

Polski Rejestr Statków

INFORMATIVE PUBLICATION NO. 2/I

PREVENTION OF VIBRATION IN SHIPS

2004



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FOREWORD

Classification Societies deal for years with the adverse effects of vibration. Many years of experience proved that vibration can cause the following effects:

- damage of the structure or shortening of its durability due to material fatigue;
- incorrect operation or failures in engines and equipment;
- fatigue of the crew and the resulting decrease of the work efficiency;
- noise with its different adverse influence upon the comfort and health of the crew.

Notwithstanding the general knowledge of the causes and effects of vibration, Classification Societies have not issued up to now any rules on the criteria influencing the ship's class due to range of vibration.

Official publications of the Societies are usually limited to general recommendations and guidelines which make the intervention of the surveyor possible when the results of vibrations are visible.

Measures adopted after sea trials can remove only some local vibrations; more serious improvements of the structure are impossible because of prohibitive costs. This is the reason why Classification Societies are more and more interested in the possibilities of successful prevention of vibration at the early stage of design.

Moreover, the necessity of issuing by PRS the informative publication with the view of its optional application in the supervision procedure is obvious because the opinions of the buyer and the manufacturer (the owner and the shipyard) are often different in the question of evaluation of vibration levels. Therefore, the publication would enable an objective estimation of vibration as to the assumed exposure limits as well as to the scope of necessary measures to be undertaken to prevent the vibrations at the whole period of the ship's design stage.

This publication suggests the procedure which gives a considerable chance to avoid vibrations as “Act of God”, thus to avoid the costs for faults removing after sea trials. Publication 2/I should not be treated as a codified summary for a reliable design because it would make an illusory certainty that the ship of a novel construction in particular will not experience the excessive vibrations. The dynamic loads of modern ships can exceed two or three times the design static load, the reason being the dense field of hazardous resonances in the low-frequency band (from 0.5 to about 50 Hz) at simultaneous low vibration damping in the steel structures. Generally, two reasons of excessive vibration should be analysed:

- excitations from the propeller induced variable forces and moments, i.e. variable component of the thrust and pressures oscillating on the hull surface above the screw, as well as the forces and moments generated by main and auxiliary engines (mainly the piston engines), transferred to the hull structure;
- response of the structures i.e. reaction to the a.m. forces presenting a particular danger at resonances. Therefore it is necessary to calculate the free (natural) vibrations of the hull girder as well as of the sub-assemblies of the structure (the so called “local vibrations”).

The publication includes several appendices containing detailed information more or less useful for the main procedure described in Chapters 2 and 3. The necessity for appendices was due to the lack of relevant references in the supervision and checking field.

Publication 2/I may become useful as a preliminary approach to the quite complex problems of prevention of vibration, until the proper Rules are issued. PRS perceives the importance of proper co-operation between the owners and the shipyards, and between the equipment manufacturers and the shipyards in resolving the ship vibration problems.

1 GENERAL

1.1 Scope of application

1.1.1 The recommendations of present publication may be applied to the classification activity of PRS in those cases when under separate agreement they are referred to contracts concluded between the parties: the owner and the shipyard, or between the shipyard and the equipment manufacturer (see General Survey Regulations, 4.7, 4.8, 5.4).

The supervision activity is carried out by PRS at a special request of one of the parties.

1.1.2 The recommendations included in the present publication may be also applied as a whole or in part, if PRS, on the basis of examined documentation, decides that it is necessary and appropriate to extend the requirements listed in *Rules for the Classification and Construction of Sea-Going Ships* over the recommendations of this publication.

1.1.3 The recommendations are intended for:

- .1** ships of 1600 gt and more
- .2** cargo ships of less than 1600 gt only in those particular cases when applying the recommendations, as a whole or in part, will provide for a more correct construction with regard to vibration; it could be particularly important for evaluation of vibration of engines and machinery.

1.1.4 Compliance with the criteria values for evaluation of vibration specified hereby does not exclude the necessity of compliance with the standards or state regulations for protection of health against vibration during continuous, periodical or occasional exposures on duty stands and in recreation spaces for the crew.

Reference to this publication in agreements mentioned in 1.1.1 is not sufficient for stating health criteria for the crew. The latter is to be mutually assented by shipyard and owner. Appendix 2 gives some basic useful information on which helps to state precisely the textual form of contract.

All commitment dealing with comfort in accommodation or on duty posts, vibration phenomena included, are beyond the scope of PRS supervision.

1.1.5 The subject of the present publication is vibration in ships and some of the methods of prevention. It is well known that the reducing of vibration level reduces the noise; the measures listed in Chapter 2 should not be considered as structural means preventing noise, agreed upon or recommended by PRS.

1.1.6 The range of vibration phenomena covered by the present publication is defined in 1.3.2.1.

1.2 Scope of supervision

1.2.1 The supervision of PRS in accordance with 1.1.1 and 1.1.2 is carried out in accordance with “General Survey Regulations”, Chapter 4.

1.2.2 When ordering the supervision of PRS it is necessary:

- .1** to submit to PRS, for information, the technical documentation within the scope agreed previously with PRS in each particular case. The scope of this documentation should enable to ascertain that the main recommendations of Chapter 2 of present publication are complied with:
 - in general, avoidance of significant resonances, and excessive values of excited vibration;
 - the application of means preventing harmful and damaging vibration
- .2** to facilitate PRS a full evaluation of vibration by providing PRS with measurements carried out during mooring and sea trials and, if necessary, during normal service conditions (see 3.2.2).

1.3 Definitions and explanations of terms used in the present publication and appendices

1.3.1 General

- .1** On the whole, the definitions concerning mechanical vibrations and vibration metrology are the same as commonly used in Standard publications issued by various standardization committees (see Appendix 1).
- .2** Because the ship vibrations have not established terminology, the definitions of measured values have been specified in 1.3.2, and the relevant explanations in 1.3.3. Their interpretation and the usage in the survey practice must be uniformed.

That does not mean however, that these definitions are in respect of their character, range of meaning and form the exact scientific definitions.

- .3** The information specified under 1.3.2 and 1.3.3 does not exhaust the problem, for example, the definitions of kinds of vibration, such as free (natural), forced, resonance, non-resonance, self-exciting, damped, parametric, non-linear etc., can be found, if necessary, in publications issued by the Polish Standardization Committee (PKN), or in other Standards, or in proper books.

This publication does not introduce any terminology that differs from that already accepted or commonly used.

1.3.2 Definitions

1.3.2.1 **Vibration** – the variation with time of the magnitude of a quantity which is descriptive of a motion or position of a mechanical system, when the magnitude is alternately greater or smaller than some average value or reference.

This magnitude varies with time in a determinable manner or at random as the so called “stochastic process” that can be characterized through statistical properties.

The Publication 2/I takes into accounts:

- **deterministic vibration** i.e. the one values of which can be determined from knowledge of its behaviour at previous times (e.g. periodic process).
- **stationary vibration** i.e. that from stochastic class which has statistical properties invariant with respect to translations in time i.e. samples averaged over sufficiently long, but finite, time-intervals are independent of the time at which the sample occurs (in particular their mean values and autocorrelation function are independent of the time).

In most cases the vibration in ships will in practice be equivalent to periodic vibration or commonly known “steady-state vibration”.

1.3.2.2 Random non-stationary and transient process designate such kinds of vibration whose magnitude cannot be precisely predicted for any given instant of time, i.e. mean value, mean square value, correlation function etc. are not invariant with respect to translations in time.

For ships it means vibration excited by forces from environment e.g. by impacts of waves on the hull (among others: slamming, slapping, whipping) or by motions caused by the sea state as well as vibrations from other stochastic forces appearing inside the ship. The a.m. kinds of vibration are not the subject of this publication.

1.3.2.3 Mechanical vibration – the vibration whose magnitude kinematic or dynamic, characterizing the state of the mechanical system, as a whole, its point, or element are a function of time. The parameters of mechanical vibration are the measurable quantities that specify the motion, such as: oscillation of magnitude (usually vectors), its discrete components, frequency, phase etc.

1.3.2.4 Active vibration – the vibration existing in the operating machine and in its direct surrounding (e.g. in foundations, brackets etc.) and disappearing after its stopping.

1.3.2.5 Passive vibration – the vibrations which appear in any mechanism when it operates or not and is excited by forces from many sources near and far (e.g. from vibrating hull or engine).

1.3.2.6 Amplitude – the maximum value of a sinusoidal quantity. The use of the term is deprecated by ISO to describe mechanical vibrations met in the engineering practice.

The terms describing mechanical vibration are named “value” in several different meanings listed under 1.3.2.7 to 1.3.2.11 and Fig. 1.3.2.6.

1.3.2.7 Peak value – a peak value of an oscillating quantity which is usually taken on board as the maximum repeated deviation of that quantity from the mean value, during a given examined interval.

1.3.2.8 Peak to peak value – of an oscillating quantity is the algebraic difference between the extreme values of the quantity.

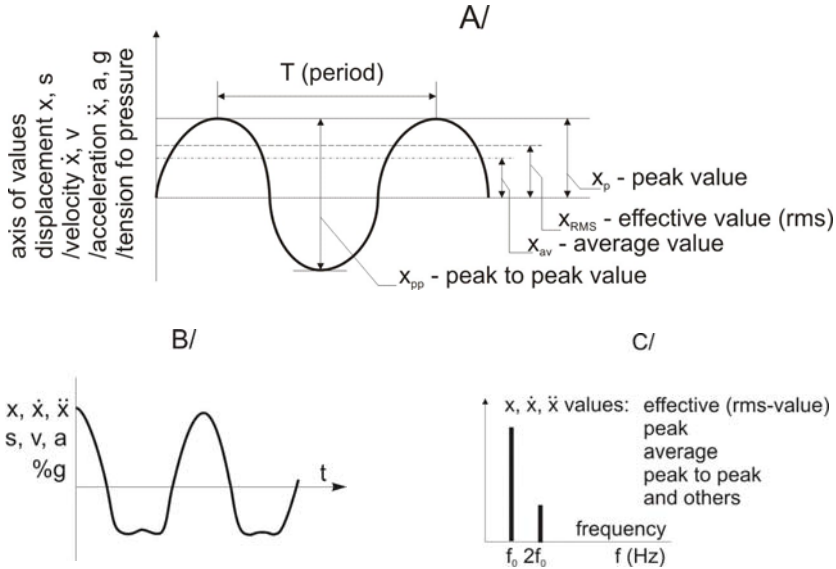


Fig. 1.3.2.6 Graphical explanation of symbols:

- A) signal of simple vibration; variation with time
- B) signal of complex vibration; variation with time
- C) the B case but viewed in frequency domain (as a function of frequency)

1.3.2.9 Average value (so called “rectified value”) is explained by the formula:

$$x_{av} = \frac{1}{T} \int_0^T |x(t)| dt \quad (1.3.2.9)$$

T – time of averaging
 $|x(t)|$ – absolute value of quantity

This value is also known as the “average absolute value” or the “mean absolute value”.

1.3.2.10 Root mean square (RMS) value is expressed by the formula:

$$x_{RMS} = \sqrt{\frac{1}{T} \int_0^T x^2(t) dt} \quad (1.3.2.10)$$

T – time of averaging

This value is also known as the “effective mean value” or the square root of the average of the squared values of the function over the interval. In vibration theory this value is equal to the so called “standard deviation” and the expression under the root is equal to the variance of a vibration process.

