J. Klein Woud

Noise and vibration on board pleasure crafts

Literature research

Januari 2002

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OvS 01/12

Oceanco

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NOISE AND VIBRATION
ON BOARD LARGE PLEASURE CRAFTS
- Literature Research –

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Report no.: OVS 01-12
Date: January 2002

Preface

Vibrations and noise on board ships still form the subject of many discussions having to do with naval architecture/engineering. Especially on (large) passenger ships and pleasure crafts, where the comfort of those on board counts even more than on other vessel types, strict target levels are pursued.
In spite of all available tools, like absorbing materials, double glazed windows and flexible mountings, target levels are not always met. Sometimes the cause is obvious – a badly fitted engine, for instance or ducts connected directly to the steel structure -, sometimes the cause is not so obvious and guesses have to be made.

Oceanco Shipyards BV in Alblasserdam builds pleasure crafts with lengths of about 40[m] and above. Of course they want to create the most comfortable situation for their customers. Although they already do everything in their power to accomplish this situation, they felt the need to gain more knowledge of noise and vibration propagation and of factors that influence this propagation, so that they could create even more comfortable situations than they do now.
This need gave the yard reason to ask me to perform some more research on noise and vibration propagation, especially on board yachts.

The intention of the research was definitely not to be complete, but to offer sufficient insight in the scope of the issue and the complexity of noise and vibration propagation. More over it should serve as an introduction to further investigation on one of the many aspects of noise and vibration problems: influence factors of sound propagation on board large pleasure crafts.

Angélique M.A. de Vries
Alblasserdam, January 2002

Summary

Sound and vibrations are important aspects when designing a ship. Most of all on board passenger ships and pleasure crafts a "silent" vessel is essential. To realise this, it has to be known where the waves, because that is what sound and vibration are about, come from, how strong they are and how propagation through the vessel takes place.
With that purpose an extensive literature research was performed, using amongst others measurement data that were made available by Oceanco Shipyards. The research was performed with respect to the comfort of people on board.

For humans the audible range is the frequency range between 20-20 000[Hz]. Within this range the frequencies between 1000-8 000[Hz] are most important, since they encompass speech. Frequencies below 80[Hz] are observed as vibration rather than sound. On board ships frequencies between, roughly estimated, 0-10 000[Hz] are present.
Two kinds of noise are important: structure borne and airborne. Structure borne will have the most influence, because it can spread over a full ship's length, where airborne noise only has a local effect. As was expected in advance, the most dominating sources are found in the engine room and just outside: the propeller, main engines, gearboxes and auxiliary engines.
To prevent or reduce the noise from propagating, the machinery is fitted with flexible mounts and heavy foundations and where possible acoustic enclosures. The accommodations, crew cabins, work spaces etcetera are fitted with appropriate wall and floor isolation; the deck just above the engine room is mostly treated with a special high density material for extra reduction. Double glazed windows keep out outside noise.
Still, with all these measures taken, the measurement results sometimes are astonishingly different from the predictions that were based on the design data. An explanation is not always found.

No definite conclusions can be drawn yet, but it seems likely that either the accuracy of the predictions is not good enough or that design and practice are too far apart from each other to reach the predicted results. Further investigation will thus be performed to find the cause of the differences and to think about how the situation could be improved.

CONTENTS

INTRODUCTION

Extreme noise and vibration levels measured on board a 95[m] yacht, for which no precise explanation was found and similar problems on smaller yachts, raised questions on how such excessive noise and vibration levels could occur, in spite of all the precautions that were taken.
A lot of research has been performed over the years, most of which is based on empirical formulas and a lot of experience. In the majority of the research projects either cargo ships or cruise ships/large passenger vessel were investigated. Yachts, unfortunately, were seldom objects of study. Acceptable values for noise and vibration levels are – until now at least – always derived from the rules and regulations for large passenger ships.
Two reasons, thus, for the yard to be interested in further investigation on noise and vibration on board yachts (or pleasure crafts). The investigation consists of two parts:

1. A literature research providing a basis of general theory and different aspects to reckon with on board pleasure crafts.
2. A more profound research focused on one of the aspects, which should lead to recommendations to the yard for future designs.

This report contains the results of the literature research. The three objectives of the research were:

1. To gain understanding of the phenomena noise and vibration;
2. To put together the most dominating trouble causes on board, which makes it possible to strike where it is mostly needed;
3. To investigate what is already done to create an optimal situation.

The structure of the report is as follows. Chapter 1 will start with an introduction to the phenomena noise and vibration. Some of the basic theory, terminology and calculation rules are discussed. After that the effects of noise and vibration on ship, machinery and people on board are discussed in chapter 2, introducing noise rating curves, dB(A) and root mean square values. Chapter 3 will focus on an inventory of dominating sources and dominating propagation paths (structure borne or airborne). Methods to handle noise and vibration transmission are discussed in chapter 4, which also gives a plan how to perform a good analysis and take action against the inconvenience caused by noise and vibration.
Chapter 5 includes a conclusion of the foregoing chapters and a proposal for further investigation in the field of sound and vibration level prognosis with help of models.

1 INTRODUCTION TO THE PHENOMENA SOUND AND VIBRATION

All machines having moving/rotating parts, small or large, generate vibrations and because of that sound. Vibration as well as sound is a waveform and is characterised by a certain amplitude and frequency.
This chapter will be a brief introduction to the subject and give a summary of the most important aspects to be considered when dealing with sound and vibration problems. Noise and vibrations induced by engine room equipment, which includes the main engines, diesel generators, auxiliary equipment and propellers, will be the main theme throughout the entire report.
In section 1 a definition of sound and vibration, in the physical sense of the words, is given. After that some basic terminology is discussed. Section 2 focuses on some specific sound wave properties, such as energy dissipation and some calculation rules. Section three covers the human aspects of sound, introducing octave bands and noise rating curves, etcetera. Section 4 discusses the use of a computer program by example of the calculation of natural and resonance frequencies of an engine sub-frame. Relevant formulas are taken up in ***Appendix 1****.*

1.1. Basics of Sound and Vibration

To avoid misunderstandings it is important to have a clear definition of the subject under discussion. In this case a physical definition is desirable. The word "physical" is explicitly added here, because the definition may change completely when based on a psychological point of view, where personal perception rather than physical fact is dominant. Throughout this chapter the terms "sound" and "vibration" will only be used in the physical sense of the words, unless otherwise mentioned.
Secondly the basic forms of sound and vibration will be discussed in the way they generally occur on board ships and in engine rooms.

1.1.1. Definition of Sound and Vibration

A wave is a physical phenomenon characterised by amplitude and frequency. Sound and vibration are waveforms as well. A convenient definition would be one that describes the observable effect of the phenomenon, such as the definition given next.

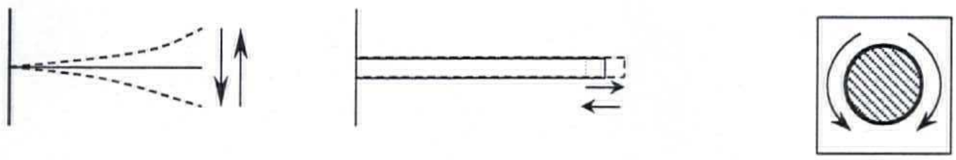
Definition: Vibration is a waveform defined as a periodically alternating displacement of (parts of) a structure caused by forces and moments working on (parts of) that structure. It is propagated as a waveform through the structure and its foundation, connections in the form of bolts, couplings and piping to components next to the vibrating object and through air or water.

Definition: Sound is a waveform just as well, obeying the definition given above. This specific waveform is referred to as sound, because the waves excite the human senses.

It is hard to make a strict separation between sound and vibration. What can be said is that lower frequencies, up to 80[Hz], are likely to be experienced as vibration; higher frequencies, above 80[Hz], are more likely to be experienced as sound. On board ships frequencies varying from 0[Hz] up to 2000[Hz] and higher occur. The lower part of this range causes the most trouble.
In the following subsection the kinds of sounds and vibration, which most frequently occur on board pleasure crafts, will be subject of discussion.

1.1.2. Kinds of Sound and Vibration

There are basically three types of vibration that are of importance for vibration problems due to engine room machinery. First of all there exists lateral vibration. This occurs in long, relatively slender elements such as shafts, rods, beams, but also in plating as a result of load acting perpendicular to the surface.
Secondly longitudinal vibration can occur. This is vibration in direction of the applied load and causes alternating stretching and shrinking of the element, such as shafts, beams, etcetera.
Thirdly there exists torsional vibration, which primarily occurs in rotating structural elements. It is a consequence of inertia loads and plays a significant part in examining vibration in long slender construction parts, such as shafts and piping.



a. b. c.

Figure 1.1. Basic kinds of vibration; a. Lateral vibration; b. Longitudinal vibration; c. Torsional vibration

Sound occurs in three forms: airborne, water-borne and structure-borne sound. Airborne sound is caused by waves in air, which are experienced by the human ear. The influence of airborne sound on ships is generally restricted to the area close to the source.
Water-borne sound is caused by sound waves in water as a consequence of propeller rotation. Depending on the frequency, the influence of the waves is restricted to the direct surroundings of the propeller or to the complete hull. Influence on the propeller shaft may be obvious.
Sound waves in solid materials are referred to as structure-borne sound. Machinery, cabin floors, bulkheads, plating, etcetera are solid materials to be thought of. The waves are experienced by the entire human body.

Barber [1992] acknowledges three sound-subcategories:
- impulse noise
- single event noise: bow thruster
- continuous noise: main engines, air conditioning systems

The first category includes very intense, short sounds, which could be associated with, for example, noise induced by an explosion.
The second category refers to short time exposure, not necessarily of high intensity.
The third category is usually associated with long-time, relatively low-level sounds.

1.2. Propagation of Sound Waves

Before any validation or comparison between two sound levels is possible at all, a suitable unit has to be defined to express this sound level or intensity. In this section basic theory and terminology will be discussed as well as several ways to describe and calculate sound levels.

1.2.1. Energy Dissipation

Sound as a waveform contains a certain amount of energy, which is gradually dissipated in all directions obeying the *Law of Radiation*. According to that law intensity, which is the energy that the wave possesses, decreases with distance in the following way: $I \propto \frac{1}{r^2}$, where I stands for intensity in [W/m^2] and r is the distance from the source in [m].

This dissipated energy is passed on to obstacles, for instance walls and machinery, in the surrounding of the source. By "moving" through these obstacles part of the energy will convert into heat. The remaining part of the energy will be dissipated further to adjacent spaces, walls, machine components, and so on. Energy dissipation will thus cause changes in either the amplitude or length of the wave, or both. This process is called wave *propagation*. Knowledge of the propagation process is essential in designing sound reduction plans.

1.2.2. Basic Terminology

The most common way to express airborne sound level is by means of the sound pressure level. Another frequently used notation is the sound power level. A third notation is the sound intensity, the energy dissipated per square meter. The notations will be discussed hereafter.

Decibel as unit for sound level

The unit used to describe sound levels is the *bel*, or rather *decibel*, named after Alexander Graham Bell. This could either be sound *pressure* level, sound *power* or sound *intensity* level. The bel is defined as the logarithm to the base 10 of the ratio of two quantities. The numerator is the estimated or measured quantity, the denominator a reference quantity. It is common practice to relate the reference quantity to the threshold of human audibility (this subject will be discussed later on in this chapter). Taking power as example, the reference value is $P_{ref} = 1 \cdot 10^{-2}\,[W]$. To obtain a value in decibel, the logarithmic value is to be multiplied by 10. There are two reasons for the use of a logarithmic scale:

1. The large frequency range over which a vibration is perceived as sound (20-20 000[Hz] for humans);
2. The response of the human hearing tends to be logarithmic rather than linear.

In case a sound *pressure* level is wanted the powers in numerator and denominator are to be replaced by source pressure and reference pressure respectively. In formula this looks like:

$$L_W = 10 \cdot {}^{10}\log \left| \frac{P}{P_{ref}} \right| \qquad (1.1)$$

$$L_p = 10 \cdot {}^{10}\log \left(\left| \frac{p}{p_{ref}} \right| \right)^2 \; or \; L_p = 20 \cdot {}^{10}\log \left(\left| \frac{p}{p_{ref}} \right| \right) \qquad (1.2)$$

$L_W = sound\ power\ level\,[dB]$ $\quad$ $L_p = sound\ pressure\ level\,[dB]$

$P = power\ level\,[W]$ $\quad$ $p = pressure\ as\ measured\,[Pa]$

$P_{ref} = reference\ power\ level\,[W]$ $\quad$ $p_{ref} = reference\ pressure\ of\ 2 \cdot 10^{-5}\,[Pa]$

Calculation Rules

When two or more sound sources are present, the total sound power level can be determined as follows:

$$L_{tot} = 10 \cdot {}^{10}\log \left\{ \sum_{i=1}^{n} {}^{10}\log^{-1} \left| \frac{L_i}{10} \right| \right\} \qquad (1.3)$$

$L_i = sound\ power\ level\ of\ the\ i^{th}\ sound\ source\,[dB]$

$Divide\ by\ 10\ to\ go\ from\ decibel\ to\ bel$

The formula stated above may only be used under strict conditions:
1. The sounds should all be in the same frequency band;
2. The sound waves should be in phase.

If the sound waves would not be in phase, anti-sound could occur, resulting in lower total sound pressure level. The explanation for a lower level is interference. Theoretically complete silence could be realised if two sounds having the same wavelength and same amplitude would be in anti-phase. In practice this will not occur and if so, this effect would be extremely local.

Since there are always more than two sources on board ships, which will not be perfectly in phase with each other, an alternative method [**Buiten, De Regt**, 1983], is applied for practical purposes. A brief step-by-step description of how to perform such a calculation will be given.

- The sound levels of all sources have to be known (either by measurement or estimation) and ordered in increasing order.
- Determine the difference between the two lowest levels by subtracting them linearly.
- Use **table 1.1.** to determine the required correction factor and add this factor to the higher of the two levels under consideration. The result will be referred to as L_{sum1}.
- Determine the difference between the second smallest level and the L_{sum1}, such that the difference is a positive value.
- Determine the required correction factor and add this factor to the higher of the two levels under consideration to obtain L_{sum2}.
- Repeat this procedure until all levels are treated.

Level Difference ΔL [dB]	0	0.5	1	1.5	2	3	3.5	4	5	6	7	8	11	12
Correction Factor L_{corr} [dB]	3	3	2.5	2.5	2	2	1.5	1.5	1	1	1	0.5	0.5	0

Table 1.1.Correction values applied when adding sound levels of more than two sources

It should be emphasised that the levels added are sound *pressure* levels and not sound power levels.

The third way to express sound level is the intensity of the field, again a logarithmic relation:

$$L_I = 10 \cdot {}^{10}\log\left|\frac{I}{I_0}\right| \tag{1.4}$$

$L_I = sound\ intensity\ level\ [dB]$

$I = power\ intensity\ of\ the\ field\ \left[W/m^2\right]$

$I_0 = reference\ power\ intensity\ of\ 1 \cdot 10^{-12}\ \left[W/m^2\right]$

1.3. Ways to Express the (Annoyance) Level of a Sound

In the previous section the decibel was introduced and a few calculation rules were given. This is all lovely theory, but quite worthless without knowing how to judge a level of 1, 10 or 100[dB]. In other words, what does a person experience when subjected to these noise-levels. This section will provide some more background information on the human hearing sense, octave bands, noise rating curves and measurement methods.

1.3.1. Characteristics of the Observation Window of the Human Hearing

Perception of sound by humans is possible by transformation of air vibrations into mechanical vibrations, hydraulic vibrations and finally into pulses. Different frequencies are experienced as different tone heights.
Not all sounds are actually heard, just the ones that fall into the so-called observation window defined by width, height and depth. The width of this particular window is the audible frequency range from 20 – 20 000[Hz], [**Pronk et al.**, 1995]. In the handbook of ship noise control [**De Regt**, 1983] an upper limit of 15 000[Hz] is used. The height gives information about the energy in the signal and the depth gives information on the duration of the signal.
The frequency range given above is based on a young, healthy ear. Ageing for example will narrow the width, especially on the upper side of the range.
The functioning of the ear is optimal in the range from 150 – 8 000[Hz]. For speech the frequencies between 1000 and 3 000[Hz] seem to be most important. This explains why 1000[Hz] is taken as the reference frequency to express the levels in [dB].

Sound levels expressed in [dB] always represent relative power, pressure or intensity levels. Thus, a level of 0[dB] indicates that the measured level equals the reference level, not that there is absolute silence. Usually the lower limit of human audibility is taken as reference value.
For this particular frequency of 1000[Hz] the *phon* may be used to describe the loudness [**Hoyland,** 1976]. The number of phons equals the sound pressure level of a pure tone at 1000[Hz] that is judged to be of the same loudness as the experienced sound; doubling the loudness will lead to an increase of 10[phon].
An audiogram (figure 1.2.) illustrates the relation between pressure level and frequency by showing curves of equal loudness. At 1000[Hz] the equivalent phon values are added.

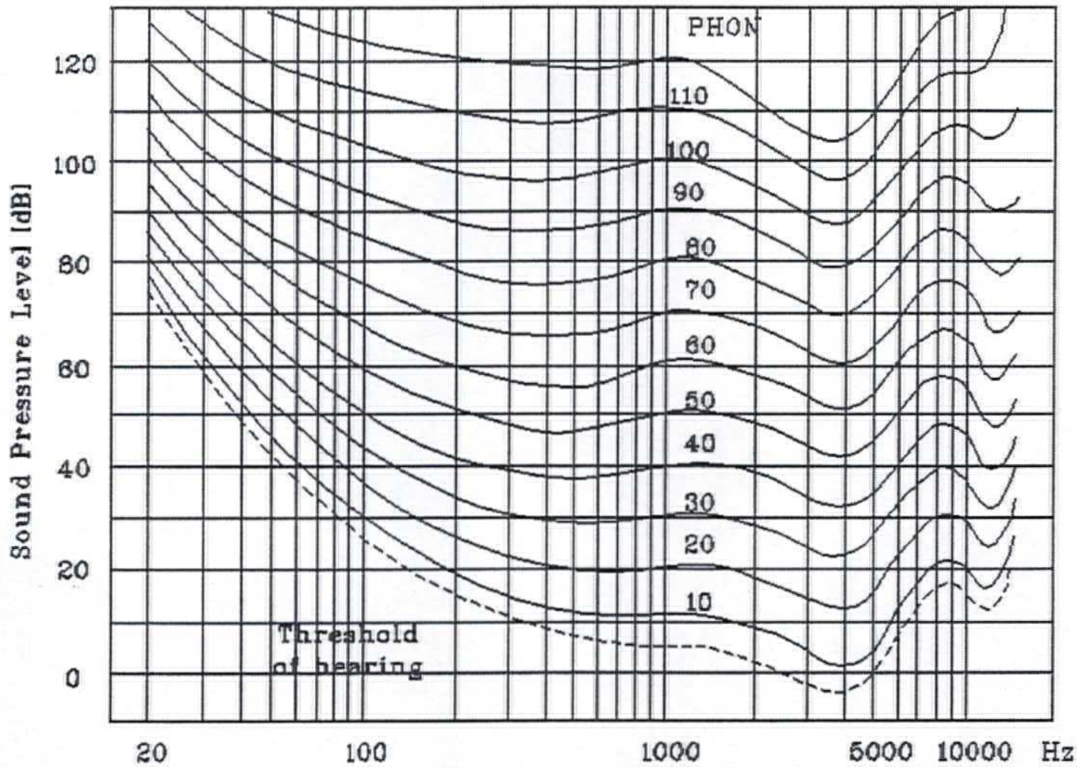


Figure 1.2. Audiogram for pure tones for the human audibility range

From the audiogram it can be derived that the most relevant frequency range for humans is the range from 1000 to 10 000[Hz]. In this range exists a high sensitivity to frequency changes.

To avoid communication problems three more terms must be explained:

- Doubling of sound sensation means an increase of 10[dB] at the same frequency;
- Doubling of sound power means an increase of 3[dB];
- Doubling of the pressure amplitude of the sound means an increase of 6[dB].

Instead of the phon the *sone* may be used to express pressure level. The sone is defined as the loudness that is experienced by a typical listener when listening to a tone of 1000[Hz] at 40[dB]. Doubling of the loudness (sensation) means doubling of the sone-number.

Octave Bands

The width of the observation window from 20 – 15 000[Hz] is initially divided into nine parts, called octave bands. Each band is referred to by its centre frequency, as visualised in **table 1.2.**. The total bandwidth of each octave band is encompassed by

$\dfrac{centre\ frequency}{\sqrt{2}}$ and $centre\ frequency\sqrt{2}$.

Number of Octave Band	1	2	3	4	5	6	7	8	9
Centre Frequency [Hz]	31.5	63	125	250	500	1000	2000	4000	8000

Table 1.2. Division of audible frequency range in octave bands

If more detailed information is required 1/3 - octave bands are used. As the word already implies this is a further division of each octave band into three parts. Each part is again indicated by it's centre frequency. The minimum and maximum frequency of each band are determined by $\frac{centre\ frequency}{\sqrt[6]{2}}$ and $centre\ frequency\sqrt[6]{2}$.

The hearing range is now divided into a total of 27 bands, see **table 1.3.**.

Octave Band Number	Centre Frequency [Hz]
1	25
2	31.5
3	40
......	......
16	800
17	1000
18	1280
......	......
25	6400
26	8000
27	10000

Table 1.3. Division of audible range into 1/3-octave bands

In case initial measurements do not lead to satisfactory results, narrow band analysis provides a suited tool for further investigation. With this analysis discrete components correlated with particular sources can be identified.

Noise Rating Curves

Besides frequency, intensity counts as an important criterion for the judgement of a sound. Higher frequencies are annoying at lower intensities. People of course do not judge sounds on their frequency and intensity, at least not directly. People will judge a sound on its annoyance level. One way of establishing the annoyance level is the previously discussed phon. Noise rating curves (NR-curves), which describe the relationship between frequency and intensity, are a more common way to express annoyance levels. A curve connects points of equal annoyance, again based on a frequency of 1000[Hz]. That is, the 80NR-curve is the line that crosses 80[dB] at a frequency of 1000[Hz].

The Noise Rating value of a sound is determined by intersection with the highest NR-curve over the frequency range. The red curve in figure 1.3. illustrates this. For this curve the probable NR-value will be 57 or 58, though the total noise level may be either lower or higher than 57[dB(A)].

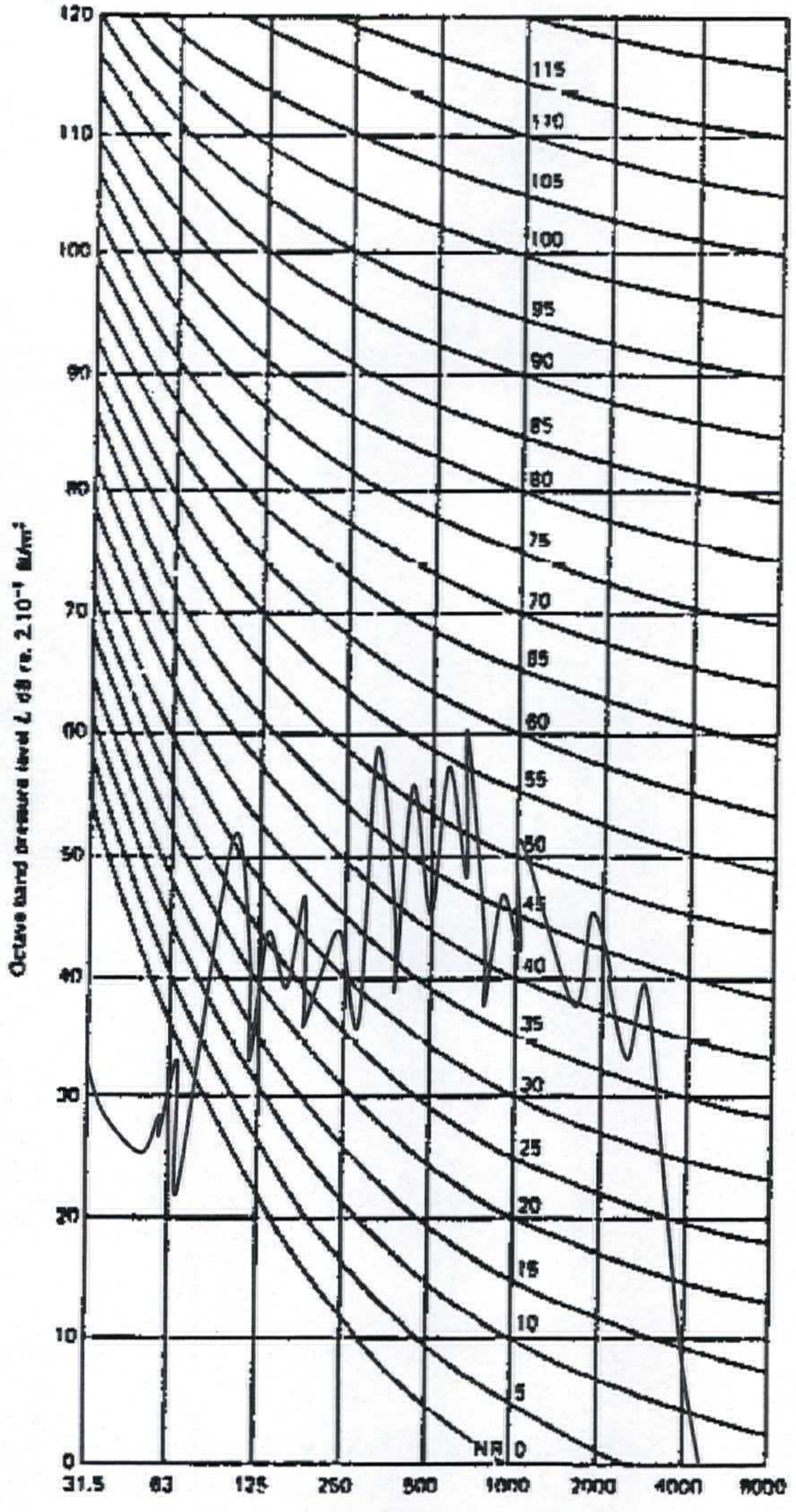


Figure 1.3. Noise Rating Curves for human audibility, connecting points of equal annoyance

Sound Weighting Scales

There are four weighting networks: A, B, C and D-weighting [**Pearsons, Bennet**, 1974]. These weighting methods are used to express the annoyance level of a sound in a single number. It can thus be used instead of NR-curves. For practical purposes A-weighting

(dB(A)) is mostly used. This measurement method takes into consideration that higher frequencies are more irritating at lower intensities than low frequent sounds. The method therefore uses a filter to weaken lower tones with respect to higher. Level-A measuring does not take into account that the difference between high and low frequent becomes less for higher sound levels contrary to noise rating curves. Originally it was intended to use A-weighting levels only up to 55[dB]. Nowadays A-level weighting is used for all sound levels.
Though rarely used in marine industry, the B, C and D-weighting levels will be mentioned briefly to complete the story. B-weighted levels (dB(B)) reduce the effect of low frequency noise and were intended for use between 55 and 85[dB]. Nowadays B-level is a popular weighting scale. Above 85[dB] C-weighting levels (dB(C)) were preferred. The levels are filtered to approximate the average range of human hearing. Application of C-level is restricted to the range between 31.5[Hz] and 8 000[Hz]. Finally, D-level weighting networks reduce the effect of low frequency noise and emphasise the effect of high frequency noise. The correction values for weighting are given in figure 1.4. The

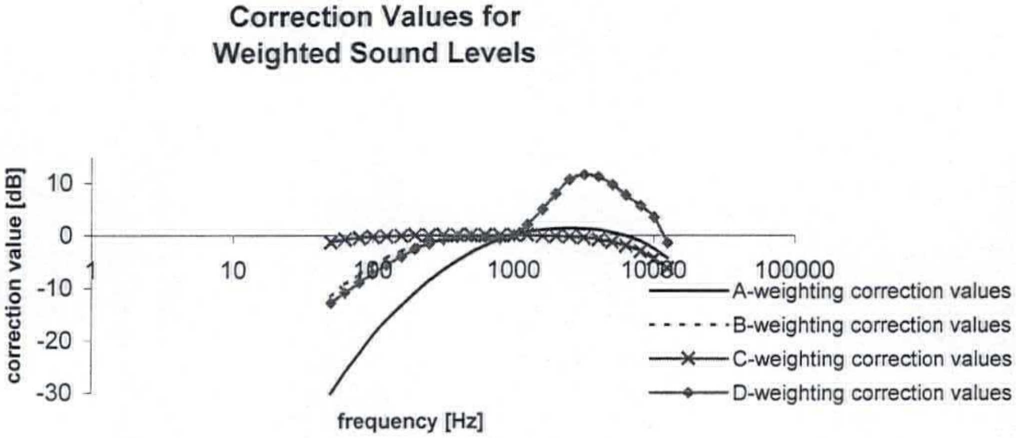


Figure 1.4. Sound level weighting networks for A, B, C and D-weighting

correction values are for the 1/3-octave band and are used to convert ordinary sound (pressure) levels into the appropriate, weighted levels. For calculation of the total sound level the squared relative pressures are used. An example will clarify the method.

Example[1]:
In the 1/3-octave band with centre frequency 315[Hz] a sound (pressure) level of 76[dB] is measured. In the 1/3-octave band with centre frequency 400[Hz] a level of 79[dB] is measured. According to figure 1.4. these levels need to be corrected for A-weighting with –6.6[dB] and –4.8[dB] respectively.
The corrected values follow by simply adding the correction value to the measured value.

Thus for the 315[Hz]-band: 76-6.6 = 69.4[dB(A)]
and for the 400[Hz]-band: 79-4.8 = 74.2[dB(A)].
The squared pressure ratios $(p/p_{ref})^2$*, denoted by* p_1^2 *and* p_2^2*, are calculated by taking the anti-log of the corrected values, after dividing them by 10 (from dB to B):*
$p_1^2 = 8.71*10^6[-]$
$p_2^2 = 2.63*10^7[-]$
Now, the total A-weighted level is calculated by adding the pressure ratios algebraically and taking ten times the log of the sum:

[1] Source: Handbook of Noise Ratings [Pearsons, 1974]

$L_A = 10log(8.71*10^6+2.63*10^7) = 75.44[dB(A)]$

The example shows that no special measurement equipment is required to determine A-weighted values. Measurement equipment will be handled in Chapter 2.

Noise Rating values and A-weighting values can be transformed into one another with help of calculation rules provided by handbooks and **Appendix 2**. This makes comparison and better judgement on acceptability possible.

To give an indication of the impact of a certain [dB(A)] level, **table 1.4.** is brought in. The table shows the pressure levels of a number of daily life phenomena.

Phenomenon	Sound Pressure Level [dB(A)] (approximately)
Moving leaves in a light breeze	10
Alarm clock ticking in quiet room	33
Noise in a large store	50
Ordinary speech at 1[m] distance	58
Ringing of (old fashioned) telephone at 2[m] distance	65
Noise from a motor highway at 7[m] distance	85
Symphony orchestra in concert hall	93
Jet-plane at 1200[m] height	110
Threshold of pain	120

Table 1.4. Sound levels of daily phenomena

1.3.2. Propagation of Sound Waves

Obstacles, change of material (structure) or geometry, influence the way in which and the distance over which sound propagates. Each obstacle will cause a loss of wave energy. The lost energy might either be absorbed by the obstacle or reflected in direction of the source.
The loss physically results in a change in amplitude and wavelength (velocity) and audibly in a lower noise level and perhaps even a deformation of the sound.
This knowledge must be used to create effective sound isolation. The efficiency of an isolation measure can be expressed in a transmissibility factor β. This factor is the ratio of the energy that passed through the isolating material and the energy that was present before the isolation. A low transmissibility factor thus implies a good isolation.

Transmissibility can be expressed in force transmissibility and in displacement transmissibility. The first expression will look like:

$$\beta = \frac{F_{trans}}{F} = \sqrt{\frac{1+(2\xi\eta)^2}{(1-\eta^2)^2+(2\xi\eta)^2}} \qquad (1.5)$$

$\beta = transmissibility\ factor[-]$ $\qquad \xi = \frac{c}{2\sqrt{km}} = damping\ ratio[-]$

$F_{trans} = transmitted\ force[N]$ $\qquad \eta = \frac{\omega}{\omega_{nat}} = frequency\ ratio[-]$

$F = resulting\ force[N]$

$c = damping\ coefficient\ [Ns/m]$ $\omega = rotational\ speed\ [rad/s]$
$k = spring\ constant\ [N/m]$ $\omega_{nat} = natural\ rotational\ speed\ [rad/s]$
$m = mass\ [kg]$

For isolation of a source $\eta > \sqrt{2}$ is needed at least, because only then β will be smaller than 1, which is derived from the definition of transmissibility. For effective isolation a minimum value of 3 is advised.
Figure 1.5. shows the relation between β and η.

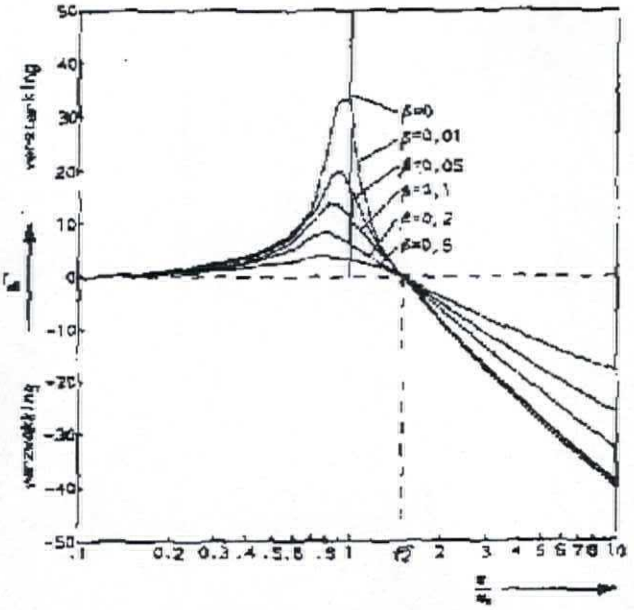


Figure 1.5. Transmissibility factor as function of frequency ratio

Application of such a theory stays limited to the simplest of systems, namely one mass and one spring. For predictions for complete propulsion systems this method will not be sufficient. For a first indication in a preliminary design stage this method may provide a decent tool. Accurate predictions in subsequent stages require more advanced programs and empirical methods, based on experience in the field.
Another restriction is the frequency range to which this transmissibility-method applies. It will be accurate only for low frequent load variations. For high frequent variations, the springs in the system will play a role as well and should be modelled as an infinite number of masses with an infinite number of springs.

