Safety at Speed - S@S

FORMULATION OF MODEL FOR HUMAN FACTOR

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CONTENTS

ABBREVIATIONS AND ACRONYMS

β	heading to waves [deg]
η_2	sway motion [m]
η_3	heave motion [m]
η_4	roll motion [deg]
η_5	pitch motion [deg]
η_6	yaw motion [deg]
σ_{Li}	mean square of lateral force estimators
τ	integration time for running averaging
a_b	ISO 2631-1 motion sickness boundary value for vertical acceleration [m/s ^{2}]
a_L	lateral acceleration [m/s ^{2}]
a_V	vertical acceleration [m/s ^{2}]
a_{0w}	frequency weighted rms vertical acceleration for unit significant wave height [m/s ^{2}]
$a_w(t)$	frequency weighted rms acceleration in vertical direction a_w [m/s ^{2}]
$a_{w,e}$	equivalent acceleration rms value corresponding to total duration of two or more periods of exposure [m/s ^{2}]
eVDV	estimated Vibration Dose Value [m/s $^{1.75}$]
g	acceleration due to gravity [m/s ^{2}]
h	height of centre of gravity of a person [m]
H_s	significant wave height [m]
H_{si}	operability-limiting significant wave height with regard to motion induced interruptions [m]
H_{ss}	operability-limiting significant wave height with regard to the motion sickness [m]
HVAC	heating, ventilation and air conditioning
ISSC	The International Ship and Offshore Structures Congress
K_m	constant in MSI formula
l	width of stance of a person [m]
L_1, L_2	lateral force estimators 1 and 2
L_A	A-weighted sound pressure level [dB]
$L_{Aeq,8h}$	equivalent continuous A-weighted sound pressure level over 8 hours interval [dB]
$L_{Aeq,24h}$	equivalent continuous A-weighted sound pressure level over 24 hours interval [dB]
$L_{ex,8h}$	noise exposure in 8 hours
m	mass of a person [kg]
m_{0Li}	area of lateral force estimator spectrum
m_{nLi}	nth moment of lateral force estimator spectrum
M_t	total number of motion induced interruptions during T_t
MII	motion-induced interruptions per minute
MSDV	motion sickness dose value [m/s $^{1.5}$]
MSI	motion sickness incidence [1/min]
MTVV	maximum transient vibration value [m/s ^{2}]
NR	Noise Rating
P	probability of exceedance

p_{peak}	peak pressure of noise [dB]
PWL	Sound Power Level [dB]
RAO	Response Amplitude Operator, transfer function
$R_{Li}(\omega)$	transfer functions of lateral force estimators L_i
rmq	root-mean-quad of vertical acceleration [$m/s^{1.75}$]
$S(\omega)$	wave spectral density
$S_{Li}(\omega)$	spectral density of lateral force estimator i
SHP	Main engine shaft horse power (PS)
SIL	Speech Interference Level [dB]
SPL	Sound Pressure Level [dB]
T, T_0, T_t	exposure times [s]
T_{zLi}	zero-crossing period of the lateral force estimator L_i [s]
V	ship speed [m/s]
VDV	Vibration Dose Value [$m/s^{1.75}$]
W_d	frequency weighting used to evaluate x- and y-axis vibration
W_f	frequency weighting used to evaluate z-axis motion with respect to motion sickness
W_k	frequency weighting used to evaluate z-axis vibration

1. EXECUTIVE SUMMARY SUITABLE FOR PUBLICATION

This report presents the work carried out in Sub-task 2.2.1 of Work Package 2 – Ship Motions Hazards of Safety At Speed (S@S). It contains the formulation of risk and cost models concerning motion sickness, safety of footing, noise, vibration and indoor climate.

Methods defining different characteristics associated with motion sickness and safety of footing of a fast passenger ship have been presented. The characteristics calculated using these methods can be compared with acceptable limit values already in the preliminary design phase of a ship. The input data for these methods will be obtained from simplified seakeeping calculation methods developed in sub-task 2.2.2 of WP2. The acceptable limit values have been stated on the basis of existing material including the latest international standards.

The hazards related to excessive noise and vibration, as well as the effect of the parameters of the indoor climate to the comfort of the passengers and the crew have been analysed. Suitable criteria ant pertinent limits are presented as a basis for simplified models for the evaluation of the effect of noise, vibration and indoor climate on passenger comfort and crew workability. Several methods for the prediction of noise and vibration levels are presented, although not all of them are suitable for an establishment of simple predictive tools.

The forthcoming work in WP2 will consist in presenting the data collected during the measurements on the high speed vessel SuperSeaCat3, which were performed in the context of the present project, presenting the analysis of them and, finally, connecting them to the aforementioned models.

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2. INTRODUCTION

The present report corresponds to deliverable 2.2.1 of Work Package 2 – Ship Motions Hazards of Safety At Speed (S@S). It presents the works carried out in Sub-task 2.2.1 for the formulation of risk and cost models for the human factors associated with motion sickness, safety of footing, noise, vibrations and indoor climate.

The models for motion sickness and safety of footing are based on existing methods presented in international and national standards and literature. One criterion for the selection of the methods has been the availability of input data already in the very early stages of ship design. The simplified seakeeping calculation method developed in sub-task 2.2.2 fulfils this requirement and provides all the necessary data needed in calculations of ship characteristics related to motion sickness and safety of footing. The standards provide also some criteria and guidelines in order to assess the ship behaviour with regard to motions and give a warning to the designers if the comfort of passengers or working ability of crew is endangered.

It should be stated from the beginning that no simplified models exist connecting the basic design parameters of the ship (like main particulars, installed SHP etc.) to the noise and vibration levels. The accurate prediction of them can be performed during the design stages of the ship only by using elaborate numerical methods. Their results should be verified by specific measurements during sea trials. Apart from the low frequency elastic vibrations (whipping, springing), it should be noted that noise, high frequency vibrations and indoor climate onboard ships don't depend significantly on the operational parameters, like ship route, heading or sea state. Thus, they actually form a rather constant environment onboard the ship. On the other hand, the aforementioned complex models can only approximate their dependence on the ship's speed.

Due to the above reasons, the models of noise, vibration and indoor climate contain mainly the limits of the parameters, which are used as a measure for their acceptability. An exception to this is the case of the prediction of the lower modes of vibration of the hull girder in vertical and horizontal modes and the analysis of the risk of resonance with the major excitation sources, as well as the analysis of whipping vibrations. For the above cases a simplified model has been developed in the context of WP2 and will be described in the next chapters.

3. IDENTIFICATION OF HAZARDS AND FORMULATION OF RISK MODELS

3.1. Identified hazards related to ship motion

The main hazards related to ship motions were identified and confirmed in Task 2.1. They were listed in the Conclusion of Deliverable 210:

1. Crew and passengers disorientation and injury
2. Large ship loading causing structural failure and foundering
3. Loss of ship control

These hazards can be related to the following events:

a. Excessive ship motions and accelerations
b. Excessive elastic ship vibrations
c. Excessive local and global wave loads
d. Dynamic capsize (broaching)
e. Excessive noise, vibration levels and bad indoor climate

Events a and e (excessive ship motions and accelerations, noise, vibration levels and bad indoor climate) are dealt in sub-task 2.2.1. The models of motion sickness and safety of footing will be used in predicting the risks involved in excessive ship motions and accelerations. The seakeeping models developed for risk prediction of the three first events a, b and c (presented in Deliverable D222) will be used to produce the necessary input data for motion sickness and safety of footing models.

3.2. Description of models

3.2.1. Motion sickness model

3.2.1.1. Introduction

The motion sickness model applied here is based solely on methods already presented in international standards. No enhancements to the existing methods have been made within this S@S-project. However, the application of even the current standards is not straightforward as they present no simple criteria for assessing the ship qualities with regard to motion sickness.

The most important international standard concerning the effects of ship motions and vibrations on passengers is ISO 2631-1:1997 [1]. It provides an evaluation method of human exposure to low frequency motions in case of motion sickness as well as vibrations in higher frequency range. ISO 2631-1:1997 is based on national standard BS 6841 [2] published in 1987. BS 6841 can be considered more consistent in its recommendations than ISO 2631 and regarded as the primary standard when it concerns the evaluation of vibration effects on comfort but in the case of motion sickness both standards are practically identical apart from slightly different frequency weighting functions.

Unfortunately neither of the standards provide any quantitative criteria, e.g. for acceptable percentage of seasick persons. In the previous edition of ISO Standard 2631/3-1985 (which was cancelled and replaced by ISO 2631-1:1997) a "severe discomfort boundary" as a function of frequency and exposure time was introduced. The magnitudes of root mean square vertical acceleration were determined so that about 10 % of unadapted, seated or standing men in normal health will experience severe discomfort and temporary disability. According to ISO 2631/3-1985 the term "severe discomfort" was used to characterize the broad spectrum of motion sickness symptoms occurring successively in order of increasing severity or progressing from pallor and dizziness through nausea to vomiting and complete disability. The standard stated furthermore: "Several additional influences, particularly vision, fear, head movement, odours and activity and the ingestion of certain foods and drink affects motion sickness sensitivity. It is not possible to quantify their effects at present. The boundaries exclude the effect of anti-motion sickness drugs, or the use of head restraints."

The ISO 2631/3-1985 boundary values have been considered too high by Karppinen [3]. Also Allen [4] and Goto [5] have recommended lower motion sickness threshold acceleration levels than were later adopted in ISO 2631/3-1985.

As it is very difficult and uncertain to determine objectively acceleration boundaries, such limits have been excluded from the latest version of ISO standard. It suggests only the calculation the Motion Sickness Dose Value (MSDV) which is linearly proportional to the incidence of motion sickness. Using the calculated MSDV value it is possible to estimate the percentage of vomiting people (Motion Sickness Incidence, MSI). These values are determined using the frequency weighted vertical accelerations measured or calculated at given location of the ship.

