GUIDANCE NOTES ON

# ONBOARD SHIP NOISE ANALYSIS

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# Foreword

These Guidance Notes provide an overview of and specific guidance on onboard ship noise analysis methodologies.

There is an increased demand for onboard ship noise analysis due to several industry requirements relating to noise exposure levels for seafarers. These include the International Labour Organization's (ILO) Maritime Labour Convention (MLC), 2006 and the IMO Code on Noise, which came into force in July 2014. To support ship designers and builders in improving the acoustic design of their ships, ABS has developed these Guidance Notes and has offered noise analysis services for years.

Included in these Guidance Notes are commonly-used modeling and analysis methods, as well as the basic concepts for the evaluation of noise analyses. These Guidance Notes can be used to assist in the noise analysis for ocean-going vessels.

These Guidance Notes become effective on the first day of the month of publication.

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**GUIDANCE NOTES ON**

# ONBOARD SHIP NOISE ANALYSIS

## CONTENTS

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# SECTION 1 General

## 1 Introduction

Noise on board ships has seen increased attention in recent years. Excessive noise levels on board ships can adversely affect the task performance and health of seafarers. Seafarers may become distracted if exposed to high noise and vibration levels and this can increase the potential for human error. Prolonged exposure to high-noise environments can lead to long-term health issues such as noise-induced hearing loss. It is expensive and difficult to fix noise-related issues after construction. Therefore, it is important and necessary for ship designers and builders to perform noise analyses and address these concerns at an early design stage. ABS has offered noise analysis service for years.

For onboard ship noise analyses, the commonly adopted “Source-Path-Receiver” modeling technique may be used. This modeling scheme takes three key elements into consideration: noise sources, transmission paths, and the receiver acoustic characteristics. Empirical methods and numerical analysis methods can be used to calculate the sound attenuation from “Sources” to “Receivers” through different transmission “Paths”. Various empirical methods have been developed. For example, The Society of Naval Architects and Marine Engineers (SNAME) published the *Design Guide for Shipboard Airborne Noise Control* [1], which provides a step-by-step empirical method to predict onboard ship noise levels. For the numerical analysis method, the statistical energy analysis (SEA) method is one of the more effective methods to predict onboard ship noise levels. It is efficient in addressing noise issues in complex structures at high frequencies.

## 3 Application

These Guidance Notes can be used to assist in the noise analysis for ocean-going vessels.

## 5 Scope

In these Guidance Notes, the following topics are discussed:

*i)* Source-Path-Receiver Modeling

*ii)* Results Evaluation

*iii)* Empirical Method

*iv)* SEA Method

The Source-Path-Receiver Modeling in Section 2 provides an overview of the modeling method of onboard ship noise analysis. It introduces methods for the modeling of the noise source, transmission path, and the receiver room.

The Results Evaluation in Section 3 provides the basic concepts of evaluating the results of the noise analyses. It introduces the concept of frequency band analysis and the most widely-used weighting method, A-weighting method, to consider the sensitivity of the human ear.

The Empirical Method in Appendix 1 provides the empirical formulae for the calculation of transmission attenuation from airborne sound, structure-borne sound, and duct-borne sound.

The SEA Method in Appendix 2 introduces the general concepts of SEA, which is a widely-used numerical method for onboard ship noise analysis.

## 7 Relevant Documents

- ABS *Guidance Notes on Noise and Vibration Control for Inhabited Spaces*
- ABS *Guide for Crew Habitability on Ships*
- ABS *Guide for Crew Habitability on Workboats*
- ABS *Guide for Habitability of Industrial Personnel on Accommodation Vessels*
- ABS *Guide for Passenger Comfort on Ships*
- ABS *Guide for Comfort on Yachts*
- ABS *Guide for Compliance with the ILO Marine Labour Convention, 2006 Title 3 Requirements*

## 9 Terminology

*Acceleration.* A vector that specifies the time rate of change of velocity (units of m/s$^2$(ft/s$^2$)). The acceleration of the vibratory motion of a structure can be specified in terms of the peak, average, or root-mean-square (rms) magnitude of the acceleration in a given direction. In this document, the acceleration levels are given in terms of the rms acceleration amplitude.

*Airborne Sound (or Noise).* Sound or noise that is transmitted through air.

*A-weighted Sound Pressure Level.* The magnitude of a sound, expressed in decibels (i.e., 20 micropascals); the various frequency components are adjusted according to the A-weighted values given in IEC 61672.1 (2004) in order to account for the frequency response characteristics of the human ear. The symbol is $L_A$, and the unit is dB(A). The measurement $L_{Aeq}$ is an equivalent continuous A-weighted sound pressure level, measured over a period of time.

*Cavitation Inception Speed.* Vessel speed at which a propeller starts to cavitate.

*Control Volume.* A volume moving with constant flow velocity through which the continuum acoustic energy flows.

*Coupling Loss Factor.* It is a parameter unique to statistical energy analysis (SEA), which measures the rate of the energy flowing out of a subsystem through a junction to another subsystem.

*Damping.* The dissipation of energy with time or distance. In this document, damping generally refers to dissipation of vibrating energy in structures.

*Damping Loss Factor.* A parameter indicating the ability to dissipate the sound energy within the structures.

*Decibel (dB).* A dimensionless unit of measure of the ratio of two quantities, $P_1$ and $P_2$, each of which is equal to or proportional to power. Ten times the logarithm to the base 10 of the ratio, $P_1/P_2$, has the dimensions of dB.

*Directivity.* A measure of the directional characteristic of a sound source.

*Direct Sound Field.* A sound field in which energy is flowing outward from the source without interference from surrounding surfaces. The sound field very close to a source, even in a reverberant room, is a direct field. Sound fields outdoors are direct fields at all distances from the source and are referred to as "free sound fields" or "free fields".

*Dynamic Positioning (DP).* A system to automatically maintain a vessel's position and heading by controlling propellers and/or thrusters. Dynamic positioning can maintain a position to a fixed point over the bottom, or in relation to a moving object (such as another vessel). It can also be used to position the vessel at a favorable angle towards wind, waves, and current.

*Excitation.* A time-dependent stimulus (force or displacement) that produces vibration. Excitation may be transient, random, and periodic. A steady-state periodic excitation, like that produced by propellers or propulsion engine, is of interest in these Guidance Notes.

*Free Field.* A sound field in a homogeneous, isotropic medium free from boundaries. The sound pressure level decreases by 6 dB per doubling of distance from the source.

*Free Velocity Level.* The velocity level of the machine measured when it is mounted on sufficiently soft isolators on a sufficiently stiff and heavy foundation.

*Frequency.* The number of complete cycles of a periodic process occurring per unit time. Frequency is expressed in Hertz (Hz), which corresponds to the number of cycles observed-per-second.

*Frequency Band.* An interval of the frequency spectrum defined between upper and lower "cut-off" frequencies. The band may be described in terms of these two frequencies, by the width of the band, and by the geometric mean frequency of the upper and lower cut-off frequencies (e.g., "an octave band centered at 500 Hz").

*Joiner Panel.* The inner side outfitting of the compartment, which covers the bulkhead and insulation.

*Level.* In acoustics, the level of a quantity is the logarithm of the ratio of that quantity to a reference quantity of the same kind. The base of the logarithm, the reference quantity, and the kind of level need to be specified. Examples are sound power levels, sound pressure levels, and acceleration levels. Levels are always expressed in decibels (dB).

*Mass Law.* The relationship between sound transmission loss and weight of the barrier. The mass law states that for every doubling of the weight of the material, a 6 dB increase in the transmission loss can be expected.

*Modal Density.* Number of modes per Hertz.

*Octave Band.* The frequency range bounded by upper and lower frequency limits $f_u$ and $f_\ell$, where $f_u = 2f_\ell$. Octave bands are usually specified by their geometric mean frequency, called the band center frequencies. The standard octave bands covering the audible range are designated by the following center frequencies: 31.5, 63, 125, 250, 500, 1000, 2000, 4000, 8000, and 16,000 Hz. The corresponding lower and upper frequencies are 22/45, 45/89, 89/177, 177/354, 354/708, 708/1416, 1416/2832, 2832/5664, 5664/11,328, 11,328/22,656.

*Radiation Efficiency.* The radiation efficiency of a vibrating surface is proportional to the acoustic power radiated per unit surface area per unit of mean-square velocity of vibration averaged over the radiating surface. It is the measure of the efficiency with which a given surface converts vibratory energy to acoustic energy.

*Reverberant Sound Field.* The part of the radiated sound field where the sound waves reflected from the boundaries of the enclosure are superimposed upon the incident field. The reverberant field may be called a diffuse field if a great many reflected wave trains cross from all possible directions and the sound-energy density is very nearly uniform throughout the field.

*Statistical Energy Analysis.* A numerical method for predicting the transmission of sound and vibration through structural acoustic systems. SEA method calculates the average response of the structure and avoids a large quantity of calculation, which is efficient for large and complex structures.

*Sound Power.* The rate at which sound energy is emitted, reflected, transmitted or received, per unit time.

*Sound Pressure Level.* SPL, in dB, is 20 times the logarithm to the base 10 of the ratio of the pressure of this sound to a reference pressure. The sound pressure, $p$, is the root-mean-square value of the instantaneous sound pressure over a time interval at the point under consideration.

*Static Pressure.* The pressure difference of air when flowing across a fan.

*Structure-borne Sound (or Noise).* Sound or noise that is transmitted through structures that are capable of supporting shear. Thus, structure-borne sound may be in the form of longitudinal or flexural waves.

