfield, for a 1 m^2 area and above the coincidence frequency, this equation reduces to

$$L_p \text{ dB re } 20\ \mu\text{Pa per } 1\ \text{m}^2 = L_a \text{ dB re } 1\ \mu\text{G} - 20\,\text{Log}(f) + 22 \tag{2.16}$$

2.3 Speed of Sound in Air, Fluids, and Structures

The speed of sound in any fluid is temperature-, pressure-, and density dependent. Thus, the speed of sound in a gas turbine exhaust is different than that in cabins. That in air is different than that in water. The speed of sound in air is $c_a = \sqrt{\gamma P/\rho}$, where γP is the bulk modulus, P is the absolute air pressure (Pa), γ is the ratio of specific heats (1.4), and ρ is the mass density (kg/m^3). This is equal to 343 m/s for air. In water, the speed of sound is $c_w = \sqrt{B/\rho}$, where B is the fluid's bulk modulus. For water, the speed of sound is approximately 1500 m/s.

Always, bear in mind that the pressure, intensity, or power level in air is different than the pressure, intensity or power level in a fluid by the ratio of their acoustic impedance, Z, which is equal to ρc.

The speed of a flexural wave in a thin plate in vacuo is $c_p = \sqrt{Eh^2\omega^2/12\rho}$, where E is the Young's modulus of the material, h is the plate thickness, ω is $2\pi f$, and ρ is the mass density of the material. Note that this speed is frequency dependent, whereas the speed of sound in a fluid or air is independent of frequency. This speed is slower if the plate is water loaded on one side or both sides (Loeser 1999). This dissimilarity leads to the 'critical frequency' component of radiation efficiency discussed below.

The wavelength, λ(m), of sound in air at 21 °C at a frequency, f, is:

$$\lambda = c/f \tag{2.17}$$

where c = 343 m/sec, the speed of sound.

2.4 Sound from a Source of Known Acoustic Power

Sound, L_p, at a distance r from a small point source of known acoustic power, L_w dB re 1 pW, is simply computed as:

$$L_p(r) \text{ dB re } 20\ \mu\text{Pa} = L_w - 20\ \text{Log}(r) + 10\ \text{Log}(Q) - 11 \tag{2.18}$$

where r is the distance from the source and Q is the source directivity (the ratio of the acoustic power radiated to the receiver to the average radiated in all directions). This equation is correct for free space, showing that the received level decreases by 6 dB for a doubling of distance. Often, the sound will come from a 'finite' sized source of dimensions a and b, with $a > b$. In this case, the received level will depend on the distance r relative to the a and b dimensions of the plate as:

$$\begin{aligned} r < b/\pi &: L_p \text{ dB re } 20\ \mu\text{Pa} = L_w + 10\ \text{Log}(\pi/4ab) \\ b/\pi < r < a/\pi &: L_p \text{ dB re } 20\ \mu\text{Pa} = L_w - 10\ \text{Log}(r) - 10\ \text{Log}(4a) \\ r > a/\pi &: L_p \text{ dB re } 20\ \mu\text{Pa} = L_w - 20\ \text{Log}(r) - 11 \end{aligned} \tag{2.19}$$

If the source is in a fluid such as water, the equation equivalent to [2.18] is[5]:

$$L_p(r) \text{ dB re } 1\ \mu\text{Pa} = L_w - 20\ \text{Log}(r) + 10\ \text{Log}(Q) + 51 \tag{2.20}$$

using the typical reference of 1 μPa for underwater noise.

It is worth noting that the supposed conversion rate between the radiated acoustic power in water and the associated mechanical power is 1×10^{-8}. In air, this ratio is between 10^{-2} and 10^{-4}. Thus, a mechanical power of 1 kW would equate to an acoustic power of 70 dB re 1 pW in water and 110 dB re 1 pW in air. The equivalent radiated pressures would be 99 dB re 20 μPa−1 m in air and 121 dB re 1 μPa−1 m in water. In essence, the same mechanical input causes a higher pressure reading in water than in air for the referenced values. This is something to be very much aware of when comparing pressure and power levels in decibels in air and in water.

[5] For underwater noise, the reference p_o would be 1 μPa. The reference pressure in water is equated to −120 dB while that in air is −94, a difference alone that adds 26 dB to the same pressure in air.