Instead of force transmissibility, displacement transmissibility can be defined. The equation looks as follows:

$$\delta = \frac{x_{rel}}{x} = \frac{\eta^2}{(1-\eta^2)^2 + (2\xi\eta)^2} \tag{1.6}$$

$\delta = displacement\ factor\ [-]$
$x_{rel} = relative\ displacement\ [m]$
$x = absolute\ displacement\ [m]$
ξ and η as before

Again, from the relation between δ and η, it is shown that the natural frequency should be at $\sqrt{2}$ times as low as the forcing frequency for the construction to be free of disturbance and at least three times as low for effective reduction.
For the same reasons as mentioned on account of force transmissibility, application is very restricted.

1.5. Summary

Noise and vibration induced by machinery on board can cause a lot of annoyance. A well-considered choice of machinery and mounts can eliminate most of the annoyance. Knowledge of sources and mounts and of sound and vibration is indispensable to come to a good solution. Therefore the basic required theory on sound and vibration was treated in this chapter.
Amplitude and frequency are the most important judgement criteria along with the duration of the sound/vibration. Weighting levels take these preferences into account. Propagation of waves is influenced by the design of the structure and can be quantified in a transmissibility factor. The lower this factor is the better.

2 EFFECT OF SOUND AND VIBRATION

Understanding of the effect of vibrations and sound on people and objects in the vicinity of the sources is essential to attain effective measures. Judgement criteria such as pressure level and frequency were explained in Chapter 1. In this chapter the judgement criteria will be put into perspective with the current topic: noise and vibration on board pleasure crafts. Besides the pure technical criteria mentioned before, a sound will be judged on its character (continuous or impulsive) and on the duration of exposure.

Strict regulations for maximum allowed noise and vibration levels on board ships should avoid the vessel and persons on board from being damaged by their effect. With respect to the vessel, vibration levels are of primary concern; with respect to the persons on board noise levels are of primary concern.
Section 1 gives a general survey of regulations, rather guidelines, for sound and vibration levels. In section 2, some sound and vibration measurement methods and the corresponding equipment are presented.

2.1. Guidelines for Sound and Vibration Levels

Depending on the environment different standards for sound and vibration levels will be desired. In a library for example the maximum acceptable sound (pressure) level will be much lower than in a production plant. In a laboratory the vibration level will be lower than in an average office in order to gain accurate test results.
Maximum acceptable values on vessels will be determined depending on the purpose of the ship: production vessel, cargo carrier, cruise ship or pleasure craft, etc.

2.1.1. Acceptable Sound Levels

Maximum permissible values are set by Classification Societies. For private yachts however, there are no official rules. This does not mean that everything is possible, on the contrary. Theoretically one could do whatever one pleases, but the chance of approval by Flag Authorities and Classification Societies will be so small, that in practice the official rules are used as binding guidelines.

On one hand, with a lot of noise making machinery in a relatively small area it is not reasonable to expect noise levels to be as low as in an average living room. The agreements are thus based on acceptable, achievable levels, in view of the special situation on board yachts. Rules for commercial vessels [**Appendix 2**] are taken as reference rules. To be on the safe side, the allowed levels on board yachts are usually agreed to be 5[dB(A)] lower than those for commercial vessels.
On the other hand the high costs involved with sound reducing measures and the limitations of mass and space are reasons why complete silence is not be realised. With the available means one tries to create an optimal situation.

Flag Authorities and IMO[2] compose general guidelines that should protect the people on board from being harmed by excessive noise levels. In consent of the yard and the owner maximum values, within the limits of these guidelines, will be established.

Agreements
Common values for maximum permitted sound pressure levels are:

Location on Ship	Sound Pressure Level [dB(A)]
engine room	110
engine control room	75
owner's stateroom/ guest rooms	50

Table 2.1. Maximum permitted sound levels on luxurious yachts

Important criteria are the well being of the persons in that area and the influence of the sound on their professional performance. In areas where high levels cannot be avoided appropriate protection should be available to reduce the noise to an acceptable level. The values are derived from the specification of one of Oceanco's yachts; they are valid for all yachts built at Oceanco.

Conditions
Keeping in mind the normal environmental circumstances of a yacht, the established levels are based on levels achievable for sea trial conditions, which are given in **Appendix 2** as well. Amongst others, a tranquil sea and closed doors are desired.

During sea trial, when sound level measurements are performed and in general, these conditions are seldom met. There is a very reasonable chance that the sea is not tranquil at all, especially in Dutch area. Besides that the ship's interior might not be ready yet. It can happen that there isn't any furniture or carpet and that not all doors are yet fixed, or if fixed that they cannot be closed.
These circumstances have to be taken into account when measuring.

The configuration of accommodations may differ per ship, but the levels given by the IMO-code are valid for passenger vessels in general.

2.1.2. Acceptable Vibration Levels

An absolutely vibration free ship is an unachievable goal, because of the nature of the equipment on board. The presence of vibration is a fact and should thus be the starting point for making guidelines. A definition of "vibration free" for practical purposes is given in *Vibration Control in Ships* [**Veritec**, 1985]. It states that a vibration free ship is a ship in which:

- vibration levels are well below those found to induce local damage, initiate cracks, cause material fatigue, etc.
- vibration levels are so low that people on board are not bothered by them during their daily activities.

[2] International Maritime Organisation

Predictions of vibration behaviour can be made with help of appropriate software. A lot of time and money is involved in predicting vibration behaviour. Therefore it is, unfortunately, not always done. This could have severe consequences, since excessive vibrations can lead to material fatigue, thus initiating cracks. Improvised solutions afterwards may not lead to the desired objective, could cause new problems and are in the end definitely more expensive than a proper design would have been.

Another cause of vibration is bad fitting between the various components of the engine room equipment.
Communication at a preliminary design stage between all involved parties is crucial for the success of the project. The parties – yard, subcontractors, owner and classification societies – have to have a uniform understanding of the problems and how to solve them. Only then will it be possible to achieve effective solutions.
The importance of this issue is often underestimated, but daily practice shows the need for it. Figure 2.1 shows a vibration-investigation-plan for an entire design process, divided in phases and stages.

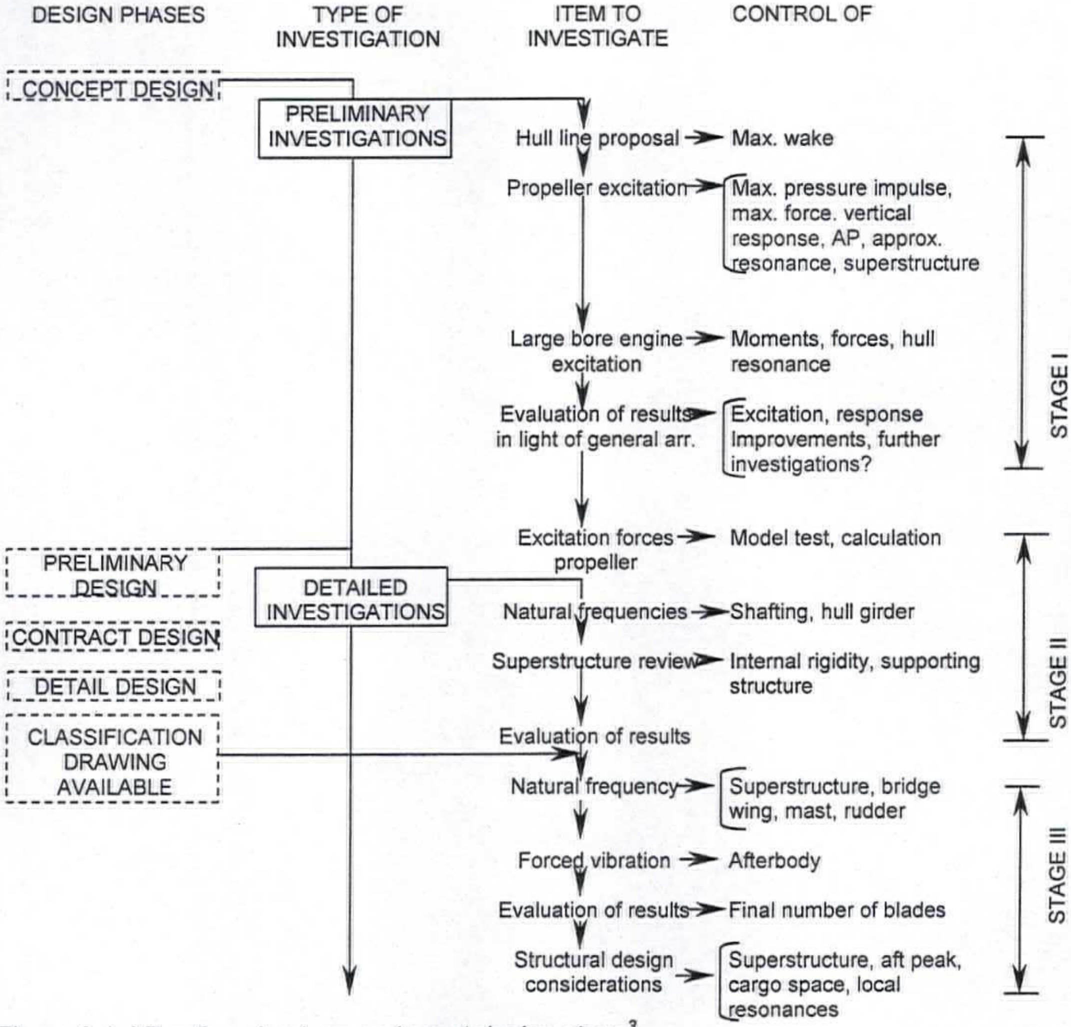


Figure 2.1. Vibration check procedure at design stage[3]

[3] Source: Vibration Control in Ships [Veritec, 1985]

Maintenance and safety of ship
Vibration levels must be low enough not to cause fatal damage to any equipment necessary for the maintainability of the ship. All bolts, bearings, ducts, etcetera have to remain in one piece and in place during operation. Vibrations may not cause material fatigue or cracks in engine foundation, ship's hull or any other supportive structure within the calculated lifetime for the materials. Vibrations may not influence any auxiliary equipment in their performance.

Well being of people on board
Under normal conditions crew and guests do not want to experience any trouble from excessive vibration. Crewmembers have to be able to perform their job in a normal way. Guests may not even want to notice that they are on board of a ship at all.

Maximum Vibration Levels
Vibration levels can be expressed in terms of displacement, velocity or acceleration. The first two quantities emphasise the lower frequencies, whereas acceleration emphasises the higher frequencies. The displacement method finds very little application, contrary to the velocity method, which is particularly suited for machinery vibration and structure borne noise. The acceleration method is found to be very useful in assessing dynamic mass forces.
It seems obvious now to define maximum values in terms of either peak velocity or peak acceleration.
Vibration level by Lloyd's (here for deck structure, but also applicable in general) is defined as:

"the single amplitude peak value of deck structure vibration during a period of steady state vibration, representative of maximum repetitive behaviour in mm/s peak over the frequency range of 1 to 100[Hz]"

Commonly applied limits, for velocity peaks, are:
- 5[mm/s] in accommodation and navigation spaces
- 6[mm/s] in work spaces

In yachting target values are applied instead of the limits mentioned above. The target values are given by Lloyd's in the table below. More information on this topic is included in **Appendix 3**. Under normal circumstances these levels are not to be exceeded. Target levels for passenger ships and high-speed crafts are included for comparison. To get approval however, target levels may be exceeded as long as maximum levels are not.

Location	**Peak Acceleration [mm/s^2]** 1-5[Hz]		**Peak Velocity [mm/s]** 5-100[Hz]	
PAC*	1	2	1	2
YACHTS				
Cabins and Lounges	31	63	1.0	2.0
Wheelhouse	47	94	1.5	3.0
Open Decks	63	110	2.0	3.5
PASSENGER SHIP				
Luxury Cabins	47	63	1.5	2.0
Standard Cabins	47	79	1.5	2.5
Public Spaces	47	79	1.5	2.5
Open Recreation Decks	79	110	2.5	3.5
HIGH SPEED CRAFTS				
Public Spaces	79	126	2.5	4.0

*PAC = Passenger Accommodation Comfort

Table 2.2. Maximum vibration levels according to Lloyd's Register

2.2. Measuring Techniques

Sound and vibration levels can be measured in many ways using many techniques. The choice for one technique or another is determined by the information one is looking for, the time and/or space available to perform measurement and again costs. During seatrials overall sound levels in closed (limited) areas are measured. Instant output of the measurements is required. The instrumentation should be fitted for this situation, like, for instance, a digital level meter or a meter linked to a (portable) computer providing a complete noise spectrum.
Techniques to measure sound and vibration will be the topic of this section.

2.2.1. Sound Measuring Techniques

The most common way to measure sound (pressure) levels is with help of a microphone. The accuracy of the results depends on the type of microphone used, the distance from the source and of the surroundings of the source. For 'normal' A-weighted measurements, which focus on the exposure of humans to machinery sound, a simple microphone would do.
A microphone responds to air pressure changes, just as the human ear. With this device it is not possible to distinguish directivity of the sound. To determine directivity of a certain sound, it is recommended to take measurements at several points around the noise source. The points should all be at the same distance from the source. Microphones are suited for sound level measurements as well as vibration level measurements.
The kind of microphone in use should fit the situation for which it is used. A few types will be discussed. First of all, a free-field microphone should be pointed as accurately as possible toward the source. It is designed to compensate for noise/disturbance caused by its presence, though. Secondly, a pressure microphone should be held as closely as possible to the sound source. For overall sound level measurements, this is not a very helpful method, because sound level decreases fast with increasing distance. A distance of 1[m] is the minimum required. In diffuse fields a random incidence microphone may be used. Directivity of the sound cannot be decided with this device, since it responds to sound in general.

Besides that there are several ways to construct a microphone, as will be explained next. The construction is linked to the intended use of the microphone.
Three main types are:
1. piezo-electric
2. electret
3. condenser

A total noise measuring system consists of more than just a microphone, as illustrated in figure 2.2. The signal coming in at the microphone is too weak to be analysed and thus needs to be amplified before any background noise (disturbances) can be filtered out and the "pure" source level can be determined. After the disturbances are filtered out, the signal has to be prepared for output, for example by putting the signal through a converter.

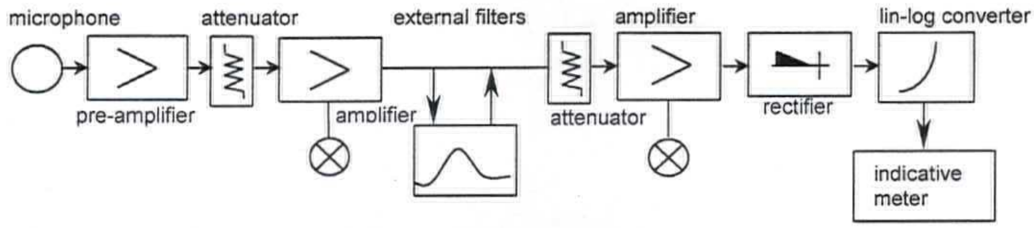


Figure 2.2. Block diagram of noise measuring system

Piezo-electric microphones are used in general-purpose instruments. Their functioning is explained here briefly. The fluctuating sound pressure will move the ceramic or crystal diaphragm of the microphone, causing it to be charged electrically. Because the charge will be so small an amplifier is built in, making the signal useful for interpretation. The microphone finds application in field measurements because of its robust design and relatively low costs. It is therefore less sensitive than the other two types mentioned.

Electret microphones consist of a thin diaphragm close to a back plate, which is covered with a dielectric foil. This foil is pre-polarised. Due to pressure changes, the distance between back plate and diaphragm change and thus capacitance changes. This then results in a voltage change. The microphone in itself is more expensive than the other two, but on the other hand it doesn't need as much circuitry. In spite of negative reports on behalf of their stability, they prove to be satisfactory and are preferred for measurements in humid environments.
Condenser microphones are pretty much the same, except they lack the foil. Instead, a polarising voltage is applied across the plates. They find application in precision instrumentation, because of their good stability and temperature characteristics. Figure 2.3. shows such an instrument.

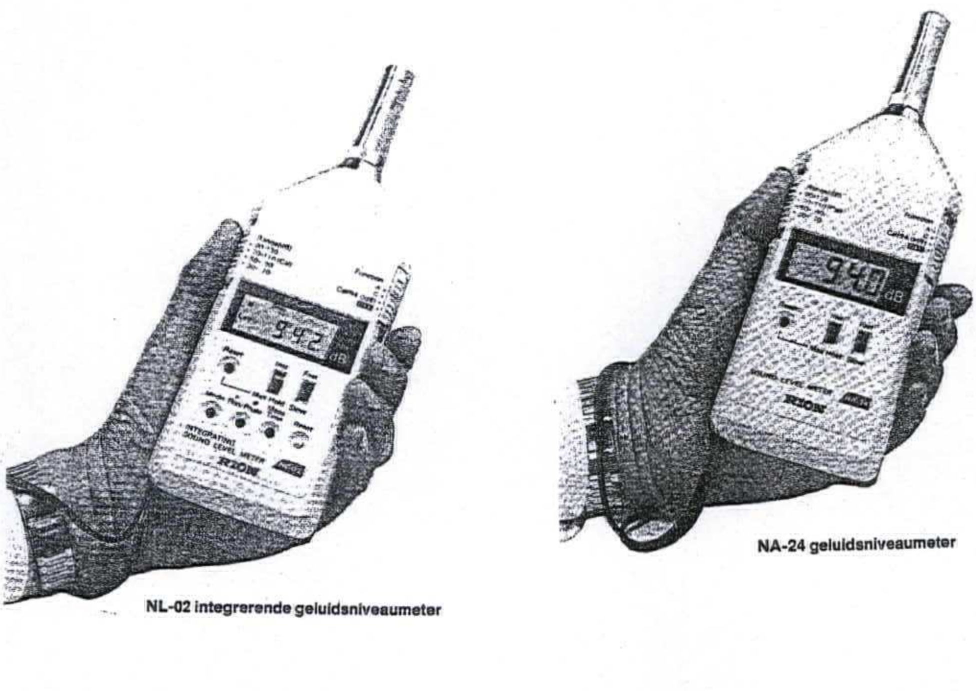


Figure 2.3. Example of sound measurement equipment

Measurement results will be presented in diagrams, such as the one below. From the diagram, the dominating source in that area can readily be derived:

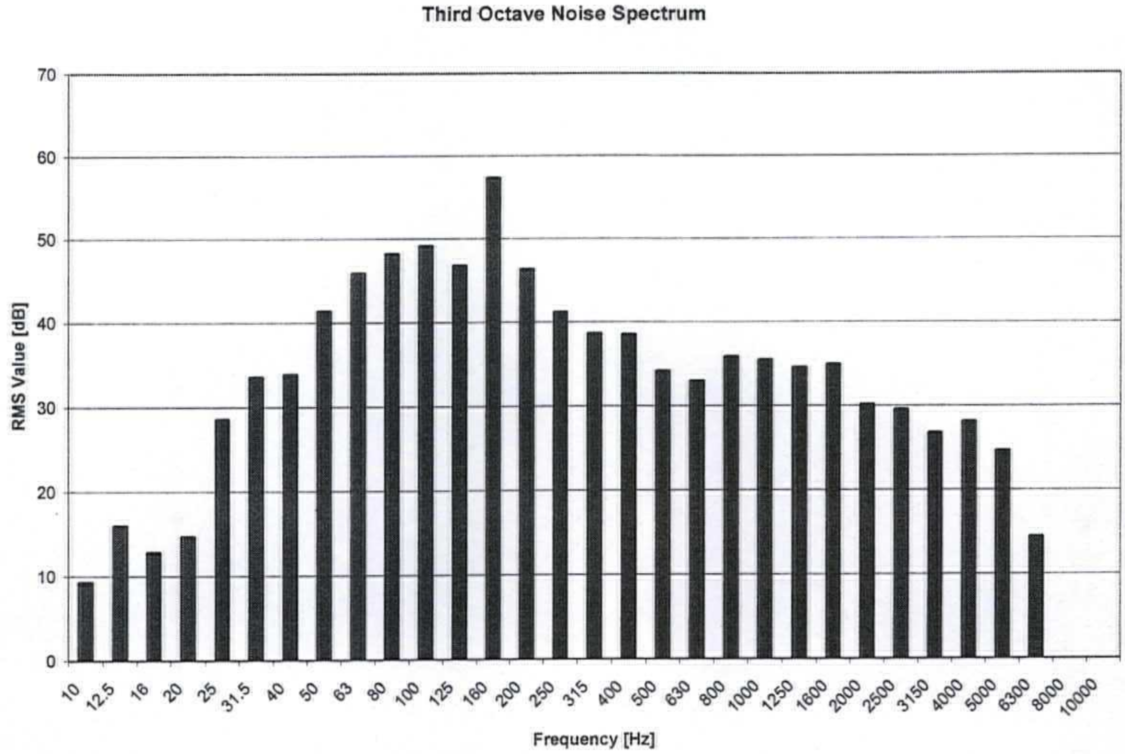


Figure 2.4. Example of sound measurement results: frequency spectrum

the highest levels determine the total sound level over the frequency spectrum. The highest peak of all represents the dominating source. The frequency at which this peak occurs will tell which source is responsible for this peak.

2.2.2. Vibration Measuring Techniques

The objective of vibration measurements in general is to establish the effect of the sources on their surroundings. If the measurements show extremities, adequate measures will have to be taken, to prevent damage of the source and its surroundings.

Choice of measurement points
Establishing the effects of a source requires measurements at several points on the source, in it's direct vicinity and in it's indirect vicinity.
Take an engine, for example. Determination of the vibration path, as it is properly called, takes place by means of specific measurement points. First of all the vibration level directly at the engine is measured, by placing a meter on its foundation. To determine the quality of the mounts, measurements are taken just above and just below. The peaks that will be visible in the frequency spectrum should be connected with the resonance frequencies of the sources. Figure 2.5. gives an example. These frequencies will also be noticed further away from the source.
Next, a number of spots on the structure in the direct surroundings are measured and finally walls and floors of adjacent rooms. In these measurement results, the influence of more than one source will show. The frequency at which peaks occur then gives a good indication on the identity of the source, provided that it's natural frequencies are known.

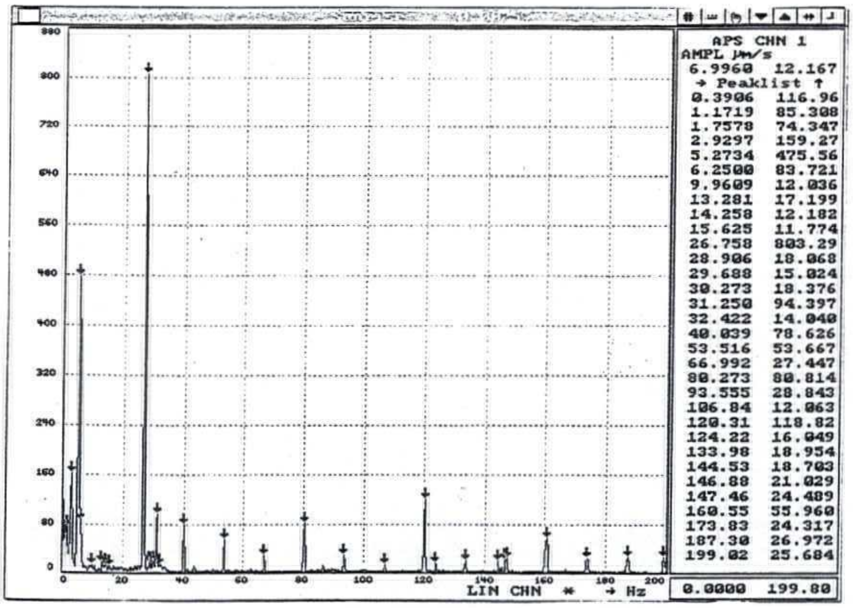


Figure 2.5. Results of vibration measurements: vibration spectrum; the peaks indicate the sources

The highest peak obviously originates from the dominating source and requires the first treatment.
Comparing the results of measurements on the various locations will give an indication of the (primary) vibration path of a certain source and on the reach of it's influence. From the height of the peaks the isolating/damping quality of the applied materials can be estimated qualitatively. This might come in handy when comparing measurement results with predictions.

Relevant formulas in vibration measuring
Vibration levels are established either by their displacement, velocity or acceleration. The applied method depends on the character of the problem. Usually electric potential is measured. That signal is then converted into [m] or rather [µm], because displacements are relatively small. The peak value is defined as the maximum positive or negative displacement relative to a certain equilibrium situation. The average value is defined by:

$$\bar{x} = \lim_{T\to\infty} \frac{1}{T} \int_0^T x(t)dt\,. \tag{2.1}$$

In this definition *x(t)* denotes the momentary displacement. From (2.1) it may be clear that the mean square value, associated with the potential energy of a wave, is defined as:

$$\bar{x}^2 = \lim_{T\to\infty} \frac{1}{T} \int_0^T x^2(t)dt\,. \tag{2.2}$$

The square root of this value, referred to as root mean square value or RMS, is commonly used in vibration analysing, where it represents the square root of the variance. RMS provides a measure of the magnitude of fluctuations of a signal.
In addition the autocorrelation function, defined as

$$R_{xx}(\tau) = \lim_{T\to\infty} \frac{1}{T} \int_0^T x(t)x(t+\tau)dt \tag{2.3}$$

gives information on how fast the signal *x(t)* is changing.

Instruments
Three types of instruments are used to measure vibration: displacement, velocity and acceleration meters. When the frequency of the vibration is known and one of the three quantities, then either of the remaining two can be derived. Simply differentiating or integrating will lead to the desired information.

First of all, a displacement meter could be used. With this device the amplitude can be estimated. Thereto a pointer may be attached to the moving part or a probe held against the moving surface. The disadvantage of such a device is of course its limited application area and accuracy. The frequencies must be low and even then, you will only get a rough estimation. For measurement on board yachts they are least appropriate.

Velocity meters are more common use. They can be applied up to frequencies of 1000[Hz].

Accelerometers are the most commonly applied instruments to measure vibrations. The instrument itself is very compact, therefore suitable for measurements on location. It can measure frequencies varying from 1[Hz] to 25 000[Hz], depending on the type of meter of course. Piezoelectric and piezo-resistive meters are very popular. The principle is based on a mass-spring construction mounted in a metal housing. The force exerted on the piezoelectric material by the mass is proportional to the acceleration. Due to the excitation a voltage will be generated.

The instrument can be linked to a computer to give direct visible feedback, showing frequency and amplitude of the vibration field. From this diagram (see figure 2.5) each source can be identified by hand of the frequency where a peak occurs.

Amplitudes away from the source are usually very small, in the order of [μm]. The signal from the accelerometer thus has to be amplified to allow a good comparison of and distinction between the sources.

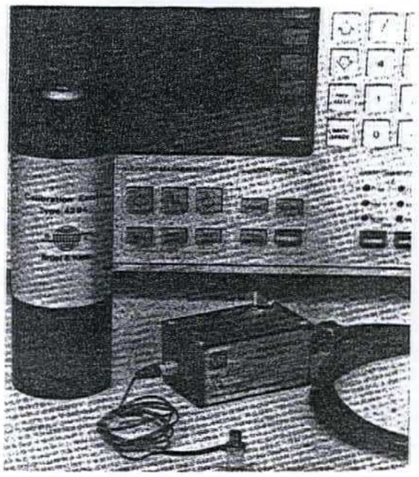


Figure 2.6. Example of an accelerometer for vibration measurement

2.3. Summary

To determine whether or not sound and vibration levels are excessive, they need to be validated. With respect to sound levels the well being of people is the most important criterion. With respect to vibration the maintenance of the ship's hull and machinery is most important.

The first section of this chapter introduced a *vibration check procedure*. Starting already in the design stage and following carefully each step, the procedure should guarantee a "vibration free ship". Furthermore guidelines for maximum sound and vibration levels were taken up.

Section two covered measuring methods and instruments for both sound and vibration levels. Sound levels are most frequently expressed in A-weighted pressure levels, vibration levels are expressed in either velocity or acceleration amplitudes.

3 SOUND AND VIBRATION ON BOARD PLEASURE CRAFTS

To protect the ship and its passengers, noise and vibration level should be kept as low as reasonably possible, or at least within the limitations set by the classification societies. In order to make adequate corrections one would want to know where the sounds and vibrations originate and what the impact on the rest of the ship is. The focus will be on sources located in the engine room.
In section 1 all relevant sources are examined. In section 2 some measurement results from measurements done on board Oceanco's yachts are given, which give an indication of the mean situation on board. Section 3 sets forth the influence of the major sources, discussed in section 1, throughout the ship.

3.1. Evaluation of Sound and Vibration Sources

Identifying the (major) sound and vibration sources is a first step towards a solution of the problem.

Major sources are generally found in or near the engine room, with the propeller, main engine and gearbox as most dominating of all. They are followed by the diesel generator and exhaust gas system. Furthermore fresh water makers as well as pumps should be mentioned, and last but not least the numerous ducts.
Some of the components mentioned, like the gearbox, act not only as a sound/vibration source but also as transmitter of sound/vibration induced by adjacent components. This topic will be discussed later on in the chapter.
Outside the engine room the laundry and air conditioning system can be seen as major trouble causes, though their relative contribution to the total problem is not likely to be very large. Kitchen equipment in the galley contributes to additional, local sound.

Not all sources deliver an equal contribution to the entire problem. To achieve a substantial improvement of the situation it is important to treat the dominant sources first. Therefore it has to be known which sources are dominating and which less.
In the following all major sources will be investigated and judged on the vibrations and sound they produce.

3.1.1. Outline of Primary Sources

Propeller
A rotating propeller pushes water through its blades thereby producing sound. Cavitation is the major cause of propeller noise and vibration. It occurs in several forms of which propeller hull vortex cavitation and sheet cavitation are the most significant. The first, also referred to as PHV cavitation, generates a lot of in board noise; the latter generates hull vibration induced by shock waves.

The amplitude of the shock waves decreases with ${}^{1}\!/_{r}$.

Amplitudes of non-cavitating waves decrease with $1/r^2$, where r is the distance from the source.
The most effective way to prevent cavitation from happening would be to improve the wake field around the propeller, which can be done by a careful design not only of the propeller but of the hull as well. Furthermore, there needs to be enough clearance between hull and blade tip. A minimum of 0.5[m] is recommended.
Bow thrusters cannot be omitted here, but since they are considered a secondary source, they will be discussed in section 3.1.2.
Background information on cavitation, propeller design and cavitation risk is given in **Appendix 4**. Figure 3.1. demonstrates a possible consequence of cavitation.

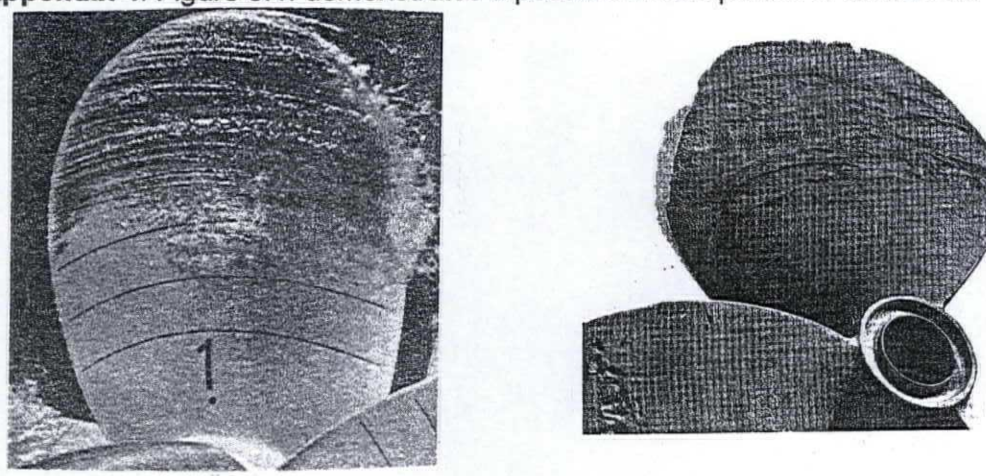


Figure 3.1. A propeller blade damaged by cavitation

Gearbox

Production of pure tones is significant for gearboxes. This explains the high annoyance level. Gear noise and/or vibration arises as a consequence of bad fitting of the teeth, not enough or too much lubrication or bad outlining. The impact of these factors can be expressed in a so-called transmission error (TE).