1.3.2.11 In the process of analysing, computing, measuring or evaluating each of the values mentioned from 1.3.2.6 up to 1.3.2.10 may concern:

- .1 some determined frequency, which is then the so called “discrete value”
- .2 determined range of frequency, the so called „bandwidth value” e.g. one-third-octave, decade, or other band values depending on the filtering gate used for (examples of those values – see Appendix 2)
- .3 when the vibration in the definite (usually rather wide) frequency range is being considered and analysed by calculation or measurements, then its magnitude is characterized by a value which has many synonymous terms: overall value, total value, global value, or linear value. This value characterizes a vibration process in the lump together with all the components (their superposition) discrete and continuous, entering the measure sensor.

Such lump signal is indicated by the gauge when any filter set is not switched on in the measuring set. The magnitude of such measured value is generally given with index of the bandwidth of amplifier, e.g. level 5 mm/s for 2-100 Hz or 10-1000 Hz, etc. The examples of such values are shown in the table 3.4 and diagram 3.4.

1.3.2.12 Spectral analysis of vibration is the description of a quantity which characterizes the vibration process as a function of frequency. The analysis is made by use of Fourier series in the way of calculation or measurement. The deterministic vibrations have the discrete (or “linear”) spectrum, but for the random processes have the continuous spectrum also known as the power spectral density of vibration (for more detail and exact definitions see items 3 to 6, Appendix 1).

1.3.2.13 Spectral band analysis of a vibration process lies in display of significant values as: r.m.s, peak or other values are defined in the whole range of frequency specific for that process. It is obtainable leading the signal proportional to the total values through bands in a filter set. These bands can have fixed widths (e.g. 3 Hz, 10 Hz) or fixed percentage bandwidth characterized by definite ratio of the filtering bands (so called “gates”) to the middle frequency of this band:

$$\frac{\Delta f}{f_o} = 23\% \text{ for one-third-octave band;}$$

$$\frac{\Delta f}{f_o} = 70.7\% \text{ for octave band.}$$

The band is characterized by the ratio of its upper limit frequency to its lower limit frequency which is equal to:

for decade	10,
for octave	2,
for half-octave	$\sqrt{2}$,
for one-third octave	$\sqrt[3]{2}$.

Usually the middle frequencies, i.e. the geometric mean of the nominal cut-off frequencies of the pass-bands are given as characteristic base of filtering.

The sequence values of these middle frequencies are ranged in the same ratios usually per logarithmic scale, on the axis of abscissae, as argument for plotting the characteristic of spectrum.

The series of this numbers had been standardized (see Appendix 1, item 6).

The one-third-octave analysis in the range from 0.5 Hz to about 200 Hz is usually sufficient for ship vibrations.

1.3.2.14 Frequency, f [Hz], is the reciprocal of vibration period equal to the number of cycles per second.

1.3.2.15 Vibration period, $T(s)$, is the smallest increment of the independent variable of a periodic quantity for which the function repeats itself.

1.3.2.16 Vibration level (in decibels, dB) expresses the logarithm of the ratio of quantity to a reference quantity of the same kind.

The definition is expressed symbolically as:

$$L = 20 \log \frac{x}{x_0} \quad \text{dB} \quad (1.3.2.16)$$

The following reference quantities are recommended by standards:

- for acceleration – $a_0 = 10^{-5} \text{ m/s}^2$,
- for velocity – $v_0 = 10^{-8} \text{ m/s}$,
- or – $v_0 = 5 \cdot 10^{-5} \text{ m/s}$,
- for displacement – $d_0 = 10^{-11} \text{ m}$.

1.3.3 Explanations

Hull beam – or: main hull girder – the conventional model of beam equivalent to the hull from its keel to the upper continuous deck (strength deck), characterized by variable geometrical displacement of mass – elastic properties (mainly: flexural and torsional rigidity).

Hull girder vibration – the states at which the hull girder vibrates in a complex way. The hull vibration can be conventionally divided into the following types: vertical flexural (lateral bending), horizontal flexural, also lateral, longitudinal (axial) and compound flexural – torsional vibration. Each of the above types of vibration has its natural frequencies and corresponding modes.

Among hull vibrations often considered separately are:

- vibration of hull beam;
- vibration of stern i.e. vibration of the hull girder but analysed separately for the length of about $0.3 L$ from the stern, mainly because of excitations caused by forces in the stern bearing and pressures acting onto shell above propeller;
- vibration of the superstructure as a separate sub-assembly of ship, fitted rigidly or flexibly to the hull girder.

Local vibration – the vibrations of structural sub-assemblies of hull (eg. of double bottom, bulkhead, deck, tween deck, partial bulkhead, shell plate, part of superstructure, etc) with levels of vibration significantly greater than that of the hull girder in this region and being also the response to the excitation from the propulsion system, auxiliary engines or from hull girder members. The local vibrations are also called the local amplification of vibration of the hull or “passive vibration” if it appears on the equipment fitted to the hull elements.

Vibration excited by propeller include:

- resonance and forced vibrations of the stern at the frequencies of:

$$f = n, 2n, nz, 2nz, 3nz, 4nz;$$

where

n – revolutions of propeller shaft, 1/s;

z – number of propeller blades;

- local vibration of the stern with one of frequencies mentioned above, taking into account the superstructure if it is placed within the distance of $0,3 L$ from the after perpendicular.

Vibration excited by main engines – the vibration which cannot be considered as caused by the propeller and having the frequencies shown in the tables of un-balanced harmonic components of inertia forces and moments as well as caused by the torque of the gas pressure forces. In most cases the dominant frequencies can be obtained from the formula:

$$f = k \cdot m \cdot i \cdot n$$

where

$k = 1, 2, 3 \dots$ up to about $4i$;

$m = 0.5$ for 4-stroke engine;

$m = 1.0$ for 2-stroke engine;

$i =$ number of cylinders;

$n =$ number of revolutions of engine shaft, 1/s.

The above formula does not concern 2-stroke engines with the so called double ignition system or of a non-typical construction. Data concerning these engines should be received from the maker.

This group includes also the vibration forced by transmission gears and other arrangements for transmission of propulsion.

Vibration of machinery equipment – vibration which occurs in:

- main joints of the engine kinematic system (so called; kinematic constrains) such as bearings, guide ways, articulated joints; etc, as well as appearing in:
- elements of timing gears and control and regulation systems,
- main static constrains placed in stream of force lines such as stands, pads, frames, foundation beds, bodies and others.

2 PREVENTION OF VIBRATION AT EARLY DESIGN STAGE

2.1 Introduction

2.1.1 The prevention of vibration at design stage of a ship consists in calculational prediction of the expected levels and in undertaking appropriate measures to avoid:

- resonance vibration of the hull and equipment
- excessive levels of forced non-resonance vibration
- objectionable, incorrect structural solutions causing the propagation of vibration; some examples of such solutions are shown in Appendix 8.

For a thorough consideration of the different kinds of vibration, a number of predictions is needed for the hull, propulsion and equipment, as pointed out in 2.2 to 2.6.

In a given design, however, the need for particular investigation is to arise from preliminary assessment and data on results of measurements made on board similar ships.

The criteria for estimation of permissible vibration levels at design stage are recommended as not greater than given in Table 3.4. In the case of hull see 3.4.3.

2.1.2 Polish Register of Shipping does not define the calculation tools for prediction purpose having in mind that they are continuously developed and the designer is familiar with them. However, it is recommended that computer programmes should be accepted by PRS.

2.2 Prediction of hull girder vibration

2.2.1 In order to aim at the sufficient approximation, the estimation of vibration should contain, at early design stage, the prediction of the following forms of hull girder vibration:

- .1 vertical flexural appearing as elastic bendings of the hull girder in the symmetry plane;
- .2 horizontal flexural, i.e. bendings of the hull in the plane of waterline;
- .3 coupled: torsional-flexural which means mutual displacements of cross sections of the hull girder around their instantaneous centres of twist (i.e. axis of shearing).

The calculation according to .2 and .3 should be made if there is a hazard of resonances in the vertical flexural vibration. For typical cargo ships the preliminary evaluation can be made by approximate methods (see Appendix 4). The application of semi-empirical formulae should be restricted for such ships, for which, taking into account their structural features and service parameters, at least 3 similar ships can be found including one ship for which the complete measurements analysis of vibration has been made.

2.2.2 The preliminary analysis of the main resonances of the hull lies in calculation of natural frequencies for the vertical flexural modes, marked with numbers of 2 to 4 (rarely more) node forms. The scope of prediction depends on the need and

estimation previously made on the basis of similar ships, and next, by comparing them with fundamental frequencies of the excitation generated by:

- regular wave (encounter frequency – see Appendix 5)
- propeller and shafting
- main engine.

If the comparison shows a possibility of occurring the resonances in service conditions, the investigation should be extended by one of the analytical methods (see Appendix 3) and next, depending on results, the horizontal flexural and coupled torsional-flexural vibrations are also to be calculated.

2.2.3 For ships of length over 150m the use of simple computer programs is advisable at the early design stage also for general preliminary assessment, chiefly with regard to more accurate calculation the frequencies of higher modes.

This procedure is also purposeful when the ship is driven by slow running diesel engine with cylinder number up to 6.

2.2.4 When resonances in calculation are considered as main and are within the range from 85 to 100 % of shafting r.p.m, it is recommended to displace them outside this range by alteration of the free or forced frequencies. If such alterations are not possible, the damping in structure is to be so increased, or the excitations are to be so reduced as not to cause the rise of vibration above the permissible level. Relevant calculation should be submitted to PRS at the design stage and should be additionally confirmed by measurements (see Chapter 3).

2.2.5 Calculation of amplitudes of forced non-resonance vibration of the hull girder is recommended when:

- it results from the estimation of the aftbearing forces and surface forces in way of the stern (for surface forces above the propeller, caused by pressures – see Appendix 6),
- ratios of principal dimensions of the hull are other than those considered as average,
- the ship is of a novel type in respect of the structural properties, speed, way of cargo handling etc; it means that there are not three similar ships out of which at least one with the vibration measurements.

2.3 Prediction of propeller excitation forces

2.3.1 More important reasons of excitation in the afterbody are as follows:

- .1** the nonuniform inflow velocity to the propeller disc area (it is assumed that wake field should be known even at the early design stage);
- .2** the afterbody lines and propeller aperture i.e. shape (dimensional ratios) of the counter stern, sternframe, rudder and propeller clearances;
- .3** non-optimum design of the propeller geometry unadjusted to the wake field velocity entering the disc area (the choice of number of blades, blade

section, rake i.e. inclination of the generating line, skewback i.e. deflection of blade central line, radial distribution of pitch, cambers of profile mean line etc.);

- .4 pulsatory (nonstationary) cavitation on blades amplifying very strongly the variable components of forces and moments on the blade.

The reasons listed under .1 to .4 above, justify the necessity of estimation of the results at the preliminary stage of design, before the geometry of the propeller is established, making use of the preliminary criteria in order to determine;

- .1 permissible values of hydrodynamic surface pressures above the propeller (see Appendix 6 and program UNCA-04 in App. 3)
- .2 permissible non-uniformity of the velocity field due to the propeller blade cavitation (see Appendix 7);
- .3 variation of forces generated on the blades and strongly influencing the aft stern tube bearings (see Appendix 3 UNCA-04).

2.3.2 Calculation of variable components of forces and moments on the propeller is especially important in case when two conditions occur simultaneously or separately:

- .1 the propeller is not the best in respect of optimum diameter or optimum number of revolutions;
- .2 the propeller inflow velocity field is of an excess non-uniformity according to the criteria specified in Appendix 7.

The preliminary criteria given in Appendix 6 and 7 may be replaced by any others, if the designer of propeller has any good reasons for that.

2.4 Prediction of shafting vibration

2.4.1 The information on vibration received from 2.2 and 2.3 should be completed with information obtained from the manufacturers of the machinery installed, concerning the magnitudes of vibration excitation, tried and recommended ways of their mounting and permissible vibration levels on important points i.e. on the kinematic and static constraints.