One goal of the sea trials conducted with SuperSeaCat 3 was to collect information of the connection of ship motions and the level of motion sickness they generate in passengers. This was done by asking the passengers how did they feel during the voyage and at the same time measuring the essential ship motions. A very useful expression of passenger discomfort in motion sickness study is vomiting as it is a very easily verifiable symptom. All the passengers were asked to fill in a questionnaire including questions about the environment in terms of annoyance due to ship motions, noise and vibrations. Unfortunately it was not allowed to include a straight question about possible vomiting as was the original intention. Though there was a question about the degree of discomfort it was not possible to decide whether the respondent actually vomited or not. As the question of vomiting had to be left out, the results of questionnaires could not be used to validate the motion sickness model.

The results of SuperSeaCat 3 sea trials containing the measurements conducted by VTT and the analysis of passenger questionnaires are presented in a separate report (S102.21.07.065.001).

3.2.1.2. Formulation of motion sickness model

This sub-chapter deals with the following issues related to motion sickness model:

- Frequency weighting
- Motion Sickness Dose Value MSDV
- Motion Sickness Incidence MSI

3.2.1.2.1. Frequency weighting

The manner in which vibration affects motion sickness is dependent on the vibration frequency. Different frequency weightings are required for the different axes of vibration and for the different effects of vibration on the body.

At higher frequencies the tolerance of human to vertical vibration improves and a general opinion is that motion sickness is not a problem at frequencies higher than about 0.5 Hz. So the relevant frequency range in motion sickness standards is 0.1-0.5 Hz. ISO 2631-1:1997 gives the parameters for Butterworth two-pole band-limiting filter for vertical acceleration. The corner frequencies of the filter are 0.08 and 0.63 Hz.

The frequency weighting curve W_f for motion sickness given by ISO 2631-1:1997 is shown in Figure 3.1.

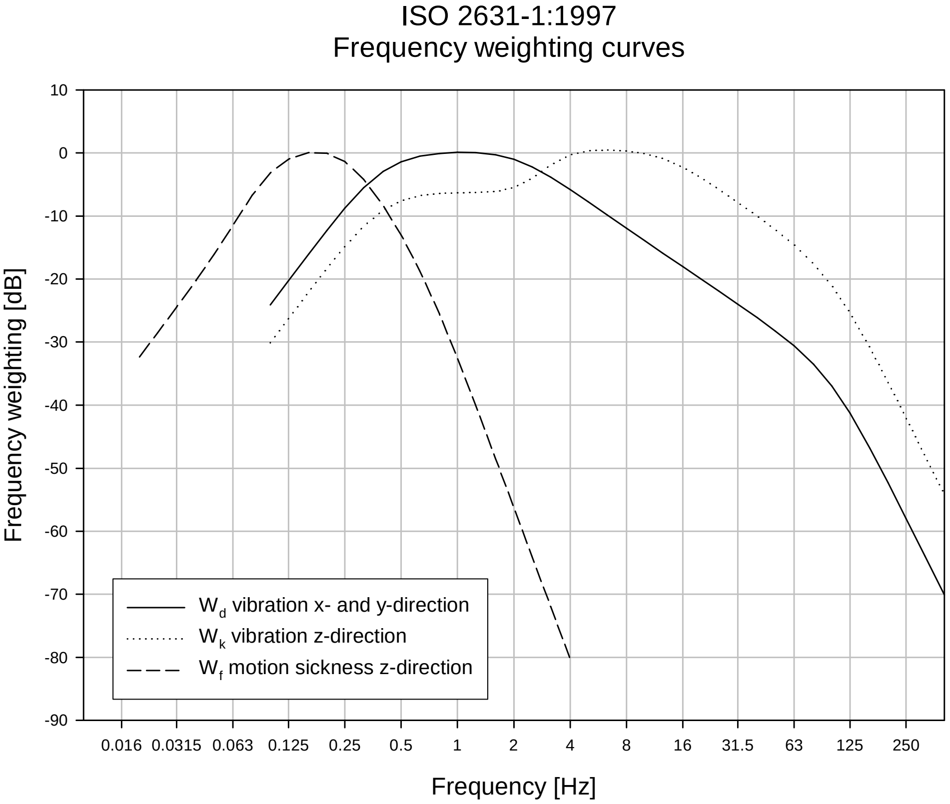


Figure 3.1 ISO 2631-1:1997 Frequency weighting curves for motion sickness (W_f) and for vibrations in x-, y- and z-directions (W_d and W_k).

3.2.1.2.2. Motion Sickness Dose Value MSDV

A motion sickness dose value is defined such that higher values correspond to a greater incidence of motion sickness. There are two alternate methods of calculating the motion sickness value:

<u>Method 1:</u>

$$\mathrm{MSDV} = \left\{ \int_0^T [a_w(t)]^2 \, dt \right\}^{1/2} \tag{3.1}$$

where $a_w(t)$ is the frequency weighted acceleration in z-direction
T is the total period [s] during which motion could occur

This method is equivalent to calculating the rms-value by true integration over the period T and multiplying by $T^{1/2}$. The unit of MSDV is $m/s^{1.5}$.

Method 2:

If the motion exposure is continuous and of approximately constant magnitude, the motion sickness dose value may be estimated from the frequency-weighted rms-value a_w measured or calculated over a short period.

$$MSDV = a_w T_0^{1/2} \tag{3.2}$$

where T_0 is the exposure period [s].

When using method 2 the measurement or calculation period of acceleration rms should not normally be less than 240 s.

3.2.1.2.3. Motion Sickness Incidence MSI

According to ISO 631-1:1997 the percentage of people who may vomit is approximately K_m*$MSDV_z$ where K_m is a constant which may vary according to the exposed population. For a mixed population of unadapted male and female adults, K_m=1/3.

So the Motion Sickness Incidence MSI as a percentage of vomiting passengers is defined as

$$MSI = \frac{1}{3} MSDV_z \; [\%] \tag{3.3}$$

3.2.1.3. Input for motion sickness model

In sub-task 2.2.2 of WP2 a simplified seakeeping calculation method has been developed. It is based on linear strip theory and provides semi-analytical formulas by which the wave-induced ship motions and accelerations in regular waves can be predicted with sufficient engineering accuracy. The method gives response amplitude operators (RAOs) and phases of heave, pitch, roll at given ship speed and angle of encounter with respect to the waves. In addition it provides the RAOs of motions and accelerations in vertical, transverse and longitudinal directions at any given point of the ship. For the calculations of motion sickness only the RAOs and phases of vertical acceleration as function of wave frequency at the selected locations of the ship will be needed.

The parameters of irregular sea used in the calculations will be selected according to the sea area where the ship is intended to operate. The irregular sea is characterised by some known wave spectrum approximation (Jonswap, ISSC, Pierson-Moskowitz). As the significant wave height and peak (modal) period is known, the wave spectrum is decomposed into several regular components. Using these wave amplitudes and the acceleration RAO the corresponding regular acceleration components can be determined. By adding these regular components with appropriate phases the time history of vertical acceleration can be composed.

Finally the acceleration time history is frequency weighted in time domain.

3.2.1.4. Motion sickness criterion

There are no limits given in ISO 2631-1:1997 or BS 6841:1987 for acceptable percentage of seasick (vomiting) people. Such limits are obviously very difficult to define. However, following the severe discomfort boundary presentation of ISO 2631/3-1985 a tentative criterion for motion sickness could be set:

MSI value in passenger spaces should not exceed 10 % for exposure period of 2 hours

The time limit of two hours has been adopted because the motion sickness incidence increases with exposure time up to about two hours. If people have not vomited within the first two hours, they rarely do so after a prolonged period of exposure to oscillatory vertical motions.

The MSI criterion of 10 % is compatible with the criterion given by ABS [31]. The COMF+ level for maximum MSDV value in ABS Guide for passenger comfort on ships is 30 $m/s^{1.5}$ which corresponds to MSI value of 10 % if the constant K_m is taken as 1/3.

The MSI criterion should be fulfilled in all passenger spaces. So it is necessary to determine the MSI value just in one location in passenger spaces if it is clear where the highest vertical accelerations are expected to occur. Such locations are the points which are farthest away from the ship center of gravity, center line and base line. If the location is not obvious all potential points should be checked.

Using (3.2) and (3.3) the corresponding MSI values can be defined also for periods shorter than two hours if MSI for two hours exposure is 10 %. Some values are given in Table 3.1. The frequency weighted vertical acceleration rms-value in these cases is 0.35 m/s^2 which is 30 % lower than the ISO2631/3:1985 severe discomfort boundary in frequency range 0.1 - 0.315 Hz.

Table 3.1 MSI criteria for different exposure times.

Exposure time [h]	Motion Sickness Incidence MSI [%]
0.5	5
1.0	7
1.5	8.5
2.0	10

3.2.1.5. Limiting significant wave height and operability

Assuming that the vertical motion of the vessel at the wave encounter frequency is linear function of the wave height the following formula may be used for estimating the operability-limiting significant wave height H_{ss} with regard to the motion sickness at a particular wave peak period:

$$H_{ss} = a_b / a_{0w} \qquad (3.4)$$

where a_b is the ISO 2631-1 boundary value = $1/3\, a_w T_0^{1/2}$

a_{0w} is the frequency weighted rms vertical acceleration for unit significant wave height

a_{0w} can be calculated using the procedure described in chapter 3.2.1.3. Setting the motion sickness limit MSI=10 % the following formula for H_{ss} at a particular wave peak period can be derived from (3.2), (3.3) and (3.4):

$$H_{ss} = \frac{30}{a_{0w}T^{1/2}} \tag{3.5}$$

Applying this formula in the whole peak period range of the scatter diagram of desired sea area, an operability-limiting significant wave height curve with regard to the motion sickness can be formed and plotted over the scatter diagram.