*Transmission Loss.* The sound transmission loss (TL) of a partition, in dB, is the difference between the incident sound intensity level and transmitted sound intensity level. It can be expressed as 10 times the logarithm to the base 10 of the reciprocal of the transmission coefficient, $\tau$.

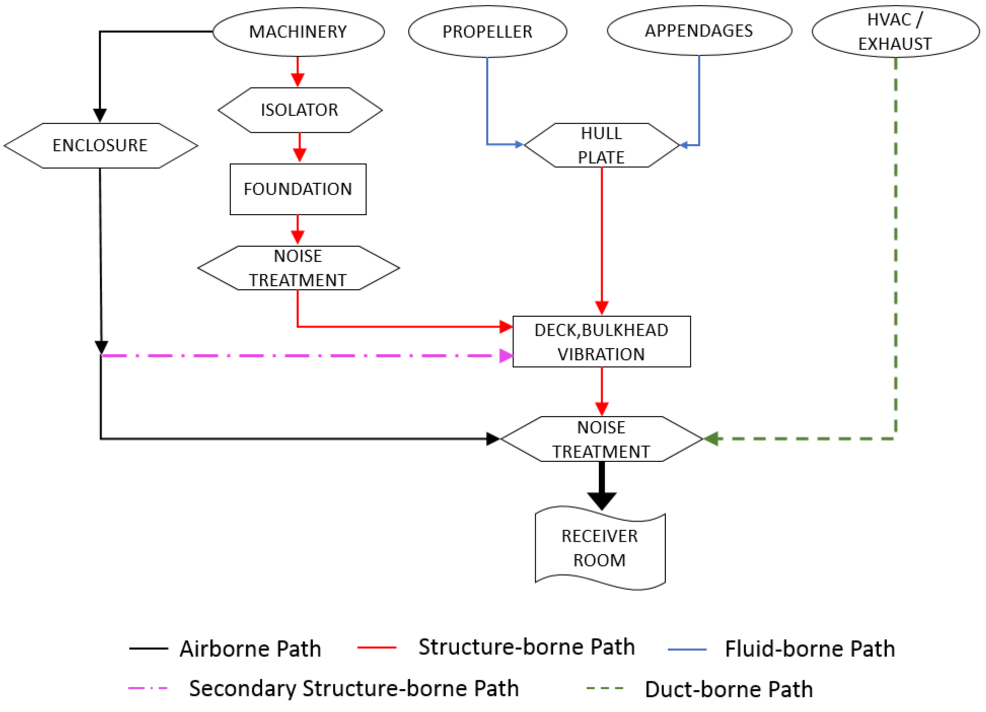
*Transit Condition.* Those conditions where the vessel is transiting (moving) from one location to another.

SECTION 2 Source-Path-Receiver Modeling

## 1 Overview

The modeling of onboard ship noise analysis requires three key elements to be considered: "Source", "Path", and "Receiver", and the procedure is described as Source-Path-Receiver modeling [1]. "Sources" are the equipment which generates airborne noise and structure-borne noise, such as main engines, propellers, compressors, and fans. "Paths" are the air, fluid, or solid structures such as decks and bulkheads through which sound propagates. "Receivers" are the compartments of interest, such as seafarer cabins, workspaces, and offices. The modeling of the receiver component addresses the location and the acoustic characteristics of these spaces. Section 2, Figure 1 is a flowchart of noise transmission from sources to receivers.

**FIGURE 1**
**Noise Transmission Flow Chart**

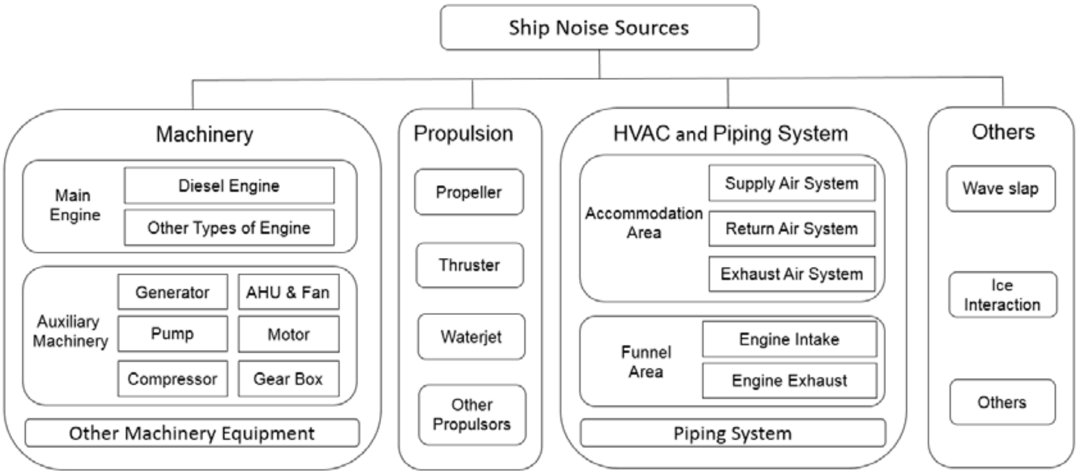


# 3 Source

## 3.1 Overview

Onboard noise sources include machinery, propulsion, heating, ventilation, and air-conditioning (HVAC) systems, piping systems, and environmental noise. Section 2, Figure 2 is an overview of noise sources on board ships. Each source can be dominant, depending on the source levels and the distance from the source(s) to the receiver room(s). Onboard noise prediction should take all dominant sources into consideration. Normally, onboard ship noise analyses considers propeller, main engine, generator, bow thruster, and the HVAC system. The need to consider other sources should be decided on a case-by-case basis. The use of actual measured source levels as the input excitation sources is recommended. When the measured noise or vibration source levels are not available, the source levels may be estimated by empirical formulae.

**FIGURE 2**
**Noise Sources on Board Ships**




## 3.3 Machinery

Typical machinery noise sources include main engines, diesel engines, generators, compressors, fans, and pumps. Among these machinery sources, main engine and diesel engines typically have the most significant effects on onboard ship noise levels. Each machinery noise source generates both airborne noise and structure-borne noise. Both the airborne noise and the structure-borne noise should be considered in noise analyses.

Airborne noise is usually the machinery casing noise. For main engines and diesel engines, besides the casing noise, airborne noise also includes the sounds emitting from air intakes and air exhausts. These noise sources could influence open deck areas and locations close to these openings. Airborne noise mainly influences the source room and the compartments adjacent to the source room.

Structure-borne noise is usually caused by machinery foundation vibration. Structure-borne noise could contribute to the noise levels in accommodation areas far away from the location of the machinery itself.

### 3.3.1 Machinery Casing Noise

Airborne noise levels can be expressed in terms of sound power level $L_W$ or sound pressure level $L_p$, which are defined as:

$$L_W = 10\log\frac{W}{W_0} \quad \text{dB}$$

$$L_p = 20\log\frac{p}{p_0} \quad \text{dB}$$

where

| | | |
|---|---|---|
| $W$ | = | sound power generated by the excitation source, in watts |
| $W_0$ | = | standard reference power, $10^{-12}$ watts |
| $p$ | = | sound pressure measured at a certain location, in Pa (psi) |
| $p_0$ | = | standard reference pressure, $2 \times 10^{-5}$ Pa ($2.9 \times 10^{-9}$ psi) [2] |

There are several ISO standards for the measurement of machinery casing noise, such as ISO 9614-1 [3], ISO 9614-2 [4], and ISO 9614-3 [5]. If actual noise measurement data is not available, the sound power level may be calculated by empirical formulae such as the estimation method described in the *Design Guide for Shipboard Airborne Noise Control* [1].

Sometimes, noise analysis requires the input to be the sound power level rather than the sound pressure level. When the sound pressure level of the excitation source is known, the sound power level of the excitation source can be calculated using the formula below [6].

$$L_W = L_p(r_0) + 10\log[12\,r_0^2 + 4r_0(L + B + 2H) + 2H(L + B) + LB] \quad \text{dB} \qquad \text{(SI units)}$$

$$= L_p(r_0) + 10\log[12\,r_0^2 + 4r_0(L + B + 2H) + 2H(L + B) + LB] - 10 \quad \text{dB} \qquad \text{(US units)}$$

where

| | | |
|---|---|---|
| $r_0$ | = | average distance between the equipment surface and the point where the noise levels are measured, in m (ft) |
| $L_p(r_0)$ | = | sound pressure level measured at a distance $r_0$, in dB |
| $L$ | = | length of the equipment, in m (ft) |
| $B$ | = | width of the equipment ($B < L$), in m (ft) |
| $H$ | = | height of the equipment, in m (ft) |

### 3.3.2 Machinery-induced Structure-borne Noise

In ship structures, machinery equipment may be installed on foundations with proper resilient mount isolators. Section 2, Figure 3 shows an example of a piece of equipment installed with resilient mount isolators on a foundation. Both the foundation and the isolator affect the structure-borne sound transmission from the equipment to the ship's structure, and thus needs to be considered in the noise analysis.