On the whole, the prediction for propulsion system can cover, among others, the following kinds of vibrations;

- .1 vibration of propeller blades;
- .2 torsional vibration of shafting;
- .3 axial (longitudinal) vibration of shafting;
- .4 whirling vibration of shafting;
- .5 vibration of foundation of the: main engine, transmission gear, thrust bearing, pitch control gear, propeller etc. (see 2.6);
- .6 vibration of propulsion and connected installations e.g. shaft generators, turboblowers, pipings for exhaust gases, steam, water and other (see 2.6).

The principles for carrying out the calculation of vibrations mentioned under .2, above, as well as their estimation are considered by PRS separately within the calculation of strength of the shafting covered by *The Rules for the Classification and Construction of Sea-Going Ships, Part VI ch. 4*).

2.5 Prediction of vibration of auxiliary engines and equipment

2.5.1 The prediction and a consequent list of preventive measures against vibration of engines and auxiliary equipment should be drawn up on the basis of data received from the makers. The data should indicate the magnitude of forces generated by these products and/or should recommend the best fittings and foundations.

One of the most important criterion in the arrangement and design of the foundation structures is the avoidance of resonance vibration which can occur as a result of the influence of the propulsion system or local members of the hull structure (free frequencies of these elements – see 2.6).

2.5.2 It is recommended that the prototypes of engine and machinery generating vibration, installed on board the ship, should be previously tested at the manufacturer's test stand, and the full information about vibration under tests (see also items 7 to 15 in Appendix 1) be taken into account at the proper design stage. If engines and machinery are to be mounted on the flexible pads, the tests are to be carried out on the flexible mountings (e.g. on the proper intermediate bed) with the same pads or flexible couplings to be installed on board the ship.

2.6 Prediction of local vibration of the hull

2.6.1 Prevention of the local resonance vibration lies in calculating the approximation of the free frequencies of the plate and frame elements of the hull (i.e. beams, frames, sub-assemblies) and next by alteration of the hull structure to shift these frequencies outside the region of suspected excitation. For this purpose, it is necessary to make the calculation of frequencies of the first form for these elements of the structure which are excited – according to preliminary evaluation – by the sources situated near them (the propulsion system, engines and auxiliary machinery).

It is recommended to take into account the following individual elements of the hull:

- .1** in way of the after peak plates of shell plating, bulkheads, platforms and web girders;
- .2** in way of the main and auxiliary engine rooms: the bottom structure for the whole space and separately for plates of the inner and outer shell, platforms, bulkheads and tweendecks particularly in the regions of piston engines;
- .3** in way of the superstructure, mast house etc. if they are situated on the stern or are influenced by other excitation sources as: deck plating, longitudinal and lateral sides plating, masts, funnels and other structures situated at a considerable distance from the hull girder;

- .4 in the superstructure: it is recommended to make the vibration prediction always when the superstructure is aft, and in other places only when it is indicated by the tests for other similar ships or by preliminary estimation. The prediction of the expected levels should be made always when the superstructure is fitted to the hull girder by flexible mountings (e.g. by pads or other elements having the flexibility selected by calculation).

2.6.2 In accordance with 2.1.2 the formulae and methods for calculation of local vibration are not compulsory. The information about the programs useful for the aims mentioned under 2.6.1 are listed in Appendix 3.

3 MEASUREMENTS AND CRITERIA FOR EVALUATION OF VIBRATION

3.1 Scope of measurements

3.1.1 According to 1.2.2, it is recommended to carry out vibration measurements for all ships under construction (the so called “new ships”) before they are put into service. The test programs for such measurements, should be as follows:

- .1 for prototypes of series – according to 3.2.1 and 3.2.2
- .2 for ships of series – according to 3.2.3.

3.1.2 The prototype of series from the vibration point of view, it is also the serial ship which has the structure and equipment altered in such a way that her mass-elastic properties could have been changed as the result of such alterations so that dynamic characteristics of the hull or equipment are different from those previously expected. Also the alteration of excitation sources or only of their location makes a serial ship “the prototype” from the vibration point of view.

3.1.3 The measurements of vibration on ships in service are recommended when the vibration levels become harmful for the crew or it is suspected that such vibration is the most probable reason of faulty operation of some machinery, or causes fatigue damages of structure.

3.1.4 Measurement data for PRS' Surveyors should be performed by laboratory or research centre recognized by PRS.

On consent of the interested parties, the measurements may be performed in the presence of PRS surveyor.

3.2 Program of measurements

3.2.1 It is recommended to deliver to PRS the program of measurements of vibration per 3.1.1 for the prototype ship of series as well as for non-serial ships. The program is to be included into the supervision technical documentation mentioned under 1.2.2 and it should contain the following items:

- .1 information about free (natural) frequencies calculated according to Chapter 2 and explained in clear graph systems or tables;
- .2 information about main parameters of machinery and equipment and about frequencies of the excitations;
- .3 technical description of the measurements and detail plan of the measuring points in the form of clear sketches;
the description should include the values which should be measured, first of all those which are considered as the least credible as well as those which are not listed in the prediction e.g. because of the lack of calculation methods;
- .4 description of the measurement procedures and service – loading conditions of the ship required by this procedure both for mooring trials and sea trials;
- .5 information about the measuring instruments and their frequency ranges, characteristic of sensors, range of dynamic properties and calibrating set-up for use on board ship, accuracy of measurements and other data important for elaboration of results and their evaluation.

3.2.2 Unless the predicting calculation and acceptance tests assure all information necessary for service and for improvement of the next ships of the series, it is recommended to enlarge the program by introducing the proper measurements and tests, e.g.:

- determination of the local natural frequencies by testing the structure with an exciter (shaker);
- determination of the parameters of vibration – if it is stated that they are excessive – at different conditions of loading at sea during the first voyages of the ship.

3.2.3 It is recommended for the consecutive ships of the series, the prototype of which has been tested according to 3.2.1, to carry out the tests including only such measurements which had been considered important in the analysis of the prototype.

3.2.4 If PRS requires direct measurements of stress of the hull, it is recommended to make the measurements in a way permitting to assess the values of stresses in the ship's hull, as well as the frequency analysis of the variable components of these stresses.

3.3 Condition for tests

3.3.1 The vibration tests of the propulsion system as well as of the hull girder and local vibrations of the hull should be carried out upon the following conditions:

- sea state up to 3;
- wind force below 4° Beaufort's degree ;
- sea depth – not lower than five times the mean draught;
- course stability of the rudder angles below 3 angle degree on side;

- loading condition: one of the typical conditions as described in “Loading Instruction”, the most recommended condition is that at which the expected vibration levels are the greatest.

The above mentioned conditions need not be fulfilled for auxiliary machines, equipments or parts of structure in the cases when these extraneous sources of excitation are not disturbing significantly the measured values, or an influence of such disturbances can be eliminated analytically.

3.3.2 The loading and operating conditions of the engine are to be such that any incidental excitation does not appear as a result of defected propeller blade, incomplete combustion in one of the cylinder, and so on.

3.3.3 The procedure of measurements and tests should provide for, as far as practicable, the simplest identification of the vibration sources, especially at points where excessive amplitudes occur. It means that at a given measuring point the vibration values should be measured separately by consecutive switching on the vibration sources.

3.3.4 When resonances outlined in Chapter 2 are being verified, the propeller revolutions should be changed at intervals not greater than 2.5 rpm in the speed range “half ahead” to “full ahead”. For ships provided with the controllable pitch propeller containing a programmed adjuster, the power feed should not be increased by more than 7% of the full power.

3.3.5 In order to avoid an unnecessarily large and impractical number of measurements and recordings in the report mentioned in 3.6, it is recommended to record the average readings from uniformly distributed points. It concerns first of all the parts of structure considered rigid (e.g. foundation beds, engine frames, etc.), but the number of measuring points should be increased if any one of the measured values exceeds the criterion value.

The increased number of measuring points should enable the recognition of the form of vibrational deflections.

For example, the value of lateral vibration (direction H) for the diesel engine heads can be treated as the mean value from measurements in at least three points situated at the ends and in the middle of the engine length at the level of combustion chamber, but only in such a case when no head vibrates too much taking into account the criterion given under item 10, Table 3.4.

3.3.6 The measuring instruments should meet the present requirements of the Polish Standards (see Appendix 1), or other national or international regulations recognized as equiponderant and sufficient. On the first ships of the series it is recommended to apply such measuring sets which enable the recording of the spectral and phase analysis of vibration processes.

3.3.7 It is recommended that the band pass spectrum analysis (if necessary) be within the frequency range as follows (see also note 3 at Table 3.4):

- lower limit f_d taken as lower of the two:
- f_{\min} = the lowest stable rotational speed of the propelling shaft (Hz)
- f_{2n} = the natural frequency of the two-node mode of flexural vertical vibration of the hull (Hz)
- upper limit taken not lower than the greatest frequency of the excitation affecting significantly the tested structure.

3.4 Criteria and application rules

3.4.1 Table 3.4 gives the criteria recommended as maximum levels of vibration in the ship during her normal service. Plots of these values are given in Fig. 3.4.

Table 3.4

Recommended values of permissible vibration levels on board ships under way, at sea state up to 3

No. of item	Name of construction machines or equipment	Directions of vibration	Frequency range	Amplitudes of vibration			Remarks
				x_p	v_{sk}	a	
			Hz(cps)	mm	mm/s	g	
1	2	3	4	5	6	7	8
1	Propeller shaft and intermediate shafts, points of measurement on bearings body	V, H	1-10	0.35			No concern of elastically supported shafts. Add direction <i>L</i> if the rolling bearings are used
			10-50			0,15.	
2	THRUST BEARINGS (on body) – slow running direct propulsion (without reduction gearbox)	L	1-5	1.0			Points of measurement on the body as high as axis of the shaftline
			5-50			0.1	
3	– for propulsive plants equipped with one-stage reduction gearbox	L	1-20	0.1			
			20-50			0.15	
4	– for propulsive plant equipped with two-stage reduction gearbox	L	1-16	0.1			
			16-50			0.1	
5	Main gearboxes, measuring points on bearing bodies or on seating pads	V, H	1-11.2	0.20			
			11.2-50			0.1	
6	Main turbines meas. points on seating pads or on bearings body	V, H	1-16	0.1			
			16-100		7.1		
7	Diesel engines up to 300 rpm meas. points on seating pads	V, H	1-10	0.16			
			10-100		7.1		

1	2		3	4	5	6	7	8
8	PROPULSIVE SYSTEM OF SHIP	Diesel engines, medium and high speed, above 300 rpm, meas. points on seating pads	V, H	5-10	0.25			Rigidly mounted. Limits concern also diesel-generator sets
				10-100		11.2		
Diesel engines as above (item 8) but mounted relatively soft or on lightweight substructure. Meas. point on seating pads of engine		V, H, L	5-10	0.4			Limits concern also all another machines with reciprocating parts. lightweight found – see note 4	
			10-60		18			
			60-100			1.0		
10		Internal combustion and all reciprocating machines, irrespective of the type of foundation: highest part of body, for diesels on covers	H, L	1-10	0.5			Limits concern also the auxiliary sets with reciproc. machines (diesel generators compressors, piston pumps e.a.)
				10-73		23		
				73-150			1,5	
11	Turbocharges for diesel, level on bearings		V, H, L	10-67		25		Practically limit is to be marked only by the maker of the bearings
				67-100			1.5	
12	Rotary machine and electrical motors up to 15 kW		V, H, L	10-200		4.5		Meas. points close as possible to bearings
13	Rotary machine and electrical motors of 15 up to 75 kW		V, H, L	10-200		7.1		Vibration of generating sets may be assessed acc. to ISO-8528 (see App. 1, item 18)
14	Rotary machine and electrical motors 75 kW and upwards, stiffly mounted		V, H, L	10-200		11.2		
15	Rotary machine and electrical motors 75 kW and upwards, elastic mounted or on lightweight substructure	V, H, L	10-200		18		lightweight substr. see note 4	