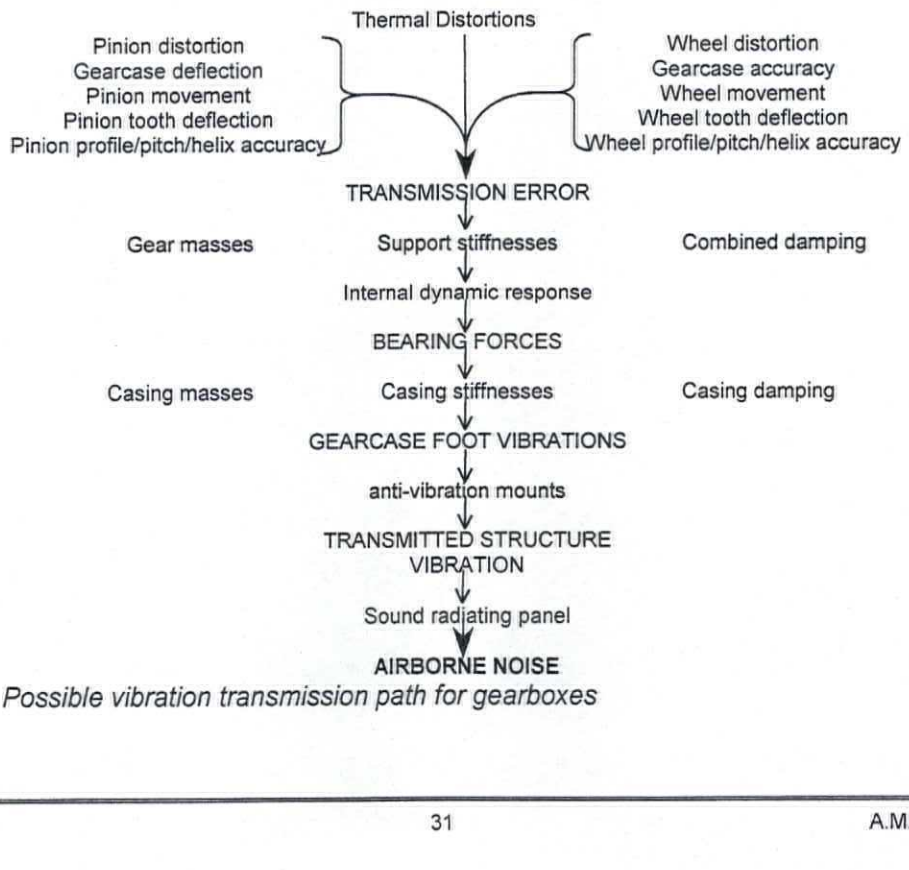


Figure 3.2. Possible vibration transmission path for gearboxes

Figure 3.2. gives a possible vibration transmission path [*Source*: Gear Noise and Vibration, by J. Derek Smith, 1999].
The figure should be read as follows: several distortions and relative movements of pinion and wheel cause a transmission error, which is the initiator of the vibration transmission. The vibrations are transmitted to connected parts being support stiffnesses, masses and damping. A resulting bearing force will remain, transmitting vibrations to the casing, etcetera. It also gives an indication for where to start solving the problem.
Additionally to internal causes, gears also transmit vibrations coming from the engine and the shaft.
The design features and mountings of the gear influence the total sound and vibration levels. The situation would be optimal if both gears were involutes. They provide in a more uniform distribution of tooth forces and in less transmission errors.

Figure 3.3. Illustration of straight tooth connection and of helical tooth connection

Main Engine
The engine is another major source of sound and vibration. Partly this is a result of the operating principle of the machine. Cylinder ignition frequency is an important influence factor here. Furthermore combustion noise, fuel injector noise, mechanical noise, inlet and exhaust and cooling fan noise are main factors of interest [**Barber**, 1992].
Unbalance is an important factor in relation to noise and vibration problems, because it stimulates the presence of free forces and moments. Free forces and moments can initiate vibration [**Appendix 5**].
The exhaust system and turbochargers are considered as part of the engine. The latter will not be discussed any further, because this involves a pure airborne noise. That means the noise will only be relevant in the direct surroundings of the source.

Diesel generator
Like the word already implies, a diesel generator is a combination of a diesel engine and a generator used for electric power delivery on board. The diesel engine driving the generator is usually smaller than the main engine and either medium or high speed.
Diesel generators run at all times, during cruising as well as during harbour time. Special care has to be given to sound and vibration phenomena here, to ensure an acceptable situation.
An extra difficulty is the location of the diesel generators. They are mostly placed on a higher level than the main engine, which means closer to the accommodations.
Besides the main diesel generator set an emergency diesel generator has to be present to take over power supply in case all other systems fail. This diesel generator is to be

placed outside the engine room at least one deck above the waterline, for understandable reasons. Restrictions on noise and vibration level are not so relevant

here, because the diesel generator will only be in use under extreme circumstances where operability of the vessel is of major concern.

3.1.2. Outline of Secondary Sources

Exhaust Gas System

The exhaust gas system, will certainly contribute to the total problem. High pressure and speed of the air passing through the ducts of the system not only generate a lot of noise (structure borne) but also considerable vibrations. Vibrations will especially cause trouble where the exhaust duct is supported. This is logical because the supports are in connection to the hull, thus forming perfect transmitters for structure borne sound into the surroundings.

High pressures and speed cause direct airborne sound and can also be held responsible for some ducting vibration. In case the exhaust is above the waterline this air borne noise will contribute a great deal to the sound on deck and on the bridge (supposing that the bridge will be near the exhaust). In yachting it is seen to that exhausts are placed below the waterline as illustrated in figure 3.4.

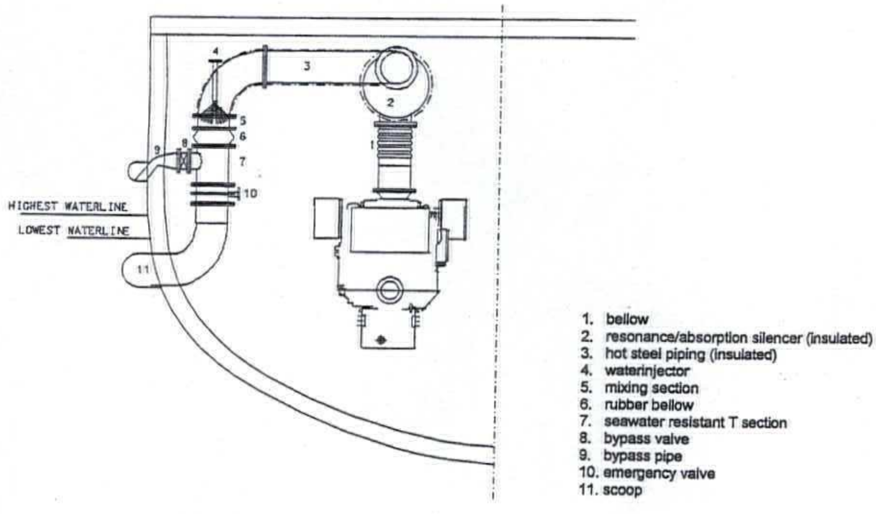


Figure 3.4. Example of an under water exhaust

Bow thrusters

Though they are smaller than the main propeller, bow thrusters rotate with higher frequency. On top of that they are installed in a tube in the front ship, where noise is transmitted directly into the ship's hull (see figure 3.5).

Bow thrusters (or bow propellers) make the ship more manoeuvrable at low speeds. They are thus needed for manoeuvring in crowded waters like harbours. Bow thrusters could be diesel driven or electrically – with an AC drive – or hydraulically. It should be kept in mind though, that these thrusters are not designed for full time use and that the noise they generate only lasts for a very limited amount of time. Besides, the noise usually remains local.

They will not be of main concern in this investigation.

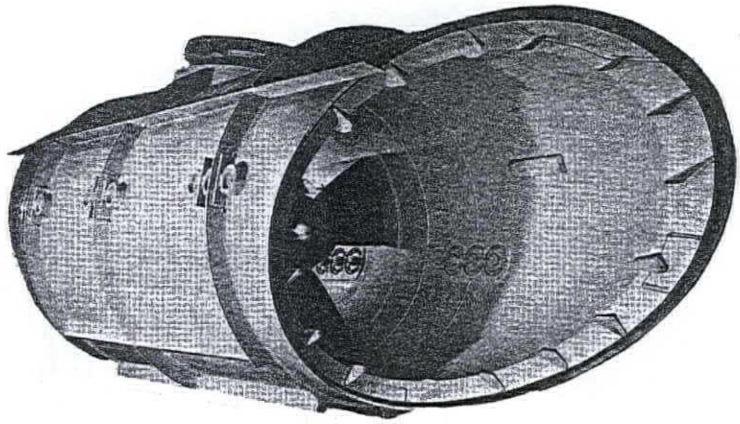


Figure 3.5. Example of a bow thruster in a duct

Pumps
On board numerous pumps are in service for the processing of fuel, oil and sewage water. For most applications on board the choice will be a centrifugal pump, known for its reliable service. The pipe system of a pump has two distinct sides, the suction and discharge side. Displacement from the contents of the pump from the suction to the discharge side takes place by mechanical variation of the chamber volume.
For optimal functioning pipe connections should be as direct as possible, avoiding sharp bends and loops to keep the flow through the pipes steady, thus resulting forces and moments to a minimum.

Separators
Like pumps, separators are numerous and placed all over the engine room and in technical spaces (i.e. spaces outside the engine room where machinery is installed; mostly used for air conditioning units and other relatively small equipment). A separator makes use of the difference in density to separate two or more liquids from each other.
A very important one is the fuel separator. It separates fuel from water and other unwanted contamination. This separator is placed between the day tank and settling tank. Its capacity can reach several thousands of litres per hour. If the day tank becomes too full, when the ship is in harbour for instance, the fuel is automatically returned to the settling tank. The settling tank is in connection with the storage tank (just for information).
The above explains why the separator is always working.

Valves and Pipelines
Valves and pipelines are perfect conductors of noise and vibration. In order to control propagation of waves (either sound or vibration) flexible couplings, decent fixation and well-considered installation with respect to bends, loops, etcetera are indispensable. It is at any time desirable to keep the amount of pipes and valves to a minimum. Lengths of pipes are to be kept to a minimum as well.

3.1.3. Outline of Minor Sources outside the Engine Room

Laundry
Of course washing machines produce sound and a little vibration, but compared to the other sources present they are completely irrelevant. They will not be investigated and are only mentioned here to complete the list.

Air Conditioning
Air handling units, or more commonly named air conditioning (systems), can be carried out in several ways. One possibility is to install one central unit where air is collected and treated properly before it is taken to other locations on board by transporting ducts. Another option is to install small units in all accommodations. An advantage of the latter system is that there is no need for transport ducts all through the ship; a disadvantage is that the units take up part of the cabin space.
The decision for one or another system will depend on the size of the ship, the available space in the engine room (or technical spaces) and in the cabins.

Kitchen Equipment
Is normally not experienced as very annoying and if so, it only lasts for a finite short time range; the noise will be local (air borne) and overruled by heavier sounds. It will not be investigated any further.

3.2. The Area Influenced by Primary Sources

The four sources mentioned as primary sources in section 1 will be discussed a little further in this section. The key point will be to establish the distance over which the influence of each particular source is noticeable.
Of course this depends on the precautions taken against transmission of noise and vibration, but this is only part of the issue. The other part of the issue is the nature of the source.
The sources will come up in the same order as they did in section 1.

Propeller
The propeller is located at the beginning of the "vibration train". Vibrations caused by the propeller can be either local or global, depending on the nature of the vibration. There is always a pressure field present round the propeller, due to rotation of the propeller blades. This pressure field is responsible for vibrations in the aft ship right above the propeller, mostly at the first order blade frequency. These vibrations tend to remain local and the forces involved are not extremely high.
When cavitation occurs, which is almost always the case, the story changes. Pressure and force variations due to cavitation cause global vibrations as well as noise. These global vibrations can affect the whole ship. The forces involved with the vibrations are relatively large compared to the "normal", local vibrations caused by the first order blade frequency.

A proper design is a first need to keep vibration to a minimum. The designer should attend to a propeller shape with a smooth and constant flow around the blades to minimise the risk of cavitation. A smooth flow can be achieved by a right choice of the number of blades, the pitch of the blades, the pitch/diameter ratio and enough clearance between blade top and ship's hull. For more background information on cavitation and propeller design see **Appendix 3**.

Gearbox
Like already said in 3.1., gearboxes produce pure tones. Usually the frequencies of the tones are low, say between 50 and 80[Hz]. The tones, which could be caused by a bad fitting of the teeth, are very penetrating. Their influence can be felt all over the ship,

because the vibrations go straight into the hull and are carried on by walls and bulkheads and so on.
Because of the low frequency they will be experienced by humans more like a vibration than like a sound, but therefore not less irritating.
Besides, gearboxes transmit vibrations originating from the engine and propeller shaft. Transmission will take place through the gearbox casing and foundation. The influence will gradually decrease with increasing distance and will probably only be relevant in spaces directly next to or above the engine room.

Reduction of gearbox vibration could be achieved by a careful design and fitting of the teeth and decent damping and foundation of course. Furthermore, insulation of the surrounding spaces is essential.

Main Engine
The influence of structure borne sound originated from the main engine can carry a long way through the ship.
The sound and vibrations basically have two opportunities to propagate, either through the structure (structure borne) or through air (air borne). Figure 3.6 illustrates the most important propagation paths that the sound waves and vibrations will follow. Numbers 1,3 and 4 indicate where structure borne propagation takes place: through the mounts and foundation and via the engine shaft. Numbers 2 and 5 indicate where air borne propagation takes place: straight from the engine into its direct surroundings and through in- and outlets. At location number 5 there will be propagation through the structure as well. All inlets and outlets are somehow connected to the ship's hull. These connections thus provide a propagation path.
Engine noise/vibrations have very varying frequencies, since this depends on the engine design and on the number of cylinders. Dominating are however first and second order frequencies and often, especially for four-stroke engines, also the 1½ th and the 2½ th order and the ignition frequency, which is of higher order, say 4th order and higher.
Appendix 5 will make clear why these frequencies are so important.

Installation of dampers and foundation is essential for effective reduction of vibration propagation. Again, good insulation of the surrounding spaces is required as well.

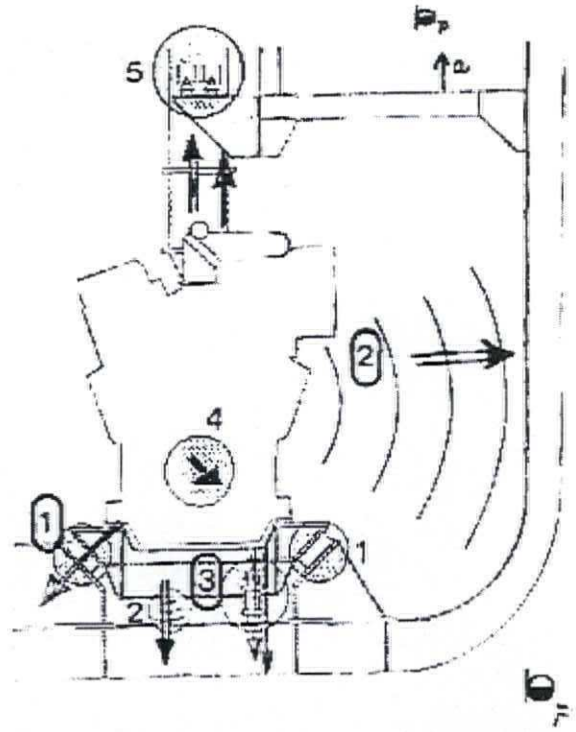


Figure 3.6. Vibration and sound propagation possibilities for an engine

Diesel Generator
Most of what was said about the main engine goes for the diesel generator as well. However, the effect of a generator will not be as far carrying as from the main engine. On one hand this can be explained by the size and power of the generator, which usually is a smaller engine with less power than the main engine.
On the other hand, the generator is often placed higher than the main engine, thus closer to the cabins. Airborne sound will then be the major cause of inconvenience. Airborne sound is a local problem and can be fixed very well by putting an enclosure round the source.

Secondary Sources
Pipelines and exhaust systems are influential secondary sources, the first as transmitter of structure borne sound, and the latter as transmitter of airborne sound.
Pipelines are present all over the ship and are perfect transmitters of sound and vibration. This is underestimated too often, though a lot could be gained if the pipes were properly treated.
Treatment of airborne exhaust sound seems to be under control by the use of silencers.

Insulation methods and materials will be the topic of chapter 4.

3.3. Presentation of Some Measurement Results

A lot has been said about the inconvenience and annoyance of sound and vibration. The dominating sources and the reach of these sources were discussed. A quantification of the inconvenience, however, is not yet given. So the question is what kind of impact should be thought about, when dealing with these problems?
The answer to that question will be given in this section on basis of measurement results obtained from seatrials of several of Oceanco's yachts.

The results, which are presented in a table, involve principal locations on board like the engine room, engine control room, captain's stateroom and the owner's stateroom. Along with the measured values the predicted values are presented, as far as available.

A few remarks are in place here.

- Results of different yachts may not be compared without further comment. The specific circumstances during measurements should be considered as well as the general arrangement of the accommodations relative to the engine room.
- Circumstances to be reckoned with are: percentage MCR, usually 80% or 100%; state of the accommodations, they might not be finished yet; sea condition, which could be anywhere between very tranquil and stormy.
- General arrangements are such that the owner and captain will find the least inconvenience from all noise and vibrations on board. Crew cabins are usually nearest to the engine room and engine control room, which is also for practical reasons, of course.
- Measurements in the accommodations give overall, mean sound levels. The source that is responsible for the total noise level can be identified only after inspection of the frequency spectrum of the measurements.
- In the engine room, overall measurements are not always performed. More often the quality of the mounts is tested, by measuring just above and just below. This shows the importance of correct and precise installation. Practice has proven that, even though the engines might be the same and on the same ship, results can vary significantly.

Unfortunately, there were no data available from measurements at the dominating sources themselves. From the noise spectra and vibration measurements however, the influence of an individual source can be estimated by studying the peaks of the graphics. The frequency, at which a peak occurs, tells a lot about the character of the source. Data on vibration measurements, as far as available, are taken up in the last column of table 3.1. and 3.2. below. More data of other Oceanco yachts can be found in **Appendix 5**.

Constellation (MY801)			
Location	**Noise Rating [dB(A)] -**	**Sound Level [dB(A)] -measured-**	**Vibration Level [mm/s] -measured-**
Engine Room*	112	114	11.44(ME)
Engine Control Room	75	-	-
Crew Cabin(PS)	57	52	0.75
Guest Accommodation (lower deck PS)	64	58	1.45
Guest Accommodation (lower deck SB)	50	51	1.85
Owner's Stateroom	50	49	0.78
Captain's Stateroom	39	43.5	-
Wheelhouse	44	48	0.54

Table 3.1. Noise and vibration levels measured on board the Constellation (80%MCR)

*vibration level was measured at the foot of the main engine(ME), that is above the mounts

Lady Christine (MY561)			
Location	**Sound Level [dB(A)] -predicted-**	**Sound Level [dB(A)] -measured-**	**Vibration Level [mm/s] -measured-**
Engine Room	110	107	-
Engine Control Room	68	72	-
Crew Cabin	49	44	0.51
Guest Accommodation (lower deck)	49	54	0.52
Guest Accommodation (upper deck)	51	-	-
Owner's Stateroom	47	50	0.62
Captain's Stateroom	53	51	0.35
Wheelhouse	51	53	0.25

Table 3.2. Noise and vibration levels measured on board the Lady Christine (1030[rpm])

Unfortunately there is no standard procedure that says which locations should definitely be measured. Therefore the available information differs per ship. The tables above are a just summary of the sea trial test results. Nevertheless they give a rough impression about the order of magnitude of the occurring noise and vibration levels on board.
From the measurement reports from the two yachts above and four other yachts the following conclusions can be drawn:

- Noise levels in the engine room are in the range of 106-111[dB(A)];
- Noise levels in the engine control room are in the range of 68-74[dB(A)];
- Noise levels in the crew accommodations are in the range of 45-47[dB(A)];
- Noise levels in the guest accommodations are in the range of 52-55[dB(A)];
- Noise levels in the owner's accommodations are in the range of 40-50[dB(A)];
- Noise levels in the captain's accommodation and wheelhouse are in the range of 46-52[dB(A)];

At first sight, it seems that length is not the determining factor for the final noise level. The ships under investigation varied in length from about 50[m] to about 80[m]. The position of the engine room (midship or aft) did not seem to influence the noise level in any way. The location of the owner's accommodation on main or upper deck does not make a difference in noise level either, although the spreading of 10[dB(A)] is large compared to the other areas.
Looking at the vibration levels the length of the ship and the installed power do seem to have influence. The four smaller vessels show much lower levels than the two 80[m]-vessels.
A well-argued judgement of both vibration and noise levels can only be made after thorough investigation of the particular situation of each yacht. This topic will be discussed in more detail in the second phase of this study, which involves determination of, amongst others, the before-mentioned influence factors.

3.4. Summary

The objective of this chapter was to identify and analyse the major sound and vibration sources located in or just outside the engine room, which was done in sections 1 and 2. The major sources are, as expected, the propeller, the gearbox, the main engines and the auxiliary engines. As principal secondary sources the exhaust gas system and the piping system were mentioned. To illustrate the impact of the sources throughout the ships, a selection of seatrial measurement results was included. From the results presented in table 3.1 and 3.2 and additional information gained from measurement reports of four more yachts it was concluded that noise levels stayed within the acceptable limits for most locations. The differences between the yachts under consideration were not extraordinary, except for the owner's staterooms.

4 REDUCTION OF SOUND AND VIBRATION

Sound and vibration are waves, that much should be clear by now. The easiest way for a wave to propagate is straight ahead through a uniform medium. On its way the wave will loose some of its energy due to friction, but this will be nothing to speak of. So, to reduce or entirely prevent propagation, obstacles or discontinuities should be inserted on the propagation path. The more obstacles the better of course, but there are restrictions. Vibration reduction is often connected with a lot of mass, which is not of advantage for yachts. Furthermore, vibration and noise control is a very expensive matter. Nevertheless, it is always better to prevent than to mend.

This chapter will go through the options for noise and vibration control starting in section 1 with a proposal for an action plan which should result in an isolation plan and finally in the implementation of that plan. Section 2 gives an abstract of the available means - elastic couplings, mounts, and special carpet - to realise the required reductions of airborne and structure borne noise and vibrations. In section 3 a few options are worked out in a little more detail. The options concern devices applied directly at the source. Section 4 treats measures that need to be taken at the accommodations. Finally, section 5 will give a recapitulation of the measures that are at presently applied at Oceanco yachts.

4.1. Making a Proper Action Plan

Optimal reduction of sound and vibrations is not obtained overnight. It is a process that needs to be thought over thoroughly, starting already in the earliest design stage.
In this section a proposal for a step-by-step plan will be presented, which should lead to satisfactory reduction of sound and vibration and to an enduring satisfactory situation. First a few important definitions will be given, to avoid misunderstandings later on.

4.1.1. Definitions of Relevant Concepts

Isolation and absorption are two different principles for reducing the effect of sound. To illuminate the difference, the definitions of isolation and absorption are given.

Definition: Isolation is a sound reducing measure based on the reflection of kinetic energy towards its source.

Definition: Absorption is a sound reducing measure based on the conversion of kinetic energy into thermal energy.

Isolation is definitely effective in spaces adjacent to the area where the source is located and at larger distances from the source. Reduction of noise in the area, where the source is located, is best done by application of absorbing materials, which also have a positive effect at spaces further away from the source.
Isolation is realised by inserting discontinuities in the transmission path. Discontinuities could be geometrical or physical, shape or material changes respectively.

Absorption is realised with help of absorbing materials. In cabins this could be, for instance, carpet, wall paper, curtains etc. For more effective absorption, materials such as glasswool or rockwool are more suitable.

Two further concepts should be defined here, active and passive (noise) control.

Definition: Active control is taking measures to protect the surroundings of the source from noise and vibrations caused by this source.

Definition: Passive control involves protection of components against vibration (or noise) from the surroundings of these components.

As example of active control the resilient mounting of engines and gearboxes can be mentioned; as example of passive control the flexible installation of sensitive equipment like computers can be mentioned.

4.1.2. Step-by-step Procedure

As said in the introduction of this section, vibration and noise control begins as soon as in the earliest design stage. A logical *first* step could thus be to make an inventory of all (possible) noise and vibration sources. In chapter 3 the most influential sources to be reckoned with were described.

The *second* step would involve a quick evaluation of the individual sources to make a distinction between primary and secondary sources.

The *third* step would be examination of the primary propagation path of the sound and/or vibrations. This is important to ensure adequate means are implemented at the places, so that no valuable time and money are wasted on useless measures.

The *fourth* step would be to decide on the total required reduction in [dB] or in [mm/s] along the propagation path necessary to meet with the standards for noise and vibration in crew cabins, guest accommodations and so on. The standards were discussed in chapter 2 and Appendix 2.

After that the *fifth* step could involve investigation of all available, usable means or devices to reach the levels established in step four. There are actually two "golden rules" to obey:
1. See to it that the largest reduction is gained directly at the source;
2. Keep it safe and simple, also referred to as KISS [**Soncini**, 1997]. By this he means that before confiding to alternative, not well-known, let alone proven products, one should explore the existing supply market. Often the simplest solutions are the best solutions, so do not make it more complicated than necessary.

Ad 1:Investigation should take place in three "phases":
Phase I: At the source
- the source itself: would it be possible to replace the component with a more silent one;
- the direct surroundings of the source: what could be achieved with mounts, enclosures, absorbing materials, etcetera;

Section 4.3. will go more into detail concerning this phase.
Phase II: Along the propagation path

- the space in which the component is installed: wall and/or floor isolation, isolation from bulkheads and girders (to prevent structure borne noise), could the component be moved to another room perhaps;
- the spaces adjacent to where the source is: wall and/or floor isolation, floating floors, double glazed windows etcetera;

This is subject of section 4.4.

Phase III: At the receiver

- the spaces where the sound and/or vibration is received: wall and/or floor isolation, carpet, curtains, etcetera.

This is subject of section 4.4. as well.

The *sixth* step would be to make a choice based on prognosis of the achievable reduction with the various possibilities. This prognosis will only be a rough estimation though, based on experience and empirical formulas. Prediction methods are usually applied at this stage.

After that the *seventh* step would be implementation (or installation) of the devices. From this point on monitoring becomes even more important. The effect of the reduction depends entirely on correct and *complete* installation.
The word "complete" is added to emphasise the importance of closing all possible leaks where vibrations and sound could enter the area. One leak could destroy the entire operation. This requires careful supervision of the whole process.

4.2. Discussion of Available Solutions

The isolation plan resulting from the step-by-step plan, which was proposed in section 4.1. should actually be divided into three categories:
1. Reduction of airborne sound;
2. Reduction of structure borne sound directly at the source;
3. Reduction of structure borne sound along the path, including the receiver location.

The solutions for each category will be discussed subsequently in the following sub-sections. At the top of the "solution list" it should say: *choice of equipment*.
This may sound trivial but is often overlooked, probably because it is so logical. The best way to start reducing sound and vibration is to install the most silent and vibration-free machinery.
Reasons not to do so may be financial or have to do with demands put on the features of the machine or maybe even with a customer's preference for a certain brand or whatsoever. In the yachting business the size, the weight and of course the power of an engine are important criteria. Before giving way to such things, one should consider the overall consequences of the choice. A less optimal alternative might cost a little less, but what will it take to compensate the "extra" noise and vibration that it produces? And will the effect of all effort be as good as expected?

4.2.1. Treatment of Airborne Noise

Airborne sound does not carry as far as structure borne sound and is usually restricted to the direct surroundings of the source. Examples of effective treatment of airborne noise are acoustic enclosures and double glazed windows, walls, doors, wallpaper and curtains.

Acoustic enclosures find frequent application in the gas turbine industry but are also broadly used to isolate diesel generator sets (gensets). They reduce the sound level by as much as 10-15[dB] or even up to 22-24[dB], according to **Gerstner** [1990].
Though this is a very effective tool to reduce the sound, acoustic enclosures have some disadvantages, which could be of overriding importance when making the final choice. These disadvantages encompass the space required for an enclosure. There needs to be a certain clearance around the enclosure for safety reasons and accessibility and between enclosure and machine for roughly the same reasons. Besides that, weight could be a problem. Figure 4.1. shows a genset fitted with enclosure.

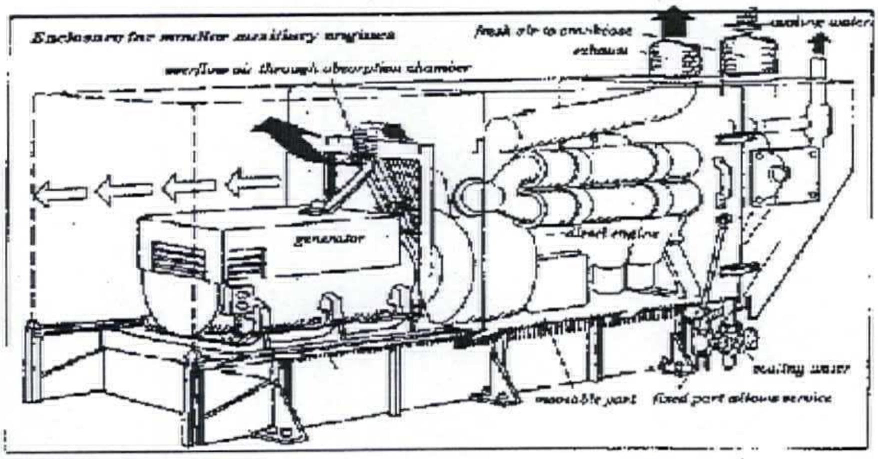


Figure 4.1. Example of a genset with acoustic enclosure, showing escape routes for sound.

Transmission of airborne noise to adjacent spaces around the source can be prevented by minimising the direct connections between source (or source foundation) and the separating wall or floor; by attaching isolating material to the wall or floor and last but not least by applying floating floors and cavity walls. It must be emphasised again that isolating material does not have much effect in the space where the source is located. Cavity walls and floating floors are fitted with absorbing materials like glass- or rockwool or something of the kind. The topic will be covered in section (4.4.).
Another form of airborne noise comes from outside. It could be exhaust noise, in case the outlet is above water (this is avoided whenever possible, especially in yacht building) or wind or vessels passing by. Isolation is in those cases provided by double glazed windows, which will be discussed later on in section (4.4.).

At receiver locations such as crew accommodations, guest and owner's accommodation airborne sound is treated with absorbing materials in the form of carpet, curtains and maybe even wallpaper. Doors take care of the isolation.

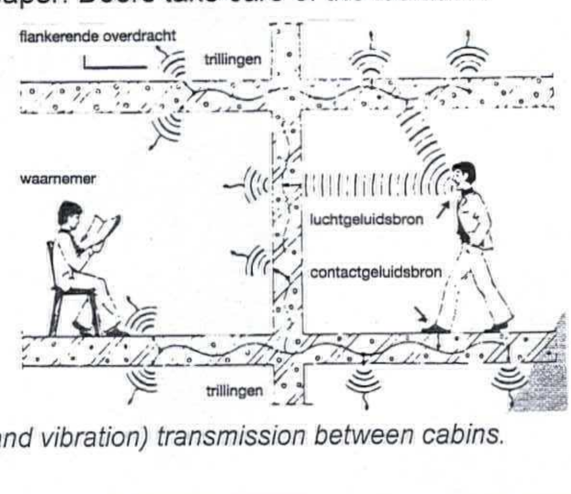


Figure 4.2. Noise (and vibration) transmission between cabins.

4.2.2. Treatment of Structure Borne Noise (Vibration)

Structure borne noise or rather vibration causes the most annoyance on board, especially when low frequent.
The principle of limiting the effect of vibrations is to disturb the propagation path of the waves. There are various ways of doing so.

Machinery
Beginning with treatment at the source means that sufficient measures should prevent as many vibrations as possible from getting through to the structure. For engines and gearboxes this means that mounts have to be chosen with great care. Knowing the characteristics and test results of the machine, vibration and noise levels can be predicted fairly well. This knowledge must be used to decide on the required reduction and to perform the necessary calculations on appropriate (elastic or flexible) mounts. A more extensive explanation and examples will be given in section 4.3.1.
Besides mounts, the foundation plays a significant role in vibration propagation. It is common knowledge amongst engineers that mass is very effective in minimising vibration effects. In yacht building mass is not very popular as solution to the problems. Additional mass means extra draught, which is disadvantageous with respect to safety since margins are kept at a minimum. Extra mass in the engine room means less mass elsewhere and thus amongst others limits the freedom of interior designers.

Geometrical barriers are very effective at larger distances from the source. Changing dimensions of parts of the construction on the main transmission path is one way to create a barrier. Putting in extra obstacles in the form of bulkheads and frames is another way to create barriers, but would once more mean unwanted weight addition. Physical barriers in the form of material changes are a third option to disturb the propagation.

At receiver locations again absorbing materials in floors and walls should provide the necessary protection. An important issue is the fitting of the double glazed windows in the walls.

Piping
Piping is a perfect transmission path for sound waves. Think about the heating system in a block of flats. If someone hits the duct at the first floor a person on the tenth floor can hear it as well. This has everything to do with the nature of ducts and the way they are fitted.

Putting an isolating/absorbing material around the ducts, rubber for instance, attenuates airborne noise quite easily.
The transmission of structure borne noise is more complicated. The points where ducts are connected to the rest of the construction (either machinery or connections to walls, ceilings, etcetera) remain weaknesses, no matter what measures are taken.
In a good piping plan, connections between ducts and machinery are flexible, as far as possible. Sharp curves and abrupt geometry changes are to be avoided in order to keep the flow through the ducts as continuous as possible. A constant (laminar) flow will have a positive effect on the noise and vibration level.

Accommodation
A good isolation of the accommodations against sound and vibration is of utmost importance, especially areas near the engine room require attention. Floors and walls are the main transmitters of sound and vibration here, thus need to be handled. Where

walls are generally provided with isolating/absorbing materials, like rockwool or a mineral wool, the floors are constructed as floating floors, of which the principle will be explained in section 4.4. For now, it will be sufficient to denote the required features of an effective floor construction[9]:

- good isolating properties with respect to mechanical vibrations;
- limited radiation of structure borne sound;
- good insulation properties with respect to impact noise;
- good properties with respect to insulation/absorption of airborne noise.