Vibration velocity level $L_v$ and vibration acceleration level $L_a$ are two commonly-used inputs to calculate the machinery-induced structure-borne noise transmitted into ship structures. They are defined as:

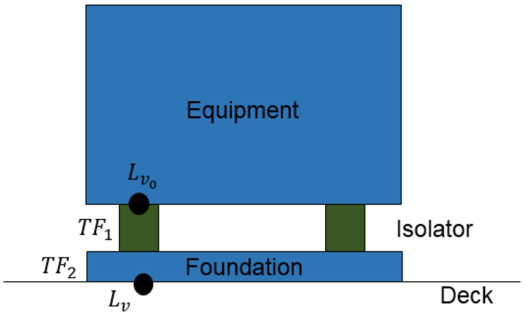
$$L_v = 20\log\frac{v}{v_0} \quad \text{dB}$$

$$L_a = 20\log\frac{a}{a_0} \quad \text{dB}$$

where

| | | |
|---|---|---|
| $v$ | = | vibration velocity generated by the excitation source, in m/s (ft/s) |
| $v_0$ | = | standard reference velocity, $10^{-8}$ m/s ($3.28 \times 10^{-8}$ ft/s) |
| $a$ | = | vibration acceleration generated by the excitation source, in m/s$^2$ (ft/s$^2$) |
| $a_0$ | = | standard reference acceleration, in $10^{-5}$ m/s$^2$ ($3.28 \times 10^{-8}$ ft/s$^2$) |

**FIGURE 3**
**Resilient Mount and Foundation Arrangement**




The structure-borne sound transmitted into the ship structures depends on the vibration velocity level asserted on the ship structures and can be calculated using the formula below:

$$L_v = L_{v0} + TF_1 + TF_2 \quad \text{dB}$$

where

$L_v$ = vibration level on ship structures, in dB

$L_{v0}$ = free velocity level of the equipment, in dB

$TF_1$ = transfer function for the resilient mount isolator, in dB

$TF_2$ = transfer function for the interaction between foundation and deck, in dB

The free velocity level of the equipment can be measured according to ISO 9611 [7]. It can also be estimated by the SNAME empirical formula [1].

The transfer functions for the resilient mount and the interaction between the foundation and the ship's structure can be obtained by experimental testing, numerical analysis, or empirical estimation. The *Design Guide for Shipboard Airborne Noise Control* [1] provides the empirical estimation method for these two transfer functions.

Sometimes, the noise analysis requires the input to be the vibration acceleration level rather than the vibration velocity level. When the vibration velocity level of the excitation source is known, the acceleration level of the excitation source can be calculated using the formula below:

$$L_a = L_v + 20\log(f) - 44 \quad \text{dB}$$

where

$L_a$ = acceleration level of the excitation source, in dB

$L_v$ = velocity level of the excitation source, in dB

$f$ = octave band center frequency, in Hz [1].

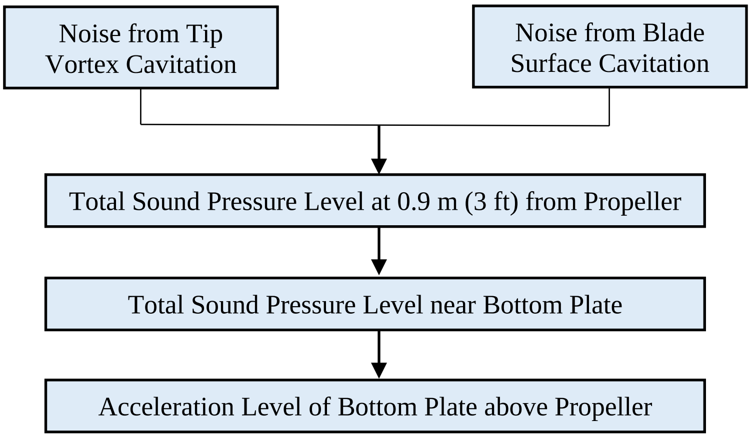
The reference for acceleration levels is $10^{-5}$ m/s$^2$ ($3.28 \times 10^{-8}$ ft/s$^2$). The reference for velocity levels is $10^{-8}$ m/s ($3.28 \times 10^{-8}$ ft/s).

## 3.5 Propulsion

### 3.5.1 Propeller

Propeller-induced hull vibration can be considered a structure-borne source as indicated in the ABS *Guidance Notes on Noise and Vibration Control for Inhabited Spaces (Noise and Vibration Guidance Notes*. The propeller-induced hull vibration level can be obtained by field measurement as well as by estimation by empirical formula. The *Supplement to the Design Guide for Shipboard Airborne Noise Control* [6] presents an empirical formula to calculate the propeller-induced hull vibration level.

## FIGURE 4
## Calculation Procedure for Propeller Induced Vibration




The propeller's radiated pressure level is a sum of two components: noise from tip vortex cavitation and noise from the blade's surface cavitation.

The tip vortex cavitation noise, $L_{pt}$, located 0.9 m (3 ft) from the propeller tip can be calculated by:

$$L_{pt} = 10\log\left[0.23 n_b \left(\frac{N_p}{60}\right)^3 D^4 \frac{1}{f_{br}}\right] + 155 + C_t \ \ \text{dB} \qquad (D \text{ in m})$$

$$= 10\log\left[0.23 n_b \left(\frac{N_p}{60}\right)^3 \left(\frac{D}{3.28}\right)^4 \frac{1}{f_{br}}\right] + 155 + C_t \ \ \text{dB} \qquad (D \text{ in ft})$$

$$C_t = \begin{cases} 10\log\left[0.67\left(\dfrac{v}{v_0} - 1\right)\right], & v/v_0 < 2.35 \\ 0, & v/v_0 > 2.35 \end{cases} \quad \text{dB}$$

The blade's surface cavitation noise, $L_{pb}$, at 0.9 m can be calculated by:

$$L_{pb} = 10\log\left[0.23 n_b \left(\frac{N_p}{60}\right)^3 D^4 \frac{1}{f_{br}}\right] + 168 + 20\log\left(1 - \frac{v_0}{v}\right) \ \ \text{dB} \qquad (D \text{ in m})$$

$$= 10\log\left[0.23 n_b \left(\frac{N_p}{60}\right)^3 \left(\frac{D}{3.28}\right)^4 \frac{1}{f_{br}}\right] + 168 + 20\log\left(1 - \frac{v_0}{v}\right) \ \ \text{dB} \qquad (D \text{ in ft})$$

where

| | | |
|---|---|---|
| $n_b$ | = | number of blades |
| $D$ | = | propeller diameter, in m (ft) |
| $N_p$ | = | propeller revolution per second |
| $f_{br}$ | = | break frequency, in Hz, which can be decided by the figures indicated in the *Supplement to the Design Guide for Shipboard Airborne Noise Control* [6] |
| $v$ | = | ship speed, in m/s (knots) |
| $v_0$ | = | cavitation inception speed, in m/s (knots) |

The total sound pressure level at the distance of 0.9 m (3 ft) from the propeller tip, $L_{p'}$, is calculated by:

$$L_{p'} = 10\log\left(10^{L_{pt}/10} + 10^{L_{pb}/10}\right) \quad \text{dB}$$

The total sound pressure level near the bottom plate, $L_p$, is calculated by:

$$L_p = L_{p'} - 10\log\left(\frac{0.1D + d_t}{0.1D + 0.9}\right) \quad \text{dB} \qquad (d_t \text{ in m})$$

$$= L_{p'} - 10\log\left(\frac{0.1D + d_t}{0.1D + 3}\right) \quad \text{dB} \qquad (d_t \text{ in ft})$$

where

$D$ = propeller diameter, in m (ft)

$d_t$ = distance from propeller tip to hull plate, in m (ft)

The octave band acceleration level of the plate structure above propeller in dB re $1 \times 10^{-5}$ Pa ($1.45 \times 10^{-9}$ psi) is equal to:

$$L_a = L_p + \Delta L_{pa}$$

where

$\Delta L_{pa}$ = sound pressure-to-acceleration transfer function, which can be found using the figures indicated in the *Supplement to the Design Guide for Shipboard Airborne Noise Control* [6]

#### 3.5.2 Bow Thruster

For those ships possessing dynamic positioning systems, bow thrusters could be part of the dominant excitation sources. The bow thruster can induce significant vibration. The thruster's tunnel wall is usually considered a structure-borne noise source. Although the airborne noise level in the bow thruster compartment is typically on the order of 100 dB(A) or greater during DP conditions [8], manned spaces are rarely adjacent to the thruster compartment. Thus, the airborne noise caused by the bow thruster is usually not a concern. However, if there are manned spaces adjacent to the bow thruster compartment, the airborne noise induced by the bow thruster will need to be considered.

The vibration level of the tunnel wall is difficult to measure. An alternative method is to measure the vibration of the bulkhead and side shell connected to the tunnel walls. This source input can be modeled as the vibration of the bulkhead and side shell instead of the tunnel wall's vibration. The tunnel wall's vibration may also be estimated by empirical method.

### 3.7 Ventilation Fans

In addition to casing and structure-borne noise, ventilation fans generate aerodynamic noise as indicated in the ABS *Noise and Vibration Guidance Notes*. Aerodynamic noise is a type of airborne noise generated by airflow from the HVAC system. The aerodynamic noise level is directly related to the fan's characteristics, such as fan type, volume flow rate, and static pressure [1]. The source level can be estimated by using the *Design Guide for Shipboard Airborne Noise Control* [1].

## 5 Sound Transmission Path

The purpose of path modeling is to calculate the sound attenuation from sources to receiver rooms. Therefore, hull structures, joiner bulkheads, joiner panels, deck coverings, HVAC duct design/arrangement, and sound insulation should be considered in the path modeling.

Sound can be transmitted from a source to a receiver in four ways as indicated in the *Noise and Vibration Guidance Notes*:

*i)* Airborne path where sound propagates through air

*ii)* Structure-borne path where sound propagates through the ship's structure [1]

*iii)* Duct-borne path where sound is transmitted through the HVAC duct system

*iv)* Fluid-borne path where the hydro-excitation sources such as propeller, thruster, and wave-slap transmit the hydro-acoustic pressure to the hull by water and result in hull pressure force

## 5.1 Airborne Path

When airborne sound propagates in a free field, it gradually attenuates over the distance it propagates. However, when it meets a solid object, such as a steel plate, the attenuation increases significantly. Section 2, Figure 5 shows the airborne sound transmission through two compartments.