Notes: Directions of Vibration:
V – vertical, H – horizontal transverse, L – horizontal longitudinal

Amplitudes of Vibration:
 x_p – displacement (peak value), V_{rms} – velocity (rms – value)
 a – acceleration (peak value) in “g” units, where: “g” – 10 m/s²

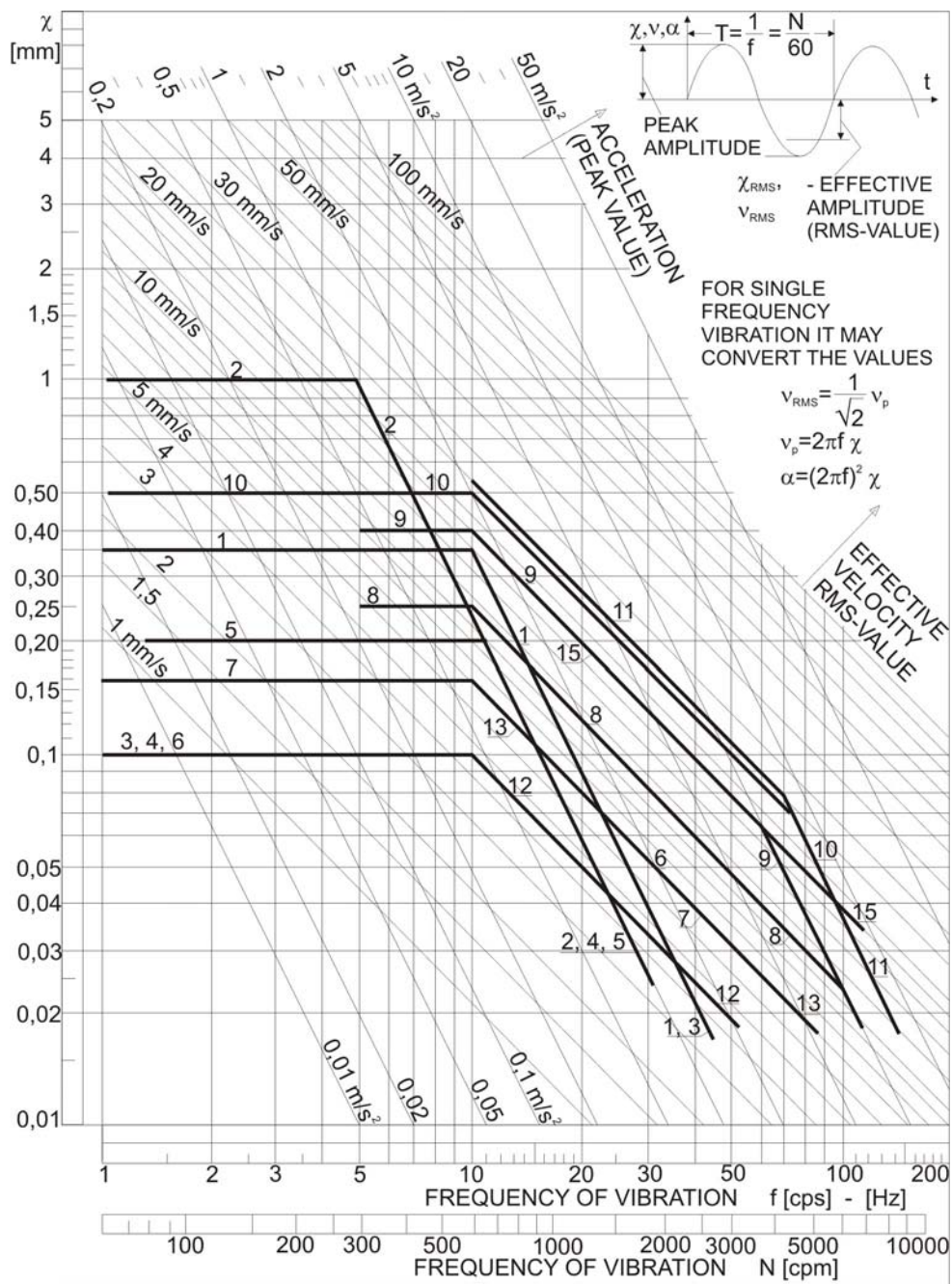


Fig. 3.4
 Graphic forms of recommended values of vibrations permissible per table 3.4
 (the curve numbers as per items in this table)

Notes to Table 3.4

- .1 Displacements, velocities, rms velocities and accelerations cannot be reciprocally converted if the vibrations have not single sinusoidal form.
- .2 Criteria listed in Table 3.4 are to be referred to separately for each of the directions defined in the column 3 of the Table,
- .3 Criteria are obligatory for the frequency bands given in column 4 of Table 3.4.

If the measured overall value is greater than the criterion, it is recommended to carry out the spectrum analysis of the signal in one-third-octave or narrower bands and to identify the dominant frequencies. This principle concerns also the wider band which cannot be altered because of the measuring instrument characteristics .

- .4 The interpretation of the definitions “lightweight substructure” used in items 9 and 15 Table 3.4 is as follows: at the design stage the light foundation is a foundation of the weight not exceeding one fourth of the machine being supported.

At the stage of measurement “light” means such a foundation on which the vibration level, measured when the supported machine operates, overpasses at least 4 dB (1.58 times) the referred level taken at the same time on the nearest framing member of the hull not directly attached to the examined foundation.

It is obvious that such comparison of levels relates to the same directions, conditions, etc.

- .5 For elastically supported propeller shaft (excluded by the note in column 8, item 1, Table 3.4) it is recommended to submit its vibration hazard estimation to PRS in a separate report including the analysis of all dynamic phenomena of the shaft line.

3.4.2 In the agreement between the owner and the shipyard the criteria of permissible vibration can be determined differently than presented in 3.4.1. However, such estimation should be based on the trustworthy tests of the shipyard and makers of equipment or on trustworthy standards or scientific publications. In such cases the details of agreement concerning the estimated criteria and their justification should be submitted to PRS,

3.4.3 Table 3.4 does not list the criterion values concerning the ship hull considered as the beam or analysed as the sum of structural sub-assemblies such as double bottom, bulkheads, superstructures, decks, stern and others. The following data can be used for guidance in case of need:

- .1 boundary values arising from the necessity of applying the health rules and recommendations (see 1.1.4 and Appendix 2);
- .2 required values involved indirectly by determination of permissible vibration levels on foundation beds of machines and equipment and on bearings of the shaftline;

- .3 values arising from the permissible fatigue stresses in the vibrating elements of the hull but when lack of such values is evident, it is recommended not to exceed the area presented within the lines (fig. 3.4):
 $x = 1 \text{ mm}$, $v_{\text{rms}} = 25 \text{ mm/s}$, $a = 15 \text{ m/s}^2$.

3.4.4 The vibration at frequencies up to 100 Hz, which cannot be identified with any of the items of Table 3.4, are to be estimated according to the permissible noise level (see Appendix 2, Fig. Z 2-4) accepted for the examined space, or according to the criteria agreed upon with PRS.

The criteria for machinery listed in Table 3.4 under items from 5 to 15 can be also valid for the estimation of the local vibrations occurring on instruments and indicators mounted on these machinery e.g. pressure gauges, thermometers and so on.

It is recommended, however, to assume a more severe criterion of accurate reading of the indications on such devices. Such a criterion can lead to changing of the place or way of the mounting of the instrument.

3.5 Evaluation of measurement results and conclusions

3.5.1 If the measurements on the prototype ship of the series indicate that the vibration levels exceed the recommended criteria, the next ship of the series should be protected against vibration by means of the measures indicated in Chapter 2, taking into account the conclusions derived from vibration test and modification of the structure carried out on the prototype.

The extended measures are to be approved by PRS in the cases covered by PRS supervision as specified in 1.1 and 1.2.

3.5.2 If the difference between the resonance frequencies measured during trials and those predicted exceed 10%, it is recommended to repeat the prediction analysis in the whole or in part for the presumed normal service conditions. The new analysis should be submitted to PRS before the next ship of the series comes into service.

3.5.3 When in normal service conditions the vibration levels on the hull girder or on the important structural sub-assemblies (stern, shafting, superstructure and others) exceed the values listed in Table 3.4 or are estimated as annoying people (see p. 1.1.4) and it is difficult or impossible to eliminate or to decrease them by modifying the structure, the range of revolutions restricted for continuous work should be defined according to the same rule as for torsional vibration. (see rule under 4.4, Part VI of the *Rules for Classification and Construction of Sea-Going Ships*, see also item 2.2.4 of this publication).

3.5.4 For the estimation of the passive vibration of the machinery it is recommended to apply the following rules:

- .1 the passive vibration which exceeds 1/3 of the active vibration level at the same point on the machine should be lowered by alteration of the foundation or by lowering the excitation level in the source;

- .2 the exceeding of the criterion value (given in Table 3.4 or defined by 3.4.2 and 3.4.3) e.g. due to passive vibration, needs separate examination before accepting the machinery into service.

The reasons for excessive passive vibrations mentioned above require detailed diagnostic tests.

The severity of oscillation or chatter of arrangement which does not generate active vibration, such as heat exchangers, pipings, electric lines, control and measurement instruments, is to be estimated as fatigue stresses hazardous to the propulsion system and safety of navigation.

3.6 Report on vibration measurements

3.6.1 The report on the trials is to be submitted to PRS before the ship comes into service. The report should include a clear set of measurements and their evaluation made on the basis of prediction and assumed criteria, being a detailed or even elaborated realization of the program agreed and accepted according to 3.2.

3.6.2 The report should include the sketches explaining alterations and modifications of the structure, if any, made during construction of the ship or after the first tests, aiming at reducing the vibration levels.

3.6.3 It is recommended that the report on a consecutive ship, includes (as an addition) the comparison of the results obtained, with the results known from prototype and former ships of the series.

APPENDICES

Appendix 1

LIST OF MORE IMPORTANT STANDARDIZATION PUBLICATIONS

1. PN-92/W-01352 Drgania na statkach. Metodyka pomiarów i rejestracji danych.
(idt¹ ISO 4867-1984: Code for the measurement and reporting of shipboard vibration data).
2. PN-92/W-01353 Drgania miejscowe konstrukcji statku i wyposażenia okrętowego. Metodyka pomiarów i rejestracji danych.
(idt ISO 4868-1984: Code for the measurement and reporting of local vibration data of ship structure and equipment).
3. PN-82/N-01350 Drgania. Terminologia.
(idt ISO-2041-1975: Vibration and Shock-Vocabulary).
4. PN-92/N-01351 Drgania mechaniczne i wstrząsy. Wytyczne do ogólnej oceny drgań na statkach handlowych.
(idt ISO-6954-1984: Mechanical vibration and shock-Guidelines for the overall evaluation of vibration in merchant ships).
5. PN-82/N-01351 Drgania. Podstawowe symbole i jednostki.
consistent with ISO-2041: Vibration. Basic symbols and units.
6. PN-91/N-01352 Zasady wykonywania pomiarów na stanowiskach pracy.
Vibration. Rules of making measurements at work places (consistent with ISO-2631/-1985).
7. PN-ISO 8579-2 : 1996 Przepisy odbioru przekładni zębatych. Określenie drgań mechanicznych przekładni zębatych podczas badań odbiorczych.
(idt ISO 8579-2 : 1993 Acceptance code for gears. Part 2. Determination of mechanical vibrations of gear units during acceptance testing).

