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Analysis of current solutions in the nautical and industrial fields: state of the art

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Contents

Summary

Since the 50s, noise and vibration issues have received increasing attention from ship owners, shipyards and customers, since they affect health ad comfort of crew and passengers. Thus, on the one hand, many regulations and recommendations were released by national and international bodies in order to safeguard the health and comfort of people on board. On the other hand, design tools capable to predict the interior noise and vibrations in ship's cabin, have been investigated and developed.

This document shortly describes the above mentioned aspects, firstly defining some general features of noise main sources and paths on board. Then some regulations both for noise and vibration are discussed, including some advanced indicators suggested to better describe the acoustic comfort for these and similar applications.

The second part of the document is focused on the analysis of the methods available in the literature used in predictive tools for numerical simulations of interior noise and vibrations on board.

Finally, some example of the current passive and active solutions adopted in the naval field to control noise and vibrations are also described.

1 Introduction

Since the 50s, the shipboard noise and vibration problem received increasing attention from ship owners, shipyards and customers, being a significant factor of health, navigation safety and comfort in sailing and mooring conditions, see e.g. [Vasconcellos and Latorre, 2001; Rutkowski and Korzeb, 2021]. In fact, noise and vibrations on board can involve problems related to the hearing, sleeping and working performances of the crew in the working areas and to the comfort perception by crew and passengers in living, recreation and resting spaces [Bouzón et al., 2015; Fischer and Yankaskas, 2011; Kurt et al., 2017; Picu et al., 2019; Sunde et al., 2016; Zhang et al., 2013].

In order to safeguard the health and comfort of people on board, many regulations and recommendations were released by national and international bodies. The normative framework is quite complex since these rules appear stratified and tangled [Badino et al., 2011, 2012; Boote et al., 2013; Smullin, 2002].

Although many guidelines address quantities to be measured and limits to determine the acoustic quality on board, more meaningful annoyance criteria, accounting for spectral content and repetition on time of noise, have been proposed in the recent literature. In particular, sound rating criteria applied in the civil engineering field [Badino et al., 2011] or psychoacoustic investigations [Volle et al., 2003; Goujard et al., 2005; Seiler and Holbach, 2013] have been proposed as advanced indicators of the acoustic quality in the nautical field.

The identification and quantification of noise and vibration sources and transfer paths constitute the main concern of most literature studies. Once the source, the transfer paths and the radiation properties of the structures at the receiving end are identified, passive control strategies (e.g., partitions, hoods, screens or absorbing panels or floating floors) can be effective, mainly, in airborne noise control [Marchesini and Piana, 2012a; Joo et al., 2009]. Active noise control has been also applied in nautical vehicles and described in [Cheer and Elliot, 2016; Ishimitsu and Elliott, 2004; Ishimitsu and Shibatani, 2007, 2008; Peretti et al., 2014; Winberg et al., 2005, 2000].

The availability of predictive tools for numerical simulations of interior noise and vibrations in ship's cabin supports and simplifies the development of control strategies, feasible and in accordance with structural and safety requirements, at the design stage. While the finite element method (FEM) is typically applied to vibration studies, for structure-borne noise simulation in large structures like ships, alternative approaches have been proposed [Parunov et al., 2012] as those based on statistical energy analysis (SEA) [Boroditsky et al., 2007; Burkwitz et al., 1994; Cabos and Matthies, 2000; Chang et al., 2006; Fischer et al., 2012; Insel et al., 2007; Ma and Li, 2003; Ming et al., 2021; Plunt, 1980, 1999; Tso and Hansen, 1997; Wang et al., 2007; Weryk, 2012], from 60s, and, in the last two decades, energy finite element analysis/method (EFEA/EFEM) [Gilroy et al., 2005; Parunov et al., 2012; Vlahopoulos, 2011; Vlahopoulos et al., 1999; Vlahopoulos and Wu, 2010], are nowadays applied for structure-borne noise predictions [Parunov et al., 2012]. Other analysis, like empirical [Plunt, 1980] and wave-guide [Nilsson, 1977, 1984] methods have been applied.

The main issues of the noise and vibration on board are given in §2, while a brief description of the present normative framework and consequent measurements is reported in §3. The main methods and tools of simulation of ship's cabin interior noise and vibration are compared in §4. Finally, the current passive and active solutions adopted in the naval field to control noise and vibrations are presented in §5. Conclusive remarks are synthesized in §6.

2 Noise and vibration on board ships

From the point of view of noise and vibration, a ship is very complex to analyze since it includes many sources, flanking parts and discontinuities. Many types of vessels were investigated in the past and recent literature, such as: recreational boats (fast boats [Chang et al., 2006] and, mainly, luxury yachts, mega-yachts and super-yachts) [Boote et al., 2013; Insel et al., 2007; Li, 2011; Marchesini and Piana, 2012a; Peretti et al., 2014; Plunt, 1999; Seiler and Holbach, 2013; Smullin, 2002], passenger ships (cruise ships and ferries) [Besnier et al., 2007; Goujard et al., 2005; Holland and Wong, 1995; Milburn, 2010; Verheij, 1982; Volle et al., 2003; Yucel and Arpaci, 2010, 2013], merchant ships (e.g. fishing vessels) [Nilsson, 1978; Wu and Yang, 2008; Zytoon, 2013], push boats [Lekic et al., 1999], training [Ishimitsu and Elliott, 2004; Ishimitsu and Shibatani, 2007, 2008] and surface ships [Vlahopoulos, 2011; Vlahopoulos and Wu, 2010].

Many of these studies deal with the identification and quantification of noise and vibration sources and transfer paths and characterization of the radiation properties of the structures at the receiving end. These three aspects are deepened in the following subsections. Ship onboard noise propagation is a serious issue especially for vessels up to 100 m [Weryk, 2012].

2.1 Sources

The noise and vibration on board are determined by the sum of the contributions of many sources, that should be identified and characterized in order to realize an effective control [Nilsson, 1978]. Typically, these sources include [Carlton and Vlasic, 2005; Weryk, 2012; Fischer and Bahtiarian, 2017]:

- main and auxiliary engines,
- shaft-line dynamics,
- propeller radiated pressures and bearing forces (including the cavitation effect),
- air conditioning systems,
- gears and gearboxes,
- pumps, compressors, maneuvering devices,
- cargo handling and mooring machinery,
- vortex shedding mechanisms,
- intakes and exhausts,
- slamming phenomena.

The propulsive engine and auxiliary machinery are generally the main steady state noise sources [Burella et al., 2019]. However, all the sources should be carefully analyzed. It can be observed that most sources are located in the aft or middle part of the hull, depending from the vessel type, and affect all other compartments on the basis of their distance from noise source [Weryk, 2012]. As an example of source of noise and vibration on board a vessel, the exhaust system investigated in [Martins et al., 2009a,b] is shown in Figure 1.

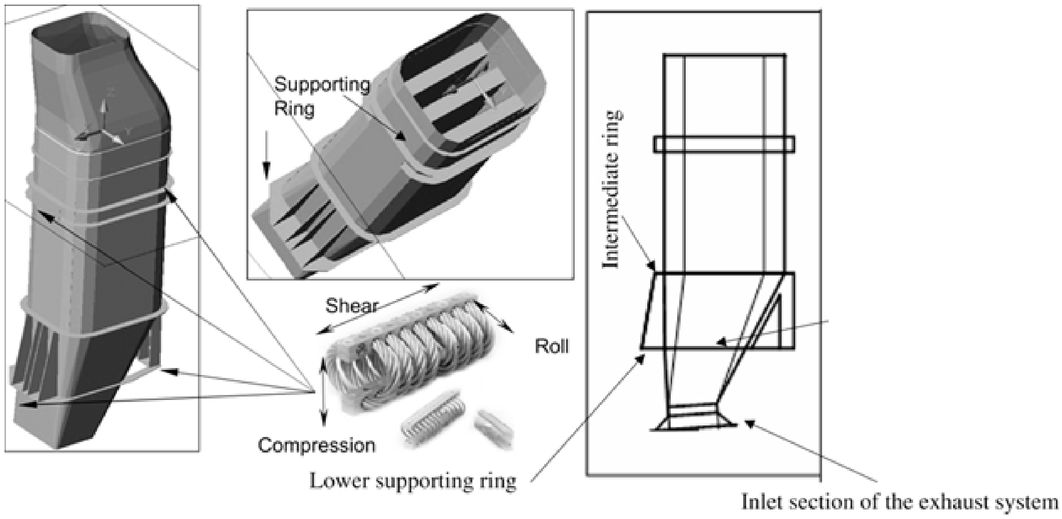


Figure 1: Overall view of the exhaust system in [Martins et al., 2009b].

2.2 Transfer paths

When generated, the noise in a ship propagates in various ways (Figure 2):

- *air-borne noise* (more generally "fluidborne") radiated by a source and transmitted through walls, bulkheads and decks;
- *structure-borne noise*, which causes the appearance of noise in ship compartments remote from the source of vibration.

The expression structure-borne noise refers to noise generated by high frequency structural vibrations in the 16 000 to 20 000 Hz frequency range, induced in the structure. These vibrations excite partitions in ship structure and cause them to radiate noise. Structure borne noise is induced by any mechanical power transmitted from a source through its connection to the foundation and propagating into the structure as flexural, longitudinal, transverse and torsional waves [Nilsson, 1978]; these kinds of wave are coupled since other wave types can be generated from a pure one at a junction. Only the flexural waves need to be considered at the receiving end since they are coupled to the sound field in a room [Nilsson, 1978].

Two structure-borne paths can be identified, contributing to the total radiated noise at the receiving end: the first structure-borne path is generated by the source vibrations while the second path is produced by the source noise that impacts to the structure through the air-borne path (Figure 2).

Generally, noise present on a ship is mostly structure-borne noise [Cabos and Matthies, 2000; Hynná et al., 1995; Janssens et al., 1999; Joo et al., 2009; Milburn, 2010; Nilsson, 1976, 1977, 1978, 1984] while air-borne noise [Fischer et al., 1985; Fischer and Pettit, 1988; Joo et al., 2008] determines almost exclusively the sound pressure in compartments, which are adjacent to the main sources [Weryk, 2012].

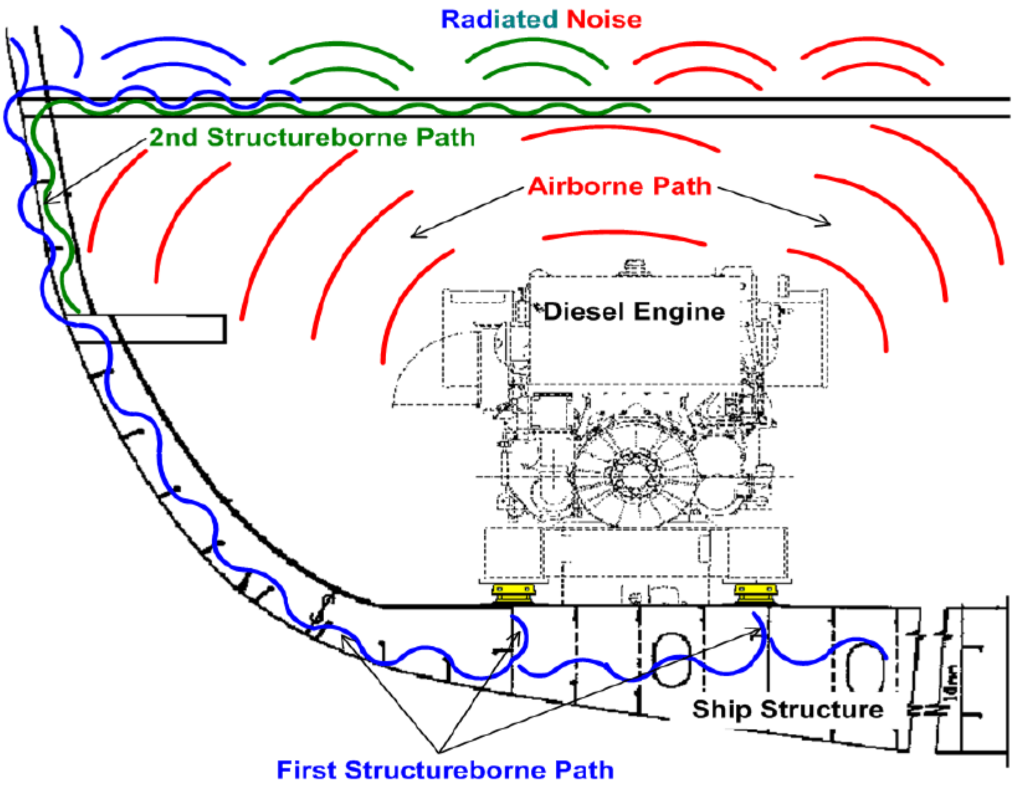


Figure 2: Noise transfer paths [Fischer and Bahtiarian, 2017].

3 Regulatory framework

In the nautical field, regulations and requirements about shipboard noise and vibrations are stratified and tangled, being released by a number of international and national bodies, such as the International Organization for Standardization (ISO), the International Maritime Organization (IMO), the International Labor Organization (ILO), Class Societies (i.e. non-governmental organizations that establish and maintains technical standards for the construction and operation of ships and offshore structures) and national authorities. Furthermore, given the whole normative and regulatory frame, each owner can obviously add its own stricter requirements.

Most of these rules address problems connected to the health and performances of the crew and to the comfort of both crew and passengers in the accommodation spaces.

Typically, all regulations include a detailed description of the technical aspects concerning the definitions of the quantities to be measured and limits to be satisfied, the instrumentation, the test environment, the procedures, the operating conditions and the test report, referring to other normative references (e.g. International Electrotechnical Commission standards), thus representing guidelines for experimental activities and measurements. The requirements represent also a crucial aspect in the design of vessels in general and, especially, of cruise ships and yachts, since acoustic and vibration comfort constitutes a primary objective for the achievement of new customers by shipyards and ship owners.