Finally the operability with regard to motion sickness can be determined by summing the sea state occurrence probabilities under the curve.

3.2.2. Safety of footing model

3.2.2.1. Introduction

Though it would be possible for the passengers to sit for short periods of time in their seats the safety belts locked when the going gets rough, the conditions on board should in general be such that the passengers are able to stand and move without an excessive danger of loosing balance. For instance, the passengers must be able to visit the toilets. The conditions on board may be assessed with regard to the safety of footing by predicting the number of loss-of-balance events per unit of time, or motion-induced interruptions (MIIs). A MII is considered to occur when the motion-induced forces on a standing (or walking) person are large enough to cause one foot to lift off the ground.

The safety of footing model here is mainly based on the work of Graham [6] and Karppinen [3] but the concept of motion-induced interruptions was introduced by Baitis et al. [7] already in 1984 in an investigation on the naval helicopter recovery operations. Baitis et al. estimated the loss-of-balance events of the two crew members hooking up the landing helicopter by a time-domain simulation method. Graham [6] presented how the MIIs can be calculated in the frequency domain and took into account the effect of vertical acceleration on the loss-of-balance events which was neglected by Baitis et al. in their study. Finally Graham et al. [8] extended the method to include the effects of wind, ship longitudinal acceleration, moment of inertia of the person, and arbitrary facing of the person in the ship-fixed co-ordinate system. However, the effects of these extensions by Graham et al. are not taken into account in the present relatively simple safety of footing model.

3.2.2.2. Formulation of safety of footing model

Graham [6] considers a person facing forwards or aft standing (or walking) on deck. The centre of gravity of the standing person is assumed to be at the height of h and the width of the stance is assumed 2xl as in Figure 3.2.

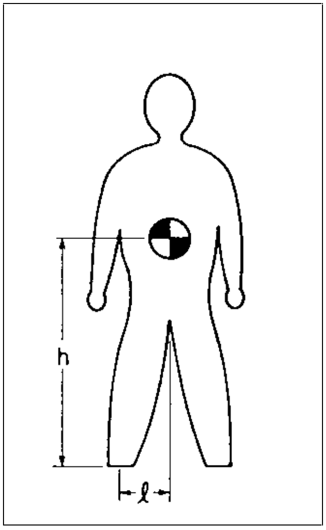


Figure 3.2 Person model.

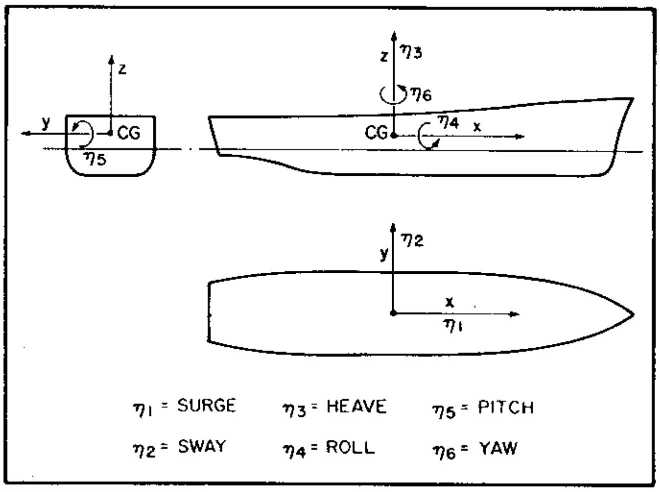


Figure 3.3 Axis system and definitions.

A reasonable estimate for the ratio of l/h is $l/h = 0.25$. The effect of the longitudinal acceleration of the ship is neglected as small. Thus, tipping is assumed to occur when the moment of the lateral force on the person with regard to either foot is larger than the moment due to the vertical force. This gives the following condition for tipping [6]:

$$hm|-a_L - g\eta_4| > lm(g + a_V) \tag{3.6}$$

where m = mass of the person
g = acceleration due to gravity
a_L = lateral acceleration
a_V = vertical acceleration

a_L and a_V are given as the second time derivatives of the displacements in the lateral and vertical directions at a point (x,y,z) on the ship by:

$$a_L = \frac{d^2}{dt^2}(\eta_2 - \eta_4 z + \eta_6 x) \tag{3.7}$$

$$a_V = \frac{d^2}{dt^2}(\eta_3 + \eta_4 y - \eta_5 x) \tag{3.8}$$

The lateral and vertical displacements include linear contributions from sway, heave, roll, pitch and yaw defined in Figure 3.3. Expanding the inequality in (3.6) tipping occurs if either

$$-a_L - g\eta_4 - (l/h)\,a_V > (l/h)\,g \tag{3.9}$$

or

$$+a_L + g\eta_4 - (l/h)\,a_V > (l/h)\,g \tag{3.10}$$

The probabilities that the events defined by (3.9) and (3.10) occur in irregular seas can be determined by assuming that the left hand sides of (3.9) and (3.10) follow the Rayleigh distribution. Thus, the following two lateral force estimators from (3.9) and (3.10) are defined:

$$L_1 = a_L + g\eta_4 + (l/h)a_V \tag{3.11}$$

$$L_2 = a_L + g\eta_4 - (l/h)a_V \tag{3.12}$$

The transfer functions of the lateral force estimators, or the amplitudes of the lateral force estimators in sinusoidal waves per unit wave amplitude as a function of wave frequency, ω, can be determined on the basis of sway, heave, roll, pitch and yaw transfer functions and phase angles as the formulas (3.7), (3.8), (3.11) and (3.12) show. If the lateral and vertical acceleration transfer functions and phase angles at the point (x,y,z) on the ship are known for instance on the basis of model tests or numerical predictions, the transfer functions of L_1 and L_2 can be determined directly by (3.11) and 3.12).

Applying the linear superposition principle and assuming the irregular waves to be unidirectional, the spectral density of the lateral force estimator is given by:

$$S_{Li}(\omega) = [R_{Li}(\omega)]^2 S(\omega) \tag{3.13}$$

where $R_{Li}(\omega)$ for i = 1 and 2 are the transfer functions of L_1 and L_2, respectively, and $S(\omega)$ is the wave spectral density. The mean square values of the lateral force estimators are obtained as the area under the spectrum (3.13):

$$\sigma_{Li}^2 = m_{oLi} = \int_0^\infty S_{Li}(\omega)d\omega, \text{ for i = 1 and 2} \tag{3.14}$$

When the root mean square (rms) value, σ_{Li}, is known, the probability that the value of the lateral force estimator exceeds (l/h)g is given by the Rayleigh distribution:

$$P\{L_i > (l/h)\,g\} = \exp\left[-\frac{1}{2}\left(\frac{(l/h)\,g}{\sigma_{Li}}\right)^2\right], \text{ for i = 1 and 2} \tag{3.15}$$

The number of motion-induced interruptions, or the number of occurrences that either (3.9) or (3.10) is satisfied during a time period of T_t is finally obtained by:

$$M_i = \frac{T_t}{T_{zLi}}\exp\left[-\frac{1}{2}\left(\frac{(l/h)g}{\sigma_{Li}}\right)^2\right], \text{ for i = 1 and 2} \tag{3.16}$$

where T_{zLi} is the zero-crossing period of the lateral force estimator L_i given by

$$T_{zLi} = 2\pi(m_{oLi} / m_{2Li})^{1/2} \tag{3.17}$$

with

$$m_{nLi} = \int_0^\infty \omega^n S_{Li}(\omega) d\omega \tag{3.18}$$

The total number of motion induced interruptions during T_t is

$$M_t = M_1 + M_2 \tag{3.19}$$

Finally the result is expressed as motion induced interruptions per minute:

$$MII = M_t / T_t \tag{3.20}$$

where T_t is time of motion measurement or simulation in minutes.

3.2.2.3. Input for safety of footing model

The input for the safety of footing model is calculated using the same simplified seakeeping calculation method as in the case of motion sickness model. This time more motion components are needed. In the mathematical safety of footing model the vertical and transverse acceleration equations are composed using the basic motion components. The seakeeping model calculates directly the transverse and vertical accelerations. In addition to these accelerations also roll motion is needed.

The calculation of time histories of accelerations and roll proceeds as in the case of motion sickness described in section 3.2.1.3. However, in safety of footing model no frequency weighting is needed.

3.2.2.4. Safety of footing criterion

There is no data available what would be the acceptable MII risk level with regard to passenger safety. Graham [6] has defined five risk levels for deck operations of naval vessels. They cannot of course be used as such in the case of passenger vessels but are shown in Table 3.2 to give some idea of the magnitudes of MII probabilities.

Table 3.2 MII risk levels for deck operations of naval vessels [6].

Risk level	MIIs per minute
1. possible	0.1
2. probable	0.5
3. serious	1.5
4. severe	3.0
5. extreme	5.0

Karppinen [3] suggests that for passenger vessels the lowest level shown in Table 3.2, 0.1 MII/minute or one MII in ten minutes would be adequate. As there seems not to be any more justified criterion for safety of footing the following criterion can be used:

MII value in passenger spaces should not exceed 0.1 / minute

The application of the safety of footing model in the sea trial measurements of SuperSeaCat 3 gave the result that in all tests the ship motions were so small that the model predicted no MIIs. As the passengers were not asked about the occurrences of loosing balance, the results of model could not even qualitatively be confirmed. The maximum significant wave heights during the tests were about 1.5 m which is half of the operational limit of the ship. As the waves were mainly coming from the bow or stern, the transverse accelerations remained quite low. So it is not very surprising that the model predicted no MIIs.