The accommodations require protection not only from noise coming from the machinery and other equipment, but also from noise coming from the neighbouring cabin. Target values for noise reduction from crew cabin to crew cabin differ from values for owner's cabins:

- crew-crew: 33[dB(A)]
- guest-guest: 40[dB(A)]
- owner-owner/guest: 40[dB(A)]

The isolating properties of a material can be determined by experiment, resulting in a *loss-factor*. This factor is defined as the ratio of the dissipated energy and the total amount of vibration energy in a system. The loss factor is frequency dependent. In formula it will look as follows [**De Regt, Akkermans,** 1995]:

$$\eta(\omega) = \frac{P_{dis}(\omega)}{\omega E_{tot}(\omega)} \qquad (4.1.)$$

$\eta(\omega)$= *internal loss factor* [-]
$P_{dis}(\omega)$= *dissipated energy as a function of* ω [J/s]
$E_{tot}(\omega)$= *total vibration energy in the system as a function of* ω [J/s]
ω= *frequency* [rad/s]

Table 4.1. gives the loss factors for some materials. The lower the value, the less dampening will take place. From the table it can thus be derived that aluminium is a very good conductor, as expected of course. Rubber is a good isolator with a loss factor varying from 0.1-0.3.

MATERIAL	LOSS FACTOR
Aluminium-pure	0.00002-0.002
Aluminium alloy-dural	0.0004-0.001
Steel	0.001-0.008
Lead	0.008-0.014
Cast Iron	0.003-0.03
Manganese Copper alloy	0.05-0.1
Rubber natural	0.1-0.3
Rubber hard	1
Glass	0.0006-0.002
Concrete	0.01-0.06

Table 4.1. Damping values for construction materials[10]

At a Dutch research institute, TNO, experiments have been performed to determine the loss factor of decks, hull and planar fields. The results are given in table 4.2.

[9] Source: ISSA '86 [J.Buiten]
[10] Source: Literatuuronderzoek naar Dempingscoëfficiënten bij Scheepstrillingen, M.Former

COMPONENT	LOSS FACTOR
Steel deck	0.005
Steel deck with floating floor	0.03
Steel deck with equalising layer	0.012
Hull	0.02
Planar fields	0.004
Planar fields with machines	0.01
Planar fields with *in plane* waves	0.001
Steel frames	0.008

Table 4.2. Loss factors of ship structure components

4.3. Implementation of Solutions at Sources

In the following the implementation of appropriate measures for the largest noise and vibration issues will be discussed. This includes flexible mounts, silencers for exhaust gas systems, acoustic enclosures for generator sets and floating floors and cavity walls for crew and guest accommodations. Examples of implementations are included as well.

4.3.1. Flexible Mounting of Engines and Gearboxes

In the last section resilient mounts were briefly mentioned as a means to isolate engines and gearboxes (and pumps also) from the ship's hull. They are indeed very effective, under certain conditions. To impose the right conditions onto the quality of the mounts a few things have to be known about the engine and reduction gearbox. Besides mechanical properties of the mounts, a strategic position of the mounts is essential to obtain the expected results.

As a thumb rule for good mounting the natural frequency of the mounts should be at least three times as small as the forced frequency, as explained in section 1.2.3. The forced frequency is the frequency of the dynamic load excited by the engine or gearbox on the mounts. Thus the loading conditions of engine and gearbox have to be known precisely, in order to choose appropriate mounting.
With respect to noise and vibration in the rest of the vessel, the effect of the system as a whole, engine and/or gearbox including the mounting, has to be studied. Measuring noise and vibration levels, for example, could be used to gather the necessary information. Measuring conditions are crucial for the results and should always be noted carefully.

Choosing an appropriate mounting system
Usually the desired reduction of sound level in [dB] or vibration amplitude in [mm/s] is known. The mounts capable to achieve this reduction are then evaluated.
There are basically two options to install an engine-gearbox combination successfully:
1. engine and gearbox could be separately mounted;
2. engine and gearbox could be single mounted together;

A third possibility is more common use for generator sets than for engine-gearbox combinations. In yacht building however, it is applied to engine-gearbox combinations as well:

3. engine and generator double mounted together.

A schematic representation of a separately mounted system is given in figure 4.3. A flexible coupling, which compensates for small misalignments, links gearbox and engine.

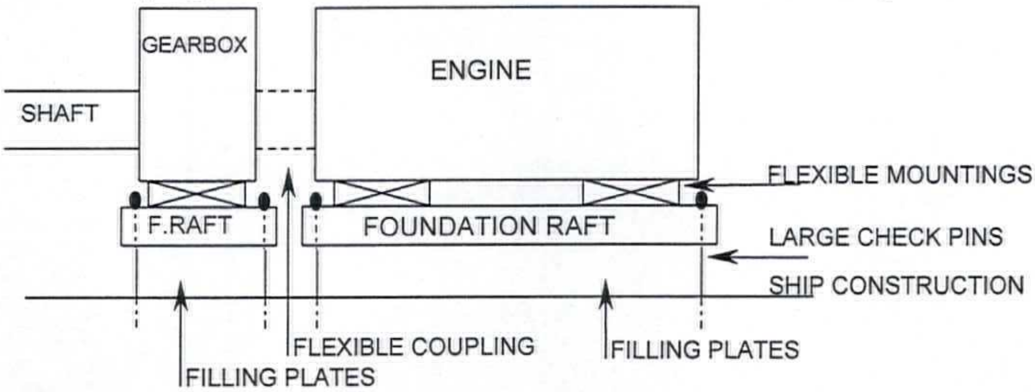


Figure 4.3. Example of a mounting system where engine and gearbox are separately mounted

All machinery is installed in the engine room when the ship is still on land. Once it has been launched, the ship will need a few days to settle into its new condition. Only after the ship has settled, the real outlining of the propulsion train can be executed. Until that time, large check pins are used to keep the foundation positioned in such a way that the mounts are under a certain pre-stress.
When the relative position of propeller shaft, gearbox and engine are found correct, filling plates are added to fill up the space between the rafts and ship's construction. These plates could be steel or composite and the connection is stiff. Finally the check pins are taken out and the mounts start functioning.

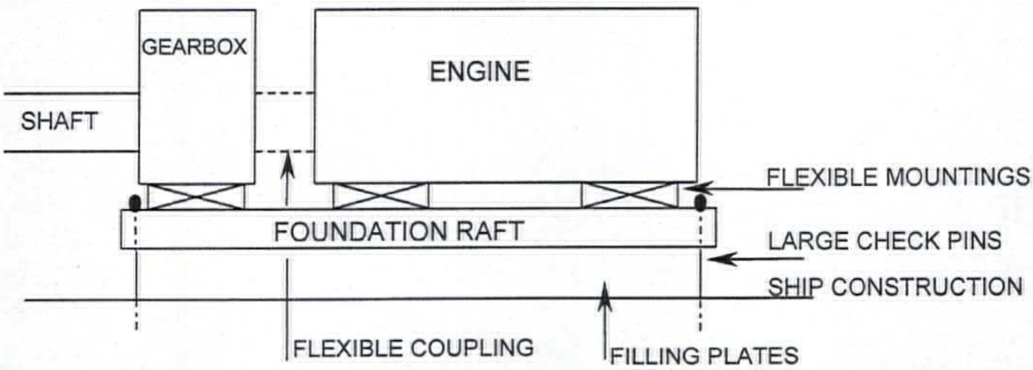


Figure 4.4. Example of a mounting system where engine and gearbox are mounted together

Figure 4.4. shows a configuration where engine and gearbox are together on one raft. This requires less strict conditions for the flexible coupling between engine and gearbox. The coupling could never be totally stiff however, because some movement will always be present. The installation and outlining will take place as described before.

Compared to the configuration presented in figure 4.3. the outline procedure for the configuration as shown in figure 4.4 requires less effort. Because engine and gearbox are already in correct position, the only issue is to get propeller shaft and gearbox into the desired position. This is of course a great advantage: it saves time and money.

A disadvantage is of course the extra mass that is added by using one large raft instead of two smaller ones. Especially when the engine sizes become considerable, the added mass will increase more than linear with engine (and gearbox) size. This is why in most cases, especially in yacht building, separate rafts are preferred.

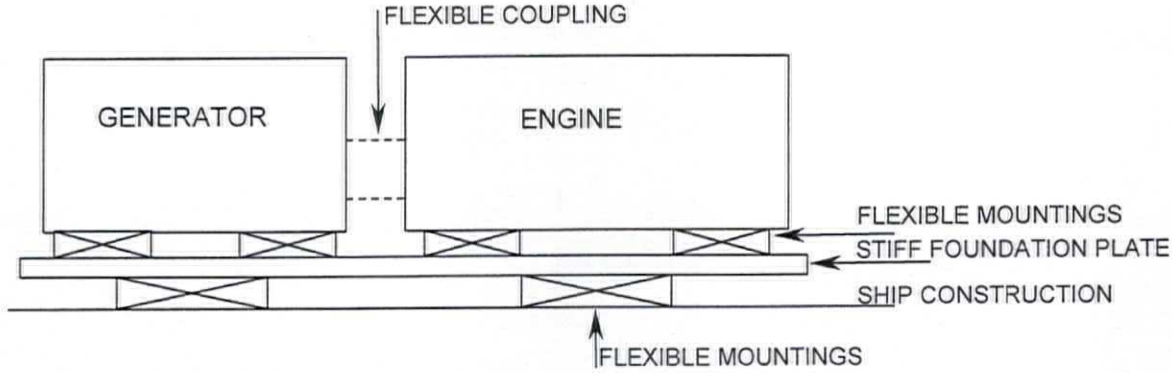


Figure 4.5. Example of a mounting system for a diesel generator set where the combination is double mounted

A great advantage of a double mounted system, shown in figure 4.5. as applied to one of Oceanco's yachts, is the reduction that can be achieved. The first reduction step may be relatively small, compared to the second step. Because there are two reduction phases the total reduction will be better controllable.
A disadvantage is the very heavy foundation plate that is placed between the mounts. This plate is necessary to create a stiff connection between the engine-generator set and the lower mounts. The mass of the plate includes so-called counterweight to compensate imbalance in the engine. At the yacht in question a counterweight of about 1000[kg] was used. Figure 4.6. shows the drawing of the double resiliently mounted generator set.

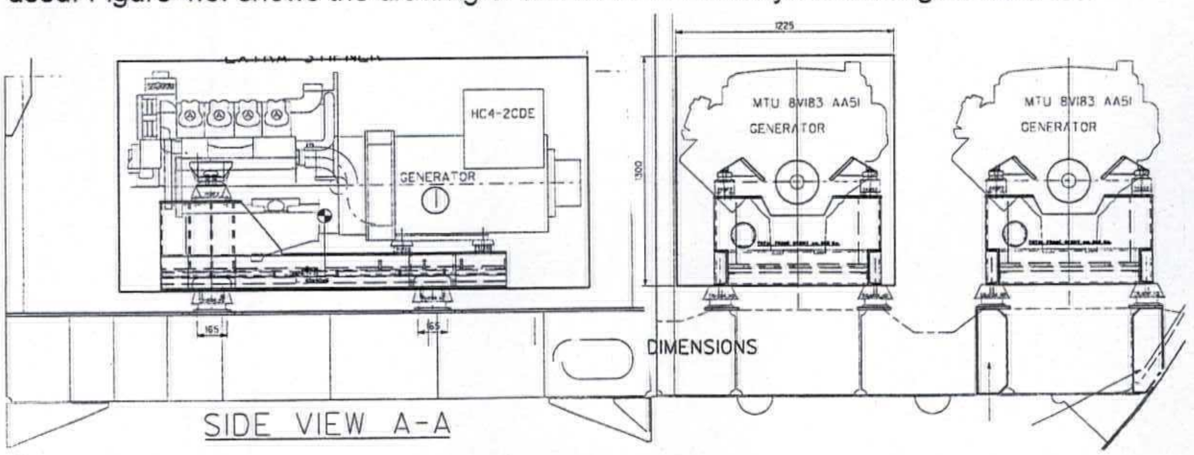


Figure 4.6. Double resiliently mounted generator set as applied on a 50[m]-yacht.

Installing the mounts under the machinery
A material that is often used in mountings is rubber. It is known for its excellent absorbing capacity of normal stress; for absorption of shear stress it is less suitable. The position of the mount should thus be so that shear stress is avoided as much as possible. Another thing is the extreme sensitivity of rubber to humidity and oily products. Finally rubber can only be applied in a low temperature range of up to 70[°C] continuous and 90[°C] peak.

For some cases higher temperatures are desirable. Then a special treated rubber suitable for temperatures of 110[°C] continuous loading and 130[°C] peak loading could be applied [**Rubber Design Catalogue**, 1996].
Analysis of the engine will give the directions and magnitudes of the free forces and moments. This knowledge and the knowledge of the mount properties are crucial for the choice of suitable mount positions.
On yachts it is common practice to use V-engines, because they do not weigh as much as the traditional in-line engines (L-configuration). Besides, the V-configuration is more compact than an equivalent L-configuration, which is a great advantage of course, considering the total available space. For the locations of the mounts this does not make a difference, however. Calculation of the free forces and moments in **Appendix 5** prove this.

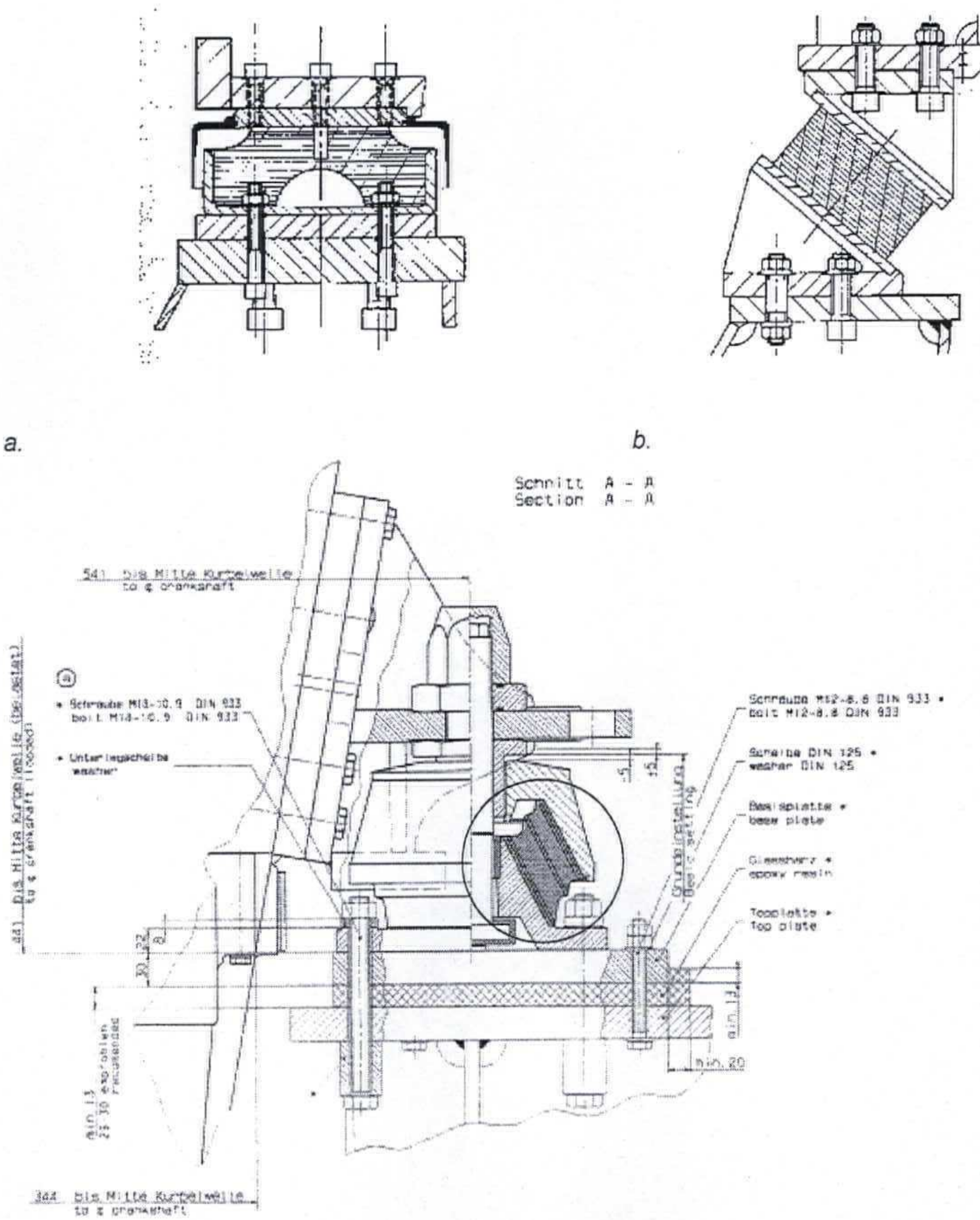
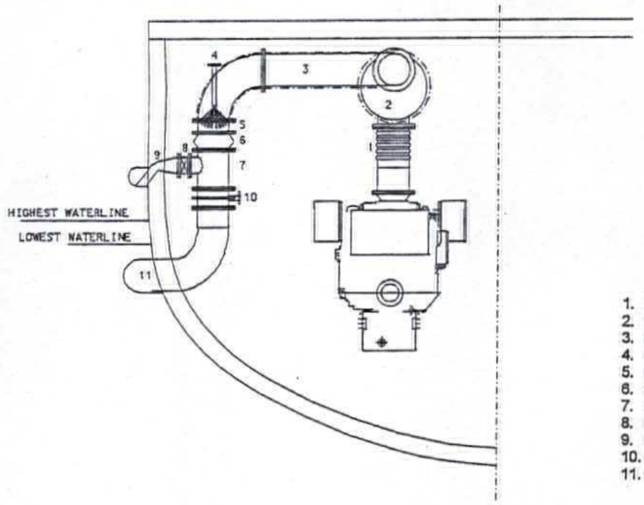


Figure 4.7. Examples of flexible mounts; a. positioned straight forward; b. under a certain angle; c. showing the cross section of a mount.

Figure (4.7.c.) shows a cross section of an engine mount, showing the extra interleaf ring that divides the rubber element into two parts, thus providing more stiffness. The circle marks the rubber element with interleaf ring.

4.3.2. Silencers for Exhaust Gas Systems

Exhaust gas systems can cause a lot of direct airborne noise. The only, really effective way to insulate the exhaust duct is a silencer. Reduction of the sound level is accomplished by absorption or resonance or by a combination of both principles.
Combining absorption and a resonance chamber is suitable for a wide frequency range.
Absorption is applicable in higher frequency ranges. It is based on energy conversion: kinetic energy in to thermal energy. The conversion is realised with help of an absorbing material, usually a mineral wool.
Resonance is based on the reflection of waves, in such a way that they eliminate each other (create anti-phases). The elimination is realised by using cells of different lengths, all connected to each other. This type of silencer mainly finds application in the lower frequency ranges.
To cover a wide frequency range the best option would be a silencer in which both principles are joined. The reductions gained with this type of silencer reach up to 35[dB(A)][11].



1. bellow
2. resonance/absorption silencer (insulated)
3. hot steel piping (insulated)
4. waterinjector
5. mixing section
6. rubber bellow
7. seawater resistant T section
8. bypass valve
9. bypass pipe
10. emergency valve
11. scoop

Figure 4.8. Example of an exhaust silencer system with underwater outlet

Optimal result will only be achieved if the following criteria are met:

- The silencer-choice must be based on the dominant frequency.
- The location of the silencer must be chosen very carefully, so that the length of the pipe after the silencer will not be too long, with respect to the wavelength. A badly chosen length could make the pipe function like an organ pipe due to resonance.
- The dimensions of the silencer are adjusted to the required damping characteristics.
- The gas speed may not exceed 50[m/s] to avoid excessive flow noise.

[11] Source: Discom Catalogue; this company specialises in silencers

The diameter of the silencer depends on the system's back-pressure. For a safe design a maximum permissible value is prescribed. This maximum value should be $\leq 60\%$ of the value permitted by the manufacturer.

4.3.3. Acoustic Enclosures for Generator Sets

Diesel generators produce high sound levels as well. On yachts they are normally put in acoustic enclosures, to eliminate at least the effect of airborne sound. An acoustic enclosure is a box large enough to wrap the complete diesel generator with clearances that are sufficient to establish the required sound level reduction and leave place enough for necessary maintenance tasks.
The enclosure consists of aluminium plates treated on the inside with a high quality absorbing material (Baryfol or something similar). In some cases this material consists of two layers of foam separated by a special type of plastic. The most inner foam layer then works as an absorber, while the outer one has a decoupling function[12].
It could also be that the diesel generator is placed in a separate room. On yachts this will seldom be the case, simply for lack of space.

The sound enclosures are always custom made. The maximum achievable reduction will depend on the space available for the diesel generator set as a whole, the available finances, and so on. Reductions up to 35[dB(A)] are not exceptional.

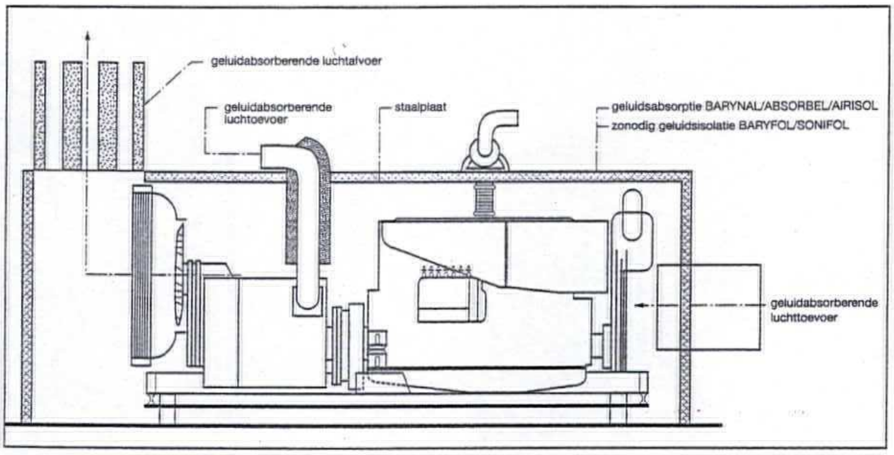


Figure 4.9. Example of a sound box with absorbing air inlets and exhausts

The acoustic barrier is meant mainly to reduce the effect of diesel generator fan noise.

[12] Source: Northern Lights Catalogue

4.4. Implementation of Solutions for Accommodation

4.4.1. Floating Floors

Between decks and accommodations another form of decoupling is implemented in the form of floating floors and cavity/insulated walls respectively. Floating floors normally consist of two layers, a lower layer of steel and a top layer of phonisol or something alike. On top of the phonisol a thin finishing layer may be constructed. An insulating material, usually rockwool or glass-wool or something similar, separates the layers from each other. This combination is referred to as *sandwich-construction.*
The principle of a floating floor is to keep the interior structure completely free from the surrounding steel construction, so that vibrations cannot be transmitted. Therefore all metal-metal connections between floor and supports need to be avoided by using layers of absorbing material in between them, as figure (4.10) illustrates. The accommodation is built on the thin finishing layer.

The effect of the floating floor is expressed in terms of a loss factor already briefly mentioned in section (4.2.2.). For practical purposes however *insertion/transmission loss and level difference* are far more common terminology. Both are expressed in [dB] and given in equations (4.2)and (4.3).

For the derivation of the equations, the floating floor was modelled as a system consisting of two masses and a spring. The equations hold for low frequencies, defined here as: $f \le 3f_0$.

With f_0 as the natural frequency of the top layer: $f_0 = \frac{1}{2\pi}\sqrt{\frac{s}{m_2}}$ (4.1)

$$\Delta L_a = 10\log\left(\frac{\left(1-(f/f_0)^2\right)^2+\eta^2}{1+\eta^2}\right) \tag{4.2}$$

$\Delta L_a = \textit{level difference between steel deck and top layer with respect to accelerations}\,[dB]$

$f = \textit{excitating frequency}\,[Hz]$

$f_0 = \textit{natural frequency of the top layer}\,[Hz]$

$\eta = \textit{loss factor of the insulating material}\,[-]$

$s = \textit{dynamic stiffness of } 1[m^2] \textit{ of the resilient layer} (\text{mod}\,\textit{elled as a spring})[N/m]$

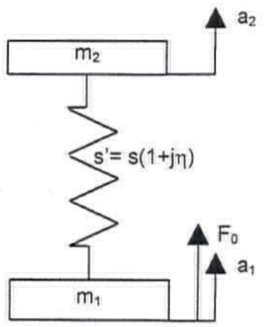
$$IL(a) = 10\log\left(\frac{\left(1-(f/f_0)^2+(m_2/m_1)\right)^2+\eta^2\left(1+(m_2/m_1)\right)^2}{1+\eta^2}\right) \tag{4.3}$$

$IL(a) = \textit{insertion loss between bare steel deck and top layer with respect to acceleration}\,[dB]$

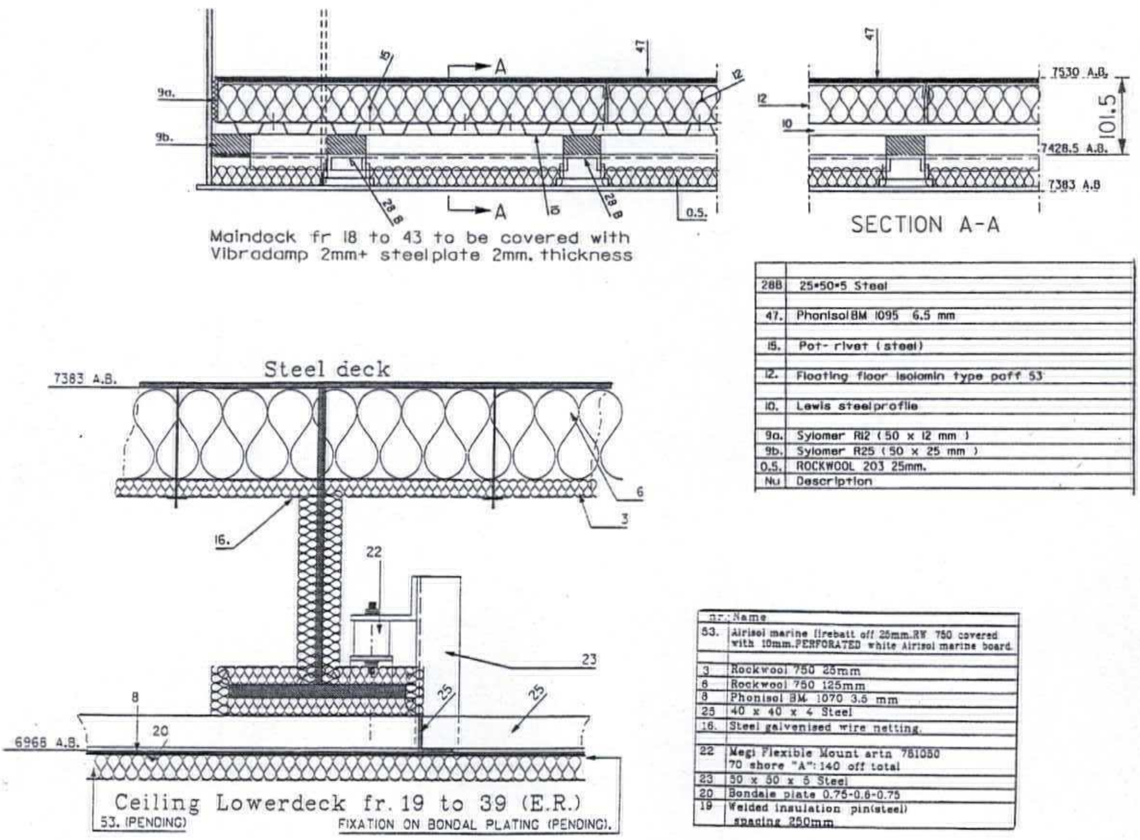
$m_1 = \textit{mass of (bare) steel deck}\,[kg]$

$m_2 = \textit{mass of top layer}\,[kg]$

$f, f_0 \textit{ and } \eta \textit{ as before}$



A floating floor modelled as a mass-spring system consisting of two masses, m_1 and m_2, representing the steel layer and top layer respectively. The spring in between represents the resilient layer. The spring constant is a complex quantity. The real part (s) stands for the dynamic stiffness of the material; the imaginary part ($s\eta$) stands for the internal damping.



Nu	Description
28B	25*50*5 Steel
47.	PhonisolBM 1095 6.5 mm
15.	Pot- rivet (steel)
12.	Floating floor isolomin type poff 53
10.	Lewis steelprofile
9a.	Sylomer R12 (50 x 12 mm)
9b.	Sylomer R25 (50 x 25 mm)
0.5.	ROCKWOOL 203 25mm.

nr.	Name
53.	Airisol marine firebatt off 25mm.RW 750 covered with 10mm.PERFORATED white Airisol marine board.
3	Rockwool 750 25mm
6	Rockwool 750 125mm
8	Phonisol BM 1070 3.5 mm
25	40 x 40 x 4 Steel
16.	Steel galvenised wire netting.
22	Megi Flexible Mount artn 781050 70 shore "A": 140 off total
23	50 x 50 x 5 Steel
20	Bondale plate 0.75-0.6-0.75
19	Welded insulation pin(steel) spacing 250mm

Figure 4.10. Example of a floating floor and of a ceiling construction as applied on board yachts

4.4.2. Wall Isolation

Cavity walls provide the necessary isolation between accommodations. The walls consist of two parts, separated by an air gap. Both parts are isolated with rockwool and covered with wooden plates, which have gone through a special treatment so that they are fire-resistant.
In situations where extra safety is required either one or both sides of the wall are of aluminium instead of wood. The walls are built as frameworks, the rockwool is fitted in each of the frames.

Dependent on how much the total reduction of noise should be, the densities of the applied materials and the air gap can be chosen larger or smaller, provided that space is available.
The walls are positioned in metal gullies on top of the floating floor to complete the floating structure.

The isolation class for specific spaces is stated by, amongst others, the IMO in its **S**afety **o**f **L**ife **a**t **S**ea book, of which a few pages are taken up in **Appendix 8** to give an idea what isolation class is about.

4.5. Résumé of Applied Techniques

Of course the yachts built at Oceanco Shipyards are fitted with the best thinkable isolating materials and mounts to fight any noise and excessive vibrations. This section should give some insight in the scope of an insulation plan.
Data required to do so were retrieved from the design data of the yachts and involve flexible mounts and couplings, materials that should provide additional mass, materials used for floating floors and cavity walls.

An isolation plan actually consists of three parts:
1. Fire protection
2. Acoustic insulation
3. Vibration insulation

In most cases a combination of at least two of the three parts is required to obtain the optimal situation. Depending on the location on board more or less fire protection is demanded. The demands could for instance be derived from the *Maritime and Coastguard Agency* codes [**Appendix 8**].

Fire protection between accommodations and between decks will mainly consist of steel and aluminium plates. The thickness of the plate determines its resistant endurance. Passages are fitted with steel fire doors, which close automatically when a fire alarm sets off at any location on board. The stairway is protected extra by two fire doors and a mainly steel construction.
To obtain the required reduction of sound and or vibration they are combined with rockwool, paroc or thermoheat, which is also fire resistant. The location determines whether one or both sides need to be treated.
Last but not least the thickness and density of the material are involved. The thickness is responsible for the total sound reduction.

Vibration isolation on other locations than at the machinery is still mainly reached by adding mass. The deck directly above the engine (room) is most important to be treated. A mass layer consisting of a material called vibradamp, in combination with either steel or aluminium tiles with a size of about 0.01-0.04[m^2], provides in that treatment. The tiles should be placed randomly otherwise the will not have any effect. Preferably they should be of different sizes as well. Figure 4.11. shows a detail-drawing.

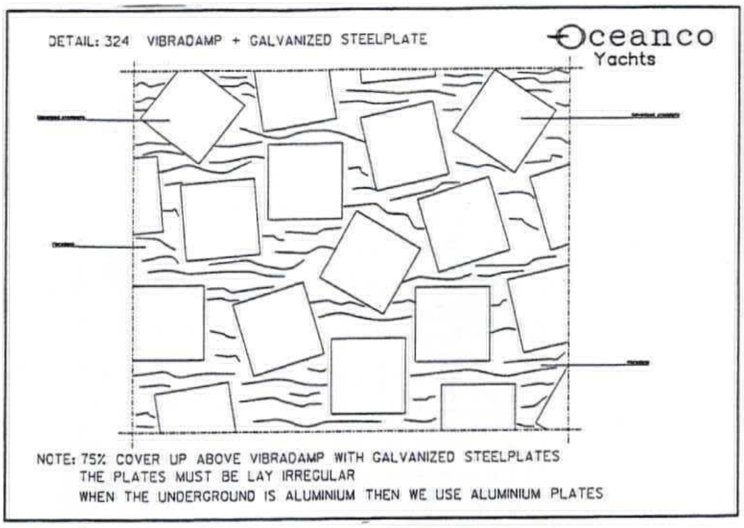


Figure 4.11. Detail drawing showing the application of vibradamp

Sound isolation is reached by installing cavity walls and floating floors throughout the entire ship, as described in section (4.4.). Floating floors are fitted on a framework, leaving an air gap between the floor and the actual deck plating, which is covered with rockwool.

Double glazed windows provide isolation from outside noise. They are described in **Appendix 7**.

4.6. Summary

Subject of this chapter was the actual implementation of noise and vibration reduction measures. The need for a proper action plan was emphasised in section one. A proposal was made, which included the right order of applying insulation. First of all at the source, then along the dominating propagation path and finally at the receiver. Another important theme was maintaining the quality standard of the insulation equipment by monitoring and controlling after installation.
In the sections following, available insulation tools were discussed. Flexible mounts, acoustic enclosures and floating floors were amongst them.
Finally a short résumé of insulation in practice at yachts was given. This résumé was not meant to be complete, but to give an idea about the scale of an insulation plan.

5 CONCLUSION

Recapitulation
Noise waves and vibration are complex physical phenomena as can be concluded from chapters 1 and 2, which only gave a superficial view on the theory behind these phenomena.
Distinction between sound and vibration is not always easy to make, so by rough estimation it can be said that transition from vibration to noise takes place at a frequency of approximately 80[Hz]. Noise is divided into structure borne and airborne. The former can have influence all over the ship, the latter is generally very local, but therefore not less important.