Steel reduces sound greatly. A typical steel plate can attenuate the airborne sound transmission by as much as 40 dB. The joiner panel can work as an additional barrier to reduce the radiation from the structural bulkhead into the receiver spaces and decrease the transmission of noise. Sound absorptive material such as fiberglass applied to the partition can also decrease the transmission of noise. Due to the high attenuation of the decks and bulkheads in ship structures, the airborne path is usually a critical factor only within a source space itself and the compartments directly adjacent [1].

When the airborne sound transmits from one compartment (source room) to its adjacent compartment (receiver room), the sound pressure level of the receiver can be calculated by the formula below:

$$L_{p,R} = L_{p,S'} - TL + 10\log\left(\frac{S}{A_E}\right) \quad \text{dB}$$

where

$L_{p,R}$ = sound pressure level in the receiver room, in dB

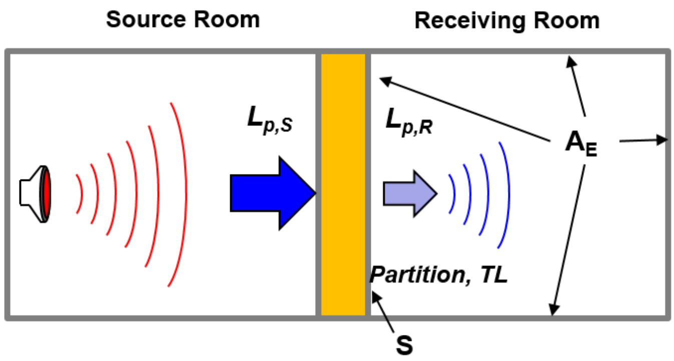
$L_{p,S}$ = sound pressure level in the source room, in dB

$TL$ = transmission loss, in dB (ft$^2$)

$S$ = area of bulkhead or deck between receiver room and source room, in m$^2$ (ft$^2$)

$A_E$ = sound absorption area of receiver room, in m$^2$ (ft$^2$) [9]

### FIGURE 5
### Sound Transmission through Compartments



According to the mass law, attenuation is highly related to the surface mass of the partition. The transmission loss increases by 6 dB for each doubling of the surface mass [9]. Therefore, when calculating transmission loss through a partition, the key parameters to be considered are thickness and density. Section 2, Table 1 shows the airborne sound transmission loss through a partition in some typical types of materials on board ships [1].

**TABLE 1**
**Airborne Sound Transmission Loss of Typical Ship Panels**

| *Material* | *Thickness mm* | *Octave Band Center Frequency, Hz* | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | *31.5* | *63* | *125* | *250* | *500* | *1k* | *2k* | *4k* | *8k* |
| Steel | 6 | 16 | 22 | 26 | 31 | 36 | 40 | 37 | 42 | 51 |
| Aluminum | 6 | 7 | 13 | 19 | 25 | 28 | 34 | 30 | 32 | 42 |
| Thermal insulation board | 25 | 0 | 0 | 0 | 0 | 2 | 9 | 18 | 27 | 35 |
| Fiberglass board | 25 | 0 | 0 | 0 | 0 | 0 | 2 | 4 | 7 | 11 |

In ship structures, certain decks and bulkheads have large openings, such as the large opening on the tween deck in the engine room. These openings located in the primary noise transmission path should be considered because a significant amount of sound energy can transmit through them. However, openings that are not located in a primary noise transmission path, such as the openings of the deck in the water tank, can usually be ignored in the analysis.

If the transmission loss of a deck or bulkhead is high, the attenuation can be no greater than $10\log(A_t/A)$ if there is an opening on the deck, where $A$ is the area of all openings of the panel and $A_t$ is the total area of the panel. For example; a bulkhead with a nominal transmission loss of 40 dB has openings representing 1% of the total bulkhead area, the effective transmission loss will be no more than 20 dB [1].

## 5.3 Structure-borne Path

Structure-borne paths often carry acoustic energy to everywhere on the vessel, including remote spaces and spaces adjacent to the area where the source is located [1]. Structure-borne sound will usually attenuate gradually in a continuous structure. However, obstacles or discontinuities, such as deck/bulkhead intersection and frames, can attenuate significantly. As indicated in the *Noise and Vibration Guidance Notes*, the greater the number of obstacles along the structure-borne path, or the longer the distance between the source and the receiver, the lower the structure-borne sound that propagates to receivers.

### 5.3.1 Structure-borne Sound Transmission in Ship Structures

The elements of the structure-borne path in ship structures can be divided into four groups, as listed below. The total structure-borne sound transmission loss from the source to the receiver is the arithmetic sum of all the losses throughout the transmission path.

*i)* Ship structures within the source room

*ii)* Ship structures beyond the source room

*iii)* Intersections of ship structures

*iv)* Pillars

Within the source room, the structure-borne transmission loss from an effective source area to the source room boundary mainly depends on its shape, orientation, and distance to the compartment boundary [6]. "Effective source area" in these Guidance Notes is defined as the region of decks or side shells within which the excitation force almost has no dissipation. For machinery equipment, the effective source area is the deck region immediately below the machinery's foundation.

For ship structures beyond the source room, the structure-borne transmission loss depends on the damping loss factor of the structure, the size of the structure, and the distance from the source to the structure of interest [6]. The larger the damping loss factor and the larger the distance from the source to the structure of interest, the higher the attenuation of the structure-borne sound. Section 2, Table 2 shows the damping loss factor for typical hull structures [6].

For the intersection of ship structures, such as the junction of two bulkheads, bulkhead and deck, or a deck and the hull, the structure-borne sound transmission loss can be significant. The transmission loss of the intersection depends primarily on three factors: the shape of the intersection, the plate thickness, and the material. Most intersections of ship structures are right-angles, cross junctions or T junctions. The higher number of intersections between the source and the receiver along the transmission path, the more the structure-borne sound attenuates [6].

Pillars usually act as a rigid coupling between decks at most frequencies of interest. The transmission loss between decks connected by a pillar is almost zero. Therefore, pillars, especially those located in the vicinity of vibration sources, cannot be ignored in the noise analysis as the acoustic energy can transmit through the pillar with almost no dissipation [1].

## TABLE 2
## Damping Loss Factor for Typical Hull Structures

| | *Octave Band Center Frequency, Hz* | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | *31.5* | *63* | *125* | *250* | *500* | *1k* | *2k* | *4k* | *8k* |
| Steel, Dry | 0.008 | 0.007 | 0.006 | 0.005 | 0.005 | 0.004 | 0.004 | 0.003 | 0.003 |
| Steel, Fluid | 0.040 | 0.035 | 0.03 | 0.025 | 0.016 | 0.013 | 0.009 | 0.007 | 0.006 |
| Aluminum, Dry | 0.013 | 0.012 | 0.011 | 0.010 | 0.009 | 0.008 | 0.007 | 0.007 | 0.006 |
| Aluminum, Fluid | 0.060 | 0.055 | 0.050 | 0.035 | 0.027 | 0.021 | 0.014 | 0.012 | 0.009 |

### 5.3.2 Damping Layer

Damping is a frequently-used, high frequency, passive control method to reduce structure-borne sound and can be applied to decks and bulkheads. It dissipates the vibration energy by converting it into heat. In order to be effective, the damping layer should have at least the thickness of the vibrating panel [9]. Damping layers are more effective in reducing free vibration caused by resonance than in reducing the response caused by forced vibration [1].

There are two main types of structural damping materials. One is called free-layer damping and the other is constrained-layer damping. For free-layer damping, the damping materials are applied uncovered to the structural surface. For constrained-layer damping, the damping material is sandwiched between two layers of structural material. The two forms of damping treatment deform in different ways. The free-layer damping deforms due to simple extension and shear strains in the material. The constrained damping layer deforms due to relative longitudinal or transverse compressional motion between the elastic face layers. This strains the viscoelastic damping core and converts a portion of the mechanical energy into heat. Therefore, the constrained damping layer acts as a better sound barrier. Section 2, Table 3 shows the damping loss factor of the ship structures with the two types of damping [1].

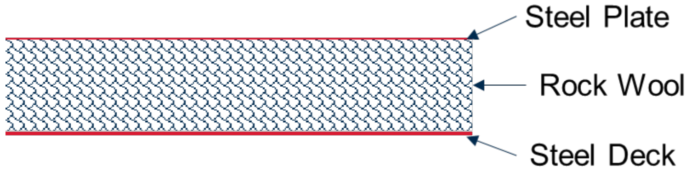
## TABLE 3
## Damping Loss Factor for Two Types of Damping

| *Ship Structures* | *Octave Band Center Frequency, Hz* | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | *31.5* | *63* | *125* | *250* | *500* | *1k* | *2k* | *4k* | *8k* |
| With free-layer damping | 0.3 | 0.3 | 0.3 | 0.3 | 0.3 | 0.3 | 0.3 | 0.3 | 0.3 |
| With constrained-layer damping | 1.4 | 1.4 | 1.4 | 1.4 | 1.4 | 1.4 | 1.4 | 1.4 | 1.4 |

#### 5.3.3 Floating Floor

A floating floor is a floor that is not affixed to the deck structure. The term "floating" refers to the way that it is installed. Floating floors are a frequently-used method to reduce structure-borne sound. The noise reduction capability of the floating floor varies from product to product. Section 2, Figure 6 shows the typical configuration of a floating floor. The key factor impacting the performance of the floating floor is the dynamic stiffness of the core material of the floating floor. The lower the dynamic stiffness, the higher the structure-borne sound transmission loss and the lower the natural frequency of the floating floor. However, when choosing the product, it is necessary to make sure that the product is dimensionally stable and has enough compression strength. Otherwise, it would be too soft and uncomfortable to walk on. Besides, the top layer of the floating floor is not to have rigid connections with steel structure to avoid flanking transmission.