Another way to assess ship behaviour with regard to safety of footing is to compare the calculated maximum horizontal acceleration values with the level 1 and 2 criteria given by IMO HSC Code shown in table 3.3. The time-history of horizontal acceleration can be calculated as the resultant of transversal and longitudinal accelerations given by seakeeping model of WP3. As the calculated maximum acceleration value depends on the chosen values of regular wave component phase angles in the seakeeping model, the most probable maximum value can be used instead when comparing the simulated horizontal acceleration with the criteria in table 3.3.

Table 3.3 Criteria for maximum horizontal acceleration for HSC (taken from IMO HSC Code).

Effect	**Criteria not to be exceeded**		**Comments**
	Type of load	**Value**	
LEVEL 1 MINOR EFFECT Moderate degradation of safety	Maximum acceleration measured horizontally **(1)**	0.20 g	0.08 g: Elderly person will keep balance when holding 0.15 g: Mean person will keep balance when holding 0.15 g: Sitting person will start holding
LEVEL 2 MAJOR EFFECT Significant degradation of safety	Maximum acceleration measured horizontally **(1)**	0.35 g	0.25 g: Maximum load for mean person keeping balance when holding 0.45 g: Mean person falls out of seat when not wearing seat belts
LEVEL 3 HAZARDOUS EFFECT Major degradation of safety	Collision design condition calculated Maximum structural design load, based on vertical acceleration at centre of gravity	Ref. 4.3.3 Ref. 4.3.1	Risk of injury to passengers; safe emergency operation after collision 1.0 g: Degradation of passenger safety
LEVEL 4 CATASTROPHIC EFFECT			Loss of craft or/and fatalities
(1) The accelerometers used shall have an accuracy of at least 5% full scale and shall not have a frequency response of less than 20Hz. The sampling frequency should not be less than 5 times the maximum frequency response. Anti-aliasing filters, if used, should have a passband equal to the frequency response. g: gravity acceleration (9.81 m/s^2).			

3.2.2.5. Limiting significant wave height and operability

Since the wave-induced motions have been assumed linear with regard to the wave height, the rms value of lateral force estimator is a linear function of the significant wave height at a particular value of the wave peak period and can be expressed as:

$$\sigma_{Li} = H_s \sigma_{0Li}(T;V;\beta) \qquad (3.21)$$

where H_s is the significant wave height

σ_{0Li} is the rms value of the lateral force estimator L_1 for unit significant wave height as a function of wave peak period T, with ship speed V and heading to waves β

Thus at each wave peak period the value of σ_{0Li} depends on the ship speed and heading. Substituting (3.21) in (3.16) the operability-limiting significant wave height with regard to motion induced interruptions, H_{si}, can be determined as:

$$H_{si} = \frac{(l/h)g}{\sigma_{oLi}\sqrt{-2\ln\left[\left(T_{zLi}/60\right)M_t\right]}} \qquad (3.22)$$

Applying this formula in the whole peak period range of the scatter diagram of desired sea area, an operability-limiting significant wave height curve with regard to the motion induced interruptions can be formed and plotted over the scatter diagram.

Finally the yearly operability with regard to safety of footing can be determined by summing the sea state occurrence probabilities under the curve.

3.2.3. Models related to noise

The hazards related to noise can be divided into the following three major categories :

- Hearing damage
- Interference with the speech
- Psychological effects

The criteria for noise limits provide the means for establishing models to evaluate their effect on the crew and the passengers and, at the same time to determine how much noise reduction will be required, to examine the feasibility of alternative proposals for noise control and to evaluate the cost of meeting the relevant criteria. These criteria are proposed as a result of relevant research concerning the adverse effects of noise and can be taken as the tools for the determination of the adverse effects of the noise.

Although the aforementioned limits may be established, the community noise levels are generally very high: In the European Union about 40% of the population is exposed to road traffic noise with an equivalent sound pressure level exceeding 55 dB(A) daytime, and 20% are exposed to levels exceeding 65 dB(A). When all transportation noise is considered, it is estimated that more than half of all European Union citizens live in zones that do not ensure acoustical comfort to residents. At night, more than 30% are exposed to equivalent sound pressure levels exceeding 55 dB(A), which are disturbing to sleep [9].

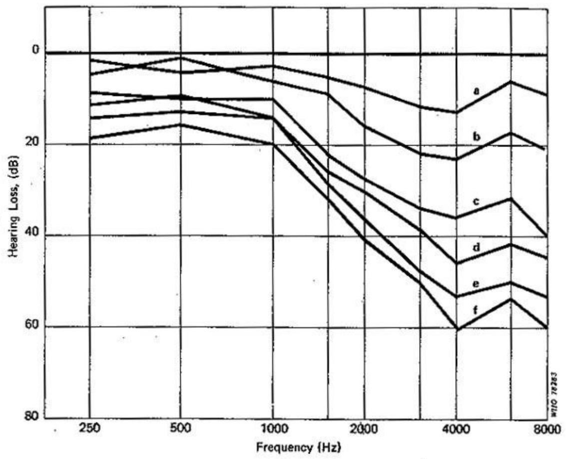
The various acoustical inputs are not necessarily disturbing or harmful. Physically, there is no distinction between sound and noise. Sound is a sensory perception and the "Noise" is defined as the unwanted sound. On the other hand, complete absence of sound can also be harmful.

In the sequel, the aforementioned hazards related to noise are presented, together with the corresponding model for the evaluation of their effect to human.

3.2.3.1. Hearing damage

For the evaluation of the risk of the hearing damage the crew members working in the machinery spaces and in the workshop are concerned.

The hearing loss caused by exposure to excessive noise usually occurs first in the frequency range from about 3000 Hz to 6000 Hz and, therefore, is primarily a loss of speech recognition (Figure 3.4)



Hearing loss as a function of duration in noise exposure in years. Mean audio-grams for 203 miners, best ear tested. [a <1 year; b = 1-5 years; c = 6-10 years; d = 11-20 years; e = 21-30 years; f > 30 years; From: B. Johansson, 1952.)

Figure 3.4 Hearing loss as a function of duration in noise exposure in year.

Exposure to excessive noise for a short period may produce a temporary loss and a permanent diminution of hearing. With increasing $L_{Aeq,8h}$ and increasing exposure time, noise-induced hearing impairment occurs even at frequencies as low as 2000 Hz. However, hearing impairment is not expected to occur at $L_{Aeq,8h}$ levels of 75 dB(A) or below, even for prolonged noise exposure.

The ISO Standard [10] gives a method of calculating noise-induced hearing impairment in populations exposed to all types of occupational noise (continuous, intermittent, impulse), defining an equivalent continuous A-weighted noise level for a nominal eight-hours working day. Some authors [11] have demonstrated that an alternative interpretation of the ISO 1999 data base is possible and some others [12] have shown that neither the formulation of the authors nor the standard ISO 1999 accounts for post-exposure loss observed in war veterans. For continuous noise, although an exposure to 80 dB(A) for 8 hours per day would ensure negligible hearing loss for speech recognition, a lower level would be required to ensure negligible hearing loss at all audible frequencies. Long term exposure to $L_{Aeq,24h}$ of up to 70 dB(A) will not result in hearing impairment. Care should be taken with respect to the following:

a. Data from animal experiments indicate that children may be more vulnerable in acquiring noise-induced hearing impairment than adults.

b. The risk for noise-induced hearing impairment increases when noise exposure is combined with vibrations, ototoxic drugs or chemicals. Under these circumstances, long-term exposure to $L_{Aeq,24h}$ of 70 dB may induce small hearing impairments

In the EU the limits imposed by Directive [13] are

- 85 dB(A) daily personal noise exposure ($L_{ex,8h}$) or 200 Pa unweighted instantaneous sound pressure: above these values appropriate measures shall be taken
- at workplaces where the daily personal noise exposure is likely to exceed the 90 dB(A) or where the maximum value of the unweighted instantaneous sound pressure is likely to exceed the 200 Pa, the information must take form of appropriate signs and the areas in question must be delimited and the access to them must be restricted.

In a Common Position [14] adopted by the European Parliament and the Council for a new Directive, the ISO 1999 Standard is used, to define the daily noise exposure level, covering all noises present at work, including impulsive noise. The limits are

- Where the noise exposure ($L_{ex,8h}$) exceeds the 80 dB(A) (or p_{peak} exceeds the 112 Pa), the employer shall make individual hearing protection available to workers.
- Where the noise exposure ($L_{ex,8h}$) exceeds the 85 dB(A) (or p_{peak} exceeds the 120 Pa), individual hearing protectors shall be used.
- Under no circumstances shall the value of 87 dB(A) for the noise exposure (and for p_{peak} the 120 Pa) be exceeded.

In the IMO Resolutions [15], it is specified (chap. 5) that the limits are designed to ensure that seafarers will not be exposed to an $L_{eq,24h}$ exceeding 80 dB(A). The values are 90 dB(A) for the machinery spaces (continuously manned), 75 dB(A) for the machinery control rooms and 85 dB(A) for the workshops. To control the emission of high (or low) frequency sound, the NR [16] is measured and it is required that the NR number does not numerically exceed the specified A-weighted value minus 5. Personnel entering spaces with noise levels greater than 85 dB(A) should be required to wear ear protectors (parag. 4.1.3)

3.2.3.2. Interference with speech

All the groups (crew and passengers) are concerned. A majority of the population belongs to groups sensitive to interference with speech perception. Most sensitive are the elderly and persons with impaired hearing. Even slight hearing impairments in the high-frequency range may cause problems with speech perception in a noisy environment.