Even if not excessive noise and vibration determine for a great deal the comfort of passengers and performance of personnel on board. They thus deserve considerable attention, especially since these aspects have become more and more important over the years.

Noise and vibration on board are mainly caused by the propulsion system, including the propeller, and auxiliary engines. The character of the installed machinery, the materials used in its surroundings and throughout the ship and the quality of the implemented insulation tools influence the final noise and vibration levels.

Reductive devices are chosen on basis of analyses, calculation results and prediction by means of model-tests and experience. They consist of mounts and heavy foundations for main and auxiliary machinery; sound insulation in all areas in the form of cavity walls, carpets, curtains, etcetera; additional mass on weak spots to create extra protection against vibration.
In practice however, the effect of the measures is not always as expected from theory. Seatrial test results learned that deviations can be impressive.

Seatrial results also showed that the largest problems were caused by main engines and gearboxes and, in some cases, propellers and auxiliary engines. In short: structure borne noise. This was expected right from the start. It was however amazing to see that there were partly great differences between portside and starboard propulsion systems on the same ship, even though they were fitted equally (or at least should be). This raises questions with respect to the fitting of anti-vibration devices.

Further Investigation
In this report it was not yet explained how deviations between expected and actual results could arise. Further investigation on this topic will be performed hereafter. The investigation will focus on prognoses of noise levels in several areas on board. The areas are located at various distances from the source. The quality of the prognoses is estimated by comparing prognosis results with practice results.
The prognoses will be performed with a computer program called SEA (Statistical Energy Analysis), which was developed at the Dutch physical research institution, TNO.

For a good prognosis, knowledge of the (dominating) propagation path is indispensable. A study of wave propagation will thus be included in the continuation of the investigation on noise and vibration.

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NOMENCLATURE

Symbol	Description	Value	Unit
A	area		[m^2]
D	stiffness		
F	force		[N]
I	power intensity		[W/m^2]
L	sound level		[dB(A)], the letter in round brackets denotes the weighting network that is used.
P	power		[W]
R	radius		[m]
	resistance		[N]
T	torque		[Nm]
Y	mobility	[Z^{-1}]	[m/Ns]
Z	impedance	[k/ω]	[Ns/m]
a	acceleration		[m/s^2]
c	damping coefficient		[Ns/m]
d	diameter		[m]
f	frequency		[s^{-1}]
h	height		[mm]
k	spring constant		[N/m]
l	length of connecting rod		[m]
m	mass		[kg]
p	pressure		[Pa] or [bar][1]
r	distance from sound source		[m]
	radius		[m]
s	displacement		[m]
t	time		[s]
	thickness		[mm]
v	velocity		[m/s]
w	width		[mm]
x	displacement		[m]
z	number of cylinders		[-]

[1] 1[bar] = 10^5[Pa]

Greek symbol		
Δ	difference between two quantities	
α	(rake) angle	[rad]
	relative crank arm angle	
β	transmissibility factor	[-]
γ	V-angle of an engine	[rad]
δ	displacement factor	[-]
	ignition angle	[rad]
ζ	internal damping of material	
η	frequency ratio	[-]
	loss factor	[-]
	efficiency	[%]
ξ	damping ratio	[-]
ρ	density	[kg/m^3]
τ	time constant	[s]
ω	angular speed	[rad/s]

Subscript	
I	indication for sound intensity level
W	indication for sound power level
a	acceleration
clear	clear glass (Appendix 6)
corr	correction
hor	horizontal
i	the ith number of a series
ign	ignition
nat	natural
p	indication for sound pressure level
ref	reference
rel	relative
req	required
rot	rotational
s	centre of gravity
smoked	smoked glass (Appendix 6)
sum	summed value of two or more quantities
tot	total
trans	transmitted
vap	vapour
vert	vertical
0	reference or initial value

Abbreviations

Acc	accommodation
ECR	engine control room
IMO	International Maritime Organisation
MCA	Maritime Coastguard Agency
MCR	maximum continuous rating
ME	main engine
MS	midship
NR(C)	noise rating (curve)
PAC	passenger accommodation comfort
PHV	propeller hull vortex
PS	portside
RMS	root mean square
SB	starboard
SOLAS	Safety of Life at Sea
TE	transmission error
acc	according (to)
approx	approximately
arr	arrangement

APPENDICES

APPENDIX 1: EQUATIONS OF MOTION FOR A (RESILIENT) MOUNTING SYSTEM[1]

A1.1. One Degree-of-freedom-system

The dynamic behaviour of such a system is described by linear equations, in principle based on Newton's second law, but represented a little differently to suit the application field. The mounting model is described with help of impedances and mobilities (which are the reverse of impedances).
The validity of such an equation has to be evaluated for each individual case, considering the assumptions that may be made.

The general model for a one-degree-of-freedom-system is illustrated in figure A1.1.

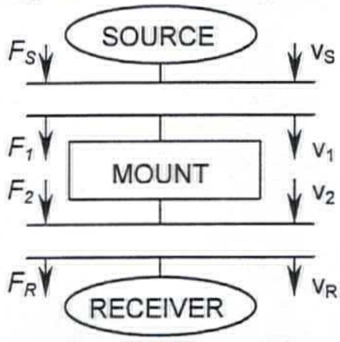

Figure A1.1. One degree of freedom-model for resilient mounting system

The system is determined by F_1 and F_2, so two equations are to be solved. A matrix denotation is most practical for these kinds of problems.
Its impedance, like mentioned before expresses the effectiveness of the mount.
Impedance is defined as:

$$Z_i = \frac{k_i}{\omega_i} \qquad \text{(a1.1)}$$

where
$k = stiffness\ of\ the\ mount\,[N/m]$
$\omega = excitation\ frequency\,[rad/s]$

The mobility is denoted by Y_i and thus defined as:

$$Y_i = \frac{\omega_i}{k_i} \qquad \text{(a1.2)}$$

For the system in figure A1.1. the matrix equation looks like:

[1] Source: Fischer, R. and L. Boroditsky, 2001

$$\begin{bmatrix} F_1 \\ F_2 \end{bmatrix} = \begin{bmatrix} Z_{11} & Z_{12} \\ Z_{21} & Z_{22} \end{bmatrix} \begin{bmatrix} v_1 \\ v_2 \end{bmatrix} \quad \text{(a1.3)}$$

with Z as defined in (a1.1.). Furthermore $v_1=v_S$, $v_2=v_R$, $F_1=-F_S$ and $F_2=-F_R$, as follows from the model.

The free velocity of the source, which depends on the internal forces in the source, is obtained from the following equation:

$$v_{FS} = v_S - Y_S F_S \quad \text{(a1.4)}$$

with Y as in (a1.2.) and v_{FS} the free velocity of the source.

At the receiver a noise velocity can be defined, which takes into account the background noise that may be present.

$$v_{NR} = v_R - Y_R F_R \quad \text{(a1.5)}$$

Combining equations (a1.1.) to (a1.5.) will, after a little rearranging, finally lead to:

$$\begin{bmatrix} 1 & -Y_S & 0 & 0 \\ Z_{11} & 1 & Z_{12} & 0 \\ Z_{21} & 0 & Z_{22} & 1 \\ 0 & 0 & 1 & -Y_R \end{bmatrix} \begin{bmatrix} v_S \\ F_S \\ v_R \\ F_R \end{bmatrix} = \begin{bmatrix} v_{FS} \\ 0 \\ 0 \\ v_{NR} \end{bmatrix} \quad \text{(a1.6)}$$

The source force and velocity as well as the receiver force and velocity can be solved by matrix inversion.

The equations derived are valid only for the case that $e^{j\omega t}$ is omitted.

In the following two specific mounting conditions are discussed for the one degree of freedom system.

Rigidly mounted

In that case the matrix equation rigorously reduces by total absence of mounts. The effect of the source vibrations is directly transferred to the receiver, applying:

action = - reaction

This results in: $F_S = -F_R$ and $v_S = v_R$

and for the matrix equation:

$$\begin{bmatrix} v_S^r \\ F_S^r \end{bmatrix} = \begin{bmatrix} v_R^r \\ -F_R^r \end{bmatrix} = \frac{1}{Y_S + Y_R} \begin{bmatrix} Y_R & Y_S \\ -1 & 1 \end{bmatrix} \begin{bmatrix} v_{FS} \\ v_{NR} \end{bmatrix} \quad \text{(a1.7)}$$

Resiliently mounted

Equation (a1.6.) of course already describes a resiliently mounted system, but only for the most general case. For a well designed system it is assumed that the products of transfer impedances of the mounts and mobilities of source and receiver are way smaller than 1, in case that the frequency of the system is above the natural frequency. The matrix equation may then be simplified, leaving:

$$\begin{bmatrix} v_S^s \\ F_S^s \\ v_R^s \\ F_R^s \end{bmatrix} \approx \begin{bmatrix} 1 & Y_S & -Y_S Z_{12} Y_R & -Z_{12} Y_S \\ -Z_{11} & 1 & -Z_{12} Y_R & -Z_{12} \\ -Z_{21} Y_R & -Y_S Z_{21} Y_R & Y_R & 1 \\ -Z_{12} & -Z_{21} Y_R & 1 & -Z_{22} \end{bmatrix} \begin{bmatrix} v_{FS} \\ 0 \\ 0 \\ v_{NR} \end{bmatrix} \qquad \text{(a1.8)}$$

A1.2. Effectiveness of the Mounting System

The objective of a mounting system is of course to reduce the transmission of vibration as much as possible. To establish the quality of a mounting system there has to be some judgement criterion. Well, there are two actually. One is called the *transmissibility* and focuses on that part of the energy that remains after the mount, the other is called the insertion loss, establishing how much of the energy is prevented from propagating any further than the mount. The mathematical formulas common to describe either transmissibility or insertion loss are given next.

Transmissibility

The formulas are valid only under the assumption of a fully rigid source and an ideal (linear) spring. The natural frequency for such a system would be:

$\omega_0 = \sqrt{\dfrac{k}{m_{rigid}}}$, which should look very familiar.

k = stiffness of the ideal spring

m_{rigid} = mass of the rigid source

Consequently resonance occurs when $\omega = \omega_0$, instead of rotational speeds, linear speeds may be used to express the so-called *critical speed.*
Doing so results in:

$$\frac{v_R}{v_S} \approx \frac{k}{j\omega} Y_R$$

It is however common practice to express transmissibility as function of the frequency ratio:

$$F_R \approx \frac{F_0}{1 - \left(\dfrac{\omega}{\omega_0}\right)^2} \qquad \text{(a1.9)}$$

From this equation it becomes clear why the forced frequency should be considerably larger than the natural frequency to be of any relevance. In the field of acoustic vibrations it seems logical to communicate in dB-level rather than in Newton. The force level denoted by L_F is defined as:

$$L_F = 20\log|F_R| \qquad [dB] \qquad \text{(a1.10)}$$

Since dB-levels are always relative, the reference force is defined as 1[N].

Insertion loss

Insertion loss is related to the power of a sound.

The expression, which describes the velocity here, is a complex equation. Therefore, the expression describing the total power of the sound is complex as well, since $P = Fv$. To determine the amount of power that is transmitted, only the real part of the complex equation is required. Thus:

$$P_R = \frac{1}{2}\mathrm{Re}(F_R v_R) \qquad \text{(a1.11)}$$

The factor ½ indicates that the mean transmitted power is used.

APPENDIX 2: CONVERSION OF WEIGHTED NOISE LEVELS

Noise levels can be expressed in octave bands, 1/3-octave bands, A, B, C or D weighted levels or even in NR numbers. Each expression focuses on a certain part of the audible frequency range. All methods have in common that they indicate how observers experience a certain sound. There could be cases of course where the applied weighting scale does not give the information wanted. Another possibility is that two situations are to be compared, but measurements took place with different weighting scales.
For these situations it is desirable to convert one scale into another scale. The calculation rules to do so follow next.

A2.1. Relation between Sound Power, Pressure and Intensity Level

First, the basic ways to express a sound level are treated. This includes sound pressure level, sound power level and sound intensity level.

Pressure and power level are related in the following way:

$$L_W = L_p + 10\log\left(\frac{A}{A_0}\right) \quad \text{(a2.1.)}$$

L_W = *sound power level in [dB]*
L_p = *sound pressure level in [dB]*
A = *area of the measuring surface in [m²]*
A_0 = *reference area of the measuring surface 1[m²]*

For this case the measuring surface is supposed to be a sphere, thus having a surface area: $A = 4\pi r^2\,[m^2]$.
The derivation of this relation becomes clear after rewriting the terms a bit. The intensity of a sound wave is defined as the power of the wave per unit area:

$$I = \frac{P}{A}\ [W/m^2] \quad \text{(a2.2.)}$$

I = *intensity in [W/m²]*
P = *power in [W]*
A = *area in [m²]*

From (a2.2.) it can easily be derived that:

$$P = IA \quad \text{(a2.3.)}$$

Intensity level can also be expressed as a function of pressure and velocity:

$$I = \frac{p^2}{\rho v} \quad \text{(a2.4.)}$$

p = *pressure in [N/m²]*
ρ = *density of the medium in [kg/m³]*
v = *velocity of the sound wave in [m/s]*

A sound level always represents a relative level. Thus, the sound power level L_W should be written as:

$$L_W = 10\log\left(\frac{P}{P_0}\right) \quad \text{(a2.5a.)} \qquad \text{or} \qquad L_W = 10\log\left(\frac{IA}{I_0 A_0}\right) \quad \text{(a2.5b.)}$$

W_0 = reference power of 10^{-12}[W]
I_0 = reference intensity 10^{-12}[W/m^2]
A_0 = reference area of 1[m^2]
Equation (a2.5b.) can be rearranged according to the calculation rules for logarithms, which state that *log(ab)=log(a)+log(b)*. This results in:

$$L_W = 10\log\left(\frac{I}{I_0}\right) + 10\log\left(\frac{A}{A_0}\right) \quad \text{(a2.6.)}$$

Now suppose that ρ and v from equation (a2.4.) are constant, or at least ρv is constant. For that case the sound intensity level would coincide with the sound pressure level, leading back to equation (a2.1).

A2.2. Sound Level Weighting

Sound levels measured in (1/3-) octave bands can be calculated in A, B, C, and D-weighted levels, by using the correction values given in Table A2.1.

1/3-Octave Band Centre Frequency	A-weighting Correction Values	B-weighting Correction Values	C-weighting Correction Values	D-weighting Correction Values
50	-30.2	-11.6	-1.3	-12.8
63	-26.2	-9.3	-0.8	-10.9
80	-22.5	-7.4	-0.5	-9.0
100	-19.1	-5.6	-0.3	-7.2
125	-16.1	-4.2	-0.2	-5.5
160	-13.4	-3.0	-0.1	-4.0
200	-10.9	-2.0	0.0	-2.6
250	-8.6	-1.3	0.0	-1.6
315	-6.6	-0.8	0.0	-0.8
400	-4.8	-0.5	0.0	-0.4
500	-3.2	-0.3	0.0	-0.3
630	-1.9	-0.1	0.0	-0.5
800	-0.8	0.0	0.0	-0.6
1000	0.0	0.0	0.0	0.0
1250	0.6	0.0	0.0	2.0
1600	1.0	0.0	-0.1	4.9
2 000	1.2	-0.1	-0.2	7.9
2 500	1.3	-0.2	-0.3	10.6
3 150	1.2	-0.4	-0.5	11.5
4 000	1.0	-0.7	-0.8	11.1
5 000	0.5	-1.2	-1.3	9.6
6 300	-0.1	-1.9	-2.0	7.6
8 000	-1.1	-2.9	-3.0	5.5
10 000	-2.5	-4.3	-4.4	3.4
12 500	-4.3	-6.1	-6.2	-1.4

Table A2.1. Correction values for A, B, C and D weighting

Especially A- and B- weighting focus on reduction of the influence of low level sounds, while D-weighting besides reduction in the lower frequency range also focuses on emphasising the influence of sound levels within the speech-range
(from 3 000 – 8 000[Hz]).According to C-weighting all frequencies up to 10 000[Hz] are equally important, significant compensations are made only above 10 000[Hz].
A calculation example is given in section 1.2.2.
The main reason for weighting levels is to create a level range that matches the human hearing sensitivity as closely as possible.

In case NR levels are preferred above A-level values or octave band levels, these values have to be converted in accordance to the method described here. Table A2.4. gives correction values for different kinds of noise, which involve structure borne noise of propellers, engines and gearboxes with or without floating floor and airborne noise for exhausts or other sources.
Parallel to that, the A-level values as obtained from the measurements should be summed to determine the overall A-weighted level. This level is to be diminished with 6. The larger of these two values will be assigned as the NR level for that sound spectrum.

A critical note to the above: the table only mentions corrections for a very limited number of possibilities and the information may be backdated, because the source that has been used was published in 1983.

A2.3. Definition of Root Mean Square Value

Determination of an overall sound level is often done by use of root mean square values, because average pressure gives a more trusting idea of the energy contents of the wave.
With respect to sound waves it is very acceptable to consider the pressure function as a sine form. The average of any sine function is of course zero, the average of the square of a sine form on the other hand is not. This square form turned out to be very useful in the field of sound measuring.
The root mean square is found from equation (a2.7.), the peak value is found by multiplying the average value by √2. This will be proven in equations (a2.8.) till (a2.11.).

$$p_{rms} = \sqrt{\frac{1}{T}\int_0^T p^2(t)dt} \qquad \text{(a2.7.)}$$

Assume that *p(t)* is a sine function of the following general form:

$$p(t) = p_{\max}\sin(\omega t + \varphi) \qquad \text{(a2.8.)}$$

With

$p_{\max}$ $= pressure\ amplitude\,[Pa]$

ω $= sound\ wave\ angular\ velocity\,[rad/s]$

t $= time\,[s]$

φ $= phase\ shift\,[rad], assumed\ 0\,[rad]\ in\ this\ case$

equation a2.8. reduces to

$$p(t) = p_{\max}\sin(\omega t) \qquad \text{(a2.9.)}$$

Equation (a2.9.) will now be substituted into the integral in (a2.7.), which will be worked out for the general case. With help of goniometry rules the square of the sine function is rewritten to a more usable form.

$$\sin^2 \alpha = \frac{1}{2} - \frac{1}{2}\cos 2\alpha$$

thus

$$\sin^2 \omega t = \frac{1}{2} - \frac{1}{2}\cos 2\omega t = \frac{1}{2}(1 - \cos 2\omega t)$$

and

$$\begin{aligned} p_{rms} &= \sqrt{\frac{1}{T}\int_0^T p^2(t)dt} \\ &= \sqrt{\frac{1}{T}\int_0^T (p_{\max}^2 \sin^2 \omega t)dt} \\ &= \sqrt{\frac{1}{2T} p_{\max}^2 \int_0^T (1 - \cos 2\omega t)dt} \end{aligned} \tag{a2.10}$$

First the integral is worked out for the case that $T = \frac{2\pi}{\omega}$, which will lead to:

$$\int_0^T (1 - \cos 2\omega t)dt = \left[t - \frac{1}{2\omega}\sin \omega t \right]_0^{\frac{2\pi}{\omega}} = \frac{2\pi}{\omega} \tag{a2.11}$$

Substitution of this result into equation (a2.10.) will give the final result for p_{rms}.

$$\begin{aligned} p_{rms} &= \sqrt{\frac{1}{2T} p_{\max}^2 \int_0^T (1 - \cos 2\omega t)dt} \\ &= \sqrt{\frac{\omega}{4\pi} p_{\max}^2 \frac{2\pi}{\omega}} = \sqrt{\frac{1}{2} p_{\max}^2} = \frac{1}{\sqrt{2}} p_{\max} \end{aligned}$$

Unless specifically mentioned otherwise, sound level meters always show root mean square values.

A2.4. Sound Level Regulations

On board vessels regulations concerning maximum permitted sound levels apply. There are no specific rules stated for private yachts. In that case however, the regulations from Lloyd's are taken as committing guidelines, especially while they have to approve the ship.
To give a better idea on the stringency of the levels, regulations for other kind of ships, such as special service crafts and cargo ships, are included.
Usually the permitted levels are set 5 [dB] below the values determined by Lloyd's to be certain at all times.

Acceptable sound levels for several vessel types according to the rules
Tables A2.2., A2.3. and A2.4. will give values of permitted sound levels on board of, amongst others, pleasure crafts and duty vessels.

LOCATION	PERMITTED SOUND PRESSURE LEVEL [dB(A)]
Crew Cabins	55
Engine Room	110
Engine Control Room	75
Exterior Aft Main Deck	75
Guest Cabins Lower Deck (forward)	55
Guest Cabins Lower Deck (aft)	50
Main Lounge	55
Owner's Stateroom	50
Sky Dining Lounge	50
Wheelhouse	55

Table A2.2. Sound Regulations for Pleasure Crafts

Another very influential body is the International Maritime Organisation. The maximum acceptable sound pressure levels stated by them are taken up in table A2.3. below. These are levels for vessels in general, not in particular for pleasure crafts.

LOCATION	ACCEPTABLE SOUND PRESSURE LEVEL [dB(A)]
Work Spaces	
Machinery spaces (not cont. manned)	110
Machinery control rooms	75
Workshops	85
Navigation Spaces	
Navigating bridge and chartrooms	65
Listening post	70
Radio rooms	60
Radar rooms	65
Accommodation Spaces	
Cabins and hospitals	60
Mess rooms	65
Recreation rooms	65
Open recreation areas	75
Offices	65
Service Spaces	
Galleys, without equipment operating	75
Pantries	75
Normally Unoccupied Spaces	90

Table A2.3. Acceptable sound pressure levels as defined by the IMO

/1/ Bekendmaking aan de Scheepvaart no. 213/1987 (Voorschriften betreffende geluidniveaus aan boord van schpen); Ministerie van Verkeer en Waterstaat, Directoraat-Generaal Scheepvaart en Maritieme Zaken, 1987

/2/ Code of practice for noise levels in ships; Department of Trade; London 1978

/3/ Unfallverhütungsvorschrift Lärm für Seeschiffe; See-Berufsgenossenschaft Hamburg, 1979

/4/ Sjöfartverkets bestämmelser och rekommendationer om skydd mor buller pa fartyg; Sjöfartsverkets Meddelanden, 1972

/5/ Forskrifter on vern mot støj om bord i skip; Sjøfartdirektoratet, 1973

/6/ Forslag til bekendgørelse om forskrifter mod støj i skibe; Lovtitend a. no. 258, juni 1975

/7/ Highest permissible noise levels in crew accommodation aboard ships; Finnish National Board of Labour Protection, 1976

/8/ Vorläufige Vorschriften für die Lärmbekämfung auf Schiffen; Deutsche Schiffs-Revision und -Klassifikation, 1964

/9/ Grenswaarden voor toelaatbare geluidniveaus op zeeschepen en voorschriften ter voorkoming van schadelijke invloeden door geluid; CCCP 24/IX 1962 r. No. 416-62

/10/ Grenswaarden voor telaatbare geluidniveaus op zeeschepen; CCCP 11/XII-1981 r, Moskou 1981

/11/ Rules of Polish Ministry of Shipping, 1973

/12/ Code of practice for noise levels in ships; IMCO-dokument DE XXIII/14, bijlage 3, 1981

/13/ SNAME-project HS-7-1, Ship Vibration and noise guidelines (draft); Society of Naval Architects and Marine Engineers, augustus 1978

/14/ SNAME-project HS-7, Ship vibration and noise guidelines; Society of Naval Architects and Marine Engineers, 1980

/15/ K.D. Harford, "Development of the proposed Canadian noise level regulations for vessels engaged in towing"; Consulting engineering division of Aero Acoustic Systems Ltd, Vancouver, BC

/16/ Recommended maximum noise and vibration levels; Shipping notice no. SHN 4/76, State of Israel, Ministry of Transport, Department of Shipping & Ports, 1976

/17/ Recommendation on methods of measuring noise levels at listening posts; IMCO-Resolutie A. 343 (IX)

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Scheidingsconstructie	IMO I_a[1]	Nederland R_w[9]	West-Duitsland R'_w[2]	Noorwegen R[3]	Zweden I_a[1]	Denemarken I_a[1]
tussen hutten onderling	30[4]	30[4]	30[5]	30[6]	35[7]	30
tussen hutten en gang	--	--	30	30	30	30
tussen hutten en mess/dagverblijf	45	45	45	45	45	45
tussen hutten en keuken	--	--	--	45	45[8]	45
tussen hutten en machinekamer	--	--	--	50	--	45
tussen hut en ruimte waarvoor geluidniveaus hoger dan 70 dB(A) zijn toegestaan	--	--	--	40	45[8]	--
tussen accommodatie en open dekken	--	--	--	40	--	40
dek in de accommodatie	--	--	--	--	--	40

Noten:

1. volgens ISO R717 te meten in laboratorium volgens ISO R 140
2. volgens DIN 52210 te meten in laboratorium
3. geluidisolatie bij 500 Hz
4. eis is dat aan boord niet significant lagere waarde optreedt
5. aan boord, gemeten aan gesloten wanden, geen lagere waarden dan R'w -3 dB
6. niet lager dan 5 minder dan de gegeven waarden ten gevolge van ventilatie-openingen, deuren etc.
7. voor vaartuigen kleiner dan 3000 BRT: I_a = 30 dB
8. ten opzichte van geluidniveaus lager dan 85 dB(A)
9. Volgens ISO R717/1-1982 te meten in laboratorium volgens ISO R140/III

Voor de realisatie van deze grenswaarden raadplege men A810020, A810021 en A810022.

010031 Grenswaarden voor het geluid aan boord van zeegaande schepen (2)

Ruimten	Nederland		Groot-Brittannië		Duitsland BRD		Zweden	Noorwegen		Denemarken		Finland	Duitsland DDR	USSR		Polen	
	/1/		/2/		/3/		/4/	/5/		/6/		/7/	/8/	/9/	/10/	/11/	
	dB(A)[16]	NR	dB(A)	NR	dB(A)[2]	NR	dB(A)[3]	dB(A)[2]	NR	dB(A)	NR	dB(A)	NR	dB(A)[3]	dB(A)[3]	dB(A)	NR
Accommodatie	1987 G*)		1978 R		1981 G		1973 G	1974 G		1975 G		1976 G	1964 G	1962 G	1975 R	1973 G	
slaaphutten	60	55	60	55	60[4]	55	55	60	55[5]	60	55[5]	60	55	50	50	60	55
dagverblijven,																60	55
eetkamers	65	60	65	60	65	60	65	65	60[7]	65	60[7]	60	55	50	55	65	60
kantoren	65	60	65	60	65	60	55[8]	65	60	--	--	60	55	50	55	60	55
hospitalen	60	55	60	55	60	55	--	--	--	--	--	60	50	50	45	55	50
keukens	75	70	70[9]	65	70	65	65[10]	70[11]	65	70	65	--	--	65	55	75	70
pantries	75	70	70	65	70	65	65[10]	70[11]	65	70	65	--	--	--	70	80	75
recreatie-ruimten	65	60	65	60	60	55	65	70	65	70	65	--	55	--	--	65	60
hobbyruimten	--	--	65	60	70	65	65	70[11]	65	70	65	--	--	--	--	--	--
gangen, badkamers	--	--	80	75	--	--	--	--	--	--	--	--	--	--	--	--	--
Machinekamers																	
indien controle-																	
ruimte aanwezig	110	105	110	105	100	105	100	110	105	110	105	--	--	100	105	--	--
permanent bezet	90	85	90	85	90	85	85	90	85	90	85	--	85	85	85	90	85
controlekamer	75	70	75	70	75	70	70	75	70	75	70	--	75	70	65	75	70
werkplaatsen	85	80	90	85	90	85	75[13]	85	80	85	80	--	85	70	70	--	--
magazijnen	90	85	90	85	--	--	75	90	85	--	--	--	85	--	--	90	85
wasruimten	--	--	80	75	75	70	--	--	--	--	--	--	--	--	--	--	--
Navigatieruimten																	
brugvleugels	70	65[14]	--	--[14]	70	--[14]	70	70	65	70	65[15]	--	--	65	--	--	--
stuurhuis, kaar-																	
tenkamer	65	60	65	60	65	60	65	65	60	65	60	--	55	65	55	65	60
radiostation	60	55	60	55	60	55	55	65	60	65	60	--	55	65	--	60	55
radarstation	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--	--
Open dekken																	
recreatiegebieden	--	--	75	70	70	65	65	70	65	--	--	--	--	--	--	--	--
werkgebieden	--	--	90	85	--	--	65	--	--	--	--	--	--	--	--	--	--
Bedrijfsconditie	80% MCR **)		80% MCR		100% MCR		80% MCR	80% MCR		80% MCR		onbekend	100%	Niet gegeven		onbekend	

010031 Grenswaarden voor het geluid aan boord van zeegaande schepen (4)

Ruimten	IMO		USA		Canada	Israël
	/12/		MARAD /13/	SNAME /14/	/15/ towing vessels	/16/
	dB(A)[16]	NR	dB(A)	dB(A)	dB(A)	NR
Accommodatie	1981 R		1978 V	1980 V	1969 R	1976 R
slaaphutten	60	55	60	60	70	55
dagverblijven,			60	65	74	--
eetkamers	65	60	65	65	74	65
kantoren	65	60	60	--	--	--
hospitalen	60	55	--	--	--	--
keukens	75	70	--	--	74	--
pantries	75	70	--	65	--	--
recreatie-ruimten	65	60	65	--	74	65
hobbyruimten	--	--	--	--	--	--
gangen,badkamers	--	--	--	--	--	--
Machinekamers						
indien controle-ruimte aanwezig	110[17]	105	--	--	--	--
permanent bezet	85	80	85	85	--	85
controlekamer	75	70	75	75	--	70
werkplaatsen	85	80	85	85	--	75
magazijnen	85	80	85	85	--	85
wasruimten	--	--	--	--	--	--
Navigatieruimten						
brugvleugels	70	65[14]	--	--	--	--
stuurhuis, kaartenkamer	65	60	65	65	--	60
radiostation	60	55	60	60	--	60
radarstation	65	60	--	--	--	--
Open dekken						
recreatiegebieden	75	70	--	--	--	--
werkgebieden	80[18]	--	--	--	--	--

010031 Grenswaarden voor het geluid aan boord van zeegaande schepen (5)

Noten:

1. alleen globale controle-waarde; voldaan dient te worden aan het NR-getal
2. indien het geluidniveau-A wordt overschreden dienen octaafbandspectra te worden bepaald en getoetst aan NR-waarden
3. tevens zijn grenswaardencurven gegeven
4. geldig voor het niveau verkregen als rekenkundig gemiddelde van de geluidniveaus in alle verblijven per dek; voor individuele hutten is een overschrijding met 3 dB(A) toegestaan
5. voor de octaafbanden 31,5 en 63 Hz is NR-60 toegestaan
6. voor eetkamers eventueel NR-65 mits het NR-getal in één van de octaafbanden 500, 1000 of 2000 Hz niet hoger is dan 55
7. voor de octaafbanden 31,5 en 63 Hz is NR-65 toegestaan
8. bij gestopte voortstuwing; tijdens laden en lossen intermitterend 65 dB(A)
9. achtergrondgeluid; veroorzaakt door (geïnstalleerde) apparatuur op 1 m gemeten maximaal 80 dB(A) resp. NR 75
10. achtergrondgeluid; veroorzaakt door geïnstalleerde apparatuur max.70 dB(A)
11. achtergrondgeluid; veroorzaakt door geïnstalleerde apparatuur max.75 dB(A) resp. NR 70
12. tevens dient aan de geluideis in dB(A) te worden voldaan
13. achtergrondgeluid; veroorzaakt door geïnstalleerde apparatuur max.85 dB(A)
14. maximaal 68 dB in 250 Hz octaafband en 63 dB in 500 Hz octaafband bij 3/4 van de normale scheepssnelheid /17/
15. voor 500 Hz octaafband en hoger
16. indien grenswaarde vordt overschreden en hinder door laagfrequent geluid of door duidelijk waarneembare tonen wordt ondervonden dient aan NR-waarden te worden voldaan
17. Leq_{24} niet hoger dan 105 dB(A)
18. indien een Leq_{24} van 80 dB(A) wordt overschreden dienen gehoorbeschermers te worden gebruikt

The differences become especially clear in the accommodation spaces, where the demands for pleasure crafts differ up to 10[dB(A)] from normal service crafts.

Acceptable vibration levels
The actual maximum level that is permitted on board for the Classification Society to approve the ship is 6[mm/s]. On pleasure crafts the levels are usually lower and a target level of 2[mm/s] is aimed at.

Desired sea trial conditions
The levels are established, based on the following conditions:

- Yacht at cruising speed and medium displacement loading condition (80% mcr[1])
- Air-conditioning in cabins in normal operating conditions
- Deep water
- Ship on even keel[2]
- Wind force at maximum Beaufort 2 (i.e. 5.56[m/s])
- All doors closed

Even though the utmost effort is given, these conditions are not always met due to various factors. The weather conditions and the situation in the accommodations are usually hard to meet. Weather simply cannot be influenced, especially during winter the North Sea can be a rough area.
Seatrials often take place before the entire ship is ready. Since the interior is always done as late as possible, there is a very reasonable chance that carpets, curtains, furniture and even doors are absent during the seatrials.
This will give a distorted picture of the final sound pressure levels.

[1] maximum continuous rating
[2] draft at front ship is equal to draft at aft ship

APPENDIX 3: LLOYD'S PROVISIONAL RULES

For those who are interested the following eight pages contain Lloyd's provisional rules for passenger and crew accommodation comfort (PCAC) on board. The rules give maximum acceptable noise and vibration levels, derived from several criteria.

Section 1 General requirements

1.1 Scope

1.1.1 These Provisional Rules set down the criteria for the assessment of noise and vibration levels on ships and special service craft 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) and the *Rules and Regulations for the Classification of Special Service Craft* (hereinafter referred to as the Rules for Special Service Craft).