**FIGURE 6**
**Typical Configuration of Floating Floor**




### 5.5 Duct-borne Path

The duct-borne path is where the HVAC system transmits sound from air-conditioning equipment such as the air handling unit (AHU) to the duct outlets. When the sound transmits via ducts, it attenuates due to the following six factors [1].

*i)* *Plenum (if any),* $T_{plena}$. A plenum is a pressurized housing containing air at positive pressure (pressure higher than surroundings) to equalize pressure for more even distribution. When the plenum is installed with sound absorption linings, it can significantly attenuate the sound energy. Therefore, a plenum can sometimes work as an acoustic silencer device.

*ii)* *Silencers (if any),* $T_{silencer}$. A silencers is a device commonly used in HVAC systems to attenuate duct-borne noise. It consists of a large amount of sound absorptive materials which dissipate acoustic energy. The sound attenuation effect of the silencer depends primarily on its type and length. There are two types of silencers: high pressure drop silencer and low pressure drop silencer. The sound attenuation effect of the high pressure drop silencer is greater than the low pressure drop silencer. The sound attenuation effect of the silencer also increases with its length.

*iii)* *Straight Duct Attenuation,* $T_{straight}$. Sound attenuation will occur even when transmitting in straight ducts. The sound attenuation effect depends on the sound absorptive effect of the inner surface of the duct and the distance it flows by.

*iv)* *Branches where Flow Divides,* $T_{branch}$. When the sound transmits through a branch, it will divides into several portions. This will result in sound attenuation for each branch.

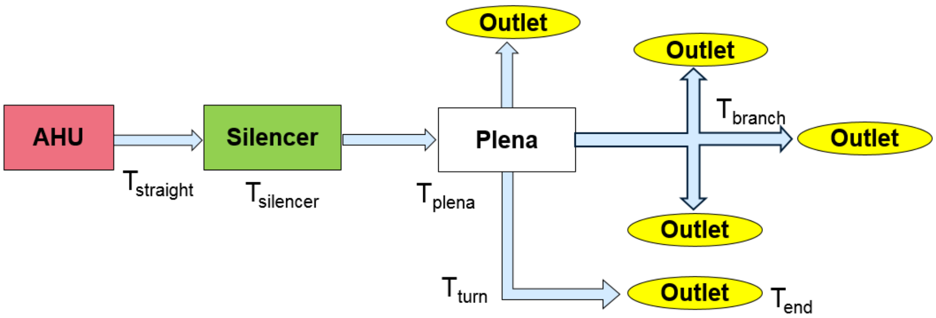
*v)* *Turns where Flow Changes Directions by More than 30 Degrees,* $T_{turn}$. When sound propagates through a turn, it will attenuate by approximately 1 to over 10 dB depending on the diameter of the cross section of the turn duct and the lining configuration. Both the increase in the cross-sectional area of the duct and the installation of sound absorptive linings will increase the attenuation.

*vi)* *End Reflections at Duct Openings,* $T_{end}$. When sound transmits into a room, part of the sound energy will reflect back rather than transmit into the receiver rooms. This reflection at the duct openings will cause sound attenuation, especially for the frequency range below 250 Hz.

Section 2, Figure 7 shows the sound transmission via HVAC ducts. The total sound attenuation from source to outlet via ducts is the sum of the attenuation caused by the six factors [1]:

$$T_{duct} = T_{plena} + T_{silencer} + T_{straight} + T_{branch} + T_{turn} + T_{end}$$

**FIGURE 7**
**Sound Transmission via HVAC Ducts**




## 5.7 Fluid Load

The fluid load of ships can be divided into two major groups. One is the ocean water outside the hull, and the other is ballast water.

Normally, the effect of ballast water in the water tank on onboard ship noise is slight and can be ignored. Although the water will increase the damping loss factor of the ship's structures by several times [6], the damping is still too small to affect the compartment structure-borne noise.

Ocean water is recommended to be considered. Although its effect on the high frequency range is slight, it may be significant in the low frequency range.

## 5.9 Path Simplification

As the accommodation blocks of ships are mostly located only in the fore (Type A) or aft part (Type B), it is expected that the analysis efficiency can be improved by modeling only part of the structures. Section 2, Figures 8 and 9 show the typical layout of Type A vessels and Type B vessels, respectively.

For Type A vessels, only engine rooms and accommodation areas (mid to fore of the vessels) are to be modeled, assuming that there is no severe propeller cavitation or poor shaft alignment. If either of the two situations occurs on a sister vessel, these conditions shall be considered in the analysis. Compared to other sources such as the main engine, bow thrusters and the HVAC system, the contribution from the aft propellers to the fore living areas can normally be ignored. Although the propeller-induced structure-borne source is on the same scale or even higher than that of the main engine, the distance from the aft propeller to the accommodation areas is usually great enough to attenuate the sound propagation .

For Type B vessels, only the aft part of the structure are to be modeled as all the compartments of interest and excitation sources are located at the aft part of the vessel. Based on industry experience, the effect on the noise levels caused by ignoring the rest of the vessel is negligible.

**FIGURE 8**
**Type A Vessel – Accommodation Area in Fore Part**

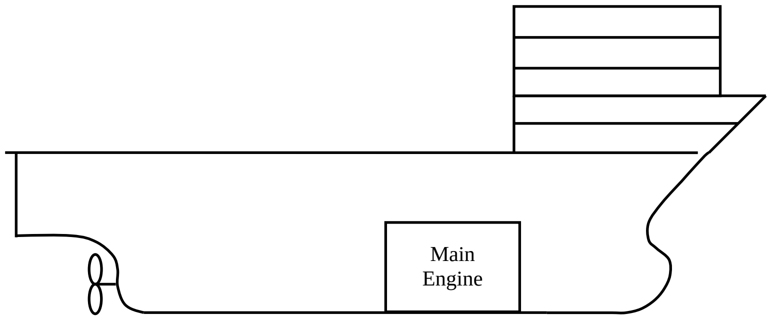
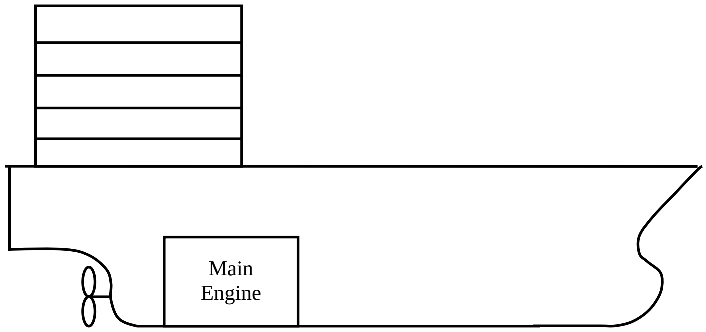



**FIGURE 9**
**Type B Vessel – Accommodation Area in Aft Part**




## 7 Receiver

### 7.1 Radiation Efficiency

Radiation efficiency, $10\log\sigma_{rad}$, is used to calculate the airborne sound radiated by the structure's vibration. It is an indicator of the capability of the structure's surface to convert the structure-borne sound energy to airborne sound. $\sigma_{rad}$ is the radiation value of the structure. The higher the radiation efficiency, the more vibration energy can be converted into airborne sound. Section 2, Table 4 shows the radiation efficiency of typical ship structures [6].

**TABLE 4**
**Radiation Efficiency of Typical Ship Structures,** $10\log\sigma_{rad}$

| *Material* | *Octave Band Center Frequency, Hz* | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | *31.5* | *63* | *125* | *250* | *500* | *1k* | *2k* | *4k* | *8k* |
| Deck with Poured Floor | –11 | –10 | –10 | –8 | 0 | 3 | 0 | 0 | 0 |
| Steel Panel Joiner Bulkhead with Mineral Wool Core | –15 | –15 | –15 | –12 | –10 | –8 | –2 | 0 | 0 |

### 7.3 Room Constant

The reverberant sound pressure level of the receiver room depends on the total sound power level transmitted into the compartment and the room constant of the compartment.

$$L_p = L_{W'} - 10\log R + 6 \quad \text{dB} \qquad (R \text{ in m}^2)$$

$$= L_{W'} - 10\log R + 16 \quad \text{dB} \qquad (R \text{ in ft}^2)$$

where

$L_p$ = sound pressure level of the receiver room, in dB

$L_W$ = sound power level transmitted into the receiver room, in dB

$R$ = room constant of the receiver room, in m$^2$ (ft$^2$) [1].