The clarity of speech in the presence of masking noises is depending on their magnitude, the loudness of the speaker voice (at the level of the listener's ear), the pronunciation and the listener's hearing acuity. The influence of the reverberation can be quantified as an additional noise. The speech level of a talker will depend on his own subjective response of what he is listening (the level is increased in noisy environments). The noise spectrum is also determining the interfering effect. The usual method involves taking the arithmetic average of the sound noise pressure levels in the three octave bands, 500 Hz, 1000 Hz and 2000 Hz. This average is known as the speech interference level (SIL).

Conforming to the American Standards [17], the upper noise (SIL) limits for just acceptable speech communication (60% word-out of context recognition) may be estimated as shown in the table 3.4.

Table 3.4. Upper noise (SIL) limits according to ANSI 77.

Distance	SIL (for male voice)	SIL (for female voice)	Expected voice level
0.5	65	60	74-86
1	59	54	67-78
2	53	48	58-68
4	46	41	50-55

The above analysis does not examine the influence of noise with important amount of information. It is important to notice that the adaptation of the individual talker's voice to the background noise level creates an important increment of the noise.

Conforming to other references:

a. For complete sentence intelligibility in listeners with normal hearing, the signal-to-noise ratio should be 15–18 dBA [18].

b. Earlier recommendations suggested that, in a typical living room, sound pressure levels as high as 45 dBA would be acceptable [19].

c. With raised voice (increased vocal effort) sentences may be 100% intelligible for noise levels of up to 55 dBA.

d. Sentences spoken with straining vocal effort can be 100% intelligible with noise levels of about 65 dBA.

e. Reverberation times below 1 s are necessary for good speech intelligibility in smaller rooms; and even in a quiet environment a reverberation time below 0.6 s is desirable for adequate speech intelligibility for sensitive groups.

3.2.3.3. Other physiological and psychological effects

These effects are often related to some information or even fear (for example the shock of the waves creates the fear of a shipwreck).

The more often mentioned effects are:

- Sleep disturbances
- The effects of noise on performance
- Cardiovascular and physiological effects
- Mental health effects

These effects are discussed in the following sub-chapters.

3.2.3.3.1. Sleep disturbance

Field studies conducted with people in their normal living situations are scarce. Most of the more recent field research on sleep disturbance has been conducted to examine the effects of aircraft, road traffic and railway noise (1994-1998).

The primary sleep disturbance effects are: difficulty in falling asleep and alterations of sleep stages or depth, especially a reduction in the proportion of rapid eye movement. Other primary physiological effects can also be induced by noise during sleep (increased blood pressure, increased heart rate, changes in respiration, increase in body movements). For each of these physiological effects, both the noise threshold and the noise-response relationships may be different. Different noises may also have different information content and this also could affect physiological threshold and noise-response relationships [20].

Exposure to night-time noise also induces after effects. These secondary effects include increased fatigue, depressed mood and decreased performance.

A recent research [21] observed some habituation of sleep to the continuous noise of a ship with a sound level of 60 dB(A).

3.2.3.3.2. The effects of noise on performance

It has been documented that noise affects task performance. Accidents may also be an indicator of performance deficits. The few field studies on the effects of noise on performance and safety showed that noise may produce some task impairment and increase the number of errors in work, but the effects depend on the type of noise and the task being performed [22]. Studies showed that noise could act as a distracting stimulus. Also, impulsive noise events may produce disruptive effects. In the short term, noise-induced arousal may produce better performance of simple tasks, but cognitive performance deteriorates substantially for tasks that require sustained attention to details or to multiple cues or tasks that demand a large capacity of working memory, such as complex analytical processes.

Noise exposure consistently produces negative after-effects on performance. An important finding was that some of the adaptation strategies for dealing with aircraft noise, such as tuning out or ignoring the noise, and the effort necessary to maintain task performance heightened sympathetic arousal, as indicated by increased levels of stress hormone, and elevation of resting blood.

3.2.3.3.3. Cardiovascular and physiological effects

Studies, involving workers exposed to occupational noise and general populations, indicate that noise may have both temporary and permanent impacts on physiological functions in humans. Acute noise exposures activate the autonomic and hormonal systems, leading to temporary changes such as increased blood pressure, increased heart rate and vasoconstriction. After prolonged exposure, susceptible individuals in the general population may develop permanent effects, such as hypertension and ischaemic heart disease associated with exposures to high sound pressure levels. The magnitude and duration of the effects are determined in part by individual characteristics, lifestyle behaviors and environmental conditions. Sounds also evoke reflex responses, particularly when they are unfamiliar and have a sudden onset.

If noise exposure is temporary, the physiological system usually returns to a normal state within a time in the range of the exposure duration. If the exposure is of sufficient intensity and unpredictability, cardiovascular and hormonal responses may appear, including increases in heart rate and peripheral vascular resistance, changes in blood pressure, blood viscosity and blood lipids, shifts in electrolyte balance (Mg/Ca) and hormonal levels (epinephrine, norepinephrine, cortisol). The first four effects are of interest because of noise-related coronary heart disease.

The greatest number of occupational and community noise studies have focused on the possibility that noise may be a risk factor for cardiovascular disease. The overall conclusion is that cardiovascular effects are associated with long-term exposure to $L_{Aeq,24h}$ values in the range of 65–70 dB or more, for both air- and road-traffic noise. However, the associations are weak and the effect is somewhat stronger for ischaemic heart disease than for hypertension. Other observed psychophysiological effects, such as changes in stress hormones, magnesium levels, immunological indicators, and gastrointestinal disturbances are too inconsistent for conclusions to be drawn about the influence of noise pollution.

3.2.3.3.4. Mental health effects

Environmental noise is not believed to be a direct cause of mental illness, but it is assumed that it accelerates and intensifies the development of latent mental disorder.

Exposure to high levels of occupational noise has been associated with development of neurosis and irritability and with deteriorated mental health [23]. However, the findings on environmental noise and mental health effects are not conclusive [24]. Various studies show the importance of taking vulnerable groups into account, because they may not be able to cope sufficiently with unwanted environmental noise This is particularly true of children, the elderly and people with preexisting illnesses, especially depression. Despite the weaknesses of the various studies, the possibility that community noise has adverse effects on mental health is suggested by studies on the use of medical drugs, such as tranquilizers and sleeping pills, on psychiatric symptoms and on mental hospital admission rates.

3.2.3.4. The IMO Resolution

In the above mentioned IMO Resolution, the noise level limits are specified in dB(A) as follows:

Space	Room	Noise Limit (dbA)
Navigation Spaces	Navigation Bridge	65
	Radio Rooms	60
	Radar Rooms	65
Accommodation spaces	Cabins	60
	Mess rooms	65
	Recreation rooms	65

The limits recommended by the Ship Classification Societies follow the same lines. In Appendix1 the ABS recommendations are summarized [31] [32].

3.2.3.5. Noise level prediction

The conventional methods used in the prediction of the noise levels in a space of a ship follow the so-called 'Source-Path-Receiver' approach (Figure 3.5 [30]). This method can be split into the following three steps:

a. Identification of the noise sources and determination of the corresponding Sound Power Level (PWL). The PWL values for machinery depend on its specific characteristic (like machine type, power, rotational speed, no of blades for a turbine or meshing frequency for a gear box etc.). It is preferable to use measured data if available, but in the absence of them, empirical formulae combining the main characteristics of the noise source and estimating a baseline PWL or Acceleration Level L_A, may be found in the literature. However, the estimated accuracy of these formulae is may vary in a range from ±3 db to ±10 db, depending on the machine type considered. After this point, the PWL and L_A are subjected to a conversion, in order to obtain the octave band spectral distributions.

b. Estimation of the attenuation caused by the path (or paths) from the source to the receiver. Two different types of paths can be distinguished here: the airborne path and the structureborne one (Figure 3.6 [30]).

c. The final step consists of estimating the Sound Pressure Level (SPL) in the receiver space. This is done by computing the total PWL in the receiver space, considering all the noise sources and all the paths (both airborne and structureborne), and then, by applying the Room constant (R) correction to the computed total PWL in order to obtain the Sound Pressure level in the receiver space.

It must be noted that the method described above suffers from the uncertainties inherent in the prediction of the source PWL, the inaccuracies in the calculations of the airborne and structureborne paths through the complicated metallic structure of the vessel, and finally the inaccuracies in the prediction of the room constant corrections. Due to these problems, some other methods have been developed, which attempt to predict the SPL by defining experimentally the noise transfer function between the source and the receiver. However these methods can't be used in the early design stages, since they are based on complicated noise measurements that must be performed onboard an existing ship or a similar one.

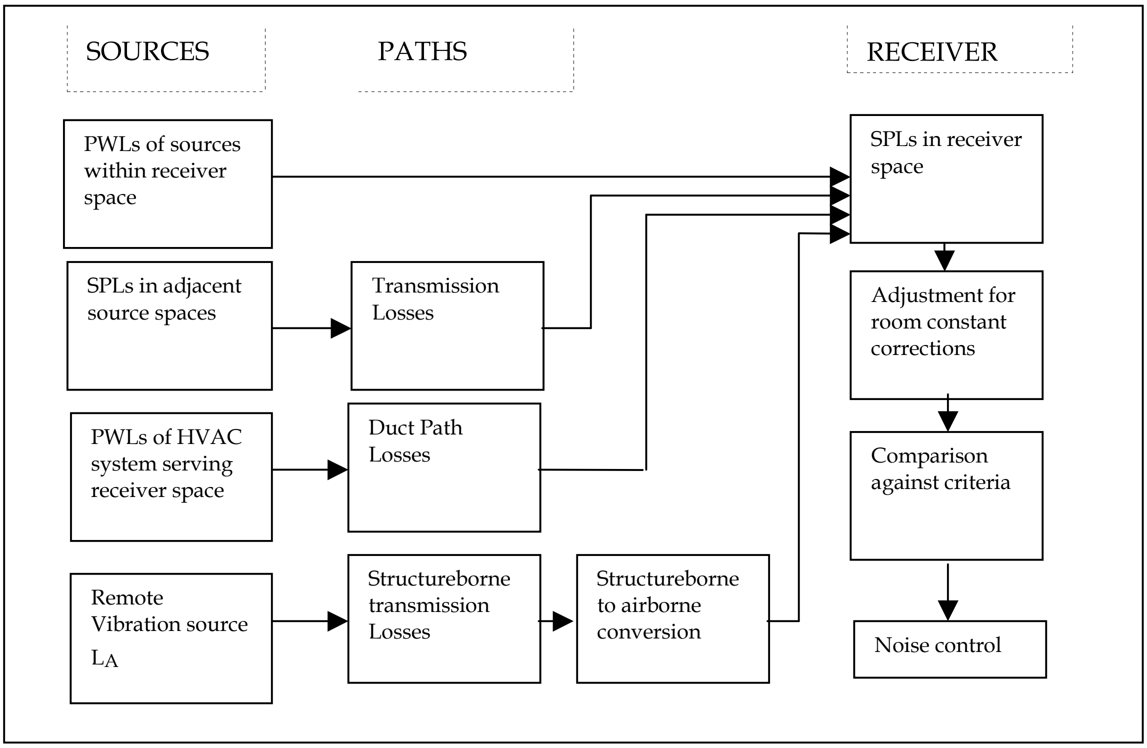


Figure 3.5 Noise prediction following the Source-Path-Receiver method.