2.5 Direct Versus Reverberant Field

Engine or equipment rooms typically have a variety of acoustic sources. If the acoustic power level, L_w dB re 1 pW, is given for each item, the resulting noise in the compartment can be calculated. A "small" source will radiate uniformly in all directions, and the pressure in a free field decays at a rate of 6 dB per doubling of distance from the source. Placing this source inside an enclosure will result in a reverberant field, consisting of reflections from the enclosure walls. The pressure will no longer decay at a rate of 6 dB per doubling of distance; however, it will decay to a set level depending on the type and quantity of absorptive materials inside the enclosure or room. This results in the following equation used to determine the direct and reverberant sound level, L_p, in a compartment due to a source in the compartment that emits a known sound power level, L_w:

$$L_p \text{ dB re } 20\ \mu\text{Pa} = L_w + 10\ \text{Log}\left(Q_\theta / \left(4\ \pi\ r^2\right) + 4/R\right) \tag{2.21}$$

where Q_θ is the directivity factor depending on the position of the source in the room (+3 dB if placed in the floor, +6 dB is placed at the intersection of two surfaces, and +9 dB if placed in a corner); r is the distance from the source, m, and R is the Room Constant, m^2, of the room.

The room constant, R, a measure of the absorption in a room, is computed as:

$$R = S\ \alpha_{sab}, m^2 \tag{2.22}$$

where S is the surface area of the compartment, m^2, and α_{sab} is the average Sabine absorption coefficient.

The closer the Sabine absorption coefficient is to one, the lower the reverberant pressure level in the room due to lower reflections from the boundaries. The $1/r^2$ term in Eq. 2.21 denotes the 6 dB reduction per doubling of distance in the free field portion of the room.

For a 'typical' berthing compartment with a modicum of absorption on the overhead and bulkheads, the reverberant field exists within approximately 1 meter of a point source, as shown in Table 2.1. In this case, the sound pressure would only decrease by 2 dB going from 0.5 to 1 m from the source rather than by 6 dB if the source were in a free field. If this 90 dB re 1 pW source were in the free field, the pressure level at 1 m would be 79 dB re 20 μPa versus the reverberant level of 89 dB. This point, where the reverberant field equals the direct field, is called the Hall Radius, r_H, and is equal to $r_H = R/(16\ \pi)$.

As shown in Table 2.2 for an engine room, the reverberant field starts approximately 2 meters from a point source. In this case, the sound pressure would only decrease by 3 dB going from 0.5 to 1 m from the source rather than by 6 dB if the source were in a free field. This is due to the larger room dimensions. Note that at low frequencies, the absorption value significantly decreases, resulting in a higher received pressure both in the free field and reverberant fields, as shown in Table 2.2.

Table 2.1 Representative free versus reverberant sound calculations – Berth and Engine Room

	Berth Dimensions, m^2	α_{sab} [a]	Distance r, m	Received L_p dB re 20μPa @ r
Deck	16.7	0.02	0.5	87.8
Overhead	16.7	0.7	1	85.7
Bulkheads	12.5	0.15	2	
			4	84.7
	Engine Room Dimensions, m^2	α_{sab}	Distance r, m	Received Lp dB re 20 μPa @ r
Deck	41.8	0.02	0.5	86.4
Overhead	41.8	0.7	1	83
Bulkheads	19.7	0.15	2	81.5
			4	81

[a]For mid- to high-frequency

Table 2.2 Free versus Reverberant Sound for Engine Room with Little Absorptive Treatment

	Engine Room Dimensions, m^2	α_{sab}	Distance r, m	Received Lp dB re 20 μPa @ r
Deck	41.8	0.01	0.5	90
Overhead	41.8	0.1	1	89
Bulkheads	19.7	0.05	2	88.7
			4	88.6

Note that in the free field, for $r < 1$ m, the received pressure for the same source is decreased by 5 dB, and the reverberant field is increased by 5 dB due to the decreased absorption over the case shown in Table 2.1.

The absorption coefficient is affected not only by the absorption on the boundaries but also by the type and extent of furnishings in the compartment. Additionally, the bulkheads, decks, and overheads themselves affect how much energy is transmitted into adjacent compartments through the boundaries. This 'apparent absorption' is significant at low frequencies (Fischer and Boroditsky 2001). Finally, in very large rooms, atmospheric absorption[6] may be important above 2000 Hz; atmospheric absorption is critical with respect to propagation in the free field – such as from exhaust stacks to nearby port facilities.

For an on-deck or free-field acoustic source, Eq. 2.22 can be used without the *4/R* term. In this case, the Q_θ term accounts for the directivity of the source. Furthermore, ducted openings are often highly directive[7].

Representative absorption coefficients are provided in Table 2.3 (Fischer et al. 1983). Included in this table is the NRC – Noise Reduction Coefficient, which is an arithmetic average of the absorption coefficient, α_{sab}, between the 125 and 4000 Hz

[6]ANSI/ASA S1.26-2014 Method for Calculation of the Absorption of Sound by the Atmosphere
[7]See SNAME T&R 3-37, Table 7.1 and SNAME Supplement, Appendix E: Design Guide Errata