Room constant, which indicates the acoustic absorption capability of the compartment, is a key acoustic characteristic parameter of a receiver room. It is determined by the average absorption coefficient of the compartment. Besides compartment boundary surface absorption, acoustic absorption also occurs at the exposed surfaces of furniture, curtains, pillows, etc. Such non-boundary surfaces have a significant portion of the total absorption coefficient and also need to be considered [2]. Section 2, Table 5 shows the acoustic absorption coefficients of typical materials on ships [1].

$$R = \frac{S\overline{\alpha}}{1-\overline{\alpha}} \quad \text{m}^2 \text{ (ft}^2\text{)}$$

where

$R$ = room constant, in m$^2$ (ft$^2$)

$S$ = total area of the compartment, in m$^2$ (ft$^2$)

$\overline{\alpha}$ = average absorption coefficient of the compartment

$$= \frac{\sum S_i \alpha_i}{\sum S_i}$$

$S_i$ = individual surface area in the compartment, in m$^2$ (ft$^2$)

$\alpha_i$ = absorption coefficient for individual surface in the receiver room

**TABLE 5**
**Acoustic Absorption Coefficients of Typical Materials**

| *Material* | *Octave Band Center Frequency, Hz* | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | *31.5* | *63* | *125* | *250* | *500* | *1000* | *2000* | *4000* | *8000* |
| Steel or Aluminum Plate | 0.01 | 0.01 | 0.02 | 0.03 | 0.03 | 0.03 | 0.02 | 0.02 | 0.02 |
| Glass | 0.30 | 0.20 | 0.16 | 0.04 | 0.03 | 0.02 | 0.02 | 0.02 | 0.02 |
| Carpeted Deck | 0.02 | 0.04 | 0.08 | 0.10 | 0.15 | 0.20 | 0.25 | 0.20 | 0.15 |

# SECTION 3 Results Evaluation

## 1 Frequency Analysis and Octave Bands

The frequency range of onboard ship noise analyses is usually from 31.5 Hz to 8000 Hz. Considering the large span of the frequency range, it is usually subdivided into frequency bands. The most common one is the octave frequency band. The center frequencies of the octave bands for onboard ship noise analyses are typically: 31.5 Hz, 63 Hz, 125 Hz, 250 Hz, 500 Hz, 1,000 Hz, 2,000 Hz, 4000 Hz and 8,000 Hz. The following equations show the relationship between center frequency and upper/lower bound frequency of an octave band:

$$\frac{f_u}{f_\ell} = 2$$

$$f_c = \sqrt{f_u f_\ell}$$

where

$f_u$ = upper bound frequency, in Hz

$f_\ell$ = lower bound frequency, in Hz

$f_c$ = center frequency, in Hz.

## 3 A-weighting Evaluation

The human ear is not equally sensitive to all frequencies of sound. To the human ear, a low frequency sound is quieter than a high frequency sound of the same level. Therefore, in onboard ship noise analyses, A-weighting is used to take this into consideration.

A-weighting is the most commonly used weighting curve, defined in IEC 61672 [10]. It approximates the sensitivity of the human ear by filtering these frequencies. Section 3, Figure 1 shows the A-weighting filtering curve. Section 3, Table 1 shows the octave band A-weighting filtering values.

The overall A-weighted sound pressure level from 31.5 to 8000 Hz can be calculated by the formula below:

$$L_p(A) = 10\log\left[\sum_{i=1}^{9} 10^{0.1(L_{p,i}+A_i)}\right] \quad \text{dB}$$

where

$L_p(A)$ = A-weighted overall sound pressure level, in dB

$L_{p,i}$ = sound pressure level under each octave band center frequency, in dB

$A_i$ = A-weighting correction under each octave band center frequency, dB

**FIGURE 1**
**A-Weighting Filtering Curve**

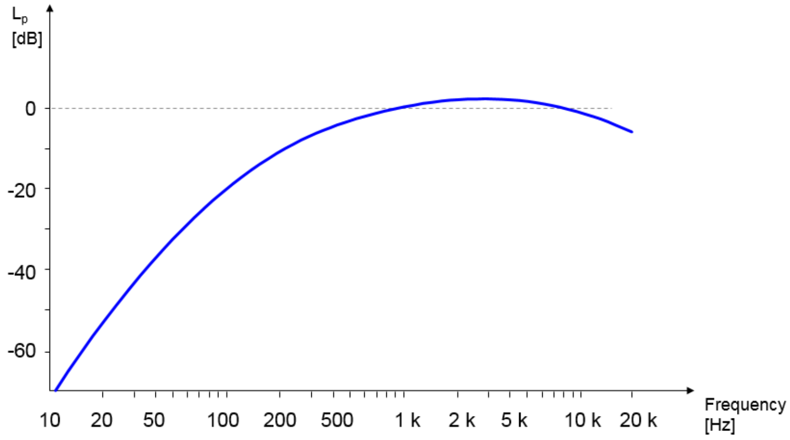



**TABLE 1**
**A-filter Values**

| *Frequency (Hz)* | *31.5* | *63* | *125* | *250* | *500* | *1k* | *2k* | *4k* | *8k* |
|---|---|---|---|---|---|---|---|---|---|
| A-Weighting Correction (dB) | –39.4 | –26.2 | –16.1 | –8.6 | –3.2 | 0 | 1.2 | 1 | –1.1 |

# 5 Criteria

## 5.1 ABS Criteria

ABS provides optional classification notations for habitability onboard ships. For details on the acceptance criteria and measurement procedures, please refer to:

- ABS *Guide for Crew Habitability on Workboats*
- ABS *Guide for Crew Habitability on Ships*
- ABS *Guide for Habitability of Industrial Personnel on Accommodation Vessels*
- ABS *Guide for Passenger Comfort on Ships*
- ABS *Guide for Comfort on Yachts*
- ABS *Guide for Compliance with the ILO Marine Labour Convention, 2006 Title 3 Requirements*

## 5.3 ILO MLC 2006 and IMO Noise Code

The ILO MLC 2006 [11] was ratified and came into force in August of 2013 by port States having adopted the Convention. MLC requires vessels to comply with the requirements of applicable international instruments on the acceptable levels of exposure to workplace hazards on board ships and on the development and implementation of ships' occupational safety and health policies and programs. The more common international instrument related to the "ambient factors" of noise is the IMO *Code on Noise Levels On-board Ships* (Resolution MSC 337(91)) [12]. This code came into force in July 2014 to provide standards to prevent the occurrence of potentially hazardous noise levels on board ships and to provide standards for an acceptable environment for seafarers. The Code is applicable to new ships of a gross tonnage of 1,600 and above. It may be applied to existing ships of a gross tonnage of 1,600 and above, or new ships of a gross tonnage of less than 1,600, as far as reasonable and practical, to the satisfaction of the Administration. Section 3, Table 2 is the criteria for noise levels onboard vessels. A-weighted sound pressure levels of the vessel need to be used for the comparison with the criteria.

# TABLE 2
## Noise Level Limits, dB(A)

| *Designation of Rooms and Spaces* | *Ship Size* | |
|---|---|---|
| | *1,600 up to 10,000 GT* | *≥ 10,000 GT* |
| *Work Spaces* | | |
| Machinery spaces | 110 | 110 |
| Machinery control rooms | 75 | 75 |
| Workshops other than those forming part of machinery spaces | 85 | 85 |
| Non–specified work spaces (other work areas) | 85 | 85 |
| *Navigation Spaces* | | |
| Navigating bridge and chartrooms | 65 | 65 |
| Look-out posts, incl. navigating bridge wings and windows | 70 | 70 |
| Radio rooms (with radio equipment operating but not producing audio signals) | 60 | 60 |
| Radar rooms | 65 | 65 |
| *Accommodation Spaces* | | |
| Cabin and hospitals | 60 | 55 |
| Messrooms | 65 | 60 |
| Recreation rooms | 65 | 60 |
| Open recreation areas (external recreation areas) | 75 | 75 |
| Offices | 65 | 60 |
| *Service Spaces* | | |
| Galleys, without food processing equipment operating | 75 | 75 |
| Serveries and pantries | 75 | 75 |
| Normally unoccupied spaces | 90 | 90 |

# APPENDIX 1 Empirical Method

## 1 Airborne Sound Transmission

### 1.1 Airborne Sound Transmission in Source Room

For the source room, the sound pressure level of the total airborne noise in the space at a given place is a sum of the direct field sound pressure level and the reverberant sound pressure level. The reverberant sound pressure level can be calculated by the formula in 2/7.3. The direct sound pressure level in the source room can be calculated by the empirical formula below:

$$L_p = L_W - 20\log r + 10\log[Q] - 11 \quad \text{dB} \qquad (r \text{ in m})$$

$$= L_W - 20\log r + 10\log[Q] - 1 \quad \text{dB} \qquad (r \text{ in ft})$$

where

$L_p$ = sound pressure level at receiver point, in dB

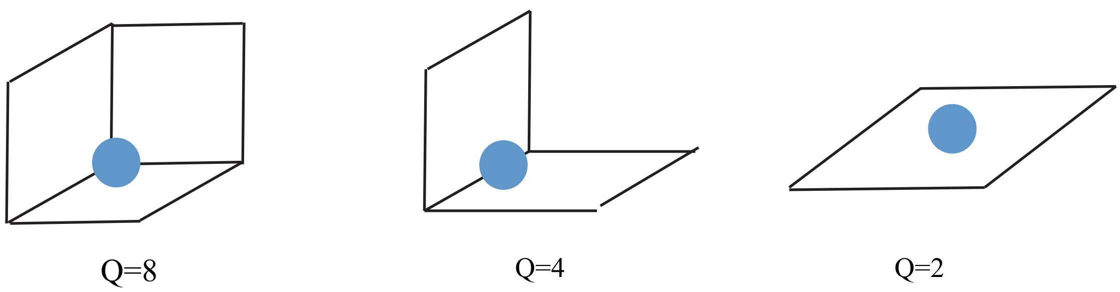
$L_W$ = sound power level of the source, in dB

$r$ = distance between the acoustic center of the noise source and the receiver, in m (ft)

$Q$ = directivity factor of the source [1]

As shown in Appendix 1, Figure 1, for a source located in the center of the source room, $Q$ is 2. For a source located against to a bulkhead, $Q$ is 4. For a source located in the corner of the source room, $Q$ is 8.