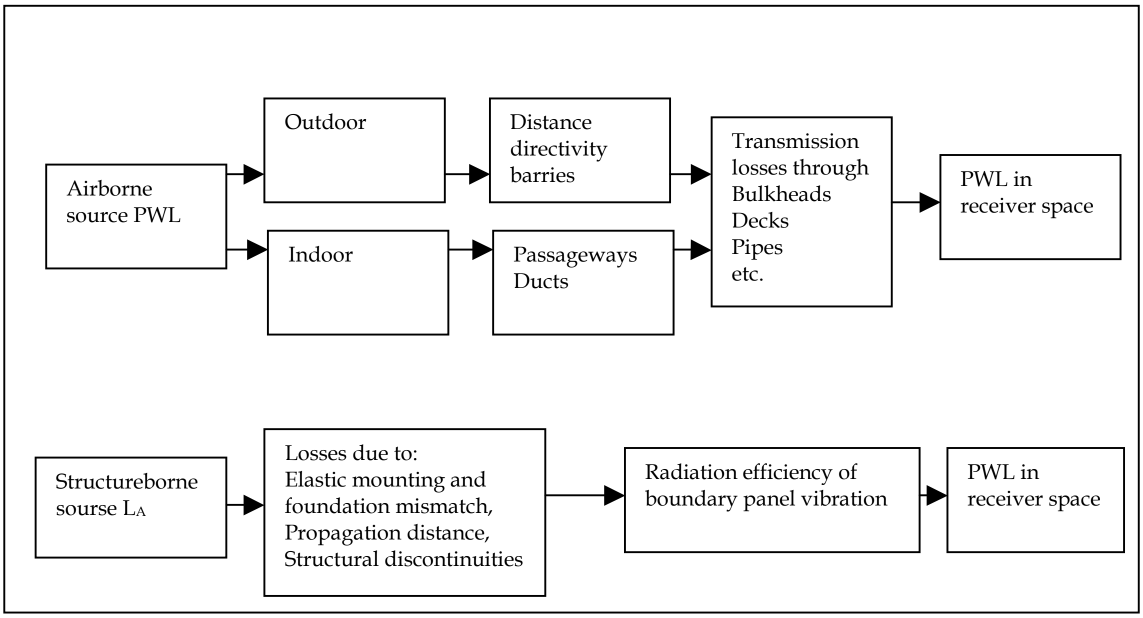


Figure 3.6 Attenuation factors in airborne to airborne and structureborne to airborne paths.

3.2.4. Vibration models

3.2.4.1. Physiological effects

These effects concern mainly the crewmembers. The whole-body vibration may have physiological effects. However it is not established with any precision how the damage depends on the physical characteristics of the vibration, the characteristics of the exposed person or other environmental aspects (the health effects of the vibration transmitted from work processes into workers hands and arms are not examined here).

The belief that exposure to vibrations induce health effects is based to epidemiological studies, subjective data, biodynamic models and a knowledge of the physical properties of the body. The main physiological effects are:

- Spinal column impairment: This is the most common result of the long-term exposure to whole-body vibration [25], where the back is especially sensitive to the 4-12 Hz vibration [26].
- Digestive system troubles: Observed in persons exposed to whole-body vibration over a long period of time. Associated with the resonance movement of the stomach to the 4-5 Hz vibration.
- Cardiovascular system effects: Prolonged exposure to whole-body vibration at frequencies below 20 Hz may influence the heart rate, the oxygen intake and the respiratory rate.

3.2.4.2. The effects of vibrations on performance.

These effects may concern both crewmembers and passengers. The whole-body vibrations may produce negative effects to the task performance. They may affect [10]:

- the senses, and create problems with collecting information
- the processing of information
- the level of arousal, motivation or fatigue
- the intentional actions.

Some of the effects are frequency related as certain parts of the body are in resonance with the vibration received. Vision, for example, is mainly affected by vibration at frequencies between 20 to 90 Hz.

The whole-body vibrations cause imbalance, disorientation and lack of co-ordination, which then leads to stress (even non-human subjects [27]), fatigue, interference with instrument readings, operation of tools etc. They can also result in impaired reflex action, distraction and annoyance.

3.2.4.3. The effects on comfort

These effects may concern both crew and passengers. Although some motion can be a source of pleasure, some other may cause dissatisfaction, discomfort and displeasure. The whole-body vibration impact on comfort was studied mainly by the transport industry to establish how to improve ride comfort in vehicles such as buses and trains.

The discomfort is related to the vibration frequency: at low frequencies, (1-2 Hz) the vibration is transmitted, without noticeable changes, throughout the whole-body; at slightly higher frequencies various body resonances tend to amplify the motion and magnify the overall discomfort. If the frequency is increased further, the body provides an increasing attenuation of vibration reducing the discomfort.

The discomfort tends to increase with increasing duration of vibration. It is reflected by the "fourth power" formula (root-mean-quad, rmq, or vibration dose value, VDV) These effects are influenced by other factors, including body posture, age, gender, and the presence of noise [28].

Vibration at frequencies of about 1 Hz and below, which occur in many forms of transport, might induce motion sickness (kinetosis). It results in nausea, dizziness, vomiting and can affect the performance of tasks. The symptoms are worst between approximately 0.125 and 0.25 Hz and rarely occur to frequencies above 0.5 Hz. Motion sickness is dealt in more detail in chapter 3.2.1.

The next sub-chapters deal with assessment the effects of vibration on humans containing:

- parameters and criteria used in vibration analysis
- measuring and evaluating the whole-body vibration
- vibration limits recommended by the Ship Classification Societies

3.2.4.3.1. Parameters and criteria used in vibration analysis

The following parameters given in ISO 2631-1 [1] can be used when analysing and estimating the vibration characteristics of ships:

1. Frequency weighted rms acceleration in vertical direction a_w

$$a_w = \left[\frac{1}{T} \int a_w^2(t) dt \right]^{1/2} \tag{3.23}$$

where $a_w(t)$ is the frequency weighted vertical acceleration as a function of time
T is the duration of the measurement or simulation, in seconds

The frequency weighting function is W_k defined in ISO 2631-1:1997 and shown in Figure 3.1.

When the vibration exposure consists of two or more periods of exposure to different magnitudes and durations, the equivalent vibration magnitude $a_{w,e}$ corresponding to the total duration of exposure can be evaluated according to the formula:

$$a_{w,e} = \left[\frac{\sum a_{wi}^2 T_i}{\sum T_i} \right]^{1/2} \tag{3.24}$$

where a_{wi} is the vibration magnitude (rms acceleration in m/s^2) for exposure duration T_i.

The rms value a_w or $a_{w,e}$ can be compared with the guidance presented by ISO 2631-1 and presented in Table 3.5 though no limits are defined. They give approximate indications of likely reactions to various magnitudes of overall vibration total values in public transport. As stated in ISO 2631-1 the reactions at various magnitudes depend on passenger expectations with regard to trip duration and the type of activities passengers expect to accomplish (e.g. reading, eating, writing, etc.) and many other factors (acoustic noise, temperature, etc.).

Table 3.5 Guide for assessing vibration magnitude with regard to passenger comfort (ISO 2631-1:1997).

Rms acceleration level	Rating
Less than 0.315 m/s^2	not uncomfortable
0.315 m/s^2 to 0.63 m/s^2	a little uncomfortable
0.5 m/s^2 to 1 m/s^2	fairly uncomfortable
0.8 m/s^2 to 1.6 m/s^2	uncomfortable
1.25 m/s^2 to 2.5 m/s^2	very uncomfortable
Greater than 2 m/s^2	extremely uncomfortable

An approximate strength of perception of vertical rms acceleration given by Griffin [34] is shown in Table 3.6.

Table 3.6 Strength of perception of vertical rms acceleration [34].

Rms acceleration range	Perception level
0.005 - 0.01 m/s^2	Perception improbable
0.01 - 0.02 m/s^2	Perception probable
0.02 - 0.04 m/s^2	Clear perception
0.04 - 0.08 m/s^2	Very clear perception
0.08 - 0.16 m/s^2	Strong perception
0.16 - 0.315 m/s^2	Very strong perception

The criterion for whole-body vibration levels of ABS [31], [32] are based partly on the ISO 2631-1 guidelines. In the ABS guide for passenger comfort the maximum acceptable vibration level for COMF notation is 0.315 m/s^2 and 0.20 m/s^2 for COMF+ notation. ABS Guide for crew habitability gives the maximum acceptable vibration level of 0.4 m/s^2 for HAB notation and 0.315 m/s^2 for HAB+ notation.