3.1 Vibration exposure regulations

Concerning the vibration control on board of passenger and merchant ships, the international standard ISO 20283-5:2016, (that replace the previous ISO 6954:2000), entitled "*Mechanical vibration —- Measurements of vibration on ship Part 5: Guidelines for the measurement, evaluation and reporting of vibration with regard to habitability on passenger and merchant ships*", constitutes a basic reference.

Since the 2000 edition, the attention of the International Organization for Standardization is explicitly turned to the evaluation of ship vibrations with respect to "habitability", i.e. the effect and comfort on human beings [Biot and De Lorenzo, 2007]. The limit values are expressed in terms of the overall frequency-weighted r.m.s. acceleration (mm/s^2) and overall frequency-weighted r.m.s. velocity (mm/s) in the range 1 Hz to 80 Hz. The human sensitivity curve on which the frequency-weighting curves are based is also reported. Thus, vibration limits, specified for three classification areas (denoted A, B, C) of a ship, concern an overall value characterizing the spectrum [Biot and De Lorenzo, 2007]. These and other aspects of the 2000 edition of the ISO rule are extensively discussed in [Biot and De Lorenzo, 2007].

The UNI ISO 6954:1990, the Italian regulation which adopts the ISO 6954, was withdrawn and not replaced in 2016.

The most representative and specific ILO document covering noise and vibration effects on workers on board ships with working areas are the "*Maritime Labor Convention*"

[ILO, 2006], the ILO Convention no. 188 — "*Work in Fishing Convention*" [ILO, 2007a] and ILO Recommendation no. 199 — "*Work in Fishing Recommendation*" [ILO, 2007b]. They all follow the general requirements of the ILO 2001 document entitled "*Ambient factors in the workplace*". However, the ILO documents are not very technical [Badino et al., 2012] since they do not include quantitative information about checking the acceptability of shipboard conditions.

3.2 Noise exposure regulations

The ILO regulations, mentioned in the previous section, indicate qualitative measures addressing the mitigation of excessive noise and vibration on board ships including working areas.

The ISO 2923:1996, entitled "*Acoustics — Measurements of noise on board vessels*", is a general purpose International Standard containing techniques and conditions for the measurement of noise on-board vessels both inland and seagoing [ISO, 1996]. It substitutes the 1975 edition; it was revised and confirmed in 2011 and it represents the current primary reference in analyzing on-board noise.

The Italian regulation which adopts the international ISO 2923:1996 (and its corrigendum published in 1997) is the UNI ISO 2923:2006 [UNI, 2006]. This regulations specifies techniques, (i.e. instrumentations and procedures) and operating conditions for the measurements of noise on board vessels, both inland and seagoing, assuming as basic quantities to be measured the equivalent continuous A-weighted sound pressure level, the C-weighted peak sound pressure level (when there is a risk that it may exceed 130 dB), the equivalent continuous sound pressure levels in octave bands from 31.5 Hz to 8 kHz, the presence of impulsive noise and tonal sound.

The first indications that mention noise control requirements are included in the SOLAS International Convention for the Safety of Life at Sea [IMO, 1974].

The most specific IMO code that extensively deals with the noise exposure on-board of commercial ships, with the aim of protect the sailors' hearing health, is the Resolution A.468(XII) [IMO, 1981] entitled "*Code on Noise Levels on Board Ships*", well known as "Noise Code". In this code, the threshold values of maximum noise levels for all space normally accessible to seafarer are indicated, and recommendations on the use of personal protection equipment are provided. Furthermore, the Noise Code includes recommendation about noise measurements and exposure evaluations. The Noise Code should be applied to all kinds of vessels with the exception of dynamically supported craft, fishing vessels, pipe-laying barges, crane barges, mobile offshore drilling units, pleasure yachts not engaged in trade, war ships and ships not propelled by mechanical means. It is important to note that, "the Code is not intended to apply to passenger cabins and other passenger spaces except in so far as they are work spaces and are covered by the provisions of the Code" [IMO, 1981].

The Noise Code basically consists in a set of A-weight Sound Pressure Level (SPL) limits fixed for different spaces, with only some minor considerations about the noise spectrum in terms of Noise Rating curves; it also includes some exposure time limits, in terms of sound equivalent level, and limits to the airborne insulation index for bulkheads and decks.

From 2011, started an updating processing involving the limit values. The final result was the Resolution MSC. 337 (91) "Adoption of the Code on Noise Level on Board Ship", adopted starting from 30 November 2012 [IMO, 2012]. This document is not very different from the previous one, even if a reduction has been foreseen for some spaces 5 dB (A) of the limit values. The limit values introduced by the various documents are shown in Table 1.

In 2012, within the EU FP7 SILENV Project (SILENV2009), following the methodology represented in Figure 3, a new classification for noise limits on board has been set, linked with the award of a Green Label "notation" (SILENV2012). This notation is intended to be a symbol of the environmental acoustic quality of the ship, subjected to the fulfillment of requirements not only in internal spaces, but on noise emissions from the ship into air and water, too. As concerns the internal part of the ship, the noise limits regard both rest and work spaces. In comparison with the previous regulatory framework, the maximum acceptable values are reduced. This means that the proposed SILENV limits are on the average more restrictive and in some cases much more restrictive than the compulsory Noise Code. Further, a more detailed classification of the spaces is introduced: in particular, workspaces are divided in four categories and different limit levels were assigned to each one [Borelli et al., 2015a]. The SILENV limit values were set to balance the required acoustics performance with the actual feasibility, in details three criteria were used: a) ensuring at least 90% of passengers and crew satisfaction; b) ensuing that a significant percentage of the existing vessels reaches the targets (at least 25%); c) feed-back from the end-users [Borelli et al., 2015a].

Table 1: Limit values for crew spaces indicated in the IMO Noise Codes and SILENV Green Label.

Space type	Spaces	Res. A468(XII) 1981	Res.MSC337(91) 2012		SILENV Green Label
		> 1600 GT	1600-10,000 GT	> 10,000 GT	
Work spaces	Machinery spaces (continuously named)	90	110	110	85
	Machinery spaces (not continuously named)	110	–	–	105
	Cargo handling spaces	–	–	–	80
	Fan rooms	–	–	–	85
	Machinery control rooms	75	75	75	65
	Workshops	85	85	85	
	Non-specified work spaces	90	85	85	
Navigation spaces	Navigating bridge and chartrooms	65	65	65	55
	Listening post, including bridge wings	70	70	70	65
	Radio rooms (not producing audio signals)	60	60	60	55
	Chart rooms	–	–	–	55
	Radar rooms	65	65	65	55
Accommodation spaces	Cabins and hospital	60	60	55	50
	Mess rooms	70	70	70	65
	Recreation rooms	65	65	60	57
	Open recreation areas	75	75	75	65
	Offices	65	65	60	55
Service spaces	Galleys without food processing equipment	75	75	75	
	Serveries and pantries	75	75	75	
	Closed public spaces	–	–	–	55
	Spaces not specified	90	90	90	

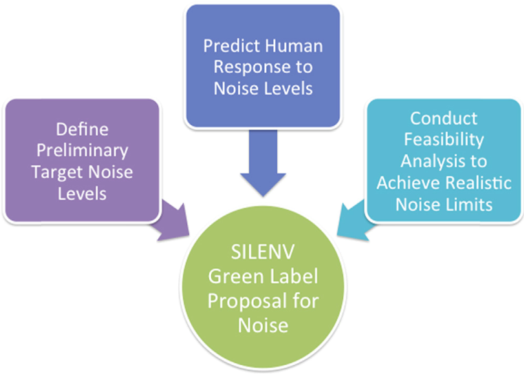


Figure 3: SILENV approach for defining noise limits [Kurt et al., 2016].

Comfort rules

Today there are a number of Classification Societies (CSs) that belong to the International Association of the Classification Societies (IACS), and that recently, released "Comfort Class Rules" for the assessment of noise and vibration comfort in the ship's compartments. The Comfort Class Rules mainly refer to energetic indexes (i.e. the A-weight SPL) for the evaluation of acoustic comfort on board with regards to noise.

Among the most important Classification Societies, members of the IACS, the American Bureau of Shipping (ABS) published in 2019 the renewed version of the "Guide for Comfort on Yacht" [American Bureau of Shipping (ABS), 2019]. The guide represents a guidance for measuring the comfort of yacht owners and their guests while occupying cabins, dining spaces, lounges, cocktail bars and other interior or exterior owner and guest spaces onboard, and it establishes two levels of noise and vibration comfort on-board denoted Comfort-Yacht (COMF(Y)) and Comfort plus-Yacht (COMF+(Y)). The latter include some additional requirements for the assessment of motion sickness. Comfort criteria for whole-body vibration and noise are different for yachts below and over 50 m in length. Limits to the multi-axis acceleration value (calculated from the root-sums of squares of the weighted rms acceleration values in each axis at the measurement point) and to the equivalent continuous A-weighted sound pressure level are reported in the vibration and in the noise sections, respectively.

The guide provides the maximum values of acceptable sound levels (L_{Aeq}, dBA) in different environments (such as cabins, public spaces and passageways, etc.) in two different conditions: boat in harbor and transit conditions, and it divides the requirements for yachts longer and shorter than 50 m, Table 2. Each maximum accepted level is indicated both for comfort level COMF(Y) and for comfort level COMF+(Y) in which, obviously, the required requirements are more restrictive (Table 2).

The guide also provides guidance on airborne sound insulation properties for bulkheads and decks within the accommodation are to comply at least with the weighted sound reduction index (R_w) according to ISO 717-1, see Table 3.

Table 2: ABS Maximum acceptable noise levels for comfort in yacht.

	Maximum Acceptable Noise L_{Aeq} Level dB(A)							
	Yacht $\leq$ 50 metres				Yacht $>$ 50 metres			
	In-Harbor		Transit condition		In-Harbor		Transit condition	
	COMF(Y)	COMF+(Y)	COMF(Y)	COMF+(Y)	COMF(Y)	COMF+(Y)	COMF(Y)	COMF+(Y)
Owner's cabin	52	50	55	53	50	48	53	51
Passageways serving Owner's cabin	52	50	55	53	50	50	53	51
Guests cabin	52	50	55	53	50	48	53	51
Passageways serving Guest cabin	55	55	58	57	52	53	55	55
Dining spaces	55	52	58	55	52	52	55	55
Indoor Guest spaces	55	52	58	55	52	50	55	53
Entertainment spaces	55	52	58	55	52	50	55	53
Passageways near Guest spaces	60	57	63	60	57	57	55	60
Gymnasiums	65	65	65	65	65	65	65	65
Outdoor spaces	70	70	73	73	67	67	70	70
Medical facilities		55	55	55	55	55	55	55

Table 3: ABS airborne sound insulation requirements.

Airborne Sound Insulation	$\mathbf{R_w}$ (dBA)
Cabin to cabin	38
Messrooms, recreation rooms, public spaces and entertainment areas to cabins and hospitals	48
Corridor to cabin	33
Cabin to cabin with communicating door	33

The Bureau Veritas (BV) released in 2020 the last version of the "NR 467 - *Rules for the Classification of Steel Ships - Part F: Additional Class Notations - Chap. 6: Comfort on board (COMF)*" [BV, 2020]. In such guide, class notation requirements are indicated for:

- Ships of less than 1600 GT (such as fishing ships, tugs, small passenger ships excluding yachts and pleasure crafts);
- Ships greater than or equal to 1600 GT (such as tankers, container ships, large fishing vessels, cruise ships, ferries, ...);
- Yachts.

Comfort criteria with regards to noise and vibration are distinguished and denoted COMF-NOISE and COMF-VIB, respectively. The notations COMF-NOISE and COMF-VIB are completed by a level 1, 2 or 3 which represents the comfort level achieved for the assignment of the notation, the grade 1 corresponding to the most comfortable (highest) class notation. The additional requirements for yachts, for example, include the description of procedures for noise, sound insulation and impact measurements;

limits for A-weighted sound pressure levels, apparent weighted sound reduction indexes and weighted normalized impact sound pressure level are tabulated for three comfort levels, at harbor and sea conditions, in various ship's areas.

Moreover, limits to the overall frequency weighted r.m.s. (vibration levels, vibration velocity (mm/s) values from 1 Hz to 80 Hz) and single amplitude peak vibration levels from 5 Hz to 100 Hz (vibration velocity (mm/s peak) values) and from 1 Hz to 5 Hz (acceleration (mm/s^2 peak) values) are reported.

This guide provides general indications valid for all types of boats and regarding the methods of measuring and calculating the comfort rating. The acoustic requirements are instead indicated in relation to the three categories of boats specified in the guide. In particular for yachts, the maximum noise levels accepted for the three comfort classes in the conditions of boat in port and boat in sea are shown in Table 4.

Table 4: BV Noise level requirements.

Locations	L_{Aeq}, T (dBA)					
	Harbor			Sea		
	Class 1	Class 2	Class 3	Class 1	Class 2	Class 3
Passengers cabins	40	45	50	50	54	58
Lounges	45	50	55	55	58	62
Open recreation areas	55	60	75	75	80	85
Crew cabins	45	50	55	55	58	60
Public spaces (closed rooms permanently manned at sea), mess rooms	55	58	60	60	63	65
Passages and closed rooms intermittently used at sea	60	63	65	-	68	72
Wheelhouse	-	-	-	65	65	65

The values of the Apparent weighted sound reduction indexes (R'_w) required between internal partitions for both passenger areas and crew areas (Table 5). Further requirements in this guide concern the Weighted normalized impact sound pressure levels requirements which are reported for three different conditions: cabins below public spaces covered with soft materials, cabins below public spaces covered with hard materials, and cabin below sport rooms or dance floors (Table 6).

Table 5: BV Apparent weighted sound reduction indexes requirements.

Locations	Apparent weighted sound reduction index R'_w (dBA)					
	Passenger areas			Crew areas		
	Class 1	Class 2	Class 3	Class 1	Class 2	Class 3
Cabin to cabin	45	42	40	37	35	32
Corridor to cabin	42	40	37	35	32	30
Stairs to cabin	48	45	45	35	32	30
Public spaces to cabin	55	53	50	45	45	45
Public spaces designed for loud music to cabin	64	62	60	-	-	-

Finally, for the COMF+ advanced comfort rating, the optimal reverberation time values are also indicated as a function of the volume of the rooms, for restaurants, bars, lounges and casinos, and for cabins, lecture rooms and libraries, see Table 7.