1.1.2 For the purpose of these Rules, the term ship applies to both ships and special service craft.

1.1.3 These Provisional Rules provide for two alternatives:
(a) A **Certificate of Compliance** which records that the ship has been designed and built to meet the noise and vibration criteria contained in these Provisional Rules. This is to be confirmed by measurements during an Initial Survey, or
(b) A **Class Notation**, which in addition to the provisions of 1.1.3(a), requires periodic survey of the noise and vibration characteristics throughout the life of the ship.

1.1.4 These Provisional Rules recognize existing national and international standards and specify levels of noise and vibration currently achievable using good engineering practice. Compliance with these requirements will be assessed by review of procedures, inspection and measurement of the relevant parameters. Pre-survey reviews, inspections and measurements will be conducted, witnessed or assessed by LR Surveyors.

1.1.5 Vibration and noise levels are a function of ship type and layout. These Provisional Rules address three types of ship:
(a) Passenger (e.g. cruise ships, Ro-Ro ferries).
(b) High speed craft (e.g. Surface Effect Ships (SES), wave piercing catamarans, hydrofoil).
(c) Yacht (e.g. motorized pleasure craft).

1.1.6 These Provisional Rules present values of noise and vibration which should be verified by measurements following completion of the ship. It is recommended that the builders undertake calculations of noise and vibration so that any potential problem areas can be identified and control measures implemented.

1.2 Definitions

1.2.1 **Passenger spaces** are defined as all areas intended for passenger use, and include the following:
(a) Passenger cabins and associated corridors.
(b) Public spaces (e.g. restaurants, hospital lounges, reading and games rooms, gymnasiums, corridors, shops).
(c) Open deck recreation areas.

1.2.2 **Crew spaces** are defined as all areas intended for crew use only, and include the following:
(a) Accommodation spaces (e.g. cabins, corridors, offices, mess rooms, recreation rooms).
(b) Work spaces.
(c) Navigation spaces.

1.2.3 **Noise level** is defined as the A-weighted sound pressure level measured in accordance with ISO 2923.

1.2.4 **Vibration level** is defined as the single amplitude peak value of deck structure vibration during a period of steady-state vibration, representative of maximum repetitive behaviour, in mm/s peak, over the frequency range 1 to 100 Hz.

1.3 Certificate of Compliance

1.3.1 The Certificate of Compliance provides ship operators with an objective assessment of a ship's noise and vibration levels in accommodation spaces at the time of the assessment.

1.3.2 The Certificate of Compliance is optional and is primarily intended to apply to passenger ships. If requested however, any ship can be assessed for compliance, using these requirements as a basis for the assessment.

1.3.3 To achieve the Certificate of Compliance, Sections 2 to 6 and 7.1 must be complied with.

1.3.4 The Certificate of Compliance will be issued after the Initial Survey, following satisfactory assessment of the measured data.

1.4 Class notations

1.4.1 The notations provide ship operators with an objective assessment of a ship's noise and vibration levels in accommodation spaces throughout its life.

1.4.2 The **PAC** (Passenger Accommodation Comfort), **CAC** (Crew Accommodation Comfort) and **PCAC** (Passenger and Crew Accommodation Comfort) notations are optional and are primarily intended to apply to passenger ships. If requested, however, any ship can be assessed for compliance, using these requirements as a basis for the assessment.

1.4.3 For ships classed with LR which achieve the comfort standards specified in these Provisional Rules, the class notation **PAC, CAC** or **PCAC** will be assigned. Following the **PAC** or **CAC** notation, numerals **1** or **2** will indicate the acceptance criteria to which the noise and vibration levels have been assessed. In the case of the **PCAC** notation, two numerals will be assigned. The first will indicate the acceptance criteria for passenger accommodation, whilst the second will indicate the crew comfort criteria.

1.4.4 The **PAC** notation indicates that the passenger accommodation meets the acceptance criteria whilst **CAC** notation indicates that the crew accommodation and work areas meet the acceptance criteria. The **PCAC** notation indicates that the passenger and crew spaces meet the acceptance criteria.

1.4.5 To achieve and maintain any of the foregoing notations, Sections 2 to 7 must be complied with.

Section 2
Noise

2.1 Assessment criteria

2.1.1 Where a space is occupied by both passengers and crew, the more stringent of the relevant requirements apply.

2.2 Maximum noise levels

2.2.1 Where the measured 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 ISO 1999.

2.2.2 For all stated noise levels the equivalent NR number ... valuated as the measured dB(A) level minus 5 dB(A).

2.2.3 Guidance on maximum acceptable sound pressure levels and noise exposure limits is given in IMO Resolution A.468(XII).

2.3 Passenger accommodation

2.3.1 When the ship is proceeding in its normal operating condition and in accordance with the provisions specified in 4.2, the applicable noise levels specified in Tables 2.1, 2.2 and 2.3 are not to be exceeded. When the ship is in harbour, the noise levels are not to exceed the specified dB(A) levels minus 5 dB(A).

2.4 Crew accommodation and work areas

2.4.1 When the ship is proceeding in its normal operating condition and in accordance with the provisions specified in 4.2, the noise levels specified in Tables 2.4 and 2.5 are not to be exceeded.

Table 2.1 Passenger ships – maximum noise levels, in dB(A)

Location	Acceptance numeral	
	1	2
Passenger cabins	45	50
Public spaces (*see* Note)	55	60
Open deck recreation areas	65	70

NOTE
Public spaces exclude discotheques and similar areas for the purposes of noise criteria.

Table 2.2 High speed craft – maximum noise levels, in dB(A)

Location	Acceptance numeral	
	1	2
Public spaces (*see* Note)	60	70

NOTE
Public spaces exclude discotheques and similar areas for the purposes of noise criteria.

Table 2.3 Yachts – maximum noise levels, in dB(A)

Location	Acceptance numeral	
	1	2
Cabins	50	55
Lounges	55	60
Open decks	60	65
Wheelhouse	60	65

Table 2.4 Accommodation – maximum noise levels, in dB(A)

Location	Acceptance numeral	
	1	2
Sleeping cabins, hospitals	50	55
Day cabins	55	60
Offices, conference rooms	55	60
Mess rooms, lounges, reception areas:		
• within accommodation	55	60
• on open decks	65	70
Alleyways, changing rooms, bathrooms, lockers	70	75

2.5 Acoustic insulation

2.5.1 Acoustic insulation of bulkheads and decks between passenger spaces is to be in accordance with the values of the airborne sound insulation index (I_a) given in Table 2.6, calculated using ISO 717/1.

2.5.2 Crew space insulation is to comply with the requirements of IMO Resolution A.468(XII).

Table 2.5 Work areas – maximum noise levels

Location	dB(A) level
Machinery space (continuously manned) e.g. stores	90
Machinery space (not continuously manned) e.g. pump, refrigeration, thruster or fan rooms	110
Workshops	85
Machinery control rooms	75
Wheelhouse	65
Bridge wings, additional limits:	
• 250 Hz octave band	68 (linear)
• 500 Hz octave band	63 (linear)
Radio room	60
Galleys and pantries:	
• Equipment not working	70
• Individual items at 1 metre	80
Normally unoccupied spaces (e.g. holds, decks)	90
Ship's whistle, on bridge wings or forecastle	110

Table 2.6 Minimum sound insulation index (I_a)

Location	Index level	
	1	2
Cabin to cabin	45	40
Cabin to public space	55	50

2.6 General alarm and public address systems

2.6.1 The general alarm and public address systems are to comply with 2.6.2 and 2.6.3 together with Pt 6, Ch 2,17.2 and 17.3 of the Rules for Ships, or Pt 16, Ch 2,17.2 and 17.3 of the Rules for Special Service Craft, as applicable.

2.6.2 During the noise measurement programme the general alarm and public address systems are to be demonstrated by tests. These tests are to be undertaken under the sea trial conditions as described in 4.2.

2.6.3 When the ship is underway in normal conditions, the minimum sound pressure levels of the public address system for broadcasting emergency announcements are to comply with the following:

(a) In interior spaces:
- 75 dB(A), and
- at least 20 dB(A) above the speech interference level.

(b) In exterior spaces:
- 80 dB(A), and
- at least 15 dB(A) above the speech interference level.

NOTE
The speech interference level is defined as the arithmetic average of the sound pressure level of the ambient noise in the four octave bands: 500 Hz, 1000 Hz, 2000 Hz and 4000 Hz.

Section 3
Vibration

3.1 Maximum vibration levels

3.1.1 Vibration limits are given in units of:

(a) peak acceleration (mm/s²), single amplitude, in the range 1 to 5 Hz, and

(b) peak velocities (mm/s), single amplitude, in the range 5 to 100 Hz.

3.1.2 Measured vibration levels are to be assessed over the frequency range 1 to 100 Hz. The limits apply to each single frequency component of vertical, fore and aft and athwartship vibration which are to be assessed separately.

3.1.3 Crew spaces are to comply with ISO 6954 requirements. The following peak velocity limits are applied in the frequency range 5 to 100 Hz:

Accommodation and navigation spaces	5 mm/s peak
Work spaces	6 mm/s peak

3.1.4 Passenger spaces are to be assessed with the ship proceeding in its normal condition and in accordance with the provisions set out in Section 4. The vibration levels specified in Tables 3.1, 3.2 and 3.3 are not to be exceeded.

Table 3.1 Passenger ship – maximum vibration levels

Location	1 to 5 Hz		5 to 100 Hz	
	Peak acceleration, mm/s^2		Peak velocity, mm/s	
	Acceptance numeral			
	1	2	1	2
Luxury cabins	47	63	1,5	2,0
Standard cabins	47	79	1,5	2,5
Public spaces	47	79	1,5	2,5
Open recreation decks	79	110	2,5	3,5

le 3.2 High speed craft – maximum vibration levels

Location	1 to 5 Hz		5 to 100 Hz	
	Peak acceleration, mm/s^2		Peak velocity, m/s	
	Acceptance numeral			
	1	2	1	2
Public spaces	79	126	2,5	4,0

Table 3.3 Yachts – maximum vibration levels

Location	1 to 5 Hz		5 to 100 Hz	
	Peak acceleration, mm/s^2		Peak velocity, mm/s	
	Acceptance numeral			
	1	2	1	2
Cabins and lounges	31	63	1,0	2,0
Wheelhouse	47	94	1,5	3,0
Open decks	63	110	2,0	3,5

Section 4 Testing

4.1 Measurement procedures

4.1.1 These requirements take precedence where quoted standards may differ.

4.1.2 The trial measurements may be undertaken by an approved technical organization or by LR. In the former case, the measurements shall be witnessed by an LR Surveyor.

4.2 Test conditions

4.2.1 Test conditions for the surveys are to be in accordance with those detailed in ISO 2923 and ISO 4868.

4.2.2 The intended operating and loading conditions of the ship during assessment surveys are to be submitted to LR for agreement, prior to commencement of surveys.

4.2.3 Surveys will only be conducted when the ship is fully outfitted and all systems contributing to noise and vibration levels are fully operational.

4.2.4 The test conditions required for the vibration and noise measurements are to be in accordance with the following conditions:

(a) The power absorbed by the propeller(s) is to be not less than 85 per cent of the maximum continuous rating of the propulsion machinery. Alternatively, by special agreement, some lesser power could be accepted if it can be demonstrated by the Owner that this would correspond to a more representative normal service condition.
(b) Auxiliary machinery essential for the ship's operating conditions together with HVAC systems are to be running at their normal rated capacity during the noise and vibration trials. Combinations of auxiliary machinery operation may be necessary.
(c) For sea-going ships, measurements are to be taken with the ship proceeding ahead, at a constant speed and course, in a depth of water not less than five times the draught of the ship. For other ships, an appropriate water depth is to be agreed with LR prior to the trials.
(d) Trials are to be conducted in sea conditions not greater than sea state 3 on the WMO sea state code.
(e) The ship is to be at a displacement and trim corresponding to the normal operating condition.
(f) Rudder angle variations are to be limited to $\pm 2°$ of the midship position and rudder movements are to be kept to a minimum throughout the measurement periods.
(g) In addition, for ships which are designed to spend a considerable period of time in harbour, the noise and vibration levels are to be measured for this condition, with the auxiliary machinery and HVAC systems running at their normal rated capacity.

4.2.5 Prior to survey, a test programme is to be submitted for approval by LR. This programme is to contain details of the following:

(a) Measurement locations, indicated on a general arrangement of the ship.
(b) The ship's loading condition during survey.
(c) The machinery operating condition, including HVAC system, during survey.
(d) Noise and vibration measuring equipment.

4.3 Noise measurements

4.3.1 Noise measurements are to be conducted in accordance with ISO 2923 and IMO Resolution A.468. Measurements of noise levels are to be carried out using precision grade sound level meters conforming to IEC 651, Type 1 or 2. Subject to demonstration, equivalent standards are acceptable.

4.3.2 Where the measured noise exceeds the relevant criteria by 3 dB(A), or contains subjectively annoying low frequency noise, or obvious tonal components, octave band readings are to be taken, with centre frequencies from 31,5 Hz to 8 kHz.

4.3.3 When outfitting is complete, and all soft furnishings are in place, airborne sound insulation indices for passenger spaces are to be determined in accordance with ISO 140/4. A minimum of six cabin to cabin indices are to be determined from three locations within the passenger accommodation.

4.4 Noise measurement locations

4.4.1 Measurement locations are to be chosen so that assessment represents the overall noise environment on board the ship. In addition to the requirements of IMO Resolution A.468(XII) for crew spaces, all public spaces and at least 50 per cent of passenger cabins in the after third of the ship, and 25 per cent elsewhere, are to be surveyed. Distribution of the measurement locations is to be agreed by LR.

4.4.2 During measurement trials, recognized noise sources are to be operated at their normal level of noise output (e.g. machinery at design rating, discotheques).

4.4.3 In larger sized spaces, where noise levels may vary considerably, such as restaurants, lounges, atria and open deck recreation areas, measurements are to be taken at locations not greater than 7 m apart.

4.4.4 For yachts and high speed craft, the noise levels in all passenger spaces are to be measured.

4.4.5 For high speed craft having large passenger saloons, measurements are to be taken along the centreline and along both sides of the saloons at locations not greater than 7 m apart.

4.5 Vibration measurements

4.5.1 Vibration measurements are to be conducted in accordance with ISO 4868.

4.5.2 Measurements are to be made with an electronic system capable of providing vibration frequency spectra in the range 1 to 100 Hz.

4.5.3 Vibration levels are to be given in maximum repetitive peak values measured over a period of not less than one minute.

4.6 Vibration measurement locations

4.6.1 Measurement locations are to be chosen so that the assessment represents the overall vibration environment onboard the ship. To minimize survey times, readings may be taken at the locations previously defined for the noise assessment part of the survey.

4.6.2 In cabins, vibration readings are to be taken in the centre of the floor area. The measurements are to indicate the vibration of the deck structure. In large spaces, such as restaurants, sufficient measurements are required to define the vibration profile.

4.6.3 Where deck coverings make transducer attachment impracticable, use of a small steel plate having a mass of at least 1 kg is permissible.

4.6.4 At all locations, vibrations in the vertical direction are to be assessed. Sufficient measurements in the athwartships and fore and aft directions are to be taken to define global deck vibrations.

■ *Section 5*
Survey reporting

5.1 General

5.1.1 The survey report is to comprise the data and analysis for both noise and vibration and is to be submitted to LR for consideration.

5.1.2 The survey report shall be prepared by the organization undertaking the trial measurements, which may be an approved technical organization or LR.

5.2 Noise

5.2.1 The reporting of results is to comply with ISO 2923, and is to include:

(a) The results of public address and general alarm system testing, in addition to the sound insulation testing.
(b) Measurement locations indicated on a general arrangement plan including, where possible, the measured dB(A) level.
(c) Tabulated dB(A) noise levels, together with octave band analysis for positions where the level exceeds the specified criterion by 3 dB(A), or where subjectively annoying low frequency or tonal components were present. The NR number is also to be given where octave band analyses have been conducted.
(d) Ship and machinery details.

(e) Trial details:
- Loading condition.
- Machinery operating condition.
- Speed.
- Average water depth under keel.
- Weather conditions.
- Sea state.

(f) Details of measuring and analysis equipment (e.g. manufacturer, type and serial numbers), including frequency analysis parameters (e.g. resolution, averaging time, window function).

(g) Copies of the relevant instrument calibration certificates, together with the results of field calibration checks.

5.3 Vibration

5.3.1 The report is to contain the following information:

(a) Measurement positions indicated on a general arrangement plan.

(b) The maximum peak vibration levels and their corresponding frequencies taken from the frequency spectra, tabulated for all measurement locations.

(c) Ship and machinery details.

(d) Trial details:
- Loading condition.
- Machinery operating condition.
- Speed.
- Average water depth under keel.
- Weather conditions.
- Sea state.

(e) Details of measuring and analysis equipment (e.g. manufacturer, type and serial number), including frequency analysis parameters (e.g. resolution, averaging time and window function).

(f) Copies of the relevant instrument calibration certificates, together with the results of field calibration checks.

Section 6
Excessive noise and vibration

6.1 Noise excesses

6.1.1 Noise levels greater than those specified in these Provisional Rules may be considered. No more than 20 per cent of the passenger cabins and 30 per cent of the public spaces are to exceed the relevant noise criteria by more than 3 dB(A).

6.2 Vibration excesses

6.2.1 Vibration levels greater than those specified in these Rules may be considered. No more than 20 per cent of all passenger spaces are to exceed the relevant vibration criteria by more than 0,3 mm/s.

Section 7
Survey requirements

7.1 Initial Survey

7.1.1 The Initial Survey shall comprise sea trial or initial in-service testing, reporting and assessment against the criteria set out in these Provisional Rules.

7.1.2 The sea trial or initial in-service testing requirements are set out in Section 4, and are to be reported in accordance with Section 5 and evaluated against the requirements of Sections 2 and 3.

7.2 Periodical Survey (first 5 years)

7.2.1 An Annual Survey is to be held between 9 and 15 months after the completion of the Initial Survey. Measurements of noise and vibration will be required at a minimum of 25 per cent of the locations specified for the Initial Survey. The locations are to be submitted to LR for agreement, prior to commencement of the Annual Survey.

7.2.2 An Intermediate Survey is to be held within three months before or after the third anniversary of completion of the Initial Survey. The percentage of measurements required is to be as specified in 7.2.1.

7.2.3 If the limiting criteria as described in 6.1 and 6.2 are exceeded and the cause of the exceedance cannot be rectified at the time of the survey, then a Renewal Survey may be required.

7.3 Periodical Surveys (subsequent years)

7.3.1 A Renewal Survey is to be held at five-yearly intervals, the first one five years from the completion of the Initial Survey. The measurements required shall be the same as those required for the Initial Survey.

7.3.2 Following each Renewal Survey, an Intermediate Survey is to be held between the second and third subsequent years. The percentage of measurements required is to be as specified in 7.2.1.

7.4 Surveys following modifications

7.4.1 A Renewal Survey may be required following modifications, alterations or repairs including replacement of major machinery items. It is the responsibility of the Owner to advise LR of such modifications.

Section 8
Referenced standards

8.1 Noise

8.1.1 The following National and International Standards for noise are referred to in these Provisional Rules:

- ISO 2923, 'Acoustics – Measurement of noise on board vessels'.
- ISO 1999, 'Acoustics – Determination of occupational noise exposure and estimation of noise-induced hearing impairment'.
- ISO 717/1, 'Acoustics – Rating of sound insulation in buildings and of building elements; Part 1: Airborne sound insulation'.
- IMO Resolution A.468(XII), 'Code on noise levels on board ship'.
- IEC Publication 651, 'Sound level meters'.
- ISO 140/4, 'Acoustics – Measurement of sound insulation in buildings and of building elements; Part 4: Field measurements of airborne sound insulation between rooms'.

8.2 Vibration

8.2.1 The following National and International Standards for vibration are referred to in these Provisional Rules:

- ISO 6954, 'Mechanical vibration and shock – Guidelines for the overall evaluation of vibration in merchant ships'.
- ISO 4868, 'Code for the measurement and reporting of local vibration data of ship structures and equipment'.

APPENDIX 4: PROPELLER DESIGN AND CAVITATION

Occurrence of cavitation depends for a great deal on proper propeller design. Design of the ship's hull, especially the aft part of it is also important to provide the propeller blades with a smooth incoming flow.
This appendix will describe which kinds of cavitation there are, how they become visible (damage the propeller and/or ship's hull) and how blade shape can influence their effect or even totally eliminate
First a few terms considering propeller design will be reviewed. After that the different forms of cavitation and their consequences for propeller and/or ship will be discussed. Finally some important checkpoints for a cavitation free propeller design will be mentioned.
[**Kuiper**, 1997]

A4.1 General Propeller Features

The backside of a propeller blade is at the shaft's side, it is also called the suction side. The face of the propeller is also called the pressure side. The edge of the blade where the water first passes is called leading edge or nose; the other edge is called trailing edge or tail. The straight line connecting nose and tail is called the chord-line (or nose-tail line), thus from chord-lines drawn at several distances from the centre the shape of a blade can be derived. The angle α between the nose-tail line and the undisturbed flow is the angle of attack. The line from nose to tail over the middle of the thickness is the camberline; it gives some information about the curving of the blade.

Another very important feature is the pitch of the propeller. The pitch states the increase in axial direction of the pitch line over one revolution. The pitch angle is successively defined as the angle between the pitch line and a plane perpendicular to the propeller shaft. As significant radius to describe the pitch 0.7R is commonly used, R being the total radius of a blade. The above can also be made clear in a diagram giving the pitch at every radius.

The generator line is called: skew. The distance between the generator line at a specific radius and the propeller plane is called rake. Positive, or backward rake increases the clearance between ship and blade tip, and is the common way to design propellers. To describe the increase in clearance the rake angle θ is defined as the angle between the generator line and the propeller reference line.

Finally there are four area definitions. First of course there is the real blade area; secondly there is the projected blade area. This is the area of the blade contour projected on the propeller plane. In a developed contour, thirdly, the blade sections are rotated around the blade reference line into a plane perpendicular to the propeller plane. From there on the fourth form, the expanded blade area can be derived where straight sections replace the curved blade sections.

Two area ratios are defined: $\frac{A_p}{A_0}$ *and* $\frac{A_e}{A_0}$ from which the latter is most important for hydrodynamical features of the propeller. In these ratios A_p stands for projected blade area, A_e for expanded blade area and A_0 for propeller plane area.
Figure A4.1 visualises the terms.

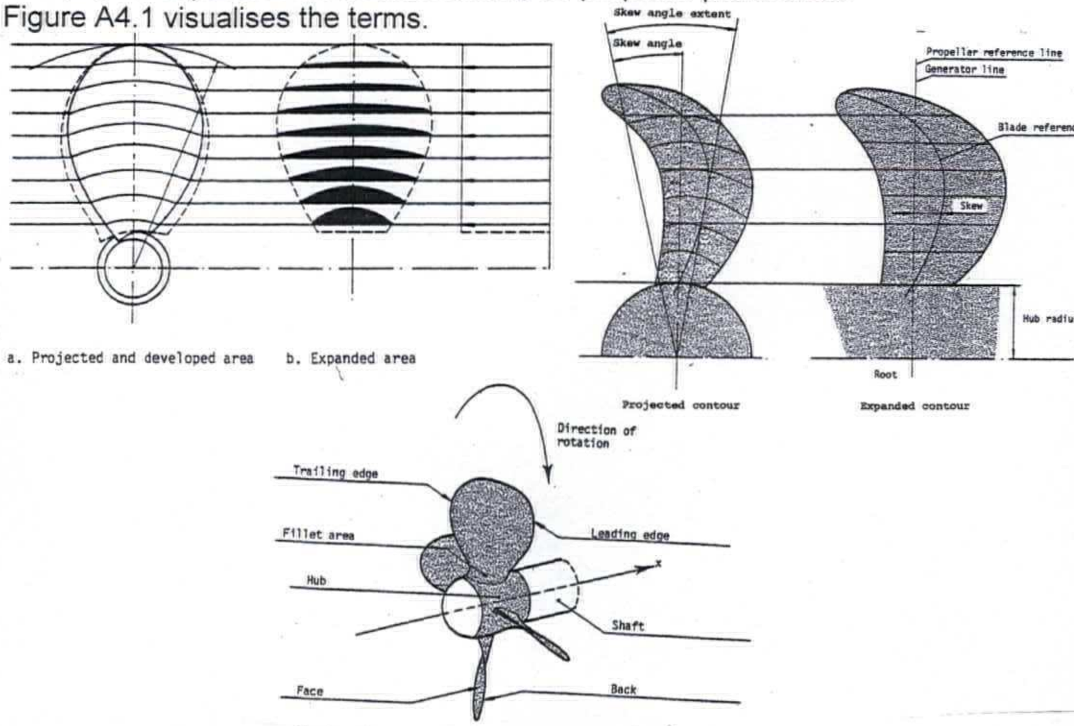

Figure A4.1. Illustration of propeller specific design terms

A4.2. Cavitation of Propeller Blades

Cavities come into being on places where local pressure is very low. These low pressures induce the forming of tiny gas bubbles, so called nuclei. Within these bubbles the pressure level will initially equal the vapour pressure of water. When the pressure around the bubbles increases enough, the bubbles will implode, leaving their traces on the propeller blade surface. The traces show in various ways, briefly described below.

Bubble Cavitation

When the pressure in the flow decreases enough nuclei will be formed and cause isolated cavities, which move with the flow. It is generally an erosive sort of cavitation. The minimum pressure will in this case be below vapour pressure and at mid-chord of a blade section of relatively thick blades and a small angle of attack.

Sheet Cavitation

This kind of cavitation occurs when the flow separates from the body because of a strong adverse pressure gradient. It is caused by a leading edge suction peak and a minimum pressure below vapour pressure. The cavity gradually merges with the tip vortex. During a revolution the angles of attack and consequently the cavitation extent change with blade position. In the wake peak cavitation will occur on the suction side, where the growth and implosion of gas bubbles causes a break up of the sheet cavitation. This is referred to as unsteady sheet or rather cloud cavitation, because of the form and the amount of vortices and bubbles. Cloud cavitation is very errosive.

Root Cavitation
This kind of cavitation obviously occurs at the root of a blade and has the shape of a wedge. Root cavitation is related to the horseshoe vortex present at the blade root.

Tip Vortex Cavitation
The pressure in the core of a vortex is low. When this pressure is below vapour pressure cavitation can occur. When this vortex is at the tip of the blade the cavitation is called tip vortex cavitation.
In extreme cases the vortex will connect with the hull and cause considerable damage to the plating and high noise levels. Usually a strong wake peak initiates the process.

The violent implosion of cavities causes noise. Depending on the amount and impact of the collapse the noise level will be higher or lower. When the implosion takes place near the surface it can cause erosion as well. It may be clear that effort should be made to avoid cavitation at all times.
The wake field at the aft ship and the character of the flow through the blades influence cavitation. The wake field is determined by the hull design of the ship. To ensure a smooth flow around the ship and into the propeller the draft of the ship (rather yacht) should be large enough and sharp edges absent. If necessary, extra ducts could be applied to lead the water through the blades in the desired way.

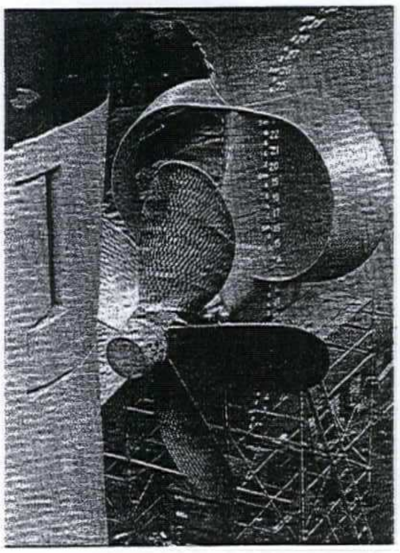

Figure A4.2. Example of a propeller with inflow ducts to reduce the cavitation risk

A4.3. The Making of a Proper Propeller Design

Some aspects influencing the occurrence of cavitation have been discussed in the previous two sections. The remaining question is of course: how should a propeller be designed to avoid cavitation problems. An answer to this question should arise in the course of this section, which will cover the basic design process of a propeller. If the process is followed securely, a cavitation-free propeller should be the result.

Propeller design is just as essential to eliminate cavitation problems. The most important design features to reckon with are:
- the number of blades
- the angle of attack, α
- the shape of the blades: diameter, area ratio, pitch, thickness distribution
- presence of an anti-singing edge
- the hub shape

A quantity of three to five blades is common practice. On Oceanco's yachts only four and five bladed propellers are applied. Choosing a certain number sets limits to the rest of the design, thus should be well considered. If the propeller is of the controllable pitch type, the blade size is limited with relation to the angle over which they are supposed to be turned.
Efficiency is another requirement that limits the number of blades. For a propeller to process a certain amount of water minimum blade sizes are required, too many blades will make the propeller look like a solid shape, increasing only resistance.
To come to an optimal design for a specific situation, a diagram of propeller series is a helpful tool. With these diagrams you can figure out what diameter you need and what efficiency belongs to that specific dimensions, etcetera.

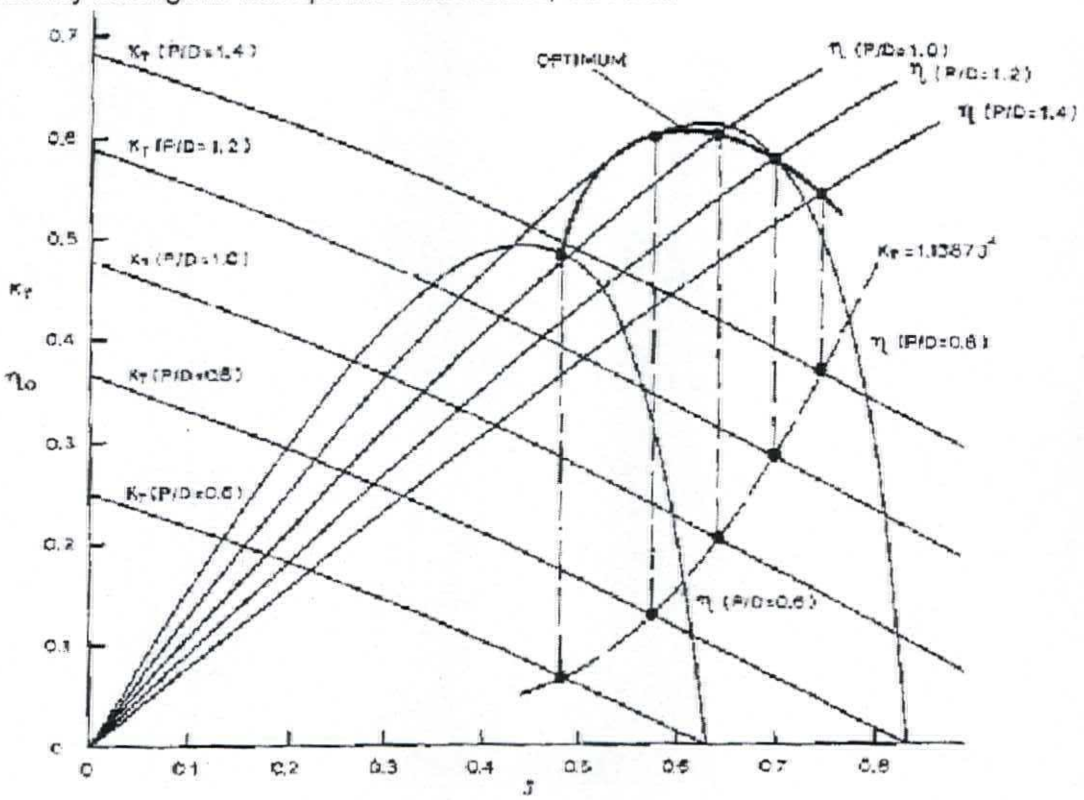

Figure A4.3. A typical open water propeller diagram

Besides the propeller diagrams the cavitation bucket is used. As long as one stays inside the bucket, with safe margins, cavitation will very unlikely occur.

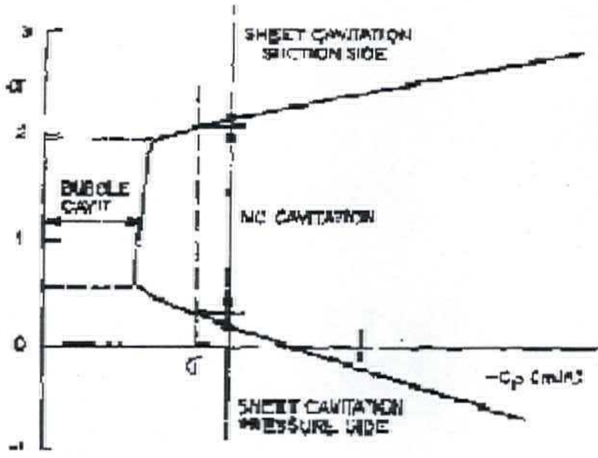

Figure A4.4. Cavitation buckets are used to determine the main dimensions of the propeller with a safe margin to avoid cavitation of any kind

Cavitation Number

The occurrence of cavitation is pressure dependent. It is initiated when the local pressure equals the vapour pressure. Cavitation of a propeller blade with varying pressure over its area will thus be initiated where the lowest pressure equals the vapour pressure.