2. Vibration Dose Value VDV

$$\mathrm{VDV} = \left\{ \int_0^T [a_w(t)]^4 \, dt \right\}^{1/4} \tag{3.25}$$

where $a_w(t)$ is the frequency weighted vertical acceleration as a function of time
T is the duration of the measurement or simulation, in seconds

The frequency weighting function is W_k defined in ISO 2631-1:1997 and shown in figure 3.1. According to ISO 2631-1 the use of VDV evaluation method will be important for the judgement of the effects of vibration on human beings when the following ratio is exceeded:

$$\frac{\mathrm{VDV}}{a_w T^{1/4}} = 1.75 \tag{3.26}$$

where a_w is the frequency weighted rms acceleration.

When the vibration exposure consists of two or more periods of different magnitudes, the vibration dose value for the total exposure should be calculated from the individual vibration dose values i using the formula:

$$\mathrm{VDV}_{\mathrm{total}} = \left(\sum_i \mathrm{VDV}_i^4 \right)^{1/4} \qquad (3.27)$$

Griffin [34] gives some guidelines for assessing measured or calculated vibration dose values in residential areas. They are presented in Table 3.7.

Table 3.7 Guidelines of VDV assessment [34].

VDV	
0.2 - 0.4 $m/s^{1.75}$	Low probability of adverse comments
0.4 - 0.8 $m/s^{1.75}$	Adverse comments possible
0.8 - 1.6 $m/s^{1.75}$	Adverse comments probable

3. Estimated Vibration Dose Value eVDV

$$\mathrm{eVDV} = 1.4 a_w T^{1/4} \qquad (3.28)$$

where a_w is the frequency weighted rms acceleration in vertical direction
T is the duration of the measurement or simulation, in seconds

This formula will underestimate the true vibration dose value when the crest factor (=weighted peak acceleration/weighted rms acceleration) exceeds about 6.0. The error will tend to increase with increases in the crest factor. The correction factor 1.4 has been determined empirically from typical vibration environments having low crest factors. Though the accuracy of eVDV may decrease in higher crest factors, it can be used when comparing discomfort of two alternative environments.

4. The running rms method

The running rms evaluation method takes into account occasional shocks and transient vibration by use of a short integration time constant. The vibration magnitude is defined in ISO 2631-1:1997 as a maximum transient vibration value (MTVV), given as the maximum in time of $a_w(t_0)$, defined by:

$$a_w(t_0) = \left\{ \frac{1}{\tau} \int_{t_0-\tau}^{t_0} [a_w(t)]^2 \, dt \right\}^{1/2} \qquad (3.29)$$

where $a_w(t)$ is the instantaneous frequency weighted vertical acceleration
τ is the integration time for running averaging
t is the time (integration variable)
t_0 is the time of observation (instantaneous time)

This formula can be approximated by an exponential integration:

$$a_w(t_0) = \left\{ \frac{1}{\tau} \int_{-\infty}^{t_0} [a_w(t)]^2 \exp\left[\frac{t - t_0}{\tau} \right] dt \right\}^{1/2} \qquad (3.30)$$

The maximum transient vibration value MTVV is defined as the highest magnitude of $a_w(t_0)$ read during the measurement period T:

$$\mathrm{MTVV} = \max[a_w(t_0)] \qquad (3.31)$$

The ISO standard recommends to use τ=1 s in measuring MTVV (corresponding to an integration time constant 'slow' in sound level meters). The standard also suggest that this method should be used in addition to the basic evaluation of rms acceleration if the following ratio is exceeded:

$$\frac{\text{MTVV}}{a_w} = 1.5 \qquad (3.32)$$

where a_w is the frequency weighted rms acceleration.

3.2.4.3.2. Measuring and evaluating the whole-body vibration

A method of measuring and evaluating the whole-body vibrations is offered by ISO 2631-1 [1]. In this standard, the effects of whole-body vibrations with respect to health are given in sections 7.1, B1, B2, B3.1 and B3.2 as follows:

- "…an increased health risk to the lumbar spin and the connected nervous system…"
- "With a lower probability, the digestive system, the genital/urinary system and the female reproductive organs are assumed to be affected".

It is, also, mentioned that:

- ".. the recommendation is mainly based on exposures in the range 4h to 8h …"
- "…health disorders are currently understood to be influenced by peak values and are possibly underestimated by methods involving r.m.s. averaging alone"

In the Figure 3.7 the 15 VDV level of the BS 6841 [2] and the two "health guidance caution zones", given in ISO 2631-1 [1] are compared. The other one of the zones is between the VDV values of 8.5 and 17. The standard does not specify which one of the zones should be used.

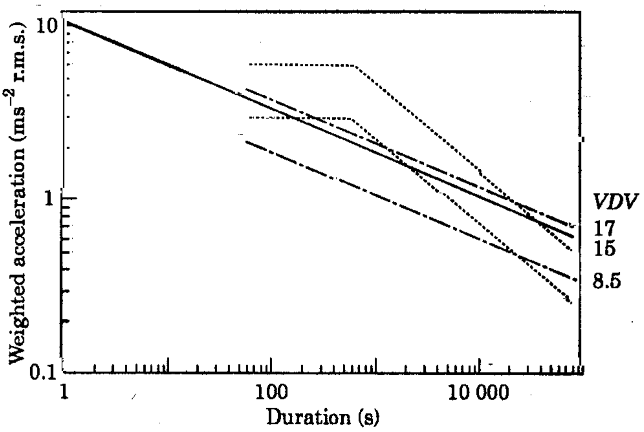


Figure 3.7 The 15 VDV action level given in BS 6841 (1987) and the two "health guidance caution zones " given in ISO 2631 (1997).
Above zone , "health risks are likely" ;
within zone, "caution with respect to potential health risks";
below zone, "health effects not documented".

3.2.4.3.3. Vibration limits recommended by the Ship Classification Societies

The limits recommended by the Classification Societies follow in general the ISO and BS rules. Based on the Class notation, the limits suggested by ABS are listed in the following tables [31] [32]:

Passenger Comfort - Whole body Vibration Levels			
Class notation	**Frequency Range**	**Acceleration measurement**	**Maximum level**
COMF	0.5-80Hz	a_w	0.315 m/sec^2
COMF+	0.1-0.5Hz	MSDVz	30 $m/sec^{1.5}$
	0.5-80Hz	a_w	0.20 m/sec^2

Crew Habitability – Maximum Weighted Rms Acceleration Level			
Class notation	**Frequency Range**	**Acceleration measurement**	**Maximum level**
HAB	0.5-80Hz	a_w	0.4 m/sec^2
HAB+	0.5-80Hz	a_w	0.315 m/sec^2

3.2.4.4. Prediction of shipboard vibration levels

The prediction of vibrations and vibration excitations are dealt below in three categories:

- prediction of hull girder free vibrations
- prediction of superstructure and local vibrations
- determination of vibration excitations

3.2.4.4.1. Prediction of hull girder free vibrations

The prediction of the vibration levels starts from the analysis of the low frequency free hull girder vibration. The method assumes the hull girder to behave like a floating free-free beam and solves the corresponding equation of dynamic equilibrium for natural frequencies and mode shapes, through the application of the so-called dynamic transfer matrix method [33].

Since the parameters entering in the dynamic equation of equilibrium equation are not constant along the length of the vessel, and, in addition some of them are not even smooth, presenting strong variations and, even more, discontinuities (like the mass distribution along the ship), a numerical procedure is necessary for the solution. The procedure starts with the division of the hull girder into segments and solves the discrete equivalent of the differential equation of equilibrium using the dynamic transfer matrix method and taking into account the boundary conditions that hold at the ends of the hull girder (free ends).

A similar to the above method of solution consists of assigning the properties of each hull girder segment to a beam finite element, and then, using the standard numerical procedures for extracting the natural frequencies and mode shapes. The hull girder is modeled as a 1D-Timoshenko beam (i.e. the influence of the shear distortions and the rotary inertia of each segment of the hull girder is accounted for). The bending strength of each segment is taken into consideration via the bending modulus, as well as the shear strength, the translational and rotational inertial of each segment and the effect of the surrounded water. The later is computed using the added mass coefficients available from the hydrodynamic analysis of each section through the application of the strip theory.

The calculated frequencies of the free vibrations can then be compared against the running speed of the major excitation sources, or the harmonics of them, and conclusions regarding the risk of resonance and of excessive vibration levels can be drawn.

Both the vertical and the horizontal vibration modes of the hull girder can be calculated by the above method, depending on the data given, which may correspond either to the vertical bending modulus, vertical added mass etc, or to the horizontal counterparts of them.

Due to its simplicity, the procedure can be used during the early stages of the ship design. The one dimensional beam method has been implemented in a software code in the context of the S@S project and is available for implementation in the project tool.

The verification of the results obtained by the model should be performed through the comparison of them against results available from more elaborate methods (i.e. results from 3D FE models of ships analyzed in other Work packages of the project if available).

Using the aforementioned beam model, the response of the hull girder in whipping vibrations can be determined, provided that the distribution of the wave loads along the ship are available from the work of the relevant work packages of the project.

3.2.4.4.2. Prediction of superstructure and local vibrations

The prediction of vibration levels in large parts of the metallic structure of the ship, as well as the prediction of the local panel vibrations can’t be done using the simplified approach presented in the previous paragraph, and more elaborate techniques must be used, which are offered by the Finite Element Analysis method.