¹ idt = identical with;

8. PN-ISO 10816-1 : 1998 Drgania mechaniczne. Ocena drgań maszyn na podstawie pomiarów na częściach niewrotujących. Wytyczne ogólne.
(idt ISO 10816-1 : 1995 Mechanical vibration. Evaluation of machine vibration by measurements on non-rotating parts. Part 1: General guidelines).
9. PN-90/N-01357 Drgania. Metody pomiarów i oceny drgań maszyn pod względem bezpieczeństwa i higieny pracy.
Vibration. Measuring methods and evaluation of vibration from machinery with regard to safety and occupational hygiene.
10. PN-90/N-01358 Drgania. Metody pomiarów i oceny drgań maszyn.
(eqv¹ ISO 2372 : 1974 Vibration. Measuring methods and evaluation of vibration of machines).
11. PN-76/M-43121 Wentylatory. Metody pomiaru drgań.
Fans. Vibration measurement methods, (consistent with ISO-2372-1974).
12. PN-EN 1299 : 2002 Drgania mechaniczne i wstrząsy. Wibroizolacja maszyn. Informacje dotyczące stosowania izolacji źródła.
(idt EN 1299 : 1997 Mechanical vibration and shock. Vibration isolation of machines. Information for the application of source isolation).
13. PN-EN 12096 : 2002 Drgania mechaniczne. Deklarowanie i weryfikowanie wartości emisji drgań.
(idt EN 12096 : 1997 Mechanical vibration. Declaration and verification of vibration emission values).
14. PN-IEC 34-14 : 1998 Maszyny elektryczne wirujące. Drgania mechaniczne określonych maszyn o wzniosach osi wału 56 mm i większych. Pomiar, ocena i wartości graniczne drgań.
(idt EN 60034-14 : 1996 idt ISO 60034-14 1996, Cor. PN-IEC 34-14 : 1998/App. 1 : 2000. Rotating electrical machines. Part 14: Mechanical vibration of certain machines with shaft heights 56 mm and higher. Measurement evaluation and limits of vibration).

¹ eqv = equivalent to

15. PN-ISO 7919-1 : 2000 Drgania mechaniczne maszyn z wyłączeniem maszyn tłokowych. Pomiary drgań wałów wirujących i kryteria oceny. Część 1: Wytyczne ogólne. (idt ISO 7919-1-14 : 1996 Mechanical vibration of non-reciprocating machines. Measurement of rotating shafts and evaluation criteria. Part 1: General guidelines).
16. PN-ISO 2923 : 2001 Akustyka. Pomiary hałasu na statkach. (idt ISO 2923 : 1996 + Cor. 1 : 1997, Measurement of noise on board vessels).
17. PN-W-01350-2 : 1996 Ochrona przed hałasem na statkach morskich. Warunki i metody pomiaru hałasu. Protection against noise on board sea-going ships. Sound level measurement conditions and measuring methods (consistent with IMO Resolution A.468 (XII) and ISO-2923 : 1966)
18. PN-ISO 8528-5 : 1997 Zespoły prądotwórcze prądu przemiennego napędzane silnikiem spalinowym tłokowym. Zespoły prądotwórcze. (idt Reciprocating internal combustion engine driven alternating current generating sets – Part 5. Generating sets).

HEALTH CRITERIA VALUES FOR ESTIMATION OF VIBRATION

All health criteria refer to vibratory motion transferred by contact from an oscillating body to the human body.

The effects of the vibration of human body depends on:

- the amplitude (or: intensity, severity) and frequencies of the vibration;
- the direction and duration of exposure (the cumulative effect);
- the sensitivity and age of the individual body.

The most frequent sickness symptoms are as follows:

- psychophysical: fatigue, depression, fear, nervousness, deconcentration, inability and indisposition towards work,
- physiological: difficulty in breathing, changes in blood pressure, changes of activity of inner glands, hemolize, stomach and chest ache, irritation of kidneys and bone marrow etc.

The susceptibility to the resonance vibration is a very important dynamic feature of the human body. The frequencies that are hazardous to individual organs of the body are as follows:

- | | |
|--|--------------|
| – head (vibration relative to body and also decay of vision acuteness) | 20 – 30 Hz |
| – backbone (bending vibrations) | 4 – 14 Hz |
| – intestines | 3 – 6 Hz |
| – eye balls (resonance) | 60 – 90 Hz |
| – joints and shin-bones of legs | 50 – 70 Hz |
| – lower jaw-bons | 100 – 200 Hz |

The criteria which the contemporary knowledge offers for estimation of the exposure to vibration are quoted with explanations on Fig. Z2-1 to Z2-5. These criteria are not mutually coherent and they leaves a pretty good margin for subjective evaluation.

The ISO recommendations for ships, quoted on Fig. Z2-3 (see Appendix 1, item 4 and 6) are applied only for an approximate estimation of the comfort of crew, taking into account: complaints (serious, medium, small) or the health criteria, listed in the agreements, for vibration of the hull girder and superstructure, in the spaces continuously manned. It means that these recommendations can be useful in quality assessment of measures adopted in prevention of vibration of the hull girder, superstructure, some machines and so on.

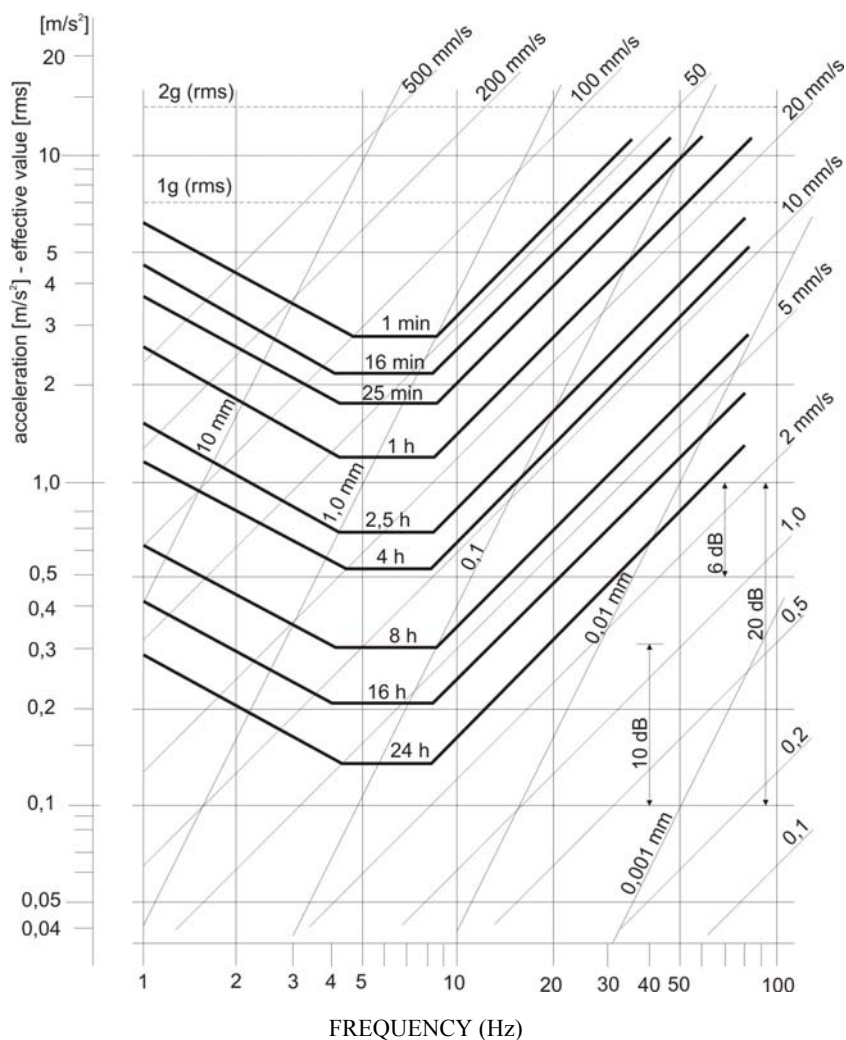


Fig. Z 2-1

The vibration influencing the human body, in a standing or sitting position depending on the frequency and duration of exposure

The curves define the fatigue-decreased proficiency boundary in any human activity which requires a continuous attention (e.g. when driving a car),

The time shown on each curve is the limit beyond which the exposure to vibration can carry a significant risk of impaired working efficiency.

The doubling of the shown levels (i.e. their increase by 6 dB) means the upper so called “exposure limit” which may be exceeded only when specially justified and when precautions measures are taken.

The level value reduced by 10 dB (i.e. lowered up to 1/3 of the shown levels) defines the so called “reduced comfort boundary” above which the trouble with eating, reading and writing occurs (after ISO-2631-1985).

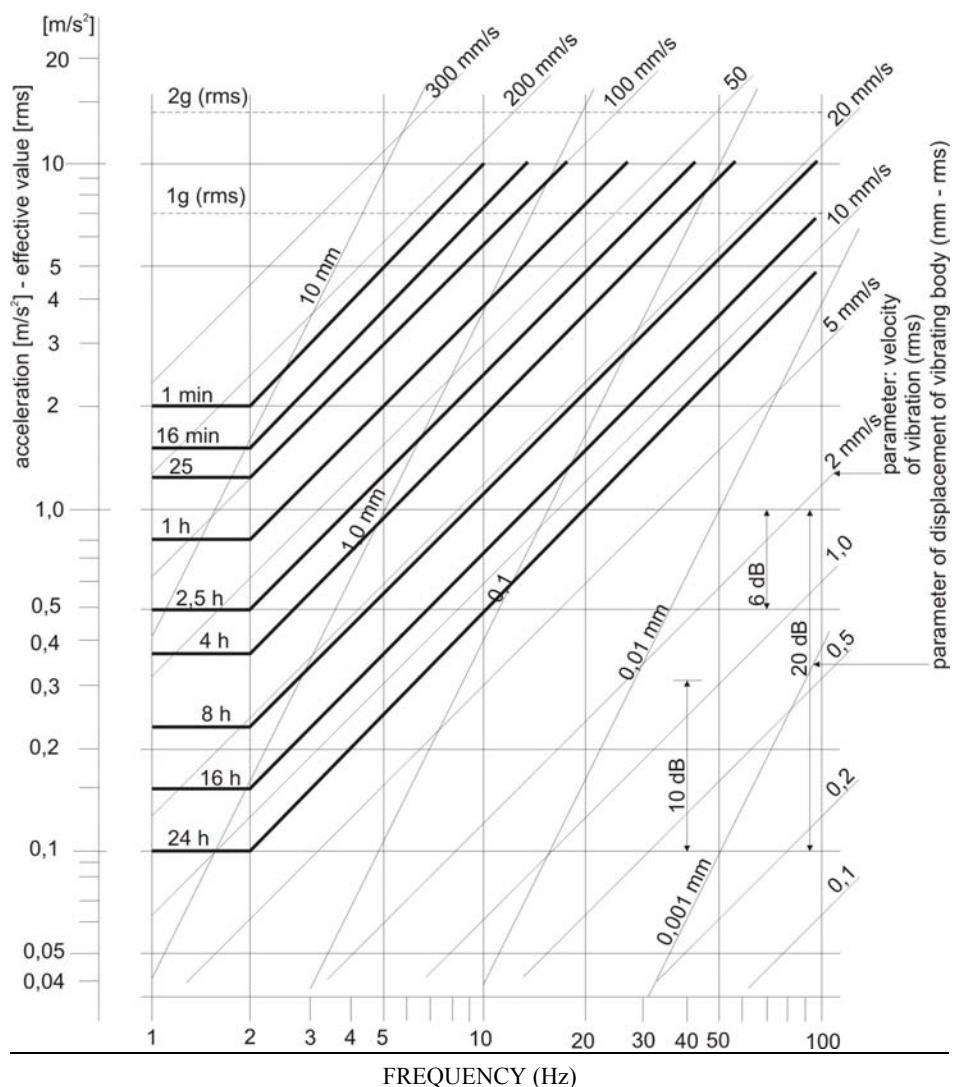


Fig. Z 2-2

The limit of “fatigue-decreased proficiency boundary” as on Fig. Z 2-1, but in the transverse direction, i.e. back-to-chest or side-to-side alike in a standing, sitting or reclining position

For the discrete (single or multiple) frequencies the evaluation lies in reference to the appropriate limit at the frequency taken from Fig. Z 2-1 or Z 2-2.