In the cavitation bucket shown above, σ represents the *cavitation index*. The index is related to the minimum pressure coefficient. The formulas for both are given below.

$$C_p = \frac{p - p_0}{\frac{1}{2}\rho v_0^2} \qquad \text{(a4.1)}$$

$p = local\ pressure\,[Pa]$

$p_0 = pressure\ in\ undisturbed\ flow\,[Pa]$

$\rho = density\ of\ the\ medium\,[kg/m^3]$

$v_0 = flow\ velocity\ of\ undisturbed\ flow\,[m/s]$

With help of the Bernoulli Law the coefficient may also be expressed as:

$$C_p = 1 - \left(\frac{v_1}{v_0}\right)^2 \qquad \text{(a4.2)}$$

$v_1 = local\ velocity\,[m/s]$

In analogue way the cavitation index σ can be expressed:

$$\sigma = \frac{p_0 - p_{vap}}{\frac{1}{2}\rho v_0^2} \qquad \text{(a4.3)}$$

$p_{vap} = vapour\ pressure\,[Pa]$

Both coefficients are dimensionless. The cavitation bucket shown in figure (A4.4) illustrates the relation between C_p and σ. On the vertical axis the angle of attack is plotted, on the horizontal axis the pressure coefficient. The dashed vertical line indicates the critical point where $\sigma = -C_{p\_min}$. Generally spoken, cavitation will not occur as long as you stay inside the bucket, with a safe margin. If the design point is chosen above or below the bucket, sheet cavitation will occur. A design point left of the bucket will result in bubble cavitation.

The shape and dimensions of the profile influence the occurrence of cavitation just as well, therefore several cavitation buckets can be plotted, each focusing on a specific feature.
The buckets are combined in the inception diagram, which should give the decisive answer to the optimal design question. It is shown in figure (A4.5) below.

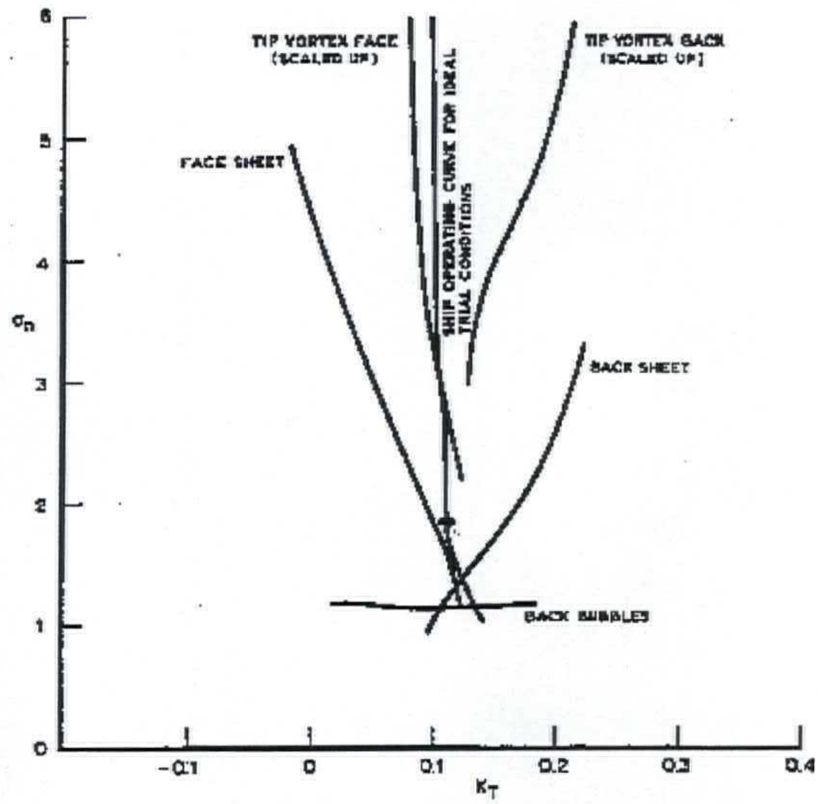

Figure A4.5. Inception diagram, the cross indicates the propeller design point

On the vertical axis the cavitation index is plotted, on the horizontal axis K_T, the thrust coefficient.

APPENDIX 5: BALANCING

The fuel combustion process of diesel engines is accompanied by large forces. The magnitude of the forces depends on the total power of the engine, the size of the cylinders and the relative position of the cylinders with respect to the central axis and to each other. A strategic choice of ignition sequence and distance and positions of the cylinders provides the possibility to cancel some or all of the forces.
In the following a method to calculate the magnitude and direction of the forces (and moments) is presented. After that some methods to eliminate free forces and moments are discussed. These methods involve the use of counterweights, on crankshaft and on gears, and flexible mounting, possibly in combination with top bracing (extra support for upper part of engine).

A5.1. Calculation of Free Forces and Moments

The inertia forces of a cylinder consist of a rotating part, due to crankshaft rotation, and a translating part, due to piston motion. Figure A5.1. illustrates this. First the method for calculating the translating and rotating forces will be discussed. After that, the method for calculating the free moments will be explained. To keep a clear view on the issue only the case for constant rotational speed will be considered.

A5.1.1. Free Forces and Moments for an In-line engine

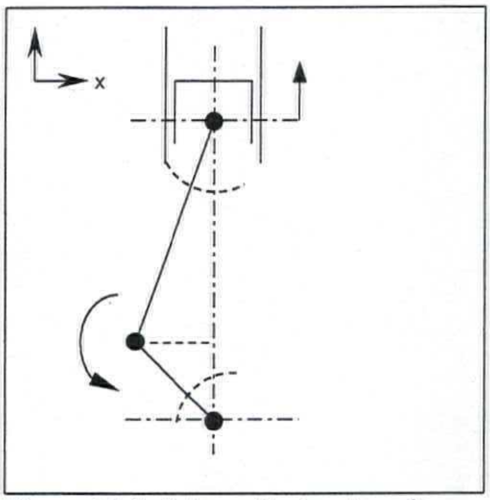

Figure A5.1. Schematic drawing of a crank rod mechanism

Inertia forces are associated with acceleration, so a description is needed for the displacement *s* of the piston as a function of time *t*. The change of piston position is determined by the rotational speed ω of the shaft. In the figure θ could be replaced by ωt. For now θ will be used to avoid confusion.
The total displacement *s* as a function of θ will then become:

$$s = R\left(1+\frac{1}{\lambda}-\cos\theta-\frac{1}{\lambda}\sqrt{1-\lambda^2\sin^2\theta}\right) \qquad \text{(a5.1)}$$

This follows from the geometry of the figure. The last term of the equation: $\sqrt{1-\lambda^2\sin^2\theta}$ can also be written as a binomial series.

The general binomial form of an expression like $(1-x)^n$ looks like:

$$(1-x)^n = 1-\frac{nx}{1!}+\frac{n(n-1)x^2}{2!}-\frac{n(n-1)(n-2)x^3}{3!}+\ldots.$$

Applying this for the particular case leads to:

$$(1-\lambda^2\sin^2\theta)^{1/2} = 1-\frac{\lambda^2\sin^2\theta}{2}-\frac{\lambda^4\sin^4\theta}{8}-\frac{\lambda^6\sin^6\theta}{16}-\frac{15\lambda^8\sin^8\theta}{384}-\ldots \qquad \text{(a5.2)}$$

It can be seen that from the third term on the right side on the values rapidly approximate zero, which makes it very reasonable to neglect them.
The next step is to convert the sine-terms into equivalent cosine-terms, so that there is only one variable left. Since: $\sin^2\theta = 1-\cos^2\theta = \frac{1}{2}(1-\cos(2\theta))$ and

$\sin^4\theta = (\sin^2\theta)^2$, which can thus be written as

$\sin^4\theta = \frac{1}{8}(3-4\cos(2\theta)+\cos(4\theta))$ and finally

$\sin^6\theta = \frac{1}{32}(10-15\cos(2\theta)+6\cos(4\theta)-\cos(6\theta))$.

Substitution of these terms into the series, replacing at the same time θ by ωt, and then into the displacement equation will give *s* as a function of ω*t*. Neglecting the higher order terms leads to an expression for *s*, which looks as follows:

$$s = R\left(1+\frac{\lambda}{4}-\cos(\omega t)-\frac{\lambda}{4}\cos(2\omega t)+\frac{\lambda^3}{64}\cos(4\omega t)-\frac{\lambda^5}{512}\cos(6\omega t)...\right) \quad \text{(a5.3)}$$

and

$$\dot{s}(t) = \omega R\left(\sin(\omega t)+\frac{\lambda}{2}\sin(2\omega t)-\frac{\lambda^3}{64}\sin(4\omega t)+\frac{3\lambda^5}{256}\sin(6\omega t)...\right) \quad \text{(a5.4)}$$

and

$$\ddot{s}(t) = \omega^2 R\left(\cos(\omega t)+\lambda\cos(2\omega t)-\frac{\lambda^3}{4}\cos(4\omega t)+\frac{9\lambda^5}{128}\cos(6\omega t)...\right) \quad \text{(a5.5)}$$

Free Mass Forces

Equation (a5.5) is the one required to calculate the vertical component of the free force, since

$F_{trans,\ vert} = m_{trans}\ a$

$$= m_{trans}\,\ddot{s}(t) = \omega^2 R\left(\cos(\omega t)+\lambda\cos(2\omega t)-\frac{\lambda^3}{4}\cos(4\omega t)+\frac{9\lambda^5}{128}\cos(6\omega t)...\right) \quad \text{(a5.6)}$$

The translating force works against the force caused by the gas pressure in the cylinder head.
The rotational component of the free force is related to the distance to the mechanism's centre of gravity measured from the heart of the crank.
The connecting rod is thus modelled as a uniform rod with two concentrated masses, one rotating and the other translating.
Now it is quite easy to calculate which amount of the total mass will be rotating and which amount will be translating.

$$m_{rod,rot} = \frac{L-L_s}{L}m_{rod}$$
$$m_{rod,trans} = \frac{L_s}{L}m_{rod} \quad \text{(a5.7)}$$

The rotating free force then becomes:

$$F_{rot} = m_{rod,rot}\,\omega^2 R \quad \text{(a5.8)}$$

This force can be resolved in a horizontal and a vertical component:

$$F_{rot,hor} = m_{rod,rot}\,\omega^2 R\sin(\omega t) \quad \text{(a5.9)}$$

$$F_{rot,vert} = m_{rod,rot}\,\omega^2 R\cos(\omega t) \qquad \text{(a5.10)}$$

In the above equation ω represents the angular speed in [rad/s] and R in [m] is the length of the crank arm.

The total vertical force thus becomes:

$$F_{vert,tot} = (F_{trans} + F_{rot})_{vert}$$

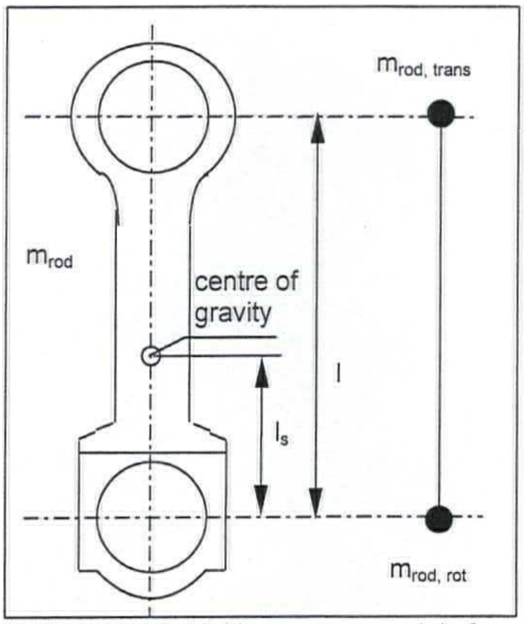

Figure A5.2. Making a mass model of connecting rod

Free Mass Moments

Next, the resulting moments are investigated.
Besides the moment caused by the presence of resulting (free) forces, Mollenhauer [1997] takes into consideration the mass moment of the rod. It is caused by the inertia of the rod due to the movement of the system. The equation looks like:

$$M_{rod,inert} = (J_s - J_s^*)\ddot{\varphi} \qquad \text{(a5.11)}$$

$$J_s = m_{rod}(L - L_s)L_s$$

with

J_s = *mass moment of inertia of the rod, as modelled for calculation*
J_s^* = *mass moment of inertia of the real rod*
s = *centre of gravity of the rod*
$\ddot{\varphi}$ = *angular acceleration of the rod*

The relation between θ and φ is:

$$\sin\varphi = \frac{R}{L}\sin\theta = \lambda\sin\theta$$

This follows from the geometry of the model. Then, applying the rules of goniometry the relation can be rearranged into a more practical form

$\sin^2\varphi = 1 - \cos^2\varphi$ (a5.12a) and $\sin^2\varphi = \lambda^2\sin^2\theta$ (a5.12b)

$$\cos\varphi = \sqrt{1 - \sin^2\varphi}$$

$$\varphi = \arccos\left(\sqrt{1 - \sin^2\varphi}\right)$$

Now substitute (a5.12b) in this last equation to obtain:

$$\varphi = \arccos\left(\sqrt{1-\lambda^2 \sin^2\theta}\right) \tag{a5.13}$$

The binomial form of the term within brackets was already derived in (a5.2.) This series can be substituted into equation (a5.13.) , which will then gain the following form

$$\varphi = \arccos\left(1-\frac{\lambda^2 \sin^2\theta}{2}-\frac{\lambda^4 \sin^4\theta}{8}-\frac{\lambda^6 \sin^6\theta}{16}-\frac{15\lambda^8 \sin^8\theta}{384}-...\right) \tag{a5.14}$$

To calculate the mass moment the second derivative of this equation has to be determined and applied in equation (a5.11).

The resulting oscillating (or translating) force can be resolved into a tangential component that is in direction of the rod, and a component parallel to the crankshaft. The tangential component will produce a moment of magnitude:

$$M_{trans,\tan} = F_{trans,\tan} R \tag{a5.15}$$

with F_{trans} as in (a5.6.) and R the length of the crank arm (in [m]).

The total resulting free forces and moments for a complete engine follow after summing the forces and moments resulting from each cylinder in the engine.

$$F_{vert,tot} = \sum_{i=1}^{z} F_{vert,i} \tag{a5.16a}$$

$$F_{hor,tot} = \sum_{i=1}^{z} F_{hor,i} \tag{a5.16b}$$

$$M_{tot} = \sum_{i=1}^{z} M_i \tag{a5.16c}$$

$z = \textit{total number of cylinders}$

It may be tempting to assume that there are no resulting moments as long as the free forces compensate each other. This is however absolutely not true, it may very well be that there are resulting moments, causing vibrations, although no resulting forces are present.

With the above equations the free forces and moments for a cylinder of an in-line engine are calculated. On yachts V-line engines are common practice. The next section will shed light on how to calculate the free forces and moments for V-configurations for different V-angles.

A5.1.2. Free Forces and Moments for a V-engine

As mentioned before, for practical reasons V-engines are common use on board yachts. The angle could theoretically be anywhere between 0° and 90°. The choice for a specific V-angle depends on three criteria:

1. Desired quiet running conditions;
2. Available space to build in the engine;
3. The cost to make the engine.

A choice based on these criteria may sometimes lead to unconventional values such as 72° for the MTU 595 Series. In most cases a compromise will be made then, in order to assign "normal" V-angle values. The discomfort (unbalance) caused by choosing a non-conventional angle will have to be compensated by appropriate *crank displacement angle (hubversatzwinkel).*

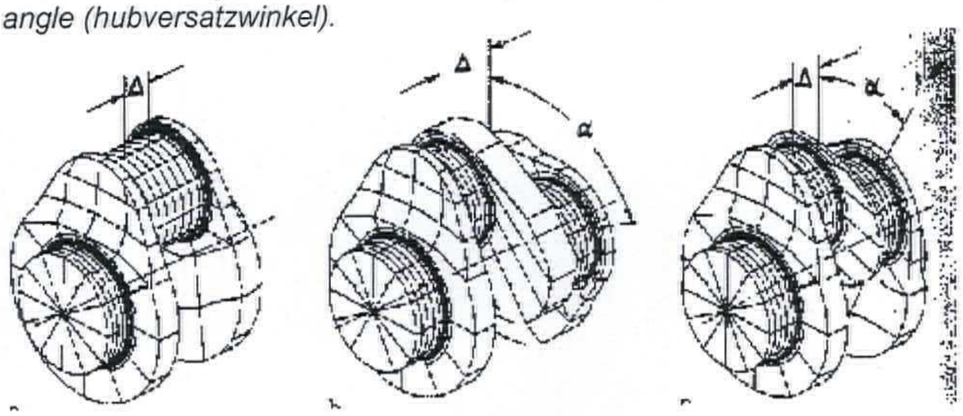

Figure A5.3. Crank displacement angle Δ

In modern V-engines the angle between two cylinders will most likely be 45°, 60° or 90°. Calculation of the free forces and moments for single cylinders is performed in the same manner as for Line-engines. To calculate the total resulting forces and moments for a complete engine, one simply has to sum the forces and moments for the single cylinders according to equations (a5.16a, b and c).

A5.2. Balance Options

Resulting free forces and moments need to be compensated to guarantee quiet running of the engine and to therewith minimise the effect of forces and moments on the foundation and ship's hull. Subsequently means to establish a good compensation for Line-engines and V-engines will be treated.

Compensation measures for line-engines
Resulting forces can very well be limited with help of counterweights, leaving two options:

1. compensation directly on the crankshaft;
2. compensation at the two ends on separate shafts.

Another option would be to adjust the design of the plunger, integrating extra mass directly in order to diminish the resulting mass moment of inertia:

3. compensation integrated in plunger design.

The engine's design plays a huge part in the decision process. Next some design aspects, which influence this process, will be given a closer look. In general the more cylinders, the less balance problems.
First of all the number of cylinders, which influences the relative position of each single cylinder. At all times a uniform distribution of cylinders over the shaft is desired.
Secondly the ignition angle should be chosen such that the ignition cycle will be steady.
Thirdly, the counterweights are to compensate at least the most dominating forces and moments, usually first and second order. Their values follow from the calculations in section A5.1.

Let us return to the three options mentioned at the beginning of this subsection. Compensation masses, or counterweights, directly on the crank web will compensate the rotating part of the mass force a great deal (never completely), but does not compensate for the dominating first order translating part of the force.
If counterweights are applied, their mass is determined with help of the following equilibrium equation:

$$m_{rod,rot}\omega^2 R = m_{bal}\omega^2 d \qquad (a5.17)$$

$m_{rod,\,rot}$ $= rotating\ mass\ of\ the\ connecting\ rod$

ω $= angular\ speed\ of\ the\ crankshaft$

R $= crank\ length$

m_{bal} $= mass\ of\ counterweight$

d $= dis\tan ce\ from\ crankshaft\ to\ centre\ of\ gravity\ of\ counterweight$

Counterweights on both outer ends of the engine will compensate first as well as higher order translating forces. These masses would have their own shafts and would by driven separately in opposite directions. This option becomes disadvantageous as engine size increases, because the masses as well as their drivers would have to increase as well. An advantage is that the first order forces and moments can be completely eliminated.

From equation a5.11. it becomes clear that compensation of the forces and moments through plunger design will be aimed at changing the mass moment of inertia in such a way that the real moment of inertia becomes equal to the substituted moment of inertia. This can be achieved by adding corresponding masses at the plunger's ends as shown in figure A5.4.

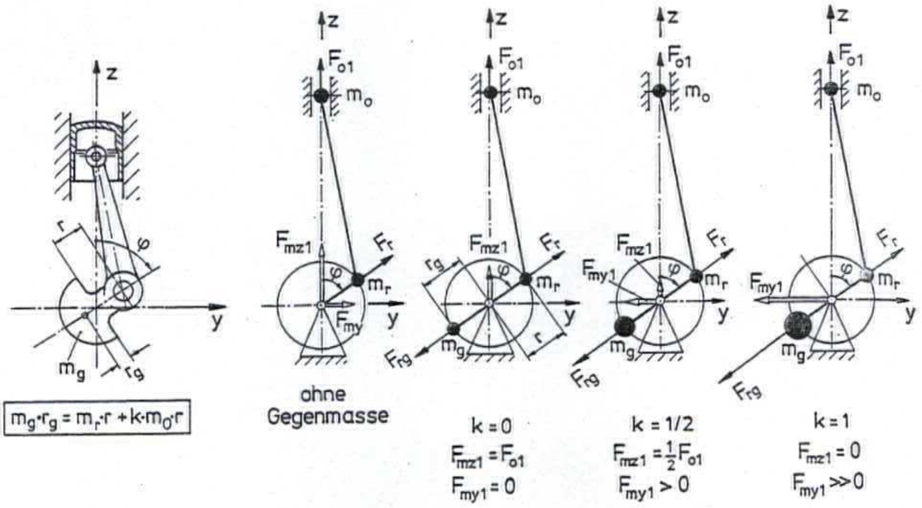

Figure A5.4. Additional masses placed at plunger's ends to compensate free forces and moments

Compensation measures for V-engines
Before implementing any compensating measures it should be seen to that the relative crank angles are chosen such that the most advantageous ignition cycle and maximum *natural* compensation are achieved. In the ideal case for a four-stroke engine this would result in a V-angle γ, which equals the ignition angle δ_{ign}.
In practice a uniform ignition distribution is realised by choosing the crank positions relative to the first crank. The relative displacement angle for the i^{th} cylinder will thus be:

$$\alpha_i = \gamma + i\delta_{ign} \tag{a5.18}$$

$i \quad = 0,1,2,...$

$\alpha_i \quad = relative\ angle$

$\gamma \quad = V - angle$

$\delta_{ign} \quad = ignition\ angle$

An illustration to clarify the theory is given in figure A5.5.

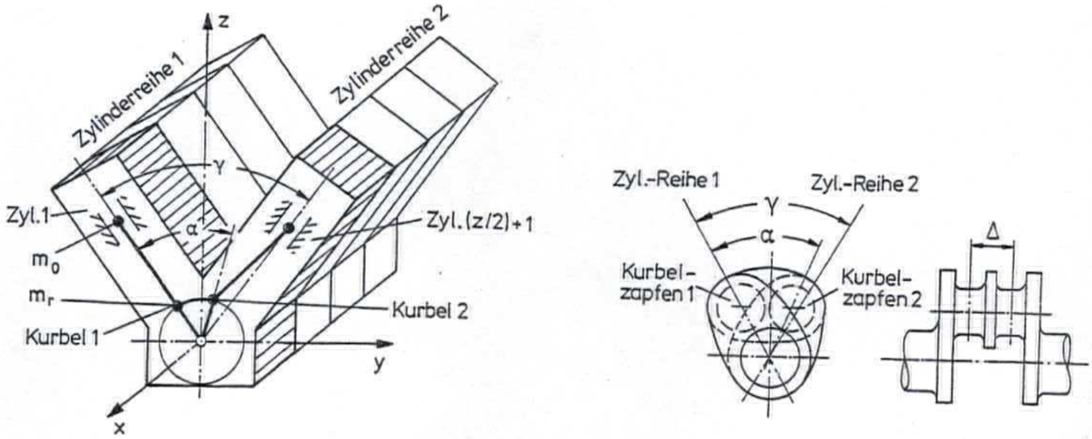

Figure A5.5. Relation between the displacement angle and the V-angle

It goes for both line- and V-engines that a symmetric design should be pursued to gain maximum balance. The symmetry affects the cylinder configuration as well as the crankshaft design.

APPENDIX 6: GENERAL DATA, NOISE & VIBRATION MEASUREMENTS

In chapter 3, section 2 some measurement results, concerning noise and vibration were given for various Oceanco yachts, to give an indication of the general situation on board. This appendix will embroider a bit more on this theme.
For a start the general ship data are given for all yachts under investigation. Secondly some more measurement results will be presented.

Before proceeding, some explanation about the yacht numbers is necessary. It was mentioned in chapter 3 that comparison was hardly possible, because the ships varied so much from each other. Here, data of other yachts belonging to the same series are presented. For two ships of the same series, comparison would be possible, though never a 100%, because specific wishes of the owner are to be respected at all times. Even if they result in a less optimal ship, considering the technical aspects.
Ships of a certain series are recognised by their number. For example: MY561 refers to Motor Yacht number 561; the number is an indication for the length and hull design of the ship. This means that all ships in this series have the same basic hull design and length, approximately 56[m] (length overall). The inward arrangement of accommodations etcetera is made by personal preference of the owner.
The final length of the ship may be increased by a few meters, this with during the hull design stage.
The 801 and 802, as insiders refer to them, form another category. Both ships have a length of approximately 80[m] (length overall).

A6.1. General Ship Data

Next ship particulars are presented for the 950, 800, 560 and 500 series as far as data were available.

MY950: Al Mirqab

Ship particulars		
Length p.p.	84.13 [m]	
Breadth	15.15 [m]	
Depth	7.38 [m]	
Max. draught	4.50 [m]	
Propulsion Machinery		
Number of units	3	
Manufacturer	MTU	
Type	20V1163 TB93L	
MCR	6720[kW]	
Speed	1310[rpm]	
Gearbox		
Number of units	3	
Manufacturer	Reintjes	
Type	WAF 6645	
Reduction	3.542:1	
Propeller		
Number of units	1	2
Manufacturer	Lips	Lips
Speed	369.8[rpm]	369.8[rpm]
Diameter	2.5[m]	2.5[m]
Number of Blades	5	4
Type	FP	AP
Auxiliary Engines		
Number of units	4	
Manufacturer	MTU	
Type	8V 183 TE52 370[kVA]	
Power	300[kW]	
Speed	1500[rpm]	

Sound reducing measures

Main and auxiliary engines resiliently mounted; floating floors

MY801/802: Constellation/Stargate

Ship particulars	
Length p.p.	70.20 [m]
Breadth	13.00 [m]
Depth	6.85 [m]
Max. draught	4.14 [m]
Propulsion Machinery	
Number of units	2
Manufacturer	MTU
Type	20V1163 TB73L
MCR	6500[kW]
Speed	1250[rpm]
Gearbox	
Number of units	2
Manufacturer	Reintjes
Type	WAF 5545 PU
Reduction	3.45:1
Propeller	
Number of units	2
Manufacturer	Lips
Speed	377[rpm]
Diameter	2.5[m]
Number of Blades	5
Type	FPP
Auxiliary Engines	
Number of units	3
Manufacturer	MTU
Type	12V 183 AA51
Power	218[kW]
Speed	1500[rpm]

MY561/562: Lady Christine/Pegasus

Ship particulars	
Length p.p.	49.80 [m]
Breadth	10.00 [m]
Depth	5.92 [m]
Max. draught	3.51 [m]
Propulsion Machinery	
Number of units	2
Manufacturer	Caterpillar
Type	3516B DI-TA/3512B DI-TA
MCR	1642[kW]/1231[kW]
Speed	1600[rpm]
Gearbox	
Number of units	2
Manufacturer	ZF
Type	BW751/BW466 G
Reduction	4.026:1/4.063:1
Propeller	
Number of units	2
Manufacturer	Lips
Speed	-[rpm]
Diameter	1.75[m]
Number of Blades	5
Type	
Auxiliary Engines	
Number of units	2
Manufacturer	Northern Lights/Caterpillar
Type	M6125T/3306B
Power	155[ekW]/171[kW]
Speed	-[rpm]/1500[rpm]

MY501/502: Accolade/Sunrise

Ship particulars

Length p.p.	45.00 [m]
Breadth	9.45 [m]
Depth	5.55 [m]
Draught at 50% load line	3.50 [m]
Propulsion Machinery	
Number of units	2
Manufacturer	MTU
Type	12V 396 TE 74
Power	1260[kW]
Speed	1900/2000[rpm]
Gearbox	
Number of units	2
Manufacturer	ZF
Type	BW466/BW461
Reduction	4.294:1
Propeller	
Number of units	2
Manufacturer	Lips
Speed	442.5/466[rpm]
Diameter	1.75[m]
Number of Blades	5
Type	FP
Auxiliary Engines	
Number of units	2
Manufacturer	MTU
Type	8V 183 AA51
Power	152[kW]
Speed	1500[rpm]

Sound reducing measures

Main and auxiliary engines resiliently mounted, gearboxes free-standing, (stiff) resilient

A6.2. Predictions and Measurements of Sound and Vibration Levels

Results of sound level predictions and seatrial measurements for the 801 and 561 were presented in chapter 3, section 2 of the main report. Results of two more ships are shown in this section, again, as far as data were available.

Pegasus- MY562			
Location	**Sound Level[dB(A)] -predicted-**	**Sound Level[dB(A)] -measured-**	**Vibration Level[mm/s] -measured-**
Engine Room		106	-
ECR		68	0.9
Crew Cabin		47	0.4
Guest Acc. (lower deck PS)		61	2.0
Guest Acc. (lower deck SB)		51	1.5
Owner's Stateroom		39	1.0
Captain's Cabin		43	-
Wheelhouse		54	1.0

Table A5.1. Noise and vibration levels for important locations on board the Pegasus (1200[rpm])

Sunrise- MY502			
Location	**Sound Level[dB(A)] -predicted-**	**Sound Level[dB(A)] -measured-**	**Vibration Level[mm/s] -measured-**
Engine Room	111	109	
PS ME Seating			6.6
PS ME Foundation			0.8
SB ME Seating			6.0
SB ME Foundation			0.6
PS Gearbox Seating			5.1
PS Gearbox Found.◊			0.8
SB Gearbox Seating			2.0
SB Gearbox Found.			0.7
ECR	72	74	0.6
Crew Cabin	48	45	0.8
Guest Acc. (lower deck PS)	53	51	0.9
Guest Acc. (lower deck SB)	54	57	0.6
Owner's Stateroom	49	43	0.4
Captain's Stateroom	56	43.5	0.8
Wheelhouse	55	52	0.3
◊Found.=foundation			

Table A5.2. Noise and vibration levels for important locations on board the Sunrise (1600-1700[rpm])

APPENDIX 7: LLOYD'S RULES FOR WINDOWS AND PORTHOLES

Windows are Portholes require special attention. They can contribute a great deal to the isolation of especially noise from outside, provided that the fitting dimensions are chosen and installation is properly performed.

Proper installation is an important factor considering dissipation of structure borne noise and vibration into the accommodations. In practice this will result in flexible connections between window casing and the wall, and between window casing and glass panels. Isolation of outdoor noise is obtained by the use of double glass. The required dimensions of the outer and inner glass plate, respectively, and the air gap between them, depends on the demands concerning the reduction and the location on the ship. Rules that are set up for this purpose supply minimum required thicknesses, for both outer and inner glass and the minimum required width of the air gap. It is always better to be on the safe side of course, so a safety margin is taken in.

The tables on the following pages are the "Rules for Windows and Portholes" supplied by Lloyd's Register for a 60[m]-yacht of Oceanco. The tables contain information about the dimensions of the windows, the required thicknesses, the actual thicknesses and the mass of the glass plates to be used.
The tables are taken up simply to give an idea of how a "window-plan" looks and how big margins could be.