**FIGURE 1**
**Directivity Factors $Q$**




### 1.3 Airborne Sound Transmission Loss through a Partition

According to the mass law, the transmission loss of a partition can be calculated by the formula below:

$$TL = 10\log\left[1 + \left(\frac{\omega m_s \cos\theta}{2\rho_0 c_0}\right)^2\right] \quad \text{dB}$$

where

$\omega$ = radial frequency, in rad/s

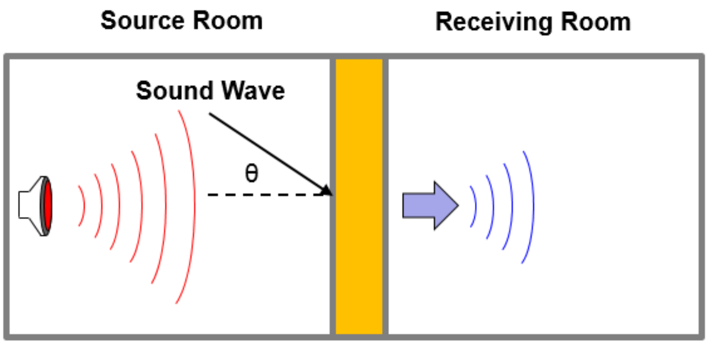
$m_s$ = surface mass density, in kg/m$^2$ (lb/ft$^2$)

$\theta$ = incident angle, in radians, as shown in Appendix 1, Figure 2

$\rho_0$ = density of air, kg/m$^3$ (lb/ft$^3$)

$c_0$ = speed of sound in air, m/s (ft/s) [9]

**FIGURE 2**
**Incident Angle**




### 1.5 Airborne Sound Transmission through Large Openings

The sound power level of the noise transmitted through the opening can be calculated by the equation below:

$$L_W = L_p + 10\log(A) \quad \text{dB} \qquad (A \text{ in m}^2)$$

$$= L_p + 10\log(A) - 10 \quad \text{dB} \qquad (A \text{ in ft}^2)$$

where

$L_W$ = sound power level transmitted into the compartment, in dB

$L_p$ = sound pressure level of the source room, in dB

$A$ = area of all openings in the partition, in m$^2$ (ft$^2$) [1]

## 3 Structure-borne Sound Transmission

The total transmission loss from the effective source area to the receiver through structure-borne path can be calculated by:

$$TF_{path} = \Delta L_a + \Delta L_d + TL_{inter} \quad \text{dB}$$

where

$\Delta L_a$ = transmission loss in the source room, in dB

$\Delta L_d$ = transmission loss caused by structure damping, in dB

$TL_{inter}$ = transmission loss caused by intersections, in dB

### 3.1 Within the Source Room

Appendix 1, Figure 3 shows the layout of the effective source area. The transmission loss can be calculated by the formula below [6]:

$$\Delta L_a = 10\log\frac{(r_1 + r_2)(1 - 0.35\varepsilon_r)}{(a + 2r_{fr})(1 - 0.35\varepsilon_0)} \quad \text{dB}$$

where

$r_i$ = distance from the ends of effective source area to the middle of the junction of interest, in m (ft)

$\varepsilon_0$ = $\dfrac{(a-b)(a+b+4r_{fr})}{(a+2r_{fr})^2}$

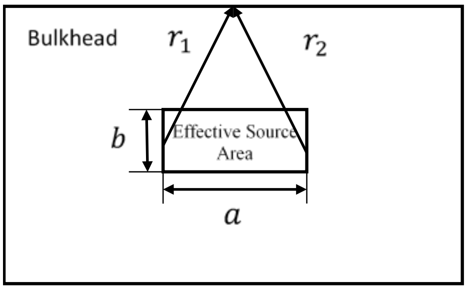
$\varepsilon_r$ = $\dfrac{(a-b)(a+b+4r_{fr})}{(r_1+r_2)^2}$

$a$ = length of the effective source area, in m (ft)

$b$ = width of the effective source area, in m (ft)

$r_{fr}$ = minimum frame spacing, in m (ft)

**FIGURE 3**
**Source Room Layout**




### 3.3 Beyond Source Room

For the ship structure beyond the source room, the structure-borne transmission loss depends on the damping loss factor of the structure, the size of the structure, and the distance from source to the structure of interest. Appendix 1, Figure 4 shows the structure-borne sound transmission in ship structures. The damping loss factor of the structure outside the source room ($\Delta L_d$) can be calculated by the formula below:

$$\Delta L_d = 0.066\sqrt{f}\sum_N \frac{\ell_i \eta_i}{\sqrt{t_i}} \quad \text{dB} \qquad (\ell, t \text{ in m})$$

$$= 0.126\sqrt{f}\sum_N \frac{\ell_i \eta_i}{\sqrt{t_i}} \quad \text{dB} \qquad (\ell, t \text{ in ft})$$

where

$f$ = octave band frequency of interest, in Hz

$N$ = number of “structures” between the source room and the structure of interest

$\ell_i$ = size of the $i^{th}$ structure located along the shortest path in the direction of wave spreading, in m (ft)

$\eta_i$ = damping loss factor of the $i^{th}$ structure including the damping treatment

$t_i$ = thickness of the $i^{th}$ structural element plating, in m (ft)

$i$ = number of the structure [6]

### 3.5 Intersections

The transmission loss caused by all intersections can be calculated by the formula below:

$$TL_{inter} = \sum_{n-1}^{N} \frac{TL_n}{\sqrt{n}} \quad \text{dB}$$

where

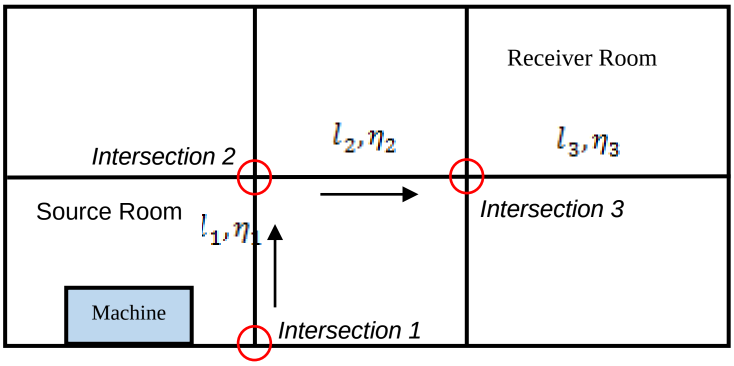
$N$ = number of connections between the source and receiver

$n$ = connection number

$TL_n$ = transmission loss for the $n^{th}$ intersection, in dB [6]

Usually, $N$ is no greater than 5. The structure-borne sound is insignificant after transmitting through large distance having 5 intersections.

**FIGURE 4**
**Structure-borne Sound Transmission Path in Ship Structures**




## 5 Duct-borne Sound Transmission

### 5.1 Plenum

The sound attenuation of a plenum, $T_{plena}$, can be calculated by the formula below:

$$T_{plena} = 10\log(R) - 10\log(A_0) - 16 \quad \text{dB} \qquad (R, A_0 \text{ in m})$$

$$= 10\log(R) - 10\log(A_0) - 6 \quad \text{dB} \qquad (R, A_0 \text{ in ft})$$

where

$R$ = room constant for the plenum, in m$^2$ (ft$^2$) [1]

$A_0$ = cross-section area of the outlet duct from the plenum, in m$^2$ (ft$^2$)

### 5.3 Silencers

Appendix 1, Table 1 shows the sound attenuation of some typical silencers [1].

## TABLE 1
## Approximate Attenuation of Typical Silencers, in dB

| *Type* | *Length* | *Octave Band Center Frequency, Hz* | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | *31.5* | *63* | *125* | *250* | *500* | *1000* | *2000* | *4000* | *8000* |
| Low Pressure Drop | 0.9 m (3 ft) | 1 | 4 | 7 | 9 | 12 | 15 | 16 | 14 | 9 |
| | 1.5 m (5 ft) | 2 | 8 | 12 | 14 | 16 | 19 | 20 | 18 | 14 |
| | 2.1 m (7 ft) | 3 | 10 | 15 | 19 | 20 | 22 | 24 | 22 | 18 |
| High Pressure Drop | 0.9 m (3 ft) | 3 | 8 | 10 | 15 | 23 | 30 | 35 | 28 | 23 |
| | 1.5 m (5 ft) | 4 | 11 | 14 | 23 | 32 | 38 | 42 | 36 | 30 |
| | 2.1 m (7 ft) | 5 | 13 | 18 | 30 | 40 | 44 | 48 | 42 | 36 |

### 5.5 Straight Duct Attenuation

The attenuation for each straight duct with the same section property, $T_{straight}$, can be calculated by:

$$T_{straight} = \beta \cdot \ell_{duct} \quad \text{dB}$$

where

$\beta$ = sound attenuation per one meter (feet) of the duct length, in dB/m (dB/ft)

$\ell_{duct}$ = length of the straight duct, in m (ft) [1]

Appendix 1, Tables 2 and 3 show the empirical value of $\beta$ for a typical types of duct [1].