Table 6: BV Weighted normalized impact sound pressure levels requirements.

Weighted normalized impact sound pressure level	$\mathbf{L'_{n,w}}$ **(dBA)**
Cabin below public spaces covered with soft materials	50
Cabin below public spaces covered with hard materials (wood, marble, etc.)	60
Cabin below sport rooms or dance floors	45

Table 7: BV reverberation times requirements.

Volume (m^3)	Reverberation time (s)	
	Restaurant, bar, lounges and casino	Cabins, lecture rooms and libraries
$V \leq 50$	0,50	0,45
$50 < V \leq 100$	0,60	0,50
$100 < V \leq 200$	0,70	0,55
$200 < V \leq 500$	0,80	0,60
$500 < V \leq 1000$	0,90	0,70
$1000 < V \leq 2000$	1,00	0,80
$2000 < V \leq 3000$	1,00	0,80
$V > 3000$	1,20	0,90

In the "*Rules for classification of ships - Special service and type additional class - Part 5 - Chapter 12 - Comfort Class*", released by Det Norske Veritas (DNV) in 2011, the maximum allowed noise levels for different ships (passenger or cargo ships, yachts, high speed and light crafts) are specified, for three comfort rating numbers, in various locations; limits for the sound insulation indexes and for the vibration levels in mm/s peak for single frequency components in the range 5 to 100 Hz in various positions are also indicated.

Harmony categories, for a graduation of noise and vibration levels on board passenger cruise ships, were introduced by the Germanischer Lloyd (GL) in 2003 issued the "*Rules for Classification and Construction - Ship Technology - Part 1: Seagoing ships - Chapter 16: Harmony Class Rules on Rating Noise and Vibration for Comfort, Cruise Ships ($v \leq 25$ kn)*"; noise level limits, sound insulation indexes (in terms of minimum required weighted apparent sound insulation index in dB) and impact sound insulation indexes (in terms of maximum permissible normalized sound pressure level index in dB), together with vibration limits (in terms of overall frequency weighted r.m.s. value in the range 1 to 80 Hz), for each category, in various spaces, are specified.

In 2013 DNV and GL merged the DNV GL into a single organization and in October 2015 a new joint release was published: "DNV GL Rules for classification ship" [DNVGL, 2015]. In the Part 6 – Additional Class Notation, Chapter 8: Living and working condition of such document, the acoustic requirements for ship are updated.

As with the previous version of the DNV Guide 2011, the DNV GL 2015 guide provides guidance for different ships: passenger or cargo ships, yachts, high speed and light crafts and identifies three different levels of comfort. In particular, the maximum admitted sound levels are indicated for the different types of boats and only those relating to yachts are reported in the present document (Table 8). These levels are indicated in relation to the three comfort classes and in the conditions of yachts in harbor and yachts at sea.

The characteristics of sound insulation between accommodation spaces are instead reported according to the type of area (and not the type of boat), dividing the environments into crew areas and passenger areas. The required values of Apparent weighted sound reduction indexes (R'_w) required are reported in Table 9.

Table 8: DNV GL Noise level requirements for yacht.

Locations	L_{Aeq}, T (dBA)					
	Harbor			Sea		
	Class 1	Class 2	Class 3	Class 1	Class 2	Class 3
Sleeping rooms	35	40	45	-	-	-
Lounge / Saloon	40	45	50	53	58	62
Outdoor recreation areas	50	55	60	75	80	85
Navigation bridge	-	-	-	60	60	65

Table 9: DNV GL Maximum apparent airborne insulation requirements.

Location	Maximum apparent airborne insulation index R'_w (dBA)		
	Class 1	Class 2	Class 3
Cabin – Cabin (passenger standard)	41	38	35
Cabin – Cabin (passenger top grade)	46	43	40
Cabin (passenger standard) – Corridor or communicating cabin	38	35	33
Cabin (passenger top grade) – Corridor or communicating cabin	41	39	37
Cabin (passenger standard) – mess rooms, recreation rooms, public spaces	51	48	45
Cabin (passenger top grade) – mess rooms, recreation rooms, public spaces	56	53	50
Passenger cabin – entertainment area	65	62	60
Cabin – Cabin (crew)	38	35	32
Cabin (crew) – Corridor or communicating cabin	37	32	28
Cabin (crew) – mess rooms, recreation rooms, public spaces	50	47	42

The Lloyd's Register of Shipping (LR) in 2020 released the "*Rules and Regulations for the Classification of Special Service Crafts*" [LR, 2020]. In Part 7, Chapter 12 is dedicated to the "Passenger and Crew Accommodation Comfort" and the related Section 2 provide specific information about Acoustics requirements. This guide set down the criteria for the assessment of the noise and vibration on ships and are applied in addition to the other relevant requirements of the Rules and Regulations for the

Classification of Ships (hereinafter referred to as the Rules for Ships). This guide address two types of ship: Passenger ships (e.g. cruise ships, ro-ro ferries), and Cargo ships (e.g. container ships, tankers), and it provides for two alternatives:

- *Class Notations* which indicate that the ship has been assessed and complies with noise and vibration criteria in these Rules and that a periodic survey regime has been established for the lifetime of the ship.
- *Certificate of Compliance* which provides evidence that the ship has been assessed and found to comply with the noise and vibration criteria in these Rules.

For passenger ships the maximum noise levels requirements are indicated for different type rooms in function of the comfort classes (see Table 10). The maximum noise levels requirements are also indicated for the crew accommodation and work areas (Table 10). When the maximum noise level exceeds the specified criterion by 3 dB(A), or contains subjectively annoying low frequency or tonal components, the noise rating (NR) number is to be established in accordance with a specific graph (Figure 4).

Table 10: LR Maximum noise levels requirements.

Location	Maximum noise levels (dBA)		
	Class 1	Class 2	Class 3
Passenger cabins – Standard	49	52	55
Passenger cabins – Superior	45	47	50
Public spaces – Excluding shops	55	58	62
Public spaces – Shops	60	62	65
Medical center	50	55	60
Theatre / Auditorium	50	55	60
Open deck recreation areas (excluding swimming pools and similar)	67	72	72
Swimming pools and similar	70	75	75
Crew sleeping cabins and hospital	50	53	55-60
Offices, conference rooms and crew day cabins	55	58	60-65
Crew Mess rooms, lounges, reception areas and recreation rooms within accommodation	88	58	60-65
Crew alleyways, changing rooms, bathrooms, lockers	70	75	75

Note 1 – Levels may be exceeded by 5 dB(A) within 3 m of a ventilation inlet/outlet or machinery intake/uptake on open decks.
Note 2 – Levels may be exceeded by 3 dB(A) in accommodation above the propellers for three decks above the mooring deck.
Note 3 – Levels for open deck recreation areas refer to ship generated noise only. On open deck spaces the noise generated from the effects of wind and waves con be considered separately to limits agreed between the Builder and Owner and advised to LR for the trial conditions.

In the guide are also indicated the minimum apparent airborne sound insulation indexes (R'_w) for the partitions between occupied areas (e.g. cabins, public spaces, stairways, corridors, etc.) and reported in this document in Table 11, and the normalized impact sound pressure level ($L'_{n,w}$) for three categories of locations: locations below public spaces covered with soft materials, locations below public spaces covered with hard materials, and locations below sport rooms or dance floors (Table 12).

In this guide no requirements are provided in relation to the impact sound pressure levels.

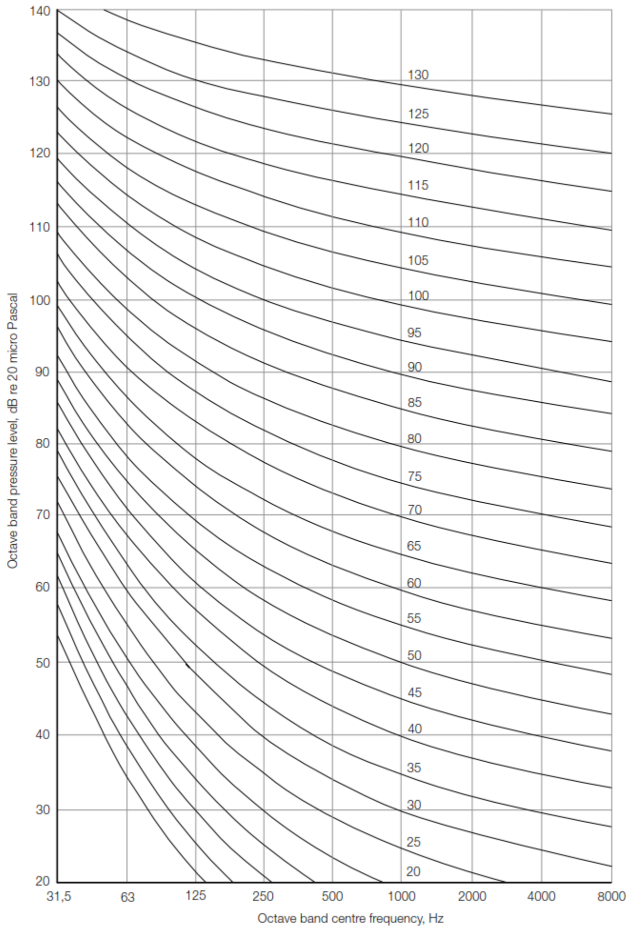


Figure 4: Noise rating curves.

Table 11: LR Maximum apparent airborne insulation requirements.

Location	Maximum apparent airborne insulation index R'_w (dBA)		
	Class 1	Class 2	Class 3
Passenger cabins – Standard	41	39	38
Passenger cabins – Superior	45	42	40
Cabin to corridor – Standard	38	36	35
Cabin to corridor – Superior	42	40	37
Cabin to stairway – Standard	47	45	43
Cabin to stairway – Superior	50	47	45
Cabin to public spaces – Standard	52	48	48
Cabin to public spaces – Superior	55	50	50
Discotheques to cabins	60	60	60
Discotheques to stairways and public spaces	52	52	42
Cabin to machinery rooms and engine casing	55	53	50

Table 12: LR Weighted normalized impact sound pressure levels requirements.

Weighted normalized impact sound pressure level	$\mathbf{L'_{n,w}}$ **(dBA)**
Location below public spaces covered with carpet soft materials	50
Location below public spaces covered with hard materials (wood, marble, etc.)	60
Location below sport rooms or dance floors	45

RINA in 2021 issued the "*Rules for the Classification of Ships*"; in "Part F: Additional Class Notations; Chapter 6: Comfort on board and in port area (COMF)"; Section 1: Comfort with regard to noise on board ships" limits for noise (in terms of A-weighted levels) and vibration (in terms of peak structural velocity, measured in mm/s from 0 to 100 Hz) at berth and in navigation are defined for various areas of a yacht [Registro Italiano Navale (RINA), 2021]. The requirements of this Rules define the limits of acceptability of noise on board, the methods for verification of compliance and the criteria for acceptance. They are based, as appropriate, on international standards and are deemed to preserve the general principles of such standards.

In this guide the spaces are divided into: crew spaces, passengers' spaces, and work spaces and each category of spaces correspond to different requirements, in particular for the work spaces there are no indicators for evaluating comfort on board, but indicators aimed at evaluating health and safety of workers.

For passenger spaces and crew spaces two level of maximum noise levels admitted are indicated (Table 13). The Comfort Class 1 must satisfy all Class 1 levels, the Comfort Class 2 must satisfy all Class 2 level, the Comfort Class 3 must satisfy different values according to the dimensions and the type of spaces; for example for passenger spaces the Class 3 must satisfy all Class 2 levels increased by 5 dB(A).

In these Rules are also indicated the Maximum apparent airborne insulation requirements (R'_w) for different inner partitions (Table 14) the normalized impact sound pressure level ($L'_{n,w}$) for three categories of locations: locations below public spaces covered with soft materials, locations below public spaces covered with hard materials, and locations below sport rooms or dance floors (Table 15).

3.3 Enhanced indicators of acoustic comfort

At present, as regards the assessment of comfort, the rules mostly refer to the sound pressure levels (dBA). In this sense, the Comfort Class rules do not substantially differ from requirements for the safeguard of crew health in the Noise Code for commercial ships. However, sound insulation between cabins, impact noise from upper decks, speech interference levels or general noise-rating curves are also taken into account.

In [Badino et al., 2011] the authors stated that the A-weighted Sound Pressure Levels and the exposure time does not necessarily correlate well with the annoyance caused by the noise and that other features such as the spectral content or the repetition over time, assume importance and should be considered in the regulations.

Table 13: RINA Maximum noise levels requirements.

Type of space	Maximum noise levels (dBA)	
	Class 1	Class 2
Passenger spaces		
Suite or mini-suite	45	50
Standard cabins	50	55
Enclosed spaces where the noise level is normally high, or where passengers are not normally expected to stay long	55	60
Enclosed spaces where the noise level is normally low	52	57
Spaces where passengers are normally expected to stay for a short period of time (e.g. corridors, stairs, etc.)	60	65
Outside spaces for prolonged and/or recreational stay of passengers (e.g. swimming pool area, open walkways, sun decks, etc.)	65	70
Crew spaces		
Crew cabins	55	60
Senior office cabins	52	55
Navigation spaces / Radar room	65	65
Radio room	60	60
Look-out posts, navigating bridge wings and windows	70	70
Hospital	55	60
Public crew spaces	60	65
Work spaces without equipment operating	70	75
Workshops other than those forming part of machinery	85	85
Offices	60	65
Mess room / recreation room	60	65
Engine control room	75	75
Crew open deck	75	75

Table 14: RINA Maximum apparent airborne insulation requirements.