The analysis of ship structures using FEM programs started around 1970. The first models used were 2D and available to obtain the main hull girder modes together with those of superstructures. The vibration levels were calculated under propeller fluctuation pressures and bearing forces, using model having about 1000 d.o.f. The results accuracy was satisfactory to get an idea of superstructures frequency in order to avoid resonance with the first harmonic of the propeller on tankers, gas carriers, container and bulk carriers for which the acceptable limits were about 6 mm/s (peak).

Around 1975 dynamic analyses appeared using FEM 3D models of the whole ship structure. At that time, the first predictive vibration calculations on cruise vessels were done. The model sizes became larger up to around 30 000 d.o.f due to the increased power of computers. All the research work achieved up to now allowed shipbuilders design teams to highly increase their know-how to design ship structures using FEM dynamic analysis results.

Nowadays, numerical simulation analyses are used for both static and dynamics. These analyses are often performed using the same large size model (about 250 000 degrees of freedom for a big passenger vessel). The use of a single model is related to the man-hours required for modeling tasks and the precision needed on the predicted vibration levels for the ship structures, regarding the ship owner requirements. The tools used for such an analysis are coming from in house development and from the market place but are not, in most cases, ready to handle in a reasonable resolution time large models in dynamic analysis.

Large FE Models are used in automotive and aerospace industries (> 1 000 000 d.o.f.) but a ship has a slender and long structure with very high modal density in the frequency range of interest (0 – 100 Hz). In fact, ship structures are 3D assemblies of stiffened panels providing a lot of local modes with very close frequency values. Some of those modes are unrealistic due to the local approximations made in modelling for both stiffness and mass distribution and then have to be eliminated. In fact, a ship constitutes a very complex elastic/mass system characterised by strong couplings between the particular sub-systems. In addition, when the dynamic analysis is done the outfitting positions are not well known and the results must be available as soon as possible in order to allow an update of the design to make the vibration levels suitable. Due to the above reasons the FE analysis of the ship structure requires the designer to have a great experience both in modelling and in conducting the analysis.

The number of modes needed to cover the second harmonic of blade frequency is very large if direct analysis is performed (several thousands). This accuracy cannot be obtained at the project stage. For these reasons it is necessary to reduce the number of d.o.f. to be kept to compute a suitable modal spectrum for the structure. The reduction of problem size is possible using condensation methods, but the corresponding computer and modelling time is still often too long regarding ship design.

3.2.4.4.3. Determination of vibration excitations

For a successful prediction of the shipboard vibrations an accurate knowledge of the excitation forces is required. These forces can be split into the following categories:

- propeller induced forces
- main engine excitation
- auxiliary machinery excitation
- wave induced dynamic forces and moments.

The wave induced dynamic forces and moments can be obtained by the models developed into the other WPs of project (i.e. seakeeping models). These forces may influence only the lower modes of vibration of the hull girder and, from the vibration point of view can be significant only if a relatively flexible hull girder, having its lower frequencies in the range of the wave excitation, is considered.

For the prediction of the main engine and auxiliary machinery excitation, data supplied by the manufacturers, normally obtained through specific vibration measurements, must be used. Since the frequency range of these excitations are relatively high, the prediction of the response of the ship structure can be made only by the elaborate methods previously defined.

For the prediction of the propeller-induced forces, apart from some empirical formulae, there are not any simple methods available. In contrast, very complicated procedures are nowadays used for the definition of the propeller excitation, which are based on modern boundary-integral equation formulation of the coupled propeller-hull-rudder unsteady flow problem including cavitation. There are also available codes for the analytic prediction of the pressure pulses induced by a propeller on hull, rudder and struts; however they present the same range of complexity.

The only alternative possibility for the development of a simple code is to use the empirical formulae and the associated graphs used for rough estimations, like those presented in [33]. However these methods are suitable only for conventional propellers and can't be used for modern propulsion systems like those incorporating waterjets or surface piercing propellers.

3.2.5. Indoor climate parameters

The ISO Standard 7730:1994 defines the thermal comfort as "that condition in mind which expresses satisfaction with the thermal environment". This definition underlines the subjective nature of the sensation of the thermal comfort for each person.

Apart from personal factors like clothing or individual activity, the main parameters that influence the thermal comfort are:

- air temperature,
- radiant temperature,
- air humidity, and
- air velocity.

The Heating, Ventilation and Air Conditioning system of the vessel should control the variations of the aforementioned parameters, in order to ensure the thermal comfort for the crewmembers and the passengers.

The ABS requirements for the Class notation pertinent to the Indoor Climate parameters are listed in the following tables:

Indoor Climate requirements for COMF and COMF+ Class notation	
Parameter	**Requirement**
Air Temperature	Adjustable range from 18 ºC to 26.5ºC
Relative Humidity	Min. 30%, Max. 70%
Vertical Gradient	0 to 3ºC
Air Velocity	Below 0.5m/sec at the center of the space
Horizontal Gradient (Berthing)	Less that 10ºC in cabins

Indoor Climate requirements for HAB and HAB+ Class notation	
Parameter	**Requirement**
Air Temperature	For HAB notation: Non-adjustable air temperature of 22ºC ±1ºC For HAB+ notation: Adjustable range from 18 ºC to 26.5ºC
Relative Humidity	Min. 30%, Max. 70%
Vertical Gradient	0 to 3ºC
Air Velocity	Below 0.5m/sec at the center of the space
Horizontal Gradient (Berthing)	Less that 10ºC in cabins

4. COST MODELS

Estimating the costs due to motion sickness and a possible injury after loosing of balance is very difficult. It is not known how often people will not travel again by ship after they have experienced motion sickness. Also the cost and availability of alternative ways of travelling have some effects on their choice. The safety of footing model predicts only the probability of loosing of balance. It doesn't tell anything about the risk or severity of subsequent injury. Due to the complexity of consequences the cost assessment of motion sickness and safety of footing seems not sensible. However, the models provide a rational means to avoid unfavourable hull designs and advice the designers to take the passenger comfort more into consideration.

The parameters of the risk models developed in Sub-tasks 2.2.1 for noise and vibrations presented in the proceeding section concern mainly ship design parameters (main particulars, lay-out of the passenger spaces and the engine room) and operational parameters (speed, route, sea states etc.).

Regarding indoor climate, modelling seems to be of no practical use, since indoor conditions can easily be adjusted to the required standards, as it is the common practise.

The cost model functions for these parameters should be formulated following extensive and systematic numerical investigation of the effect of the variation of these parameters on the respective levels. The outcome of this sensitivity analysis will ultimately be modelled using regression techniques.

However, the results of the aforementioned procedure should afterwards be verified by extensive measurement campaign, as it was the case with SuperSeaCat 3, considered for this project.

5. CONCLUSIONS

The works performed in sub-task 2.2.1 and presented in this document deal with the formulation of risk/cost model concerning hazards related to ship motion with human factor.

Methods defining different characteristics associated with motion sickness and safety of footing of a fast passenger ship have been presented. The characteristics calculated using these methods can be compared with acceptable limit values already in the preliminary design phase of a ship. The input data for these methods will be obtained from simplified seakeeping calculation methods developed in sub-task 2.2.2 of WP2. The acceptable limit values have been stated on the basis of existing material including the latest international standards and rules of Classification Societies.

The hazards related to excessive noise and vibration, as well as the effect of the parameters of the indoor climate to the comfort of the passengers and the crew have been analysed. Suitable criteria ant pertinent limits are presented as a basis for simplified models for the evaluation of the effect of noise, vibration and indoor climate on passenger comfort and crew workability. Several methods for the prediction of noise and vibration levels are presented, although not all of them are suitable for an establishment of simple predictive tools.

The forthcoming work in WP2 will consist in presenting the data collected during the measurements on the high speed vessel SuperSeaCat3, which were performed in the context of the present project, presenting the analysis of them and, finally, connecting them to the aforementioned models.

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Appendix: ABS Noise Criteria for Passenger Comfort and Crew Habitability

ABS Noise Criteria for Passenger Comfort

Vessels with Passenger Berthing Cabins	
Space	**Max. noise level L_{Aeq}(dbA)**
Cabins and Staterooms	45
Passageways near Cabins	55
Dining Spaces	55
Indoor Public Spaces	55
Indoor Recreation Spaces	55
Discotheques, Dinner Theatres, Entertainment Spaces	60
Passageways near Public Areas	60
Gymnasiums	65
Outdoor Public Areas	65
Medical, Dental and First Aid Spaces	45
Vessels without Passenger Berthing Cabins	
Indoor Public Spaces	55
Indoor Seating Areas	60
Indoor Recreation Spaces and Game Rooms	65
Passageways near Public Areas	65
Outdoor Public Spaces and Outdoor Seating Areas	70
Medical and First Aid Spaces	55

ABS Noise Criteria for Crew Habitability

Space	**Max. noise level L_{Aeq}(dbA)**
Crew Accommodation Spaces and Open Deck Recreation Areas	
Cabins, Staterooms, Berthing and Sanitary Spaces	50
Dining Spaces	55
Indoor Recreation Spaces	60
Passageways in Accommodation Areas	60
Gymnasiums	65
Open Deck Recreation Areas	65
Medical, Dental and First Aid Center	50
Navigation and Control Spaces	
Wheelhouse. Pilothouse, Bridge	55
Chart Rooms	55
Radio Room	55
Radar Room	55
Ship's Offices	55
Machinery Control Rooms	65
Cargo Control Room (on a Tanker)	65
Service Spaces	
Food Preparation (i.e., Galley, Scullery, Butcher, Shop, Thaw Room)	70
Pantries	70
Storerooms	70
Passageways between Service Spaces	70
Laundries	75
Operating and Maintenance Spaces	
Continuously Manned Machinery Spaces	85
Not Continuously Manned Machinery Spaces	108
Workshops	80
Cargo Hadling Spaces/Areas Near Cargo Handling Equipment	80
Fan Rooms	85