Narrow band vibration (up to 1/3 octave band) should be evaluated in reference to the limit at the centre frequency of that band.

Similar recommendation concerns the broad band distributed vibration; obviously the evaluation is possible when the-one-third-octave spectrum has been recorded during measurements.

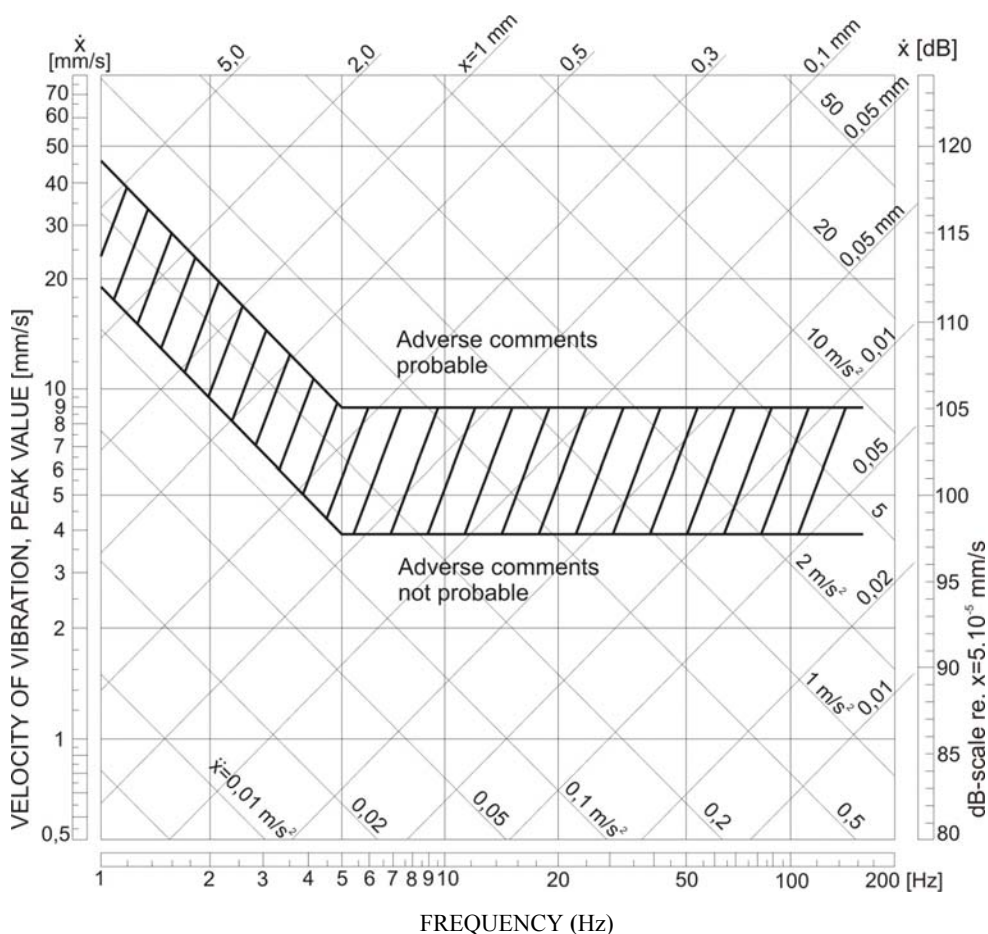


Fig. Z 2-3
 ISO-6954 “Mechanical vibration and shock-Guidelines for the overall evaluation of vibration in merchant ships”

The shadowed area delimits the values, in terms “maximum repetitive value”, which inform on response of crew: above the complaints are frequent; below, are very rare; inside, are sporadic. Values below the area comply with ISO-2631 with respect to safe and satisfying exposure to whole body vibration. The criteria are to be separately referred to each of the 3 axis of a body. When instead of peak, the r.m.s. value is measured, then the bandwidth and time-averaging should be specified, and rms value should be converted according to formula:

$$V_{peak} = F_c \sqrt{2} V_{rms}$$

when crest factor $F_c = 1,0$ for pure sinusoidal signal and $F_c = 1,8$ for commonly encountered condition on ships.

A more exact value F_c can be determined by measurement only.

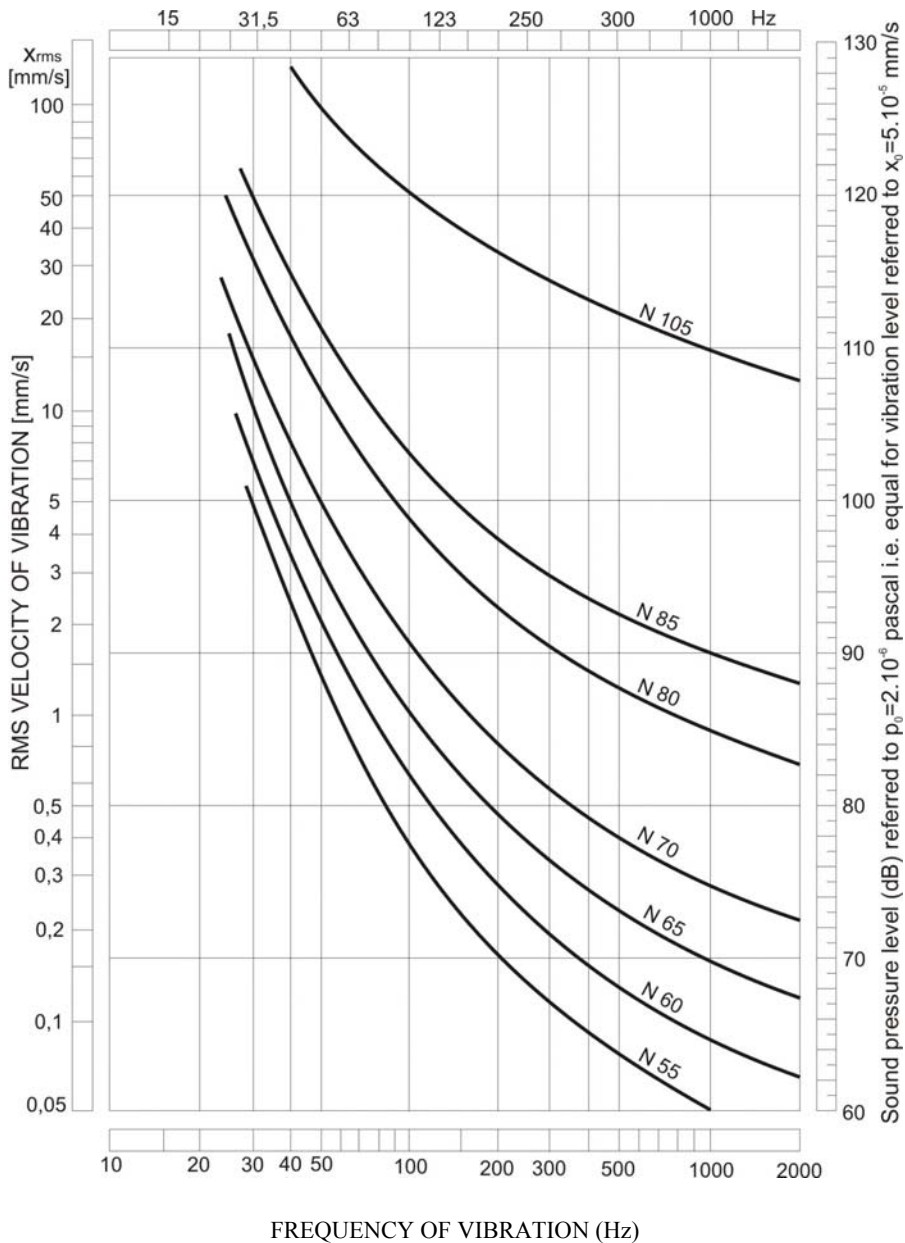


Fig. Z 2-4

The noise rating (NR) number at low and medium frequency bands as possible criteria for the estimation of vibration.

The curves NR are drawn for the analysis in 1/3 octave bands. The curve number increased i.e. by 5N60 means 65 dBA, points to A-weighted noise level (dBA) generated by vibration.

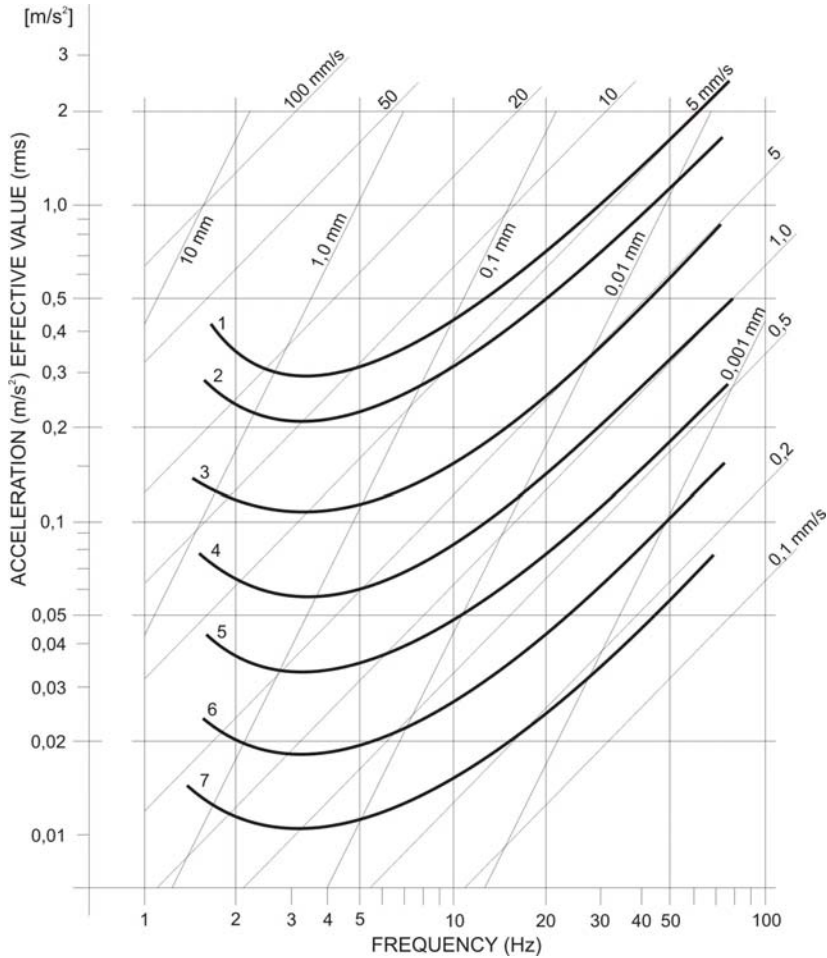


Fig. Z 2-5

The health standards for vibrations for sea-going, river and lake ships Health Ministry of the former USSR, standard No. 1103-73. Lack of information whether it is still obligatory.