Division between:	Noise insulation index R'_w (dBA)		
	Class 1	Class 2	Class 3
Suite or minor suite – cabin	43	40	35
Standard cabin – cabin	40	37	35
Disco – cabin	65	60	55
Enclosed spaces where the noise level is normally high – cabin	60	55	50
Enclosed spaces where the noise level is not normally high – cabin	55	50	45
Enclosed spaces where the noise level is normally low – cabin	50	45	40
Spaces where passengers are not normally expected to stay long – cabin	50	45	40
Corridor – suite	40	35	33
Corridor – cabin	37	33	30
Crew cabin – accommodation spaces	45	45	42
Crew cabin – crew cabin	35	35	32
Crew cabin – corridor	30	30	27

Table 15: RINA Weighted normalized impact sound pressure levels requirements.

Weighted normalized impact sound pressure level	$\mathbf{L'_{n,w}}$ **(dBA)**
Location below public spaces covered with carpet soft materials	50
Location below public spaces covered with hard materials (wood, marble, etc.)	60
Location below sport rooms or dance floors	45

In [Borelli et al., 2015a] the authors stated that the comparison of noise spectra in different operating condition of the ship gives relevant information concerning their influence on acoustic comport onboard. Therefore, the sound rating criteria provided by the American Society of Heating, Refrigerating and Air-conditioning Engineers (ASHRAE) to rate indoor background sound in the civil engineering context (Noise Criteria (NC), the Room Criteria (RC) and Balanced Noise Criteria (NCB), the Room Noise Criteria (RNC) and the RC Mark II) as new indicators to be applied in the naval field [Badino et al., 2011; Borelli et al., 2021]. The Room Criteria Mark II, in particular, allows the estimating of the occupant satisfaction and reaction through an indicator known as the Quality Assessment Index. Another index proposed in [Borelli et al., 2021] is the Speech Transmission Index for Public Address Systems (STIPA) that would be of great interest especially for the investigation of comprehension of Emergency Voice/Alarm & Communication Systems that must always be heard and understood correctly.

In [Borelli et al., 2021] it is stated that the current normative framework and regulations about noise on board ships, based on db (A) limits only, could be revised to take into account spectral components, whose effect would otherwise be neglected. In [Borelli et al., 2021] it is suggested that in the future new more sophisticated indicators would be effective in the shipping sector in order to have a better assessment of noise onboard and to guide the efforts to improve the onboard soundscape.

In order to satisfy the increasing requirements of passenger comfort, psychoacoustic models, successfully applied in other industrial sectors, could be useful tools to analyze and improve the product-sound quality and the evaluation of auditory perception on board. First studies dealing with acoustic comfort in marine vehicles appear in the literature only in 2003 [Volle et al., 2003], although many previous studies analyzed the interior noise problem in other vehicles. In [Volle et al., 2003; Goujard et al., 2005] the acoustic comfort is identified as main factor of comfort on board through a subjective questionnaire with a psychoacoustic approach. In [Seiler and Holbach, 2013], in order to find better indicators for the quantification of acoustic performance of luxury vessels, audio material was acquired at a sea trial and evaluated with the help of a psychoacoustic paired-comparison listening test.

4 Analysis methods

Predictive analysis methods, correlated with the ship layout and validated through experimental measurements, could allow the ship's layout revision with the identification of alternative solutions and definition of design procedures.

Due to the complexity of ship structures, a rigorous classical approach (e.g. the wave theory) is impractical [Tso and Hansen, 1997]. In the 70s, Nilsson [Nilsson, 1977, 1978] presented a *simplified analytical method* based on a grillage model which was made up of two parallel hull frames and the associated plate elements; some assumptions of his model and the fact that it is essentially a two-dimensional model make it not readily applicable to the general analysis of vibration transmission in ships [Tso and Hansen, 1997].

A number of *empirical methods* (i.e. empirical formulas mainly based on experimental measurements and statistical data taken on board merchant ships) have been also reported for ship noise predictions [Plunt, 1980]. Empirical methods are valuable tools in the analysis and design of a generic type of ship, when limited information are available, while become less suitable in situations where a detailed analysis on different types of ships is required [Tso and Hansen, 1997].

Transfer Path Analysis (TPA), become widely used in recent years for road vehicle applications, was applied to noise reduction in ships in [Verheij, 1982]. Although theory on mount stiffness had been around already for years, Verheij was one of the first to successfully determine interface forces and moments by experiment. Although attractive from an academical point of view, practical engineering called for less elaborate force determination methods [van der Seijs et al., 2016]. TPA is a test-based procedure which allows to trace the flow of vibro-acoustic energy from a source, through a set of known structure- and air-borne pathways, to a given receiver location.

Recently, a new *Operational TPA* (OTPA) for identifying ship noise sources was proposed from [Cao et al., 2013], and [Keizen and de Klerk, 2015]. In 1999, Janssens et al. proposed a similar method, a *multi-dimensional substitution source method*, by which structure-borne sound transmission paths on board a ship can be quantified in practical situations where it is not possible to dismount the paths [Janssens et al., 1999].

Intuitively, engineers are prone to consider noise as an extension of the low-frequency vibration (e.g. main engine and propeller induced vibration with excitation frequencies around 5 to 10 Hz) and try to analyze structure-borne noise (at frequencies in the range 16 to 20 000 Hz and, more often, above 1000 Hz) using the same methods [Parunov et al., 2012]. While the *Finite Element Analysis/Method* (FEA/FEM) is typically restricted to vibration studies at low frequencies [Boote et al., 2013; Fischer et al., 2012; Insel et al., 2007; Yucel and Arpaci, 2010, 2013; Zhang et al., 2013; Alaimo et al., 2012; Lee et al., 2011], for structure-borne noise simulation in large structures like ships, the traditional low-frequency FEA involves a dense mesh with a consequent high computational burden.

Other methods based on the vibration energy propagation, i.e. *Statistical Energy Analysis* (SEA) [Boroditsky et al., 2007; Cabos and Matthies, 2000; Chang et al., 2006; Fischer et al., 2012; Insel et al., 2007; Ma and Li, 2003; Ming et al., 2021; Plunt, 1980, 1999; Tso and Hansen, 1997; Wang et al., 2007; Weryk, 2012], from 60s, and, in the last two decades, energy finite element analysis/method (EFEA/EFEM) [Gilroy et al., 2005; Parunov et al., 2012; Vlahopoulos, 2011; Vlahopoulos et al., 1999; Vlahopoulos and Wu, 2010], are nowadays applied for structure borne noise predictions [Parunov et al., 2012].

The basic idea of SEA is to divide a complex structure into a number of coupled subsystems and model the energy flow between them in the spirit of the transport theory; energy balance equations are then set up for these subsystems in terms of their spatially averaged vibration levels, the rate of energy dissipation, exchange and input due to external forces [Parunov et al., 2012]. The ability to accurately predict noise levels in a multitude of compartments with contributions transmitted over the airborne and structure borne path from every significant source has a great potential for noise control [Fischer and Bahtiarian, 2017]. SEA has a long tradition and, thus, commercial codes based on SEA are available.

The EFEA is a new approach for simulating high frequency vibration of large-scale structures. It is based on deriving governing differential equations with respect to energy density variables, and utilizing a FEA for solving them numerically; the main advantage of the EFEA is the potential of modeling the ship structure by relatively coarse mesh of finite elements.

An extended review considering the three methods FEA, SEA and EFEA for structure-borne noise prediction was presented in 2012 by Parunov et al. In such paper the authors conclude that energy methods are preferred for the analysis of the noise propagation problems compared to the conventional FEA and, in details, the EFEA represents an emerging technology, capable of improving some drawbacks of the more traditional SEA. These methods are frequently coupled to predict the vibro-acoustic behavior of a structure [Fischer et al., 2012]; [Insel et al., 2007].

The *Boundary Element* (BEM) and the *Computational Fluid Dynamics* (CFD) methods were used in combination with the FEA in [Xu et al., 2002] and in [Ma and Mahfuz, 2009], respectively, in a model of a whole ship; while in [Xu et al., 2017] the *Boundary Element* (BEM) was used to predict to noise but modelling only a ship cabin.

4.1 Large ships

Some recent literature studies focusing on large ships (i.e., surface ships, passenger cruise ships, hydrographic echo sweeping vessels and timber container carriers) are briefly described and classified on the base of the analysis method in the following subsections.

4.1.1 FEA

In [Zhang et al., 2013], using the acoustic-structure coupling method embedded in ABAQUS®, the results of numerical analysis of aluminum or steel plate circular/rectangular plate subjected to underwater shock are obtained and compared with experimental data (Figure 5). Moreover, the FEA model of a full-scale aluminum 55 m ship is established with a mean element size of 0.5 m, resulting in 45.000 shell and beam elements in the structural domain and 260.000 acoustic hexahedral elements in the flow field model. Analysis on dynamic responses of the ship subjected to underwater shock is conducted. Responses of time history are presented, such as acceleration, velocity and displacement, and plastic zone and curves of plastic deformation.

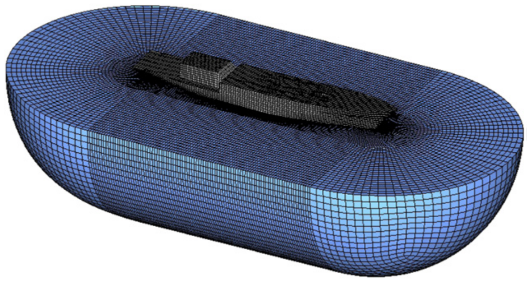


Figure 5: Whole finite element model of the fluid and a surface ship [Zhang et al., 2013].

In [Yucel and Arpaci, 2010, 2013], a three-dimensional FEA of an entire passenger ship hull, including the deckhouse and machinery propulsion system, has been developed for local and global vibration analyses. Vibration analysis has been studied under two conditions which are free-free (dry) and in-water (wet). Wet analysis has been implemented using acoustic elements. As the result of global ship free vibration analysis, global natural frequencies and mode shapes have been determined. Moreover, the responses of local ship structures have been determined as the result of the propeller-induced forced vibration analysis.

4.1.2 FEA/SEA

In [Fischer et al., 2012] a FEA, in terms of identification of hull natural frequencies and forced response, was adopted as vibration analysis of a ship, while a SEA analysis was applied to noise prediction in order to deliver an acoustically quiet vessel.

Once a SEA model is created, it can be easily used to consider trade-offs between different noise-control approaches. This is a large time savings and has accuracy advantage over most other analysis techniques. Figure 6 is a screen image from a ship-only noise-modeling program developed for use by the US Navy [Fischer and Bahtiarian, 2017].

In order to improve the vibration and noise calculation accuracy of SEA method in middle-frequency region, [Ming et al., 2021] proposed the Improved Statistical energy method. In this method, the equivalent coupling loss factors (ECLFs) between ship sub-structures were calculated based on the energy flow method (EFM), and the ECLFs

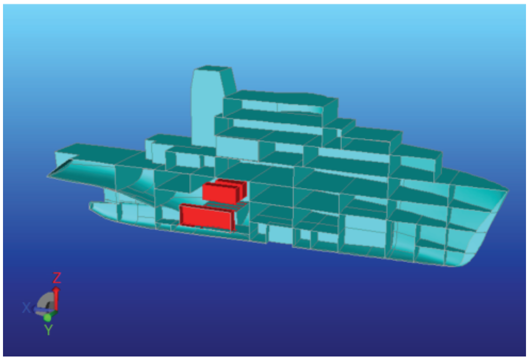


Figure 6: Cut-away view of a "Designer NOISE®" acoustic model [Fischer and Bahtiarian, 2017].

were substituted into the statistics energy balance equation (Figure 7). The obtained results shown that the predicted values of the cabin noise were in good agreement with those obtained by measurements, and the application of the statistical energy method can be extended in the ship's middle frequency cabin noise prediction.

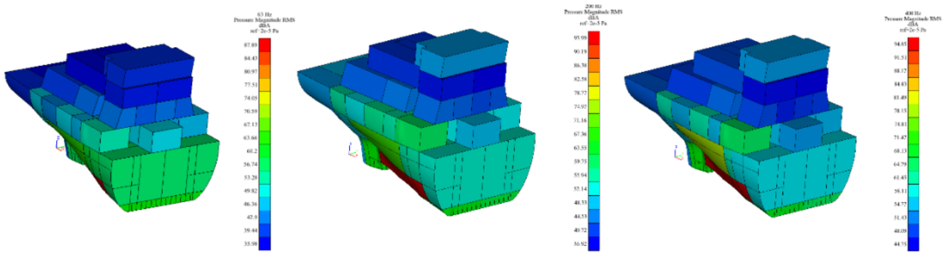


Figure 7: Sound pressure cloud maps of the tank cabins: a) 63 Hz; b) 200 Hz; c) 400 Hz [Ming et al., 2021].

4.1.3 EFEA

In [Vlahopoulos and Wu, 2010] an EFEA formulation was developed for computing the vibration and the associated radiated noise of two 1/8th scale models of a double/conventional hull design of a surface ship in the 200 to 4000 Hz frequency range. The EFEA model consisted of 1800 elements, approximately 1500 representing the structure, and 300 representing the fluid volume enclosed between the two hulls. Validation case studies are presented with correlation between computations and scaled model testing; a comparison between the energy density predicted by the EFEA model and that from the postprocessed test data is presented; as far as the experimental activity is concerned, the models were placed in a large water-filled tank and tested in two conditions: with the enclosed tanks both empty and filled with water. Provided the vibration input with electromagnetic shakers inside the hull, the resulting vibration response of the hull exterior was mapped by a scanning laser vibrometer, and the radiated SPL was measured by a single hydrophone.

4.1.4 FEA/BEM

In [Xu et al., 2002], for a whole double-shell ship, a theoretical FEA/BEM analysis of the structural vibration, fluid-structure interaction dynamics, and the exterior field pressure radiation has been summarized. A whole ship is modeled by the FEA software ANSYS®. The normal sound intensity of the outer hull is calculated and analyzed by the BEM software SYSNOISE®for predicting coupled vibro-acoustic behavior of a real naval ship.