## TABLE 2
## $\beta$, Attenuation, in dB per Meter of Duct Length

| *Lining Configuration* | *Inner Diameter, (m)* | *Octave Band Center Frequency, Hz* | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | *31.5* | *63* | *125* | *250* | *500* | *1000* | *2000* | *4000* | *8000* |
| Unlined | 0.46-0.61 | 0.33 | 0.30 | 0.20 | 0.10 | 0.07 | 0.07 | 0.07 | 0.07 | 0.07 |
| 25 mm Inside Lining | 0.46-0.61 | 0.69 | 0.79 | 0.95 | 3.44 | 6.56 | 9.18 | 8.86 | 7.54 | 4.89 |

## TABLE 3
## $\beta$, Attenuation, in dB per Feet of Duct Length

| *Lining Configuration* | *Inner Diameter, (in.)* | *Octave Band Center Frequency, Hz* | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | *31.5* | *63* | *125* | *250* | *500* | *1000* | *2000* | *4000* | *8000* |
| Unlined | 18-24 | 0.1 | 0.09 | 0.06 | 0.03 | 0.02 | 0.02 | 0.02 | 0.02 | 0.02 |
| 0.98 in Inside Lining | 18-24 | 0.21 | 0.24 | 0.29 | 1.05 | 2 | 2.8 | 2.7 | 2.3 | 1.49 |

### 5.7 Branches

The sound attenuation caused by the division of the sound energy through a branch, $T_{branch}$, can be calculated by the formula below.

$$T_{branch} = 10\log(A_{BT}/A_p) \quad \text{dB}$$

where

$A_{BT}$ = total cross-sectional area of all ducts leaving the branch point, in $m^2$ ($ft^2$)

$A_p$ = cross-sectional area of the duct associated with the transmission path of interest, in $m^2$ ($ft^2$) [1]

### 5.9 Turns

Appendix 1, Table 4 shows the empirical value of the attenuation caused by turn for a typical round duct [6].

**TABLE 4**
**Attenuation for Turns, in dB**

| *Lining Configuration* | *Inner Diameter, mm* | *Octave Band Center Frequency, Hz* | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | *31.5* | *63* | *125* | *250* | *500* | *1000* | *2000* | *4000* | *8000* |
| Unlined | 250 (98 in.) | 0 | 0 | 0 | 1 | 2 | 3 | 3 | 3 | 3 |
| Lined | 250 (98 in.) | 0 | 1 | 2 | 5 | 9 | 12 | 13 | 11 | 8 |

### 5.11 End Reflections at Duct Openings

The attenuation caused by the end reflections at duct openings, $T_{end}$, can be calculated by the empirical formula below:

- For ducts terminating in free spaces:

$$T_{end} = 10\log\left[1+\left(\frac{c}{\pi fd}\right)^{1.88}\right] \quad \text{dB}$$

- For ducts terminating flush with a boundary:

$$T_{end} = 10\log\left[1+\left(\frac{0.8c}{\pi fd}\right)^{1.88}\right] \quad \text{dB}$$

where

$c$ = speed of sound, in m/s (ft/s)

$f$ = frequency of interest, in Hz

$d$ = opening diameter, m (ft) [6]

# APPENDIX 2 Statistical Energy Analysis (SEA) Method

## 1 Overview

The SEA method calculates the diffusion of acoustic and vibration energy in complex acoustic systems using energy flow relationships. It predicts the average response of the structure, which avoids a large quantity of calculations. In the SEA method, the entire structure is considered as a system, which can be divided into a number of coupled subsystems, such as plates, beams, and cavities. Each subsystem represents a group of modes with similar characteristics and a storage of energy. The SEA subsystems can be considered to be “control volumes” for vibratory or acoustic energy flow. The successful prediction of the acoustic energy of each subsystem greatly depends on an accurate estimation of three parameters: the internal loss factor, the modal density, and the coupling loss factor.

The internal loss factor, $\eta$, is a critical parameter to predict the vibrational response of a structure by SEA and is usually obtained experimentally. It primarily incorporates three different damping losses:

*i)* Structural damping loss factor, $\eta_s$, which is associated with energy dissipation within the structural element itself

*ii)* Acoustic radiation damping loss factor, $\eta_{rad}$

*iii)* Damping loss factor at the structural boundaries, $\eta_j$, which is associated with energy dissipation at the boundaries of the structural element

Modal density is a parameter describing the energy storage capacity of each subsystem, which depends on the average speed with which waves propagate energy through the subsystem and the overall dimensions of the subsystem. For simple subsystems, such as bars, beams, flat plates, and acoustic volumes, theoretical analysis can be used to calculate this parameter. For complex subsystems, when the subsystems to be modelled are not ideal structural elements, experimental techniques can be used to obtain this parameter.

The coupling loss factor, $\eta_{ij}$, for the link between two coupled subsystems *i* and *j* is a measure of the rate of the energy flowing out of a subsystem through a junction to another subsystem. The larger the coupling loss factor, the greater the energy transmission. Theoretical expressions are available for couplings between several simple connections.

Line conjunctions between two structures are the most commonly-seen junctions. The coupling loss factor of a line junction can be calculated by the formula below:

$$\eta_{12} = \frac{2C_B L \tau_{12}}{\pi f S_1}$$

where

| | | |
|---|---|---|
| $C_B$ | = | bending wave velocity of flexural waves in the first plate, in m/s$^2$ (ft/s$^2$) |
| $L$ | = | line length of the line, in m (ft) |
| $\tau_{12}$ | = | wave transmission coefficient of the line junction from subsystem 1 to subsystem 2 |
| $f$ | = | center frequency of the band of interest, in Hz |
| $S_1$ | = | surface area of the first subsystem, in m$^2$ (ft$^2$) [2] |

The coupling loss factor between a structure and an acoustic volume can be calculated by the formula below. The subscript S and V in the formula refer to the structure and the acoustic volume respectively.

$$\eta_{SV} = \frac{\rho_0 c \sigma}{f \rho_S}$$

$$\eta_{VS} = \frac{\rho_0 c \sigma n_S}{f \rho_S n_V}$$

where

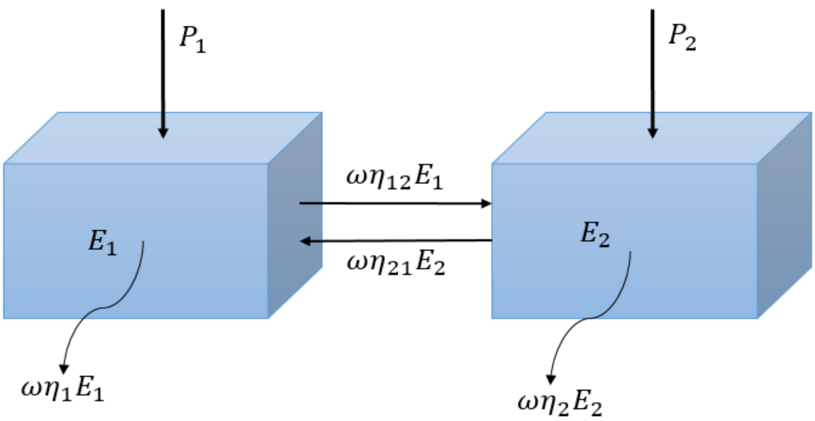
| | | |
|---|---|---|
| $\rho_0$ | = | fluid density, in kg/m$^3$ (lb/ft$^3$) |
| $c$ | = | speed of sound, in m/s$^2$ (ft/s$^2$) |
| $\sigma$ | = | radiation ratio of the structure |
| $f$ | = | center frequency of the band of interest, in Hz |
| $\rho_S$ | = | mass per unit area, in kg/m$^2$ (lb/ft$^2$) |
| $n_S$ | = | model density of the structure |
| $n_V$ | = | model density of the volume [2] |

When theoretical estimates for coupling loss factors are not available, experimental measurement techniques can also be used to determine the coupling loss factors.

## 3 Energy Flow Relationships

Appendix 2, Figure 1 shows the acoustical power flow between two subsystems.

**FIGURE 1**
**Acoustical Power Flow between Two Subsystems**




The steady-state energy balance matrix can be expressed as follows:

$$\omega[A]\begin{bmatrix} \frac{E_1}{n_1} \\ \cdots \\ \cdots \\ \frac{E_k}{n_k} \end{bmatrix} = \begin{bmatrix} P_1 \\ \cdots \\ \cdots \\ P_k \end{bmatrix}$$

where

| | | |
|---|---|---|
| $E_i$ | = | average energy of the $i^{th}$ subsystem |
| $n_i$ | = | modal density of the $i^{th}$ subsystem |
| $P_i$ | = | power transmitted into the $i^{th}$ subsystem |
| $[A]$ | = | matrix of damping [2] |

$[A]$ is expressed as follows:

$$[A] = \begin{bmatrix} \left(\eta_1 + \sum_{i\neq 1}^{k} \eta_{1i}\right) n_1 & -\eta_{12} n_1 & \cdots & -\eta_{1k} n_1 \\ -\eta_{21} n_2 & \left(\eta_2 + \sum_{i\neq 12}^{k} \eta_{2i}\right) n_2 & \cdots & -\eta_{2k} n_2 \\ \cdots & \cdots & \cdots & \cdots \\ -\eta_{k1} n_k & \cdots & \cdots & \left(\eta_k + \sum_{i\neq k}^{k} \eta_{Ni}\right) n_k \end{bmatrix}$$

where

| | | |
|---|---|---|
| $\eta_{ij}$ | = | coupling loss factor between the $i^{th}$ and the $j^{th}$ subsystem |
| $\eta_i$ | = | internal loss factor of the $i^{th}$ subsystem [2] |

By solving the energy balance equation, the diffusion of the acoustic energy can be obtained. The mean square vibration velocity of the structures $<v^2>$ and the mean square sound pressure $<p^2>$ are calculated using the following equations:

$$<v^2> = \frac{E_1}{m}$$

$$<p^2> = \frac{\rho c^2}{V} E_2$$

where

| | | |
|---|---|---|
| $E_1$ | = | vibration energy of panel, in Joules (lb·ft²/s²) |
| $m$ | = | panel mass, in kg (lb) |
| $E_2$ | = | acoustic energy of cavity, in Joules (lb·ft²/s²) |
| $V$ | = | cavity volume, in $m^3$ ($ft^3$) |
| $\rho$ | = | density of fluid, in $kg/m^3$ ($lb/ft^3$) |
| $c$ | = | sound speed in fluid, in m/s (ft/s) [2] |

# APPENDIX 3 References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