Limits concern:

- 1 – Unattended automatic engine rooms – exposure time up to 60 min per a day
- 2 – The engine rooms provided with periodically attended control stands and periodical watch – exposure time up to 120 min. per a day
- 3 – for continuous watch in the Machinery Control Room (MCR) and for industrial kind of stands (e.g. on fishing ships)
- 4 – Service rooms and cabins on ships when duration of a trip is up to 8 hours
- 5 – Cabins on sea-going ships of III Class when duration of a trip is not longer than 24 hours
- 6 – Cabins on sea-going ships of I and II class when duration of a trip is longer than 24 hours
- 7 – Medical care rooms (ship's hospitals).

INFORMATION ON SOME PROGRAMMES APPLIED FOR CALCULATION OF THE SHIP STRUCTURE VIBRATION

The programmes can be realized at the Centrum Techniki Okrętowej, Gdańsk.
Data compiled in March 2004.

1) UNCA 04 Program for propeller analysis in non-uniform wake. Concise description

Program Name: UNCA 04

Platform: PC / W9x / W2000

Algorithm: The basic theoretical model of the computational part is so called deformable lifting surface, which can dynamically change its geometry in the region of the blades covered by sheet cavitation. The lifting surfaces modelling propeller blades are composed of discrete vortex elements reflecting time-dependent hydrodynamic loading and of discrete sources and sinks reflecting the blade thickness. The intensities of the hydromechanic singularities creating the lifting surface are determined on the basis of the kinematic boundary condition applied on this surface. The vortex elements located on the blade are supplemented with the free vortex system located on the helicoidal surfaces behind the blades. These surfaces are divided into the variable zone, in which the intensities of vortex elements are time-dependent, and steady zone, in which they correspond to the circumferentially averaged inflow velocity field. The sheet cavity and the cavitating tip vortex are modelled with additional systems of sources and sinks with time-dependent intensity. Intensities of these sources and sinks are determined on the basis of the dynamic boundary condition imposed on the cavity boundary. The pressure field on the lifting surface is computed using the Bernoulli equation for unsteady flows. The same equation is employed for determination of the pressure fluctuations generated by the cavitating propeller in the surrounding space.

Range of Application: Program is used to perform analysis of ship propeller of known geometry in given 3D inflow field. The calculations may be carried out for single screw propeller as well as for twin-screw arrangement. Program allows for the calculations of blade and shaft forces, pressure distributions, cavitation phenomena on a propeller and pressure pulses in propeller surrounding.

Input: Input data consist of the following information:

- detailed description of blade geometry (radial characteristics of pitch, chord length, skewback etc.),
- geometry of profiles at particular blade sections,
- angular positions to perform calculations,

- description of inflow velocity field in form of velocity coefficient v/v_{ship} , separately for axial, tangential and radial components,
- coordinates of points for pressure pulses calculation and solid boundary factors.

Output: The program can supply the following information:

- time-dependent pressure field on the blades
- time-dependent hydrodynamic forces acting on a single blade
- time-dependent hydrodynamic forces acting on the propeller shaft
- unsteady cavitation phenomena appearing on the propeller blade, in particular:
 - sheet cavitation
 - tip vortex cavitation
 - bubble cavitation
- pressure pulses generated by the cavitating propeller in the selected points of the surrounding space, in particular on the ship hull surface.

2) Nastran/Patran – static and dynamic (linear and nonlinear) structure analysis

The Nastran program (with pre- and postprocessor: Patran) is performed for numerical analysis of complicated mechanical structures. It is based on Finite Element Method (FEM). All basic types of FEM elements are included in the software, from 0-D to 3-D. The following analysis types are possible: static, dynamic (modal method or transient integration), thermal, fatigue and contact problems. Big problems can be supported by several tools like: restarts, superelements, global-local method. As a result of calculations, the following data can be received: internal forces, reactions, stresses, deformations, normal modes and frequencies, forced vibration's amplitudes and phases etc.

The program can be applied to global and local strength analysis of ship structure as well as to natural and forced vibration analysis of ship hull and superstructure (with added water mass).

3) Maestro – strength and dynamic structure analysis, weight-cost optimization of the ship hull construction

The Maestro program is based on Finite Element Method (FEM). Specialized pre-and postprocessor (for ship structure) is build into the program.

The program is mainly used to global strength calculations of ship hull structure. Several algorithms of the program are especially dedicated to marine structure e.g. automatic mass-balance. On the base of the program, fast hull's strength analysis can be performed. Classification societies' requirements can be evaluated. Algorithm of the limiting strength (Adamchak method) is dedicated to calculation of ship hull's limiting moment. The program is fitted with the interface for transmission data to Nastran program.

The program can be also applied to optimization analysis of ship hull e.g. weight-cost or location of hull's centre of gravity.

4) LWAL – shaft line alignment analysis

The Lwal program is performed for shaft line alignment calculations with taking into account stiff characteristics of the boundary conditions (journal bearings with ship's hull). The journal bearings (mainly stem tube bearing) can be modelled as a continuous support. The program is based on Finite Element Method (shaft line) and Finite Difference Method (bearings' characteristics). Shaft line analysis can be supported by semi-automatic optimization algorithm. As a result of calculations, the following data can be received: journal bearings' reactions, shearing forces and bending moments of the shaft line, shearing and bending stresses and shaft line deformation.

5) DTOR – torsional vibration analysis of propulsion system

The DTOR program is performed for torsional vibration calculations of simple and forced power transmission systems. The program is based on Finite Element Method. Specialized procedures are performed for mass and gas harmonic tangential forces determining. Cylindrical damping and coupling's stiff characteristics can be modeled in frequency function. As a result of calculations, the following data can be received: normal modes and frequencies, forced vibration's stresses, amplitudes and phases. The program can be used for recalculations of the measured data.

6) DSLW – coupled axial vibration analysis of propulsion system

The DSLW program is performed for power transmission system axial vibration calculations. The following excitations are taken into account: axial hydrodynamic forces, torsional-bending-longitudinal kinematic coupling of the crankshaft and torsional-longitudinal hydrodynamic coupling of the propeller. The program is based on Finite Element Method. Torsional vibration analysis is performed as a first step in order to take into consideration couplings of the crankshaft. Specialized procedures are performed for mass and gas harmonic tangential and radial forces determining. The other algorithms are performed for determining of stiff-damping characteristics of the thrust bearing and axial damper. As a result of calculations, the following data can be received: normal modes and frequencies, forced vibration's amplitudes and phases and reactions of the thrust bearing and axial damper. The reactions are the dynamic excitations of ship hull and superstructure.

7) DGR – bending analysis of propulsion system

The DGR program is performed for power transmission system's bending (lateral, whirling) vibration calculations taking into account stiff-damping characteristics of the boundary conditions (journal bearings with ship's hull). The journal bearings (mainly stem tube bearing) can be modelled as a continuous support. The program is based on Finite Element Method. As a result of calculations, the following data can be received: normal modes and frequencies, forced vibrations amplitudes and phases, shearing and bending stresses and reactions of the journal bearings. The reactions are the dynamic excitations of ship hull and superstructure.

EXAMPLES OF PRESENTATION OF CALCULATION RESULTS

1. Calculation of the hull vibration

The results of calculation are shown on the diagram (see Fig. Z 3-1) where the frequencies are plotted as the ordinates, and the propeller or main engine revolutions are plotted as the abscissae.

Natural frequency of a given mode, when it is computed for two extreme conditions (ballast and full load), delimits the field for intermediate states, as shown by dotted areas in Fig. Z 3-1.

The coincidence of a.m. areas with the skew lines indicates the hazard of resonance.

These skew lines represent chief excitations from the main engine and propeller, usually 1 and 2 harmonics of main engine r.p.m. as well as 1 and 2 harmonics of blade components of the screw.

2. Calculation of shaft line vibration

The calculation of longitudinal frequencies are made as the function of stiffness of the thrust bearing since it is impossible to determine exactly its stiffness. The foreseen range of the stiffness is determined on the basis of measurements made on other ships or from the literature. For this range the analysis is made to check whether the excitation frequency (the first or second blade frequency) does not intersect the diagram of free frequencies. The intersection indicates the possibility of resonance (see Fig. Z 3-2).

Similar procedure is carried out for the flexural shaft vibration (see Fig. Z 3-3), however the calculations of free vibration are made for the presumed variable values of the stern bearing stiffness with the stiffness of the intermediate bearings being constant.

VERTICAL VIBRATION OF HULL GIRDER, RESULTS OF CALCULATION

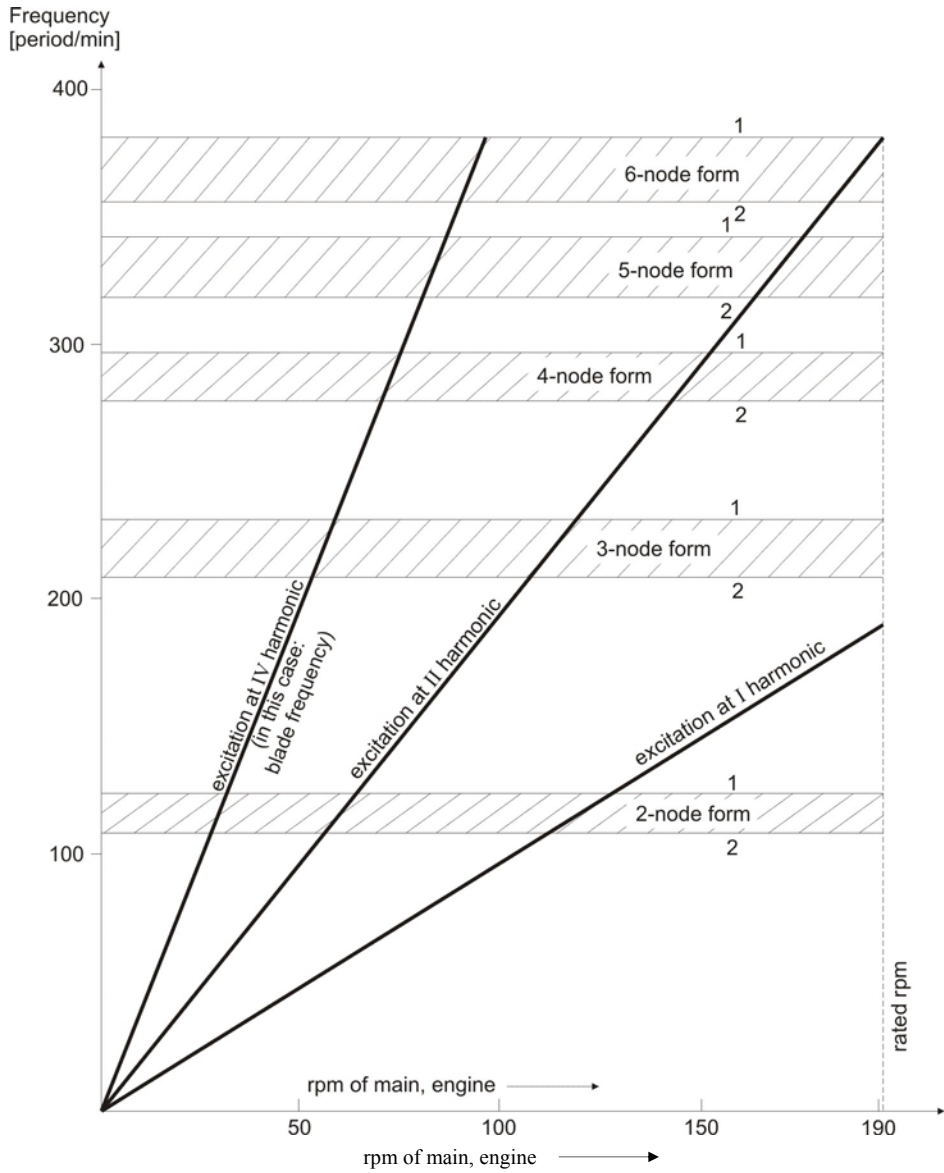


Fig. Z 3-1

Chief excitation frequencies of main engine and propeller against free frequencies of hull girder 1 – under ballast, 2 – under load