4.1.5 FEA/CFD

In [Ma and Mahfuz, 2009] the authors present a structural analysis of composite sandwich hulls of a surface effect ship using fluid structure interaction (FSI), by coupling FEA and computational fluid dynamics (CFD). The numerical model of a multi-hull ship structure in this paper consists of FE structural model and CFD model. The FE model includes a global hull structure while the CFD model includes a fluid domain and a wave motion setup. A multidisciplinary linkage connects them for fluid structure interaction (FSI). The FSI analysis is implemented through commercial FEA code ANSYS®and CFD code CFX. The sandwich design scheme with four layers with individual thickness and material properties for the structure is illustrated and imported into ANSYS (Figure 8).

The study described by the same authors in [Ma et al., 2014] focuses on predicting seakeeping loads and on how loads are transferred to 3D FE models.

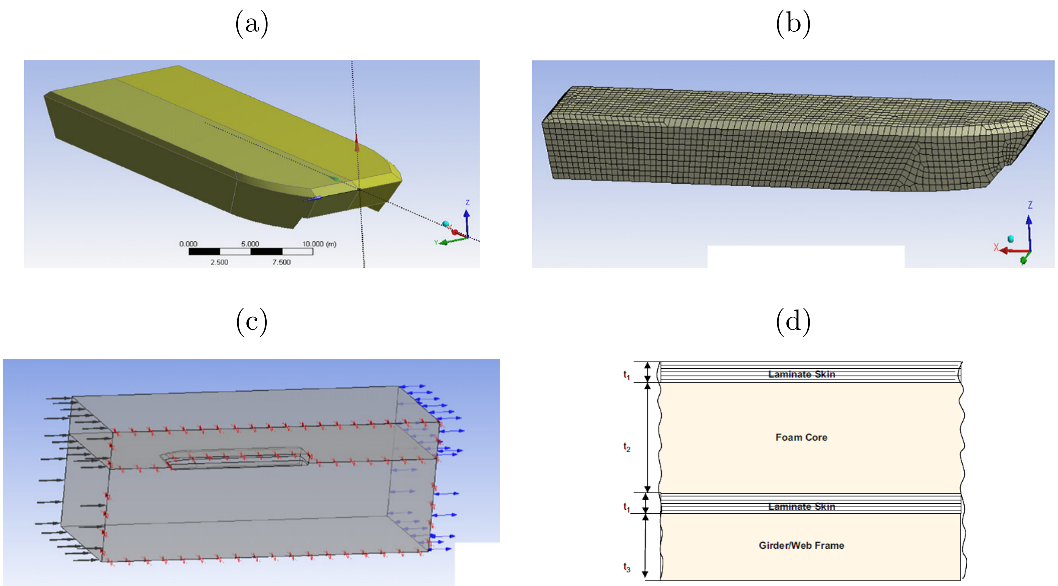


Figure 8: (a) Finite element model for multihull ship, (b) half-ship model, (c) the fluid domain in CFD model and (d) the cross-sectional view of a sandwich plate from [Ma and Mahfuz, 2009].

4.2 Recreational boats

Concerning the vessel's type, in the recent literature, the FEA and SEA were often applied specifically for predicting the vibration and noise situation, respectively, of fast boats, yacht and super-yachts.

4.2.1 FEA

The Ansys® modal and transient FEAs were applied in [Boote et al., 2013] to a 60 m long and 9.5 m wide three-decks steel super-yacht, made available by Azimut-Benetti shipyard, in order to investigate the dynamic behavior of the steel hull and aluminum superstructures. An Ansys® SHELL63 and BEAM44 element mesh was adopted (300 mm average panel diagonal), while masses and loads were modeled as SURF154 elements (Figure 9). Once computed the natural frequencies of vibration, they were compared with those measured on the same vessel during different construction phases (i.e. on the separate and assembled decks without outfitting). The structural response to the propeller exciting force at 10 Hz and 33 Hz, modeled as a pulsating sinusoidal pressure, was investigated, by varying the intensity of the exciting force and the structure damping. The structural response (in terms of 0.2 s time history displacement, velocity and acceleration) was monitored in 5 points, referring to the classification society rules.

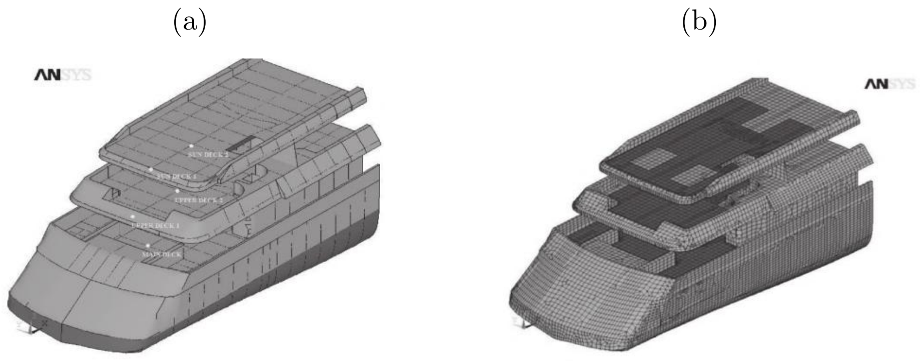


Figure 9: Model (a) geometry and (b) mesh of the stern part of a yacht from [Boote et al., 2013].

In [Alaimo et al., 2012] a structural FEA in Patran/Nastran™ of a 50 ft pleasure vessel is presented. Modal analyses to find the natural frequencies of the vessel and structural analyses to verify the strength of the vessel to design loads, computed according to RINA rules for pleasure vessels, are reported. Two different composites are used (a monolithic sequence of short fiber and balanced glass lamina and a sandwich made of glass fiber composite skins and a PVC core).

In [Lee et al., 2011], by using the commercial FEA software ABAQUS®, the response of a FRP yacht including whole boat structures (e.g. hull, bulkheads and stiffeners), supported by cradles, is compared with the measured data to validate.

4.2.2 FEA/SEA

A FEA and empirical methods were applied in [Insel et al., 2007] for vibration analysis of 30 to 70 m long motor-yachts, while SEA and empirical methods were selected for noise investigations.

The goal of [Chang et al., 2006] is to predict cabin noise level in a high-performance yacht, a fiberglass reinforced plastic (FRP) fast boat, using SEA. Twelve FRP test specimens were built with different lamination scheme according to their locations on the target boat, and vibration tests were performed to obtain the Damping Loss Factors and Coupling Loss Factors. These data were then put into the SEA model to predict the cabin noise under different operating conditions. A comparison of predicted noise levels with measured data validated SEA application.

4.2.3 SEA

In [Plunt, 1999] the authors applied SEA (LMS SEADS software) for the prediction of airborne and structure-borne transmission of noise to the cabins and saloons of a 16 to 18 m long luxury motor-yacht, built with glassfibre reinforced plastic (GRP) hulls including GRP sandwich sections. A SEA model with 150 elements and 400 subsystems was adopted.

In [Blanchet and Caillet, 2014], the authors propose a ways to improve the internal sound insulation using SEA model on a 70 m luxury yacht (Figure 10), and then they introduced ad advance d method for predict full frequency vibro-acoustic responses by coupling SEA and FEM.

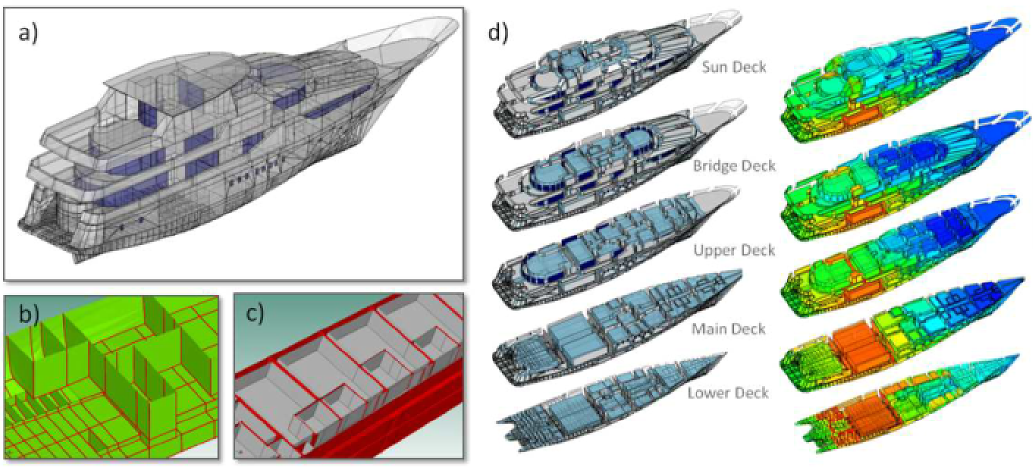


Figure 10: SEA model of a 70 m luxury yacht from [Blanchet and Caillet, 2014]: a) SEA model built automatically from 2D drawings, b) structural point and line junctions (in red) between structural panels (in green) automatically created when node connectivity is enforced, c) area junctions (in red) automatically created between panels and acoustic cavities (in grey), d) images of different decks (left) and contour plot of panel velocity and cavity SPL.

The measurements comprised airborne sound spectra in the compartments and vibration measurements in various regions at different cruising speeds as well as at harbor idle. Typical predicted A-weighted SPLs for different speed conditions were shown. The simulation of different reasonable noise reducing modifications to the hull design and the airborne sound isolation of the partitions were also investigated.

4.3 Geometrical room acoustics modelling

Given the overlapping of the acoustic comfort evaluation parameters in boats with the parameters typically used in building acoustics, in this subsection the types of simulation software used in the civil sector are briefly described.

These software use geometric acoustics, which is based on the hypothesis that the wavelength is negligible compared to the size of the environment and the objects in it. In this field the concept of sound wave is replaced by that of sound ray. A sound ray can be defined as a small portion of a spherical wave that originates at a point and is characterized by a defined direction of propagation. The total energy carried by a ray is constant (in the hypothesis of a non-dissipative medium). However, the sound intensity of a diverging beam of rays decays as $1/r^2$. The reflection of the rays is modeled, but not the refraction nor the phenomenon of curvature of the rays in a non-homogeneous medium. The propagation velocity is however considered finite, as phenomena such as reverberation, the presence of echoes, etc. depend on it. Diffraction is usually neglected, as is interference; in fact, the phase relationships between the components of the field are not considered , which are then considered inconsistent, and the energies simply add up.

The main calculation methods in geometric acoustics are:

- Image source method;
- Raytracing
- Hybrid methods.

4.3.1 Image source method

In the context of geometric approximation, if the boundary surfaces of the environment meet the conditions for specular reflection, any reflected wave can be thought of as coming from the virtual image of the real source. Therefore, it can be considered in turn as a direct wave, originating from a fictitious source coinciding with the virtual image placed behind the surface responsible for the reflection. This mechanism can be extended without difficulty also to reflections subsequent to the first: it is sufficient to consider the virtual source of the previous reflection as a real source.

The Mirror Image Sources Method (MISM) is based on this interpretation of the phenomenon of wave reflections, or rather of sound rays, whose basic hypotheses are the following:

1. Validity of the geometric acoustics hypothesis;
2. Mirror reflections on the walls;
3. An image source is associated with each specular reflection;
4. Each source emits spherical wave fronts (only in the case of small sources compared to the wavelength);
5. The propagation of the wave fronts is represented by rays;
6. The intensity that reaches a receiver is equal to that emitted by the source, real or virtual, decreased by effect: of the geometric divergence (proportional to $1/r^2$); the absorption of the α walls; of the attenuation of sound in the air.

In practice, for the search of the virtual sources (hypothesis 1, 2 and 3) the boundary surfaces of the environment are schematized through a finite number of flat surfaces, generically called "walls", identified by the coordinates of their vertices and by the succession of the themselves, which determines the normal to the wall oriented towards the inside of the room.

The geometric construction of the images continues until the order of reflection has reached a predetermined maximum value L0 or the sound intensity at the nearest receiving point does not exceed a minimum predetermined value Iε. The calculation algorithm for each source, each receiver, each surface and each order of reflection must verify that:

- The source and receiver are internal to the environment;
- The reflection point belongs to the reflection surface (first criterion of visibility);
- The ray is not interrupted by a surface not affected by the reflection (second criterion of visibility).

The information that can be obtained in this way is theoretically exhaustive, since in the environment the following can be calculated: the total sound energy density in steady state, the sound level as a function of position and time, the one-way sound intensities as a function of position and time, the delay times and the directions of arrival of the individual reflections. The main limitation of this method is the control of visibility (or "audibility") of the sources.

4.3.2 Ray Tracing

This method takes its name from the schematization adopted with regard to the propagation of sound energy: instead of dispersing on spherical wave fronts, as in the case of virtual sources, the sound energy is allowed to propagate in fractionated space along rectilinear trajectories or rays sound. The point of the first reflection is identified by calculating the intersection of the trajectory with the planes that contain the walls and selecting the closest, if this intersection belongs to a wall, then it is the reflection point. After the first reflection the particle continues according to its new trajectory until the next wall. The reflection can be specular or diffuse: in the first case the law of geometric reflection is applied, in the second a probabilistic distribution of the reflected particles is introduced. The sound absorption of the walls is represented in two ways: (a) the energy transported by the particle is reduced by a factor of $1 - \alpha$ with each reflection; (b) use a as an absorption probability. When the energy of the particle falls below a predetermined threshold or when the particle is absorbed, another particle is considered.

The basic assumptions are the following:

1. the approximations of geometrical acoustics are valid;
2. the sound is mirrored on the boundary surfaces;
3. the sound energy of the source is quantized in a finite number of packets associated with sound rays, also called sound particles 0 (improperly) phonons;
4. starting from the position of the source, the sound rays propagate in all directions according to the laws of geometric acoustics;
5. sound rays have an ideally infinitesimal and constant section;
6. the geometric divergence of the emitted sound energy is represented by the geometric divergence of the set of sound rays;
7. The sound rays lose energy due to the absorption of the impacted boundary surfaces and the attenuation of sound in the air;
8. in reception, the sound energy associated with the different rays can be added together.

Each source is characterized by the sound power emitted and the directivity factor. In Ray Tracing you choose the number of rays, the power and direction associated with each of them. The rays are generated in two different modes: deterministic and statistical.

In the deterministic mode, placed a unit sphere in the position of the source, the emission directions are identified by the position vectors of points belonging to the unit sphere. The vectors are chosen according to a geometric partition rule, which can also take into account the directivity factor of the source.