Window thickness with laminated glass

window number	w [mm] (approx.)	h [mm] (approx.)	A [m^2] (approx.)	p	t [mm] acc. to SI	t_{req} acc. to LR	t_{clear} ACTUAL	t_{smoked} ACTUAL	m [kg/pst]	
Main deck										
Dining room										
M0	1300	950	1.24	17.5	13.12	12	0	12	74	
M1,M35	1300	950	1.24	17.5	13.12	12	0	12	74	
M2,M34	1300	950	1.24	17.5	13.12	12	0	12	74	
M3,M33	1300	950	1.24	17.5	13.12	12	0	12	74	
M4	1900	950	1.81	17.5	15.52	15	8	12	181	laminated glass
M32	1000	950	0.95	17.5	10.88	12	0	12	57	
Gym										
M5	1200	1050	1.26	17.5	12.88	12	0	12	76	
M6	1200	1300	1.56	17.5	14.53	15	8	12	156	laminated glass
M7	1200	1350	1.62	17.5	14.09	15	8	12	162	laminated glass
Galley										
M31	1200	1050	1.26	17.5	12.88	12	0	12	76	
M30	1200	1300	1.56	17.5	14.53	15	8	12	156	laminated glass
Side door										
M8,M29	850	1200	1.02	17.5	11.98	12	0	12	61	
Main lounge										
M9,M28	1250	1650	2.06	17.5	16.16	15	8	12	206	laminated glass
M10,M27	1250	1850	2.31	17.5	18.15	19	15	12	312	laminated glass
M12,M25	1250	1950	2.44	17.5	18.62	19	15	12	329	laminated glass
M13,M24	1250	1950	2.44	17.5	18.62	19	15	12	329	laminated glass
M14,M23	1250	1950	2.44	17.5	18.62	19	15	12	329	laminated glass

M15,M22	1100	1950	2.15	17.5	17.26	17	12	12	257	laminated glass
window number	**w [mm] (approx.)**	**h [mm] (approx.)**	**A [m^2] (approx.)**	**p**	**t [mm] acc. to SI**	**t_{req} acc. to LR**	**t_{clear} ACTUAL**	**t_{smoked} ACTUAL**	**m [kg/pst]**	
M16,M21	1100	1950	2.15	17.5	17.26	17	12	12	257	laminated glass
M17,M20	1100	1950	2.15	17.5	17.26	17	12	12	257	laminated glass
M18,M19	900	2050	1.85	17.5	15.19	15	12	12	221	laminated glass
Upper deck										
Wheelhouse										
U0	1100	900	0.99	17.5	11.63	12	0	12	59	
U1,U34	1100	900	0.99	17.5	11.63	12	0	12	59	
U2,U33	1100	900	0.99	17.5	11.63	12	0	12	59	
U3,U32	1100	900	0.99	17.5	11.63	12	0	12	59	
U4,U31	1100	900	0.99	17.5	11.63	12	0	12	59	
Lobby, Cap. Room										
U5,U30	1100	1200	1.32	17.5	13.02	12	0	12	79	
U6,U29	1100	1200	1.32	17.5	13.02	12	0	12	79	
U7,U28	1100	1200	1.32	17.5	13.02	12	0	12	79	
State room										
U8,U27	1250	800	1.00	17.5	11.91	12	0	12	60	
U9,U26	1250	1300	1.63	17.5	14.20	15	8	12	163	laminated glass
U10,U25	1250	1600	2.00	17.5	16.72	15	8	12	200	laminated glass
U11,U24	1250	1750	2.19	17.5	17.54	19	15	12	295	laminated glass
U12,U23	1250	1800	2.25	17.5	17.85	19	15	12	304	laminated glass
U13,U22	1250	1800	2.25	17.5	17.85	19	15	12	304	laminated glass
U14,U21	1100	1950	2.15	17.5	17.26	19	15	12	290	laminated glass
U15,U20	1100	1950	2.15	17.5	17.26	19	15	12	290	laminated glass

U16,U19	1100	1950	2.15	17.5	17.26	19	15	12	290	laminated glass
U17,U18	900	2050	1.85	17.5	15.19	15	8	12	185	laminated glass
window number	**w [mm] (approx.)**	**h [mm] (approx.)**	**A [m²] (approx.)**	**p**	**t [mm] acc. to SI**	**t_{req} acc. to LR**	**t_{clear} ACTUAL**	**t_{smoked} ACTUAL**	**m [kg/pst]**	
Fly-bridge										
Lounge										
F1,F12	950	2000	1.90	17.5	15.75	15	8	12	190	laminated glass
F2,F11	950	2000	1.90	17.5	15.75	15	8	12	190	laminated glass
F3,F10	1000	1950	1.95	17.5	16.20	15	8	12	195	laminated glass
F4,F9	1000	1950	1.95	17.5	16.20	15	8	12	195	laminated glass
F5,F8	1400	1950	2.73	17.5	19.56	15	8	12	273	laminated glass
F6,F7	1400	2000	2.80	17.5	19.90	15	8	12	280	laminated glass

Window thickness with double glass

window number	w [mm] (approx.)	h [mm] (approx.)	A [m^2] (approx.)	p	t [mm] acc. to SI	t_{req} acc. to LR	t_{clear} ACTUAL	t_{smoked} ACTUAL	m [kg/pst]	
M1,M2,M20, M21	2150	1750	3.76	17.5	22.71	22	22	6	1054	double glass
M3,M4,M18, M19	1600	1750	2.80	17.5	18.99	19	15	6	882	double glass
M5,M17	1600	800	1.28	17.5	13.07	12	0	12	77	
M6,M16	1600	700	1.12	17.5	11.83	12	0	12	67	
M7,M15	1450	500	0.73	17.5	8.76	12	0	12	44	
M8,M14	1200	1050	1.26	17.5	12.88	12	0	12	76	
M9,M13	1200	1050	1.26	17.5	12.88	12	0	12	76	
M10,M12	1200	1050	1.26	17.5	12.88	12	0	12	76	
M11	1200	1050	1.26	17.5	12.88	12	0	12	38	
U1,U13	2200	1600	3.52	17.5	22.22	22	22	6	493	double glass
U2,U12	2200	1600	3.52	17.5	22.22	19	15	6	370	double glass
U3,U11	2200	1250	2.75	17.5	19.57	19	15	6	289	double glass
U4,U10	1100	1050	1.16	17.5	12.01	12	12	0	69	
U5,U9	1100	1050	1.16	17.5	12.01	12	12	0	69	
U6,U8	1100	1050	1.16	17.5	12.01	12	12	0	0	
U7	1100	1050	1.16	17.5	12.01	12	12	0	35	
MA1-MA6	1100	1800	1.98	17.5	16.59	12	0	12	0	
UA1-UA6	1300	1800	2.34	17.5	19.61	12	0	12	0	
D1-D5	400	600	0.24	17.5	0.00	12	0	12		

Window thickness with single clear (white) glass

window number	w [mm] (approx.)	h [mm] (approx.)	A [m²] (approx.)	p	t [mm] acc. to SI	t_{req} acc. to LR	t_{clear} ACTUAL
M1,M2,M20, M21	2150	1750	3.76	17.5	22.71	22	22-25
M3,M4,M18, M19	1600	1750	2.80	17.5	18.99	19	15
M5,M17	1600	800	1.28	17.5	13.07	12	12
M6,M16	1600	700	1.12	17.5	11.83	12	12
M7,M15	1450	500	0.73	17.5	8.76	12	12
M8,M14	1200	1050	1.26	17.5	12.88	12	12
M9,M13	1200	1050	1.26	17.5	12.88	12	12
M10,M12	1200	1050	1.26	17.5	12.88	12	12
M11	1200	1050	1.26	17.5	12.88	12	12
U1,U13	2200	1600	3.52	17.5	22.22	22	22-25
U2,U12	2200	1600	3.52	17.5	22.22	19	15
U3,U11	2200	1250	2.75	17.5	19.57	19	15
U4,U10	1100	1050	1.16	17.5	12.01	12	12
U5,U9	1100	1050	1.16	17.5	12.01	12	12
U6,U8	1100	1050	1.16	17.5	12.01	12	12
U7	1100	1050	1.16	17.5	12.01	12	12
MA1-MA6	1100	1800	1.98	17.5	16.59	12	12
UA1-UA6	1300	1800	2.34	17.5	19.61	12	12
D1-D5	400	600	0.24	17.5	0.00	12	12

The picture below illustrates what a double glazed window will finally look like when it is integrated into the wall's structure.

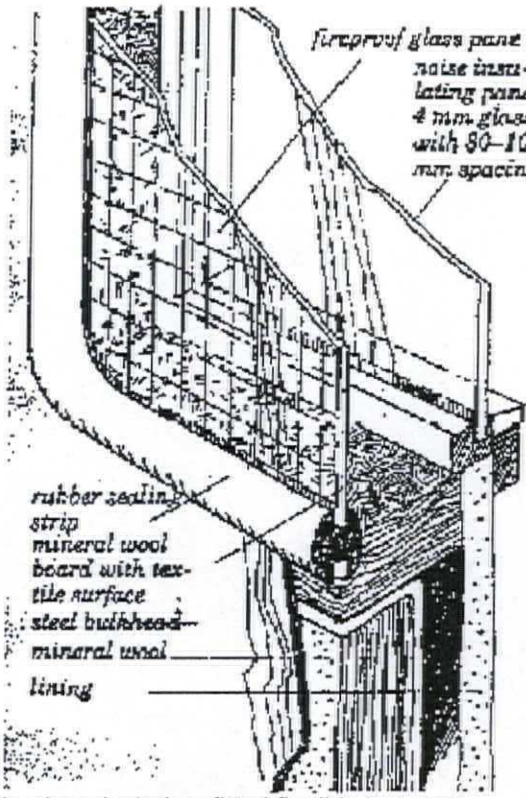

Figure A7.1. A double glazed window fitted flexibly in a wall

APPENDIX 8: INSULATION AND FIRE RESISTANT MATERIALS

A complete insulation plan consists of three parts: fire protection, acoustic insulation and vibration insulation. For each area on board specific requirements are stated. The requirements are categorised in *insulation classes*. In yacht building classes A and B are the most important.

A8.1. Lloyd's Rules on Insulation

Definitions of A and B class insulation according to Lloyd's Rules[1] are given below:

*"**2.4 Ship divisions and spaces***
***2.4.1**. "A' Class divisions' are those divisions formed by bulkheads and decks which comply with the following:*
(a) They are to be constructed of steel or other equivalent material.
(b) They are to be suitably stiffened.
(c) They are to be so constructed as to be capable of preventing the passage of smoke and flame up to the end of the one-hour standard fire test.
(d) They are to be insulated with approved non-combustible materials such that the average temperature of the unexposed side will not rise more than 139°C above the original temperature, nor will the temperature, at any one point, including any joint, rise more than 180°C above the original temperature, within the time listed below:

Class 'A-60' 60 minutes.
Class 'A-30' 30 minutes.
Class 'A-15' 15 minutes.
Class 'A-0' 0 minutes.

(e) A test of a prototype bulkhead or deck may be required to ensure that it meets the above requirements for integrity and temperature rise.

***2.4.2.** "B' Class divisions' are those divisions formed by bulkheads, decks, ceilings or linings which comply with the following:*
(a) They are to be so constructed as to be capable of preventing the passage of flame to the end of the first half-hour of the standard fire test.
(b) They are to have an insulation value such that the aver-age temperature of the unexposed side will not rise more than 139°C above the original temperature, nor will the temperature at any one point, including any joint, rise more than 225°C above the original temperature, within the time listed below:

Class 'B-15' 15 minutes.
Class 'B-0' 0 minutes.

(c) They are to be constructed of approved non-combustible materials and all materials entering into the construction and erection of 'B' Class divisions are to be non-combustible, except where permitted by other requirements of this Chapter.
(d) A test of a prototype division may be required to ensure that it meets the above requirements for integrity and temperature rise.

***2.4.3.** "C' Class divisions' are divisions to be constructed of approved non-combustible materials. They need meet neither requirements relative to the passage of smoke and flame nor*

[1] Source: Rules and Regulations for the Classification of Special Service Craft; Fire Protection, Detection and Extinction-General; Part 17 Chapter 1, Section 2

limitations relative to the temperature rise. Combustible veneers are permitted provided they meet other requirements of this Chapter.
2.4.4. *'Continuous 'B' Class ceilings or linings' are those 'B' Class ceilings or linings which terminate only at an 'A' or 'B' Class division.*

2.4.5. *'Accommodation spaces' are those spaces used for public spaces, corridors, lavatories, cabins, offices, hospitals, cinemas, games and hobbies rooms, pantries containing no cooking appliances and similar spaces.*

2.4.6. *'Service spaces' are those used for galleys, pantries containing cooking appliances, stores, mail and specie rooms, store rooms, lockers, workshops other than those forming part of the machinery spaces and similar spaces and trunks to such spaces.*

2.4.7. *'Cargo spaces' are all spaces used for cargo (including cargo oil tanks) and trunks to such spaces.*

2.4.8. *'Machinery spaces of Category A' are those spaces and trunks to such spaces which contain:*
(a) internal combustion machinery used for main propulsion; or
(b) internal combustion machinery used for purposes other than main propulsion where such machinery has in the aggregate a total power output of not less than 375[kW]; or
(c) any oil-fired boiler or oil fuel unit.

2.4.9. *'Machinery spaces' are all machinery spaces of Category 'A' and all other spaces containing propelling machinery, boilers, oil fuel units, steam and internal combustion engines, generators and major electrical machinery, oil filling stations, refrigerating, stabilising, ventilation and air conditioning machinery, and similar spaces; and trunks to such spaces.*

2.4.10. *'Control stations' are those spaces in which the craft's radio or main navigating equipment or the emergency source of power is located or where the fire recording or fire-control equipment is* centralised.

2.4.11. *'Cargo area' is that part of the craft that contains cargo tanks, slop tanks and cargo pump rooms including pump rooms, cofferdams, ballast and void spaces adjacent to cargo tanks and also deck areas throughout the entire length and breadth of the part of the craft over the above-mentioned spaces.*

2.4.12. *'Main vertical zones' are those sections into which the hull, superstructure and deck houses are divided by 'A' Class divisions, the mean length and width of which on any one deck does not, in general, exceed 40 m."*

In the following the impact of these kinds of measurements on board is illustrated by the MCA[2] code and by the IMO. Both are based on Lloyd's Rules and use the definitions quoted above.
Besides demanding minimum insulation the rules prescribe dimensions of plates and piping, required clearances, etcetera. All spaces involved in the plan are first categorised as either A or B class. The *resistance level* (0,15,30,60) can be derived from the tables.

For example: From table 1 in Lloyd's rules it can be derived that between a control station and an accommodation A-60 insulation is required. For accommodation spaces adjacent to a stairway A-0 and B-0 insulation are required.

The following pages contain the MCA code on fire protection on board new vessels (7 pages) and the SOLAS[3] code on fire protection for vessels carrying no more than 36 passengers (3 pages).

[2] Maritime and Coastguard Agency
[3] Safety of Life at Sea

A8.2. The code of practice for safety of large commercial sailing and motor vessels (MCA)

14B.3 New Vessels

New vessels should comply with the following:

14B.3.1 Ventilation systems

14B.3.1.1 Ventilation ducts should be of non-combustible material. Short ducts, however, not generally exceeding 2m in length and with a cross-section not exceeding $0.02m^2$ need not be non-combustible, subject to the following conditions:

.1 they should be of a suitable material having regard to the risk of fire;

.2 they should be used only at the end of the ventilation device; and

.3 they should not be situated less than 600mm, measured along the duct, from an opening in an "A" or "B" class division including continuous "B" class ceilings.

14B.3.1.2 Where ventilation ducts with a free cross-sectional area exceeding $0.02m^2$ pass through class "A" bulkheads or decks, the opening should be lined with a steel sheet sleeve unless the ducts passing through the bulkheads or decks are of steel in the vicinity of passage through the deck or bulkhead and the ducts and sleeves should comply in this part with the following:

.1 Sleeves should have a thickness of at least 3mm and a length of at least 900mm. When passing through bulkheads, this length should be divided preferably into 450mm on each side of the bulkhead. The ducts, or sleeves lining such ducts, should be provided with fire insulation. The insulation should have at least the same fire integrity as the bulkhead or deck through which the duct passes.

.2 Ducts with a free cross-sectional area exceeding $0.075m^2$ should be fitted with fire dampers in addition to the requirements of .1 above. The fire damper should operate automatically but should also be capable of being closed manually from both sides of the bulkhead or deck. The damper should be provided with an indicator which shows whether the damper is open or closed. Fire dampers are not required, however, where ducts pass through spaces surrounded by "A" class divisions, without serving those spaces, provided those ducts have the same fire integrity as the divisions which they pierce.

14B.3.1.3 Ducts provided for the ventilation of a machinery space of category A or a galley, should not pass through accommodation spaces, service spaces or control stations unless they comply with the conditions specified in .1 to .4 or .5 and .6 below:

.1 they are constructed of steel having a thickness of at least 3mm and 5mm for duct widths or diameters of up to and including 300mm and 760mm and over respectively and, in the case of ducts with widths or diameters between 300mm and 760mm thickness should be obtained by interpolation;

.2 they are suitably supported and stiffened;

.3 they are fitted with automatic fire dampers close to the boundaries penetrated; and

.4 they are insulated to "A-60" standard from a machinery space or galley to a point at least 5m beyond each fire damper;

or

.5 they are constructed of steel in accordance with .1 and .2 above; and

.6 they are insulated to "A-60" standard throughout accommodation spaces, service spaces or control stations;

except that penetrations of main zone divisions should also comply with the requirements of 14B.3.1.8.

14B.3.1.4 Ducts provided for ventilation to accommodation spaces, service spaces or control stations should not pass through a machinery space of category A or a galley unless they comply with the conditions specified in .1 to .3 or .4 and .5 below:

.1 where they pass through a machinery space of category A or galley, ducts are constructed of steel in accordance with 14B.3.1.3.1 & .2;

.2 automatic fire dampers are fitted close to the boundaries penetrated; and

.3 the integrity of the machinery space or galley boundaries is maintained at penetrations;

or

.4 where they pass through a machinery space of category A or galley, ducts are constructed of steel in accordance with 14B.3.1.3.1 & .2; and

.5 within a machinery space or galley, ducts are insulated to "A-60" standard;

except that penetrations of main zone divisions should also comply with the requirements of 14B.3.1.8.

14B.3.1.5 Ventilation ducts with a free cross-sectional area exceeding $0.02m^2$ passing through "B" class bulkheads should be lined with steel sheet sleeves of 900mm in length divided preferably into 450mm on each side of the bulkheads, unless the duct is of steel for this length.

14B.3.1.6 For a control station outside machinery spaces, practical measures should be taken to ensure that ventilation, visibility and freedom from smoke are maintained so that, in the event of fire, the machinery and equipment contained in the control station may be supervised and continue to function effectively. Alternative and separate means of air supply should be provided; air inlets of the two sources of supply should be so disposed that the risk of both inlets drawing in smoke simultaneously is minimized. These requirements need not apply to control stations situated on, and opening on to, an open deck, or where local closing arrangements would be equally effective.

14B.3.1.7 Exhaust duct(s) from a galley range should be constructed of "A" class divisions where it passes through accommodation spaces and/or spaces containing combustible materials. An exhaust duct should be fitted with:

.1 a grease trap readily removable for cleaning;

.2 a fire damper located in the lower end of the duct;

.3 arrangements for shutting off the exhaust fans, operable from within the galley; and

.4 fixed means for extinguishing a fire within the duct.

14B.3.1.8 When it is necessary for a ventilation duct to pass through a main vertical zone division, a fail-safe automatic closing fire damper should be fitted adjacent to the division. The damper should also be capable of being manually closed from each side of the division. The operating position should be readily accessible and be marked in red light-reflecting colour. The duct between the division and the damper should be of steel or other equivalent material and, if necessary, insulated to comply with the requirements of SOLAS regulation II-2/18.1.1. The damper should be fitted on at least one side of the division with a visible indicator showing whether the damper is in the open position.

14B.3.1.9 Inlets and outlets of ventilation systems should be capable of being closed from outside the space being ventilated.

14B.3.1.10 Power ventilation of accommodation spaces, service spaces, control stations and machinery spaces should be capable of being stopped from an easily accessible position outside the space being served. This position should not be readily cut off in the event of a fire in the spaces served. The means provided for stopping the power ventilation of a machinery space should be entirely separate from the means provided for stopping ventilation of other spaces.

14B.3.2 Structure

14B.3.2.1 The hull, superstructures, structural bulkheads, decks and deckhouses should be constructed of steel or other equivalent material.

14B.3.2.2 However, in cases where any part of the structure is of aluminium alloy, the following should apply:

.1 Insulation of aluminium alloy components of "A" or "B" class divisions, except structure which, in the opinion of the Administration, is non-load-bearing, should be such that the temperature of the structural core does not rise more than 200°C above the ambient temperature at any time during the applicable fire exposure to the standard fire test.

.2 Special attention should be given to the insulation of aluminium alloy components of columns, stanchions and other structural members required to support lifeboat and liferaft stowage, launching and embarkation areas, and "A" and "B" class divisions to ensure that for members:

(a) supporting lifeboat and liferaft areas and "A" class divisions, the temperature rise limitation specified in .1 above should apply at the end of one hour; and

(b) supporting "B" class divisions, the temperature rise limitation specified in .1 above should apply at the end of half an hour.

14B.3.2.3 Crowns and casings of a machinery space of category A should be of steel construction adequately insulated and openings therein, if any, should be suitably arranged and protected to prevent the spread of fire.

14B.3.2.4 In a vessel of less than 1000 GT, crowns and casings of a machinery space of category A need not be of steel provided they are "A-60" divisions and provision is made for boundary cooling through two fire hoses supplied simultaneously from the emergency fire pump with drainage of cooling water overside through scuppers of suitable capacity.

14B.3.3 Main vertical zones and horizontal zones

14B.3.3.1 Hull superstructure and deckhouses in way of accommodation and service spaces should be subdivided into main vertical zones by "A" class divisions. These divisions should have insulation values in accordance with tables 1 and 2.

14B.3.3.2 As far as practical, the bulkheads forming the boundaries of the main vertical zones should be in line with watertight subdivision bulkheads.

14B.3.3.3 Such bulkheads should extend from deck to deck and to the shell or other boundaries.

14B.3.3.4 When a main vertical zone is subdivided by "A" class divisions for the purpose of providing an appropriate barrier between sprinklered and non-sprinklered spaces, the divisions should be insulated in accordance with the fire insulation and integrity values given in tables 1 and 2.

14B.3.4 Bulkheads within a main vertical zone

14B.3.4.1 All bulkheads within accommodation and service spaces which are not required to be "A" class divisions should be at least "B" class or "C" class divisions as prescribed in the tables 1 and 2.

14B.3.4.2 All such divisions may be faced with combustible materials in accordance with the provisions of 14B.3.11.

14B.3.4.3 All corridor bulkheads where not required to be "A" class should be "B" class divisions which should extend from deck to deck except:

.1 when continuous "B" class ceilings or linings are fitted on both sides of the bulkhead, the portion of the bulkhead behind the continuous ceilings or linings should be of material which, in thickness and composition, is acceptable in the construction of "B" class divisions but which should be required to meet "B" class integrity standards only in so far as is reasonable and practical in the opinion of the Administration;

.2 throughout spaces protected by an automatic sprinkler, fire detection and fire alarm system complying with the provisions of 14B.3.13.1.2, the corridor bulkheads of "B" class materials may terminate at a ceiling in the corridor provided such a ceiling is of material which, in thickness and composition, is acceptable in the construction of "B" class divisions. Notwithstanding the requirements of 14B.3.5, such bulkheads and ceilings should be required to meet "B" class integrity standards only in so far as is reasonable and practical. All doors and frames in such bulkheads should be so constructed and erected to provide substantial fire resistance.

14B.3.4.4 All bulkheads required to be "B" class divisions, except corridor bulkheads, should extend from deck to deck and to the shell or other boundaries unless continuous "B" class ceilings or linings are fitted on both sides of the bulkhead, in which case the bulkhead may terminate at the continuous ceiling or lining.

14B.3.5 Fire integrity of bulkheads and decks

14B.3.5.1 In addition to complying with the specific provisions for fire integrity of bulkheads and decks mentioned elsewhere in this section, the minimum fire integrity of bulkheads and decks should be as prescribed in tables 1 and 2.

14B.3.5.2 The following requirements should govern application of the tables:

.1 Tables 1 and 2 should apply respectively to the bulkheads and decks separating adjacent spaces.

.2 For determining the appropriate fire integrity standards to be applied to divisions between adjacent spaces, such spaces are classified according to their fire risk as shown in categories (1) to (9) below. The title of each category is intended to be typical rather than restrictive. The number in parentheses preceding each category refers to the applicable column or row in the tables.

(1) *Control stations*

Spaces containing emergency sources of power and lighting.
Wheelhouse and chartroom.
Spaces containing the vessel's radio equipment.
Fire-extinguishing rooms, fire control rooms and fire-recording stations.
Control room for propulsion machinery when located outside the machinery space.
Spaces containing centralized fire alarm equipment.

(2) *Corridors and lobbies*

(3) *Accommodation spaces*

Spaces so defined, excluding corridors.

(4) *Stairways*

Interior stairways, lifts and escalators (other than those wholly contained within the machinery space(s)) and enclosures thereto.

In this connection, a stairway which is enclosed only at one level should be regarded as part of the space from which it is not separated by a fire door.

(5) *Service spaces (low risk)*

Lockers and store-rooms not having provisions for the storage of flammable liquids and having areas less than $4m^2$ and drying rooms and laundries.

(6) *Machinery spaces of category A*

Spaces so defined.

(7) *Other machinery spaces*

Spaces so defined, excluding machinery spaces of category A.

(8) *Service spaces (high risk)*

Galleys, pantries containing cooking appliances, paint and lamp rooms, lockers and store-rooms having areas of $4m^2$ or more, spaces for the storage of flammable liquids, and workshops other than those forming part of the machinery spaces.

(9) *Open decks*

Open deck spaces and enclosed promenades having no fire risk. Air spaces (the space outside superstructures and deckhouses).

.3 In determining the applicable fire integrity standard of a boundary between two spaces within a main vertical zone or horizontal zone which is not protected by a sprinkler system complying with the provisions of 14B.3.13.1.2 or between such zones neither of which is so protected, the higher of the two values given in the tables should apply.

.4 In determining the applicable fire integrity standard of a boundary between two spaces within a main vertical zone or horizontal zone which is protected by a sprinkler system complying with the provisions of 14B.3.13.1.2 or between such zones both of which are so protected, the lesser of the two values given in the tables should apply. Where a sprinklered zone and a non-sprinklered zone meet within accommodation and service spaces, the higher of the two values given in the tables should apply to the division between the zones.

14B.3.5.3 Continuous "B" class ceilings or linings, in association with the relevant decks or bulkheads, may be accepted as contributing, wholly or in part, to the required insulation and integrity of a division.

14B.3.5.4 External boundaries which are required to be of steel or other equivalent material may be pierced for the fitting of windows and side scuttles provided that there is no requirement for such boundaries to have "A" class integrity elsewhere in this section. Similarly, in such boundaries which are not required to have "A" class integrity, doors may be of combustible materials, substantially constructed.

Table 1 - Fire integrity of bulkheads separating adjacent spaces

Spaces	(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
Control stations (1)	A-0_c	A-0	A-60	A-0	A-15	A-60	A-15	A-60	*
Corridors and lobbies (2)		C_e	B-0_e	A-0_a B-0_e	B-0_e	A-60	A-0	A-15 A-0_d	*
Accommodation spaces (3)			C_e	A-0_a B-0_e	B-0_e	A-60	A-0	A-15 A-0_d	*
Stairways (4)				A-0_a B-0_e	A-0_a B-0_e	A-60	A-0	A-15 A-0_d	* *
Service spaces (low risk) (5)					C_e	A-60	A-0	A-0	*
Machinery spaces of category A (6)						*	A-0	A-60	*
Other machinery spaces (7)							A-0_b	A-0	*
Service spaces (high risk) (8)								A-0_b	*
Open decks (9)									

Table 2 - Fire integrity of decks separating adjacent spaces

Spaces above Spaces below	(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
Control stations (1)	A-0	A-0	A-0	A-0	A-0	A-60	A-0	A-0	*
Corridors and lobbies (2)	A-0	*	*	A-0	*	A-60	A-0	A-0	*
Accommodation spaces (3)	A-60	A-0	*	A-0	*	A-60	A-0	A-0	*
Stairways (4)	A-0	A-0	A-0	*	A-0	A-60	A-0	A-0	*
Service spaces (low risk) (5)	A-15	A-0	A-0	A-0	*	A-60	A-0	A-0	*
Machinery spaces of category A (6)	A-60	A-60	A-60	A-60	A-60	*	A-60$_f$	A-60	*
Other machinery spaces (7)	A-15	A-0	A-0	A-0	A-0	A-0	*	A-0	*
Service spaces (high risk) (8)	A-60	A-30 A-0$_d$	A-30 A-0$_d$	A-30 A-0$_d$	A-0	A-60	A-0	A-0	*
Open decks (9)	*	*	*	*	*	*	*	*	-

Notes: To be applied to both tables 1 and 2, as appropriate.

a For clarification on which applies, see 14B.3.4 and 14B.3.7.

b Where spaces are of the same numerical category and subscript $_b$ appears, a bulkhead or deck of the rating shown in the tables is only required when the adjacent spaces are for a different purpose, e.g in category (9), a galley next to a galley does not require a bulkhead but a galley next to a paint room requires an "A-0" bulkhead.

c Bulkheads separating the wheelhouse and chartroom from each other may be "B-0" rating.

d See 14B.3.5.2.3 and 14B.3.5.2.4.

e For the application of 14B.3.3.1, "B-0" and "C", where appearing in table 1, should be read as "A-0".

f Fire insulation need not be fitted if the machinery space in category (7), in the opinion of the Administration, has little or no fire risk.

* Where an asterisk appears in the tables, the division is required to be of steel or other equivalent material but is not required to be of "A" class standard.

For the application of 14B.3.3.1 an asterisk, where appearing in table 2, except for category (9), should be read as "A-0".

A8.3. Fire Integrity of Bulkheads and Decks (SOLAS)

- in ships carrying not more than 36 passengers-

(Paragraphs 2.2(5) and 2.2(9) of this regulation apply to ships constructed on or after 1 February 1992)

1 In addition to complying with the specific provisions for fire integrity of bulkheads and decks mentioned elsewhere in this part, the minimum fire integrity of bulkheads and decks shall be as prescribed in table 27.1 and table 27.2.

2 The following requirements shall govern application of the tables:

.1 Tables 27.1 and 27.2 shall apply respectively to the bulkheads and decks separating adjacent spaces.

.2 For determining the appropriate fire integrity standards to be applied to divisions between adjacent spaces, such spaces are classified according to their fire risk as shown in categories (1) to (11) below. The title of each category is intended to be typical rather than restrictive. The number in parentheses preceding each category refers to the applicable column or row in the tables.

(1) *Control stations*

Spaces containing emergency sources of power and lighting.

Wheelhouse and chartroom.

Spaces containing the ship's radio equipment.

Fire-extinguishing rooms, fire control rooms and fire-recording stations.

Control room for propulsion machinery when located outside the machinery space.

Spaces containing centralized fire alarm equipment.

(2) *Corridors*

Passenger and crew corridors and lobbies.

(3) *Accommodation spaces*

Spaces as defined in regulation 3.10 excluding corridors.

(4) *Stairways*

Interior stairways, lifts and escalators (other than those wholly contained within the machinery spaces) and enclosures thereto.

In this connection, a stairway which is enclosed only at one level shall be regarded as part of the space from which it is not separated by a fire door.

Table 27.1 – Fire integrity of bulkheads separating adjacent spaces

Spaces	(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)
Control stations (1)	A-0[c]	A-0	A-60	A-0	A-15	A-60	A-15	A-60	A-60	*	A-60
Corridors (2)		C[e]	B-0[e]	A-0[a] B-0[e]	B-0[e]	A-60	A-0	A-0	A-15 A-0[d]	*	A-15
Accommodation spaces (3)			C[e]	A-0[a] B-0[e]	B-0[e]	A-60	A-0	A-0	A-15 A-0[d]	*	A-30 A-0[d]
Stairways (4)				A-0[a] B-0[e]	A-0[a] B-0[e]	A-60	A-0	A-0	A-15 A-0[d]	* *	A-15
Service spaces (low risk) (5)					C[e]	A-60	A-0	A-0	A-0	*	A-0
Machinery spaces of category A (6)						*	A-0	A-0	A-60	*	A-60
Other machinery spaces (7)							A-0[b]	A-0	A-0	*	A-0
Cargo spaces (8)								*	A-0	*	A-0
Service spaces (high risk) (9)									A-0[b]	*	A-30
Open decks (10)											A-0
Special category spaces (11)											A-0

Notes: To be applied to both tables 27.1 and 27.2, as appropriate.

a For clarification as to which applies, see regulations 25 and 29.

b Where spaces are of the same numerical category and superscript b appears, a bulkhead or deck of the rating shown in the tables is only required when the adjacent spaces are for a different purpose, e.g. in category (9). A galley next to a galley does not require a bulkhead but a galley next to a paint room requires an "A-0" bulkhead.

c Bulkheads separating the wheelhouse and chartroom from each other may be "B-0" rating.

d See 2.3 and 2.4 of this regulation.

e For the application of regulation 24.1.2, "B-0" and "C", where appearing in table 27.1, shall be read as "A-0".

f Fire insulation need not be fitted if the machinery space in category (7), in the opinion of the Administration, has little or no fire risk.

* Where an asterisk appears in the tables, the division is required to be of steel or other equivalent material but is not required to be of "A" class standard.

For the application of regulation 24.1.2 an asterisk, where appearing in table 27.2, except for categories (8) and (10), shall be read as "A-0".

(5) *Service spaces (low risk)*

Lockers and store-rooms not having provisions for the storage of flammable liquids and having areas less than 4 m^2 and drying rooms and laundries.

(6) *Machinery spaces of category A*

Spaces as defined in regulation 3.19.

(7) *Other machinery spaces*

Spaces as defined in regulation 3.20 excluding machinery spaces of category A.

(8) *Cargo spaces*

All spaces used for cargo (including cargo oil tanks) and trunkways and hatchways to such spaces, other than special category spaces.

Table 27.2 – Fire integrity of decks separating adjacent spaces

Space below / Space above	(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)
Control stations (1)	A-0	A-0	A-0	A-0	A-0	A-60	A-0	A-0	A-0	*	A-30
Corridors (2)	A-0	*	*	A-0	*	A-60	A-0	A-0	A-0	*	A-0
Accommodation spaces (3)	A-60	A-0	*	A-0	*	A-60	A-0	A-0	A-0	*	A-30 A-0[d]
Stairways (4)	A-0	A-0	A-0	*	A-0	A-60	A-0	A-0	A-0	*	A-0
Service spaces (low risk) (5)	A-15	A-0	A-0	A-0	*	A-60	A-0	A-0	A-0	*	A-0
Machinery spaces of category A (6)	A-60	A-60	A-60	A-60	A-60	*	A-60[f]	A-30	A-60	*	A-60
Other machinery spaces (7)	A-15	A-0	A-0	A-0	A-0	A-0	*	A-0	A-0	*	A-0
Cargo spaces (8)	A-60	A-0	A-0	A-0	A-0	A-0	A-0	*	A-0	*	A-0
Service spaces (high risk) (9)	A-60	A-30 A-0[d]	A-30 A-0[d]	A-30 A-0[d]	A-0	A-60	A-0	A-0	A-0	*	A-30
Open decks (10)	*	*	*	*	*	*	*	*	*	–	A-0
Special category spaces (11)	A-60	A-15	A-30 A-0[d]	A-15	A-0	A-30	A-0	A-0	A-30	A-0	A-0

See notes under table 27.1.

(9) *Service spaces (high risk)*

Galleys, pantries containing cooking appliances, paint and lamp rooms, lockers and store-rooms having areas of 4 m^2 or more, spaces for the storage of flammable liquids, and workshops other than those forming part of the machinery spaces.

(10) *Open decks*

Open deck spaces and enclosed promenades having no fire risk. Air spaces (the space outside superstructures and deckhouses).

(11) *Special category spaces*

Spaces as defined in regulation 3.18.

.3 In determining the applicable fire integrity standard of a boundary between two spaces within a main vertical zone or horizontal zone which is not protected by an automatic sprinkler system complying with the provisions of regulation 12 or between such zones neither of which is so protected, the higher of the two values given in the tables shall apply.

.4 In determining the applicable fire integrity standard of a boundary between two spaces within a main vertical zone or horizontal zone which is protected by an automatic sprinkler system complying with the provisions of regulation 12 or between such zones both of which are so protected, the lesser of the two values given in the tables shall apply. Where a sprinklered zone and a non-sprinklered zone meet within accommodation and service spaces, the higher of the two values given in the tables shall apply to the division between the zones.

3 Continuous "B" class ceilings or linings, in association with the relevant decks or bulkheads, may be accepted as contributing, wholly or in part, to the required insulation and integrity of a division.

4 External boundaries which are required in regulation 23.1 to be of steel or other equivalent material may be pierced for the fitting of windows and sidescuttles provided that there is no requirement for such boundaries to have "A" class integrity elsewhere in this part. Similarly, in such boundaries which are not required to have "A" class integrity, doors may be of materials to the satisfaction of the Administration.