In the statistic mode, the directional vectors are oriented on the basis of a pair of random numbers in order to ensure uniform coverage, in a statistical sense, of the surface of the unitary sphere.

The result is collected through predefined surface or volume “counters”. When a particle passes through one of these counters, its energy and travel time (and possibly the direction it came from) are stored.

Ray Tracing is a direct, statistical method and converges as the number of rays increases. The algorithm requires that the following checks be carried out: the rays must hit the inner face of the walls; the rays must travel in front of the walls, the reflection points must belong to the walls, each ray must follow the minimum path.

The computational cost of Ray Tracing increases linearly with the number of rays and their duration. This means that there is a constant relationship between the length of the echograms and the calculation time. The most expensive part of the method is the representation of the geometric and physical characteristics (dimensions of the room, positions of sources and receivers, reflectivity characteristics of the walls). Finally, it is possible to represent the diffusion of the sound field by inserting a statistical distribution of the angle of reflection of the beam.

The main limitation of ray tracing is the lack of a rule for choosing the number of beams and the size of the receiver. In particular, the size of the receivers is a critical factor and causes systematic errors of two types.

– Multiple uptake errors: the number of beams picked up depends on the reciprocal position of source and receiver.

– Variable or invalid collection errors: a small displacement of the receiver causes a significant variation in the energy received.

4.3.3 Hybrid methods

The limitations of image source methods and raytracing methods have led to the development of hybrid methods, which combine their best features. Some acoustic simulation software using a hybrid method introduces the concept of *secondary source*. In this method, the low-order reflections (early reflections) are modeled using the image source method, while the tail of the reverberation (due to later reflections or late reflections) by secondary sources positioned on the walls in the points of the last reflections. The secondary sources are to all intents and purposes the new sources.

Another hybrid method is the so-called *Cone tracing*. In this method, a cone having one of the rays as its axis is substituted for a beam of rays coming out of the source. After each reflection the vertex of the cone coincides with a virtual source. If a receiving point belongs to the cone between two successive reflections, then the corresponding image source is potentially visible and its contribution to the total sound field is considered. Cone tracing is essentially based on the model of virtual sources, but has the typical speed of ray tracing and eliminates the uncertainties due to the statistical character of the rays. The cones are partially overlapped, without leaving empty spaces, with the risk, however, of considering the same source-receiver path several times (need for checks). In practice, only the first reflections are used. The cone method also suffers from capture errors, which are essentially due to the abrupt transition between areas of visibility and shadow areas.

The *beam tracing* method weighs the energy associated with the front of the cone according to a certain distribution function, thus solving the problems of overlapping between the cones and multiple identifications. Typically a Gaussian type distribution is used.

Other methods are based on the generation of beams starting from suitable partitions of a spherical surface (for example the *Pyramid tracing*).

Appropriate strategies are developed to model the tail of the impulse response, for which the contributions of the single beams are no longer distinguishable. In this case, the purely geometric approach is no longer sufficient. A possible solution uses beam tracing for early reflections, while the tail is evaluated with a statistic model. The transition between the two methods occurs according to the order of reflection.

The problem of including diffusion in beam tracing methods is still not completely solved. Ideally, in correspondence with the reflection, each beam should generate a set of secondary beams (split up), distributed according to the statistical model adopted for the diffusion. However, this involves an exponential increase in the calculation time. There are various solutions based for example on the discretization of the reflecting surface or on adaptive beam tracing mechanisms.

4.4 Ship building material analysis

Some studies of the recent literature focus on innovative ship building materials, and most papers deal with the applications of composite sandwich construction which are widely used due to their superior performance at weight saving, corrosion resistance and high stiffness [Ma and Mahfuz, 2009].

The prediction of shipboard noise in Fiber reinforced plastic (FRP) ship is more difficult than in boat with steel or aluminum structures, since there is a variety of combination of fiber sheet, resin, and core material in use to form the major hull and cabin structures in FRP boats [Chang et al., 2006]. An in-depth analysis on these innovative materials applied to marine vehicles is reported in the present subsection.

In [Gaiotti and Rizzo, 2012a] FEA modeling of rectangular specimens of sandwich composite laminates under compressive loading is proposed. The core of the specimens is modeled via brick solid elements, while the skins by layered shell elements (Figure 11).

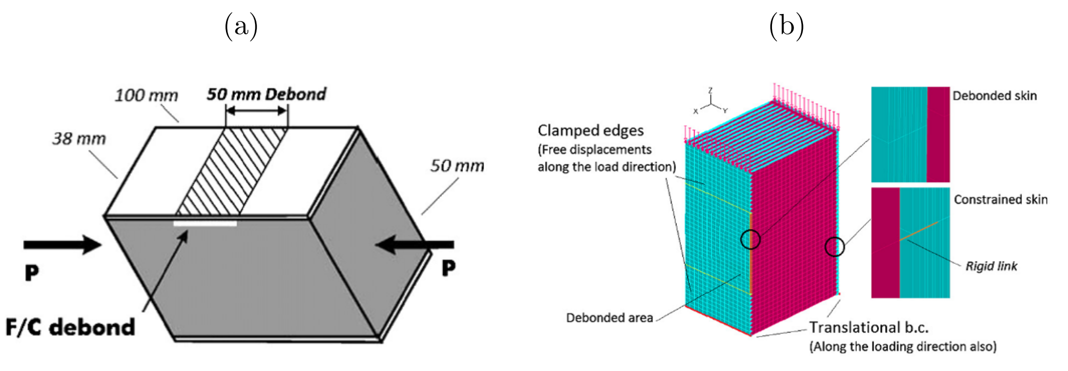


Figure 11: (a) Sandwich specimen geometry and (b) FE model from [Gaiotti and Rizzo, 2012a].

The authors state that simulation of sandwich panels of larger and more geometrically complex structures (e.g. hull of pleasure crafts) is crucial since pleasure craft industry is now facing the challenge of thinner and thinner skin laminates and can be easily implemented in FEA. The results of a test case were compared with experimental and other numerical estimates. The same authors in 2012 [Gaiotti and Rizzo, 2012b] investigated the buckling strength of typical ship stiffened panels built in fiberglass composite laminates. The results obtained applying analytical formulations based on equilibrium and potential energy equations were compared with the ones obtained by linearized eigenvalue analyses of rather detailed FEA models. A sensitivity analysis were also presented.

The work of [Balsamo et al., 2008] focuses on the sound transmission properties (in terms of transmission loss, TL) of glass reinforced plastic (GRP) structures, widely used in yachts. The predictions performed by a FEA based procedure at low-medium frequencies and by energy based models at higher frequencies were compared with the results of an experimental test campaign in a small acoustic facility on a set of GRP panels.

Analytical models based on classical lamination theory, FEA, ship motions, probability and wind and wave mechanics were used in [Miller, 200] to predict hull laminate strains, and fatigue tests were used to determine S-N residual stiffness properties of coupons. These predictions and test data were compared against two cored fiberglass sisterships having significantly different fatigue histories and undamaged laminates representing a new vessel.

In [Kumar Satish and Mukhopadhyay, 2002] a new ship structural analysis software named "ASSA" was developed for the 3D FEA of FRP boats and ships using a new stiffened plate element. The analysis of a rectangular box shaped vessel was also carried out and results compared with a general purpose FEA software (NISA). A high speed FRP patrol boat was analyzed.

The application of fiber-reinforced polymer composites to naval ships is reviewed in [Mouritz et al., 2001]. In [Son et al., 1999] vibration experiments of a FRP sandwich plate and structure are performed. The authors compared the experimental data from vibration tests and simulation results with analytical solutions. In [Shenoi and Hawkins, 1992] the design of tee connections in single skin FRP ships and boats is investigated. FEA models are developed to highlight significant influences of geometry and material variations on the performance of the tee joints. Also, in [Dodkins et al., 1994] the authors examine the problems of forming efficient joints between the major structural components of fiber-reinforced plastic (FRP) ships/boats. The development of FE numerical models to assess failure modes is discussed.

4.5 Analysis of the exhaust system

A FEA of a ship exhaust system is presented in [Martins et al., 2009a] in order to perform a failure analysis. Design changes to the original exhaust system were also analyzed (Figure 12).

The types of finite elements used were: mass – MASS 21, spring – COMBIN14, elastic foundation, as well as structural biquadratic elements – SHELL93 – and thermal – SHELL57, that allowed membrane, shear and bending loading. The flexible supports of the exhaust system were simulated through elastic foundations, along the longitudinal/compression direction, and springs, along the rolling and shear directions. The exhaust system is connected to the ship structure by several wire rope isolators —- WR20-200-08 —- that have different responses in compression, shear and roll; they are specifically applied to protect the structures from shocks and vibrations, dissipating this energy by wire friction. The wire rope isolators shows a nonlinear response for compression, shear and roll inputs and support the load indicated for low frequencies ($< 4\,\mathrm{Hz}$). When higher frequencies of loading are applied, the load support capacity lowers very quickly. Several elastic foundation areas at the support rings were defined in order to simulate the effect of the springs in compression. In the other two directions – shear and roll – a spring element type, with different rigidity values, was used. Besides the elements described, mass type elements were used in the modal analyses to simulate the weight of each of the three silencers existing inside the exhaust system. As the thickness to radius ratio (t/R), in the different cross sections, was much smaller than 0.05 (1/20), the walls of the exhaust system were modeled as shells, for the structural, thermal, transient and modal analysis performed. The thermal expansion coefficient and the thermal conductivity of the material changed with temperature, making the thermal analyses performed of nonlinear type.

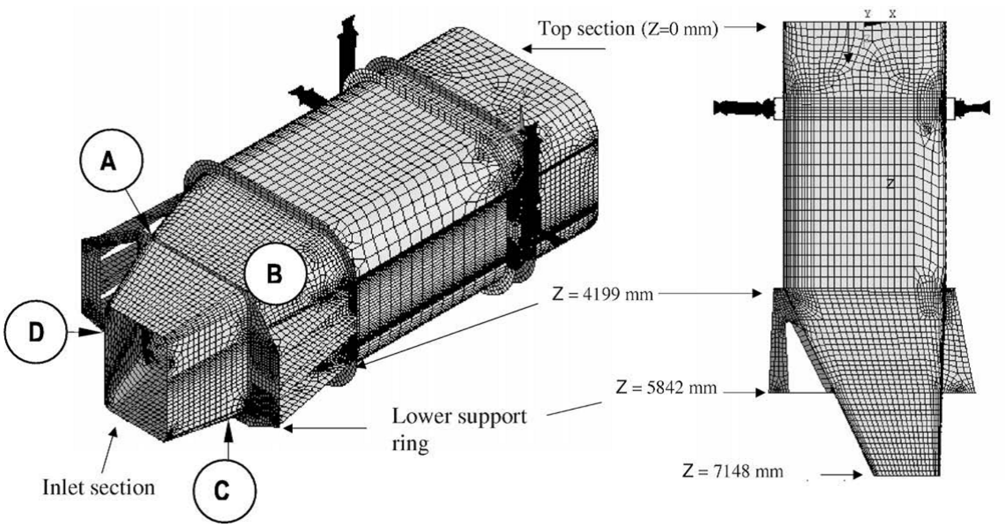


Figure 12: Overall view of the finite element mesh of the exhaust system in [Martins et al., 2009a].

The assessments of a muffler's optimal shape design that would simultaneously overcome a broadband noise hybridized with multiple tones within a constrained machine room is addressed in [Chiu, 2013]. In order to promote the best acoustical performance in mufflers, five kinds of hybrid mufflers were examined, using a simulated annealing method (Figure 13).

(a)

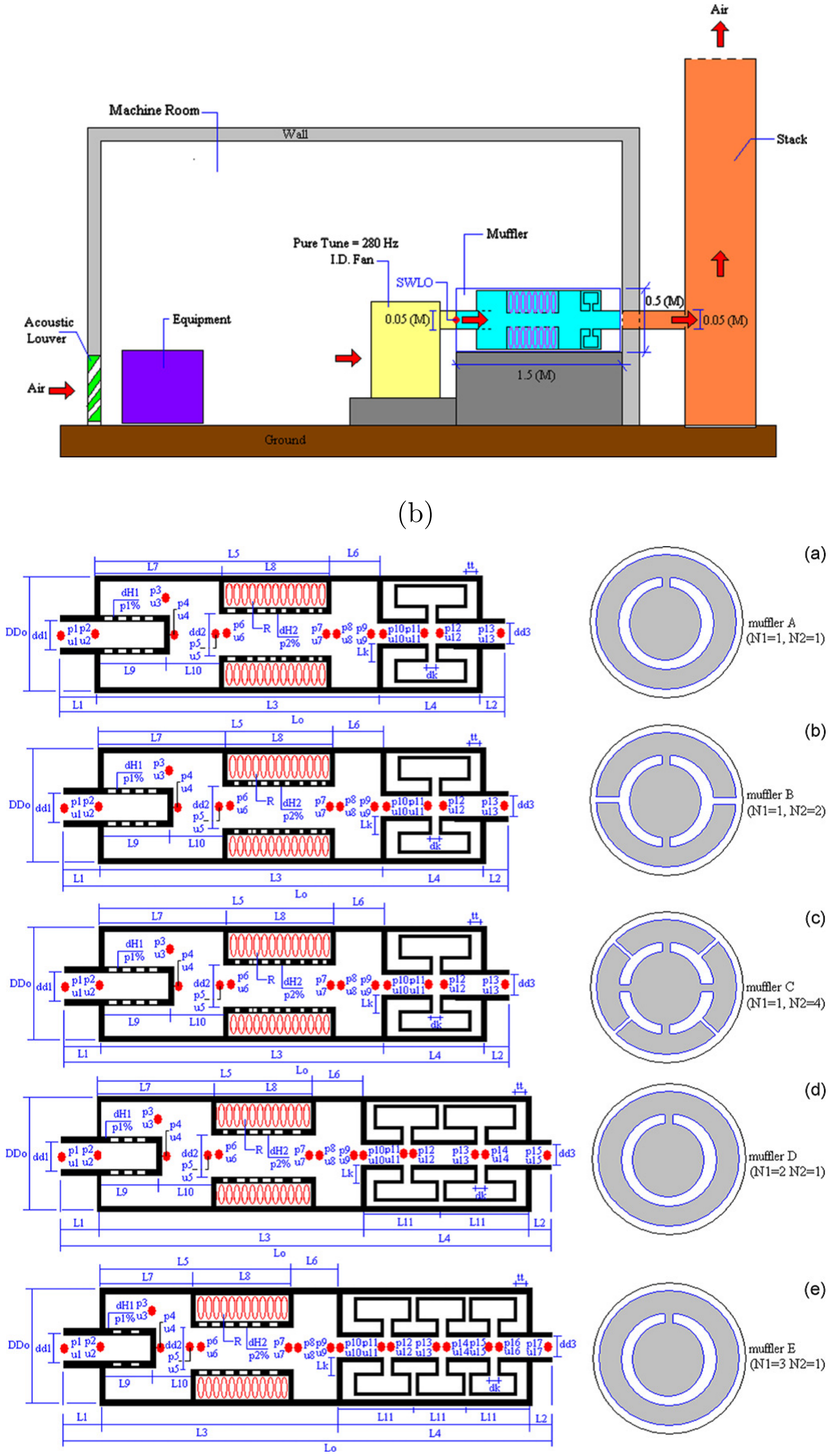


Figure 13: (a) Example of machine room and (b) five kinds of mufflers forms [Chiu, 2013].

In Bayraktar et al. [2007] a CFD and an analytical flow and heat transfer analysis of exhaust system of a gas turbine used in a ship is presented. The results consist in velocity vectors, temperature and pressure fields and pressure losses.

The work of [Wong and Wang, 2003] describes the development of an automatic design and optimization system for industrial silencer primarily used on diesel engines in the marine, generator, construction vehicle, and military vehicle industries. In [Sultanian et al., 1999] both experimental and 3D CFD investigations are carried out in a scale model of an industrial gas turbine exhaust system.

The automotive exhaust system was mainly investigated in the literature studies. The applied methodologies are briefly described in the following.

In [Montenegro et al., 2013] the prediction of the acoustical performances of a silencer for internal combustion engines is obtained by 1D (software GASDYN), 1D–3D and quasi-3D (software OpenFOAM) non-linear approaches with the aim of optimization of complex shape silencing systems. A comparison between predicted results in terms of transmission loss and experimental measurements is reported (Figure 14).

(a) (b) (c)

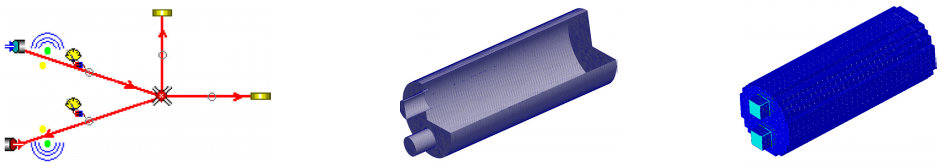


Figure 14: (a) 1D, (b) 3D and (c) 3Dcell models from [Montenegro et al., 2013].

In [Piscaglia et al., 2011] a high resolution central scheme for multi-dimensional non-linear acoustic simulation of silencers in internal combustion engines in automotive applications has been used, with ad-hoc developed boundary conditions for the generation of different acoustic perturbations (white noise, sweep, impulse), in the OpenFOAM technology (Figure 15).

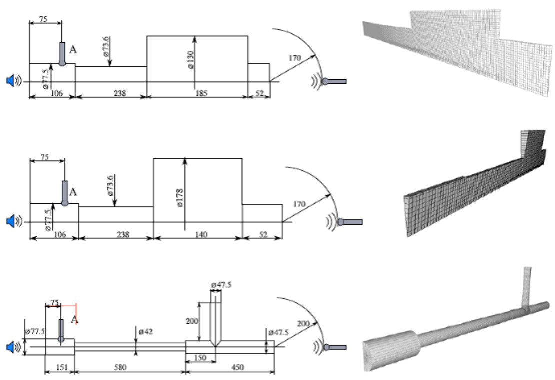


Figure 15: Silencer geometry from automotive application from [Piscaglia et al., 2011].

Based on the typical structure, a muffler with an interconnecting hole on the tail pipe (an automobile muffler with the acoustic characteristic of low-pass filter and Helmholtz resonator) was proposed to improve its acoustic performance in [Yasuda et al., 2013]. Acoustic performances of the proposed muffler were studied experimentally and theoretically in frequency and time domain. The tail pipe noise from a commercial automotive muffler was studied experimentally and numerically in [Yasuda et al., 2010]. The transient acoustic characteristics of its exhaust muffler were predicted using 1D CFD. To validate the results of the simulation, the transient acoustic characteristics of the exhaust muffler were measured in an anechoic chamber (Figure 16).

(a) (b)

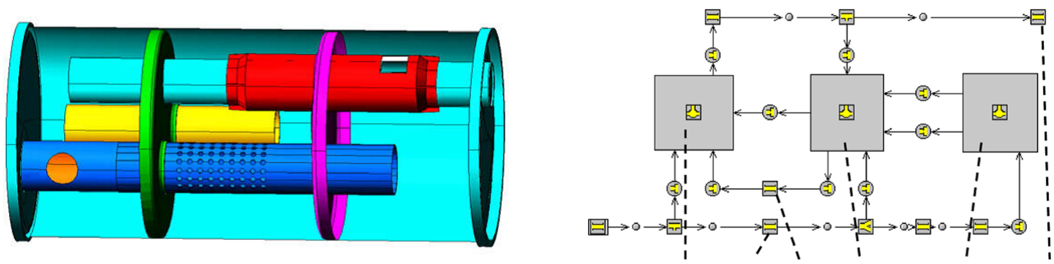


Figure 16: (a) Schematic diagram of the structure and (b) simplified 1D CFD model of the specimen muffler from [Yasuda et al., 2010].

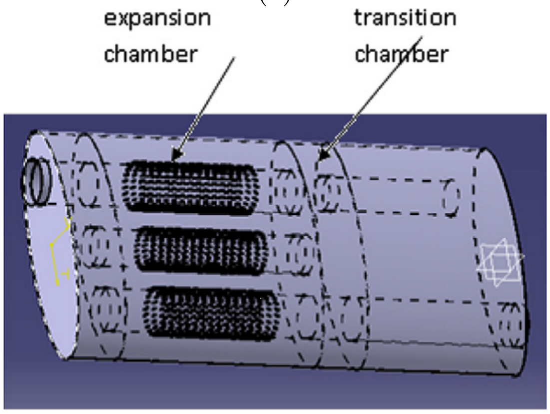
In [Rana et al., 2011] the analysis of flow induced noise in a passenger car exhaust system (muffler or silencer) with an experimental and a numerical approach is proposed. The flow analysis is carried out in commercial CFD solver Star CCM+.

The prediction of flow and acoustical performance of an automotive exhaust system using 3D CFD using Fluent (V12.0) is presented by [Sen, 2011]. The simulation results of flow field i.e. back pressure and SPL were compared with experimental results.

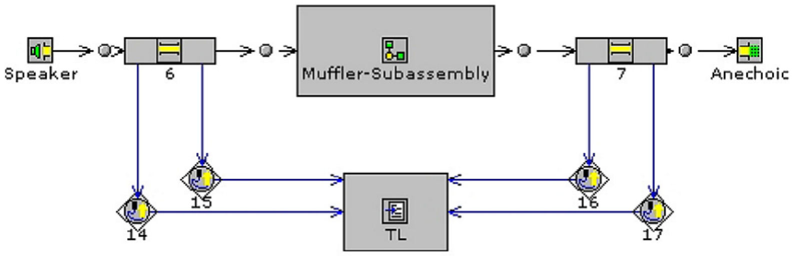
The development of an automotive exhaust silencer for improved sound quality and optimum back pressure is described in [Wagh et al., 2010]. The design constraints were the silencer shell dimensions, volume of silencer, inlet pipe and outlet tailpipe positions. The numerical simulation involves 3D computational fluid dynamics (CFD).

In [Siano, 2011], the noise attenuation characteristics of a typical perforated muffler for automotive applications are investigated. Acoustic performances are quantified by the Transmission Loss (TL) parameter, which only depends on the geometrical characteristics of the device. A 1D simulation code (GT Power™) is used to predict the TL profile in a low frequency range under the hypothesis of a planar wave propagation. A more complex 3D FEM/BEM approach is also realized using the VNOISE™ code (Figure 17).

(a)



(b)



(c)

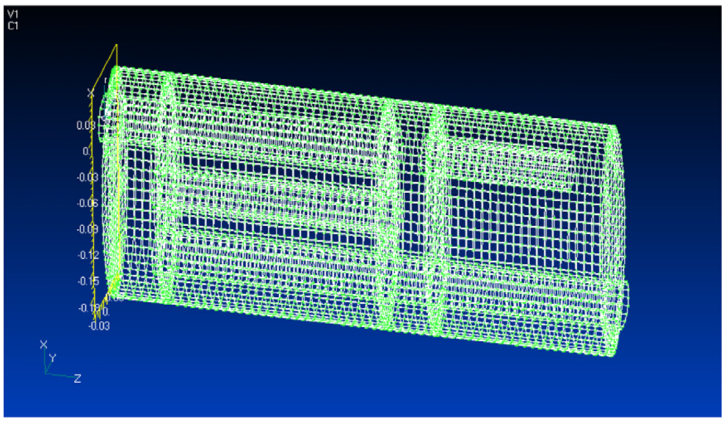


Figure 17: (a) Diagram, (b) scheme for the 1D TL analysis and (c) 3D FE model of a three-pass perforated tube muffler from [Siano, 2011].

5 Solutions

Passive and active control strategies to reduce noise and vibration are presented in the literature in order to improve the comfort on board ships.

5.1 Passive control

Passive methods to control and reduce airborne noise, such as partitions, screens, hoods and sound absorbing materials, are well known and extensively discussed in the literature. Different treatments may be uniquely applied to reduce airborne sources, structure borne sources, airborne paths, structure borne paths, and HVAC-induced noise, and so on; however, some treatments can be applied to treat multiple paths. For example, a floating floor can be used to reduce both airborne- and structure borne-transmitted noises. Each treatment type depends on an understanding of the prevailing airborne or structure borne noise components [Fischer and Bahtiarian, 2017].

Space for propulsion machines is generally the noisiest on-board compartment. Diesel engines, gas turbines or steam turbines are commonly used in most ships.

Looking at the paths of the structure borne noise from the engine into the surrounding, the engine mounts used to be the most critical components. The sound pressure waves going from the engine via the mounts of the elastic or even double elastic mounted engine frame into the ships structure, and then further on as airborne noise into the cabins and as under water noise into the environment.

But the full potential of a resilient mounting system is achieved only if all other transmission paths are considered. The secondary path goes along the powertrain via coupling, gearbox, gearbox mounts, into the ship's hull, as shown by the blue lines in Figure 18.

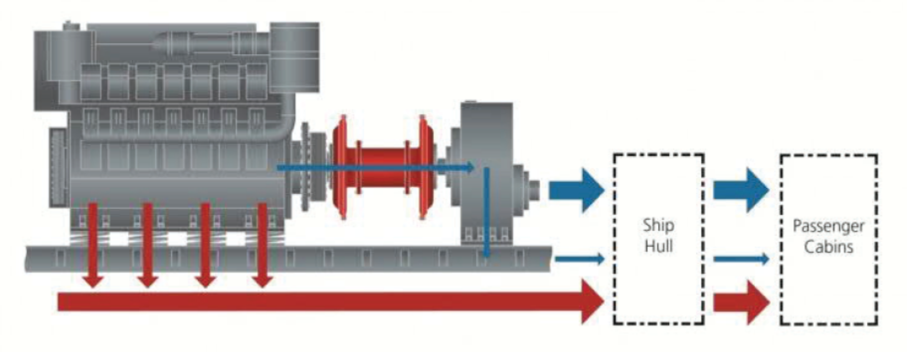


Figure 18: The path of structure borne noise on a ship. Red: primary path, blue: secondary path [Kurtze, 2019].

The "silent" engine turbines through the installation of factory mounted noise shields. However, it should be noted that a closed gas turbine has 10 to 20 dB lower airborne noise levels than an equivalent diesel engine. Likewise, the vibration levels of gas turbines are much lower than that of diesels due to the inherent differences between rotating and reciprocating machines. However, if a propulsion diesel engine is rigidly mounted, the overall sound levels in compartments adjacent to the engine room will be controlled by structure-borne noise rather than airborne noise. The most effective treatment in this case is motor vibration isolation. Alternatively, extensive application of damping material, floating floors and insulated mounted joint panels is possible. When the propulsion diesel engine is mounted non-rigidly (resiliently), the noise contribution in an adjacent compartment of the noises transmitted through the structure and the air can be of the same order of magnitude. For compartments further away, a good solution can be the use of damping materials, floating floors and elastically fixed junction panels with High Transmission Loss (HTL). Generally, compartments separated by 2 or 3 decks or 2 to 5 bulkheads from an engine room do not require additional acoustic treatment [Fischer and Bahtiarian, 2017]. In [Marchesini and Piana, 2012a] the noise transmitted from the engine room and the crew cabin to the owner's cabin was reduced, assuming as reference the DNV comfort rules, through passive noise control strategies: in order to improve the transmission loss of the bulkheads, a rigid foam was replaced by low density mineral wool, metal profiles were used instead of wooden studs and the thickness of two layers within the partition was changed to make the panel asymmetric. These improvement to the bulkhead transmission loss caused an increase of 13 dB in the airborne sound insulation index value. Moreover, the authors analyze the noise contributions, put in evidence by the improvement of the bulkhead transmission loss and deriving from the structure-borne noise coming from the floor; again, they propose a passive control solution consisting in the substitution of the stiff glue between the deck and the hull, used as resilient material for the floating floor, with silicon. Similar strategies were applied by the same authors in [Marchesini and Piana, 2012b].

When airborne noise transmission is significant, it may be convenient to increase the Transmission Loss (TL) of surfaces. The lightest treatment (in bulk) to achieve this increase is the addition of sound insulating materials generally 50 to 100 mm thick to cover the structural surface (bulkhead, deck or bridgehead). This material is generally made of glass fiber or mineral wool with a density usually around 50 to 80 kg/m^3. This intervention, in addition to the attenuation from the added mass, also provides decoupling between the airborne noise in the engine space and the compartment surfaces and also provides an additional TL through the insulation [Fischer and Bahtiarian, 2017]. The sound insulation material can be placed on both the source side and the room side of the receiver. The noise reduction of this treatment is between 3 and 7 dB in the low-mid frequency range and up to 12 dB at the higher frequencies. Approximately the same noise reduction can be achieved with thicker wood panels. The use of a heavier material, of the order of 160 kg/m^3, can further improve the TL of the bulkheads [Fischer and Bahtiarian, 2017].

In order to effectively decrease the noise levels in superstructures due to structure-borne noise, the most common solution is obtained by resiliently mounting a structure on the main deck, that consists in floating deck constructions [Nilsson, 1978]. With the aim of attenuation of structure-borne sound in the propagation path between source and receiver, resilient mounts between superstructure and main deck were introduced by [Nilsson, 1978] with a consequent noise level reduction in the superstructure by the order of 10 dB(A).

The structural path is usually treated by applying a damping layer or coating. Anti-vibration treatments have the ability to reduce the vibration level of the vibrating plates due to the unrecovered loss of energy converted into heat. The effectiveness of the damping treatments or coatings depends on the following factors: physical characteristics of the materials and location and surfaces of coverage.

Two types of anti-vibration coatings are currently in use: free surface/unconstrained and constrained layer damping. Tests have shown that the greater the deformation of the damping layer, the greater the effectiveness of the damping treatment. A relatively thin bonded metal or composite layer causes the damping material to work more effectively [Fischer and Bahtiarian, 2017].

One of the more effective treatments used to reduce the noise in the receiver space is a floating floor. Floating floors for marine applications can be usually divided into two groups (see Figure 19): floating floors in which a continuous layer of decoupling material creates the discontinuity between the ship structures and the upper floor, and those ones in which resilient mounts are used as decoupling elements [Badino and Rizzuto, 2015; Moro et al., 2016].

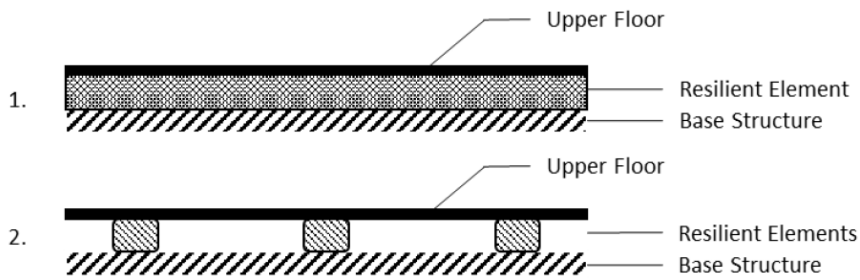


Figure 19: Two types of floating floor: 1. Floating floor with a continuous layer of decoupling material, 2. Floating floor with resilient mounts [Moro et al., 2016].

The Damping treatments reduce the vibration levels of the plate from 3 dB for the frequencies of 100 to 300 Hz up to 10 dB for the frequencies above 2000 Hz. It should also be noted that the damping treatments allow to simultaneously reduce both components of both airborne and structural noise. With this treatment, the finish floor is resiliently mounted on the structural deck. The resilient element may be a uniformly distributed layer, such as a mineral wool layer with sheet metal pan top surface. Another option is the use of individual resilient mounts placed between the sole and the structural deck. To be fully effective, it is recommended to install joiner panels on top of the floating floor without a hard connection of the joiner panel with the hull structure. The sole may be covered with a damping coating in order to increase the overall effectiveness. Noise reduction on the order of 10 to 15 dB is expected from the standard floating floor. This significant level of attenuation will only occur for a floating floor if the majority of the compartment's radiation is coming from the floor, a common situation. To achieve this level of reduction when surfaces other than the floor are major radiators of noise requires the use of the "floating room configuration" where in the joiner panels are mounted on top of the floating floor [Fischer and Bahtiarian, 2017].

In 2016, Moro et al. developed a rational approach, that includes numerical simulations and experimental tests, for the design of new floating floors for ships taking into account their capability to mitigate structure borne noise level generated by steady sources. The numerical simulations aim at the optimization of resilient mounting elements of floating floors in terms of dynamic stiffness and weight containment. The optimized configurations are then built and tested in laboratory [Moro et al., 2016].

Multi-layer floating floor systems were applied on the steel deck to reduce the radiated noise from the floor structure in a ship's cabin in [Joo et al., 2009]; the authors found that the cabin noise reduction of a multi-layer floating floor can be greatly improved over the entire frequency range if the viscoelastic deck covering is installed between the steel deck and the mineral wool of floating floor.

Another important issue related the noise reduction on ship regards the HVAC systems. Individual compartment fan coil units and central air-conditioning systems require different approaches to noise control. As a rule, the fan coil unit is a source of airborne noise, and the acoustical data for the units should be 5 dB lower than the compartment's criteria. Central ventilation and air-conditioning system may have noisy units as compressors, chillers, and fans. If located in an engine room, the noise from these units is masked by noisier sources, such as the diesel engines. If they are located in a dedicated fan room, this room should be treated as a machinery room, as compressors and fans are sources of airborne and structure borne noises. Treatment should be applied to reduce noise in both in the fan room and in adjacent rooms (resilient mounts, absorptive insulation, damping, etc.). Central HVAC systems usually have well-developed duct system with many branches and turns. Depending on the fan noise level and air-flow speed, the noise levels at the terminal (diffuser) may be significant. The application of good "acoustic" design practices will go a long way toward controlling HVAC-induced noise. It is still recommended to use low noise fans and/or low speed flow rates and to conduct flow and fan noise predictions [Fischer and Bahtiarian, 2017]. Furthermore, the use of innovative materials with increased sound absorption properties that allow the creation of high-performance advantageous systems, has been tested by [Borelli et al., 2015b].

As with the HVAC system, piping systems produce noise at the pump, which should be treated like a machinery item. In addition, consideration has to be given to the piping system itself. These systems can be designed to be quiet by reducing the flow speed and avoiding sharp bends. When treatment is needed, there are limited choices. Various hydraulic silencers exist that reduce the pump pressure pulsations getting into the fluid, and hence the pipe wall. These can either be tuned silencers or broadband silencers. A flexible hose will attenuate both the fluid borne and structure borne noise from the pump entering the piping system [Fischer and Bahtiarian, 2017]. Finally, for piping systems with high vibrations, resilient attachment points should be used, as shown as example in Figure 20.

(a)

(b)

Figure 20: Antivibration pipe supports: (a) support types, (b) example of application (source: www.otc.com).

5.2 Active control

Passive noise control treatments are an effective means of reducing the levels of noise and vibration experienced by humans in a variety of applications. However, due to both weight and size restrictions their performance in practice is generally limited to the control of higher frequency noise and vibration. Low frequency sound and vibration experienced in a ship often cannot be reduced by the use of ordinary passive methods or often leads to an unwanted increase in weight, which is in contrast with the requirements for lower weight to increase the maximum speed as well as fuel economy [Ishimitsu and Shibatani, 2008; Peretti et al., 2014; Winberg et al., 2005]. To overcome this limitation and achieve significant levels of low frequency noise attenuation, active control methods have been widely investigated [Cheer and Elliot, 2016]. Active noise control (ANS) can be a more advisable solution when structural modifications are unwanted or in presence of unpleasant low-frequency noise and vibrations.

An adaptive multichannel feedback active noise control system, also implementing a psychoacoustic correction, for a luxury yacht cabin has been recently proposed in [Peretti et al., 2014] as an alternative to common passive solutions (e.g. acoustic absorbing panels) to reduce low-frequency background noise in a specific point of the room (i.e. pillows of the bedroom) using microphones and sub-woofers for anti-noise generation.

A phase corrected filtered-error least mean square (LMS) algorithm was recently applied to the active control of ship interior noise by [Ishimitsu and Shibatani, 2008, 2007; Ishimitsu and Elliott, 2004].

An active vibration isolation, based on a combined passive/active engine mounts, was proposed by [Winberg et al., 2000] to improve comfort on board a luxury cruiser; the employment of active engine mounts, is especially important in marine applications, since the engines are usually mounted on flexible and light structures (i.e. the hull is not very stiff). In [Winberg et al., 2005] a pre-analysis of sound and vibration problem in a leisure boat was conducted, aiming at selecting the most suitable kind of approach of Active Noise and Vibration Control (ANVC) system; the authors also present an optimized passive engine mount, with a stiffness adapted to hull mobility and the engine vibration level, resulting in an A-weighted saloon sound pressure level reduction of 10 dB compared to the standard engine mounts, and in a reduction of vibration levels at the hull by up to 15 dB at the main harmonic components.

In [Cheer and Elliot, 2016] the potential of applying an active noise control system to reduce the levels of noise produced by a diesel generator in the master cabin of a luxury yacht (Figure 21) was investigated. The noise control problem has first been investigated and it has been shown that the noise and vibration due to the generator produce a sound pressure spectrum in the master cabin containing a full series of both integer and non-integer engine orders. It is difficult to control this further using passive control treatments due to both weight and size limitations. Global active noise control is also not feasible due to the relatively wideband frequency content and the modally dense nature of the master cabin. Therefore, a practical active control system has been implemented which focuses a zone of control at the head of the bed, where the generator noise is most disturbing when occupants are trying to sleep.

In [Mylonas et al., 2020] the potential of applying an active non-linear noise control system to reduce the noise levels produced by two asynchronous diesel generators in the double cabin of a luxury yacht was investigated. Due to the "beat" components and their harmonics present in the disturbance noise spectrum, and the need to control multiple tones to achieve a significant level of narrow band attenuation, a non-linear feedforward control system was studied.

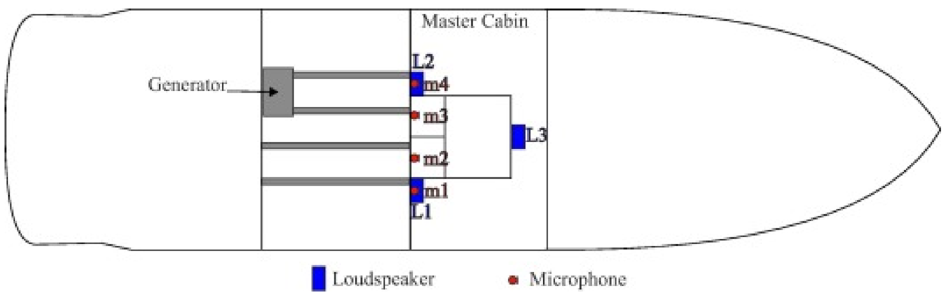


Figure 21: The layout of the standby diesel generator and the control system components on the yacht studied from [Cheer and Elliot, 2016].

6 Conclusive remarks

Noise and vibration in ship is very a complex issue and it includes many sources and not simple ways to sound and vibration propagation. When the noise generated in a ship it propagates in various ways: air-borne noise radiated by a source and transmitted through walls, bulkheads and decks, and structure-borne noise, which causes the appearance of noise in ship compartments even remote from the source of vibration due to transmission of sonic vibration through the hull structures.

In the nautical field, regulations and law that establish shipboard noise and vibrations requirements are stratified and not harmonized. There several national and international national bodies, such as the International Organization for Standardization (ISO), the International Maritime Organization (IMO), the International Labor Organization (ILO), and national authorities that publish their regulations. Most of these rules addresses problems connected to health and performances of the working crew and to the comfort of crew and passengers in accommodation spaces. Furthermore, there are the Classification Societies (CSs) that belong to the International Association of the Classification Societies (IACS), and that recently, released "Comfort Class Rules" for the assessment of noise and vibration comfort in the ship's compartments. Such comfort classes are defined in function of noise and vibration requirements that are often more stringent than those provided by national and international organization for standardization.

At present, as regards the assessment of comfort, the rules mostly refer to the sound pressure levels (dBA). The Comfort Class rules add requirements related to the apparent airborne insulation of vertical partitions and the weighted normalized impact sound pressure levels for horizontal partitions. However, the sound rating criteria provided by the American Society of Heating, Refrigerating and Air-conditioning Engineers (ASHRAE) typically used in the building engineering (Noise Criteria (NC), the Room Criteria (RC) and Balanced Noise Criteria (NCB), the Room Noise Criteria (RNC) and the RC Mark II) should be use as new indicators to be applied in the naval field. The Room Criteria Mark II, in particular, allows the estimating of the occupant satisfaction and reaction through an indicator known as the Quality Assessment Index. In any case, in the future new more sophisticated indicators would be effective in the shipping sector in order to have a better assessment of noise onboard and to guide the efforts to improve the onboard soundscape.

Concerning the predictive analysis methods, they are extremely important because, due to the complexity of ship structures, a rigorous classical approach (e.g., the wave theory) is impractical. Many different analysis methods were proposed in literature: the simplified analytical method based on a grillage model, the Transfer Path Analysis (TPA), the new operational TPA for identifying ship noise sources, the multi-dimensional substitution source method, the finite element analysis/method (FEA/FEM), the statistical energy analysis (SEA), the energy finite element analysis/method (EFEA/EFEM), the Boundary Element (BEM) and the Computational Fluid Dynamics (CFD).

The choice of the properly predictive analysis method is extremely important because it can allow to simulate the real conditions, and consequently, to identify the intervention of mitigation of noise and vibration. Regarding the mitigation interventions and so the noise control, passive and active control strategies to reduce noise and vibration are presented in the literature in order to improve the comfort on board ships. Passive methods are generally used to control and reduce airborne noise, such as partitions, screens, hoods and sound absorbing materials, are well known and extensively discussed in the literature, while Active control can be a more advisable solution when structural modifications are unwanted or in presence of unpleasant low-frequency noise and vibrations.

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