VIBRATION OF SHAFTING ON SHIP B76/ ... RESULTS OF CALCULATION

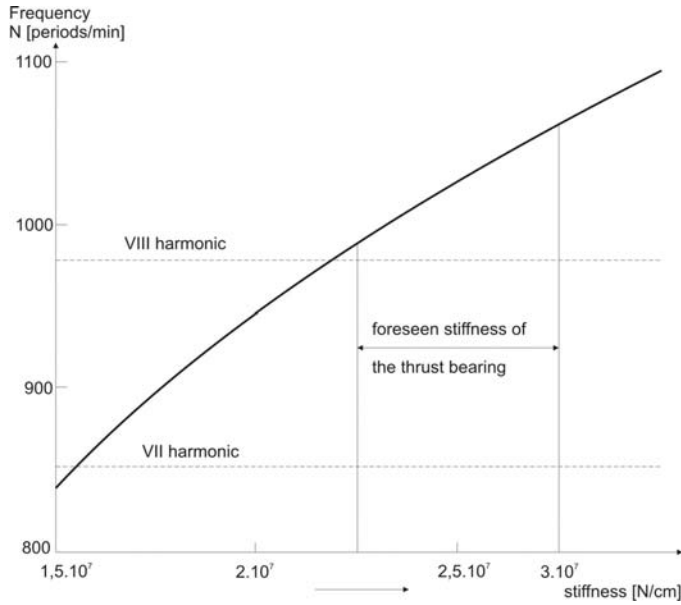


Fig. Z 3-2

The free (i.e. natural) longitudinal frequency of shafting as the function of the thrust bearing stiffness

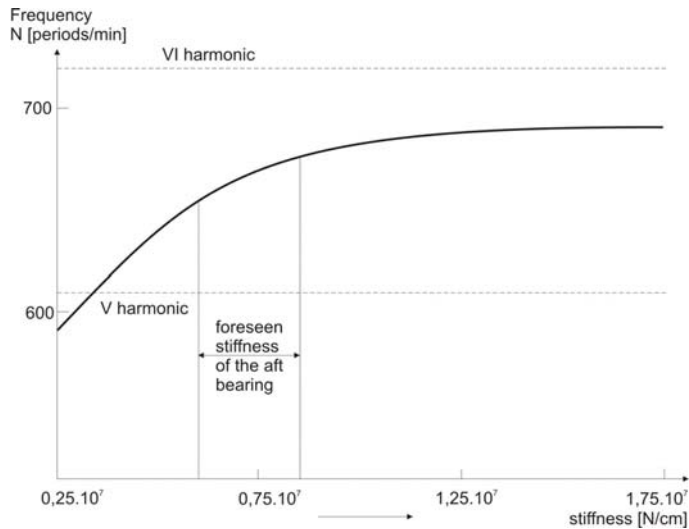


Fig. Z 3-3

Free frequency of flexural vibration of shafting as the function of the aft bearing stiffness

APPROXIMATE CALCULATION OF THE NATURAL HULL GIRDER FREQUENCIES

The approximate frequency values for the 1-st two-node form of the vertical vibration of the hull can be calculated from the Schlick formula. It is convenient to use such a form of this formula that contains coefficients verified for the given type of ship.

The formula with a shape coefficient introduced by P.Y. Chang is as follows:

$$f_1 = 63350k \sqrt{\frac{I}{\Delta L^3}} \quad (\text{Hz})$$

where:

I – moment of inertia of the midship cross section (m^4)

Δ – ship's displacement (t)

L – ship's length (m)

and shape coefficient:

$$k = \left\{ \left(1 + \frac{B}{2T} \right) \left[1 + 21.5 \left(\frac{B}{H} - 0.275 \right) \left(\frac{H}{L} \right)^2 \right] \right\}^{-\frac{1}{2}}$$

where:

L, B, H, T – main dimensions of the hull, respectively; length, breadth, depth, draught.

The values of shape coefficient for the most common dimensional proportions are shown in Fig. Z 4-1.

The free frequencies for modes from 3-node to 7-node can be approximately estimated from Fig. Z 4-2.

References:

1. P. Y. Chang – The effect of Varying Ships Hull Proportions and Hull Materials on Hull Flexibility, Bending and Vibratory Stressess, SSG-288, Final Report. US Coast Guard Headquarters, Washington 1979
2. F. H. Todd – Ship Hull Vibration, London 1961.

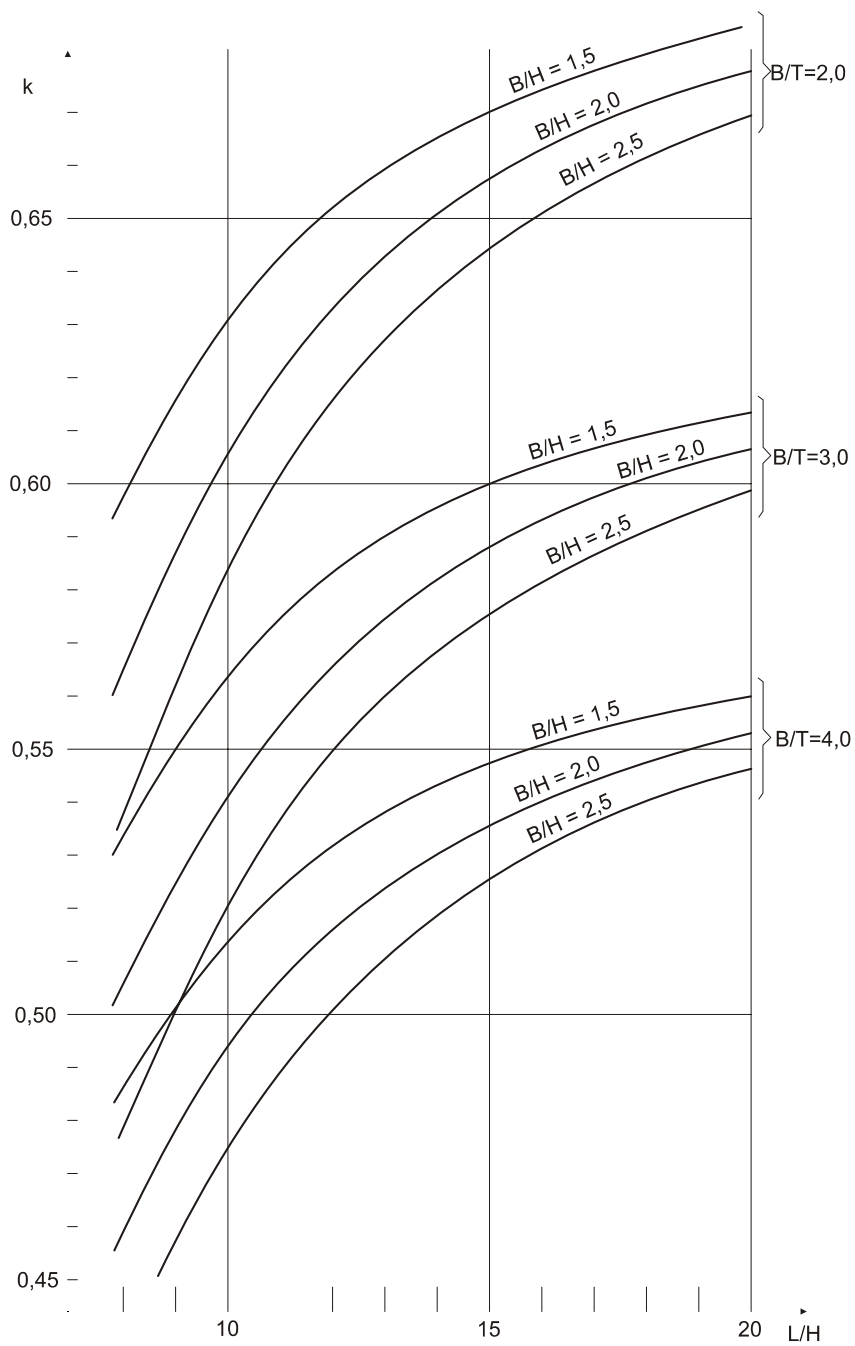


Fig. Z 4-1
 The values of the shape coefficient "k" for the most common dimensional proportions of the hull

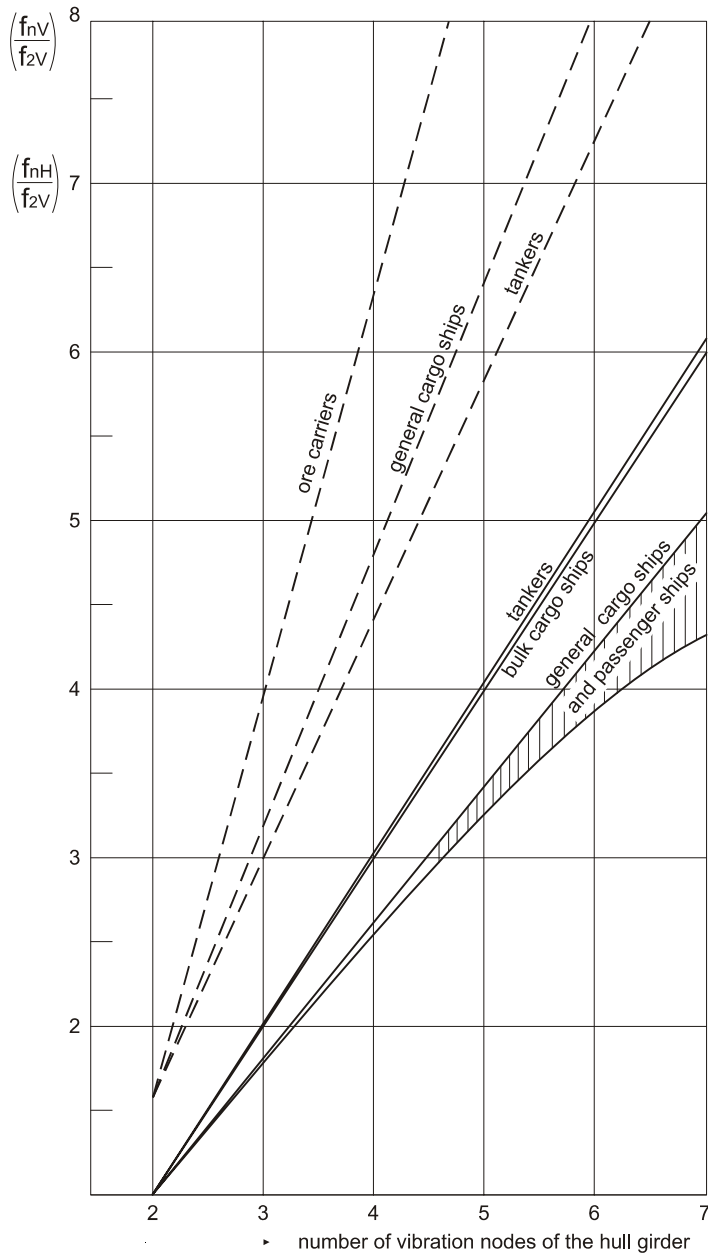


Fig. Z 4-2

Rough estimation of the frequencies for the modes greater than two-node.

Full lines for vertical vibration, broken lines – for horizontal vibration, f_{2V} – frequency of I-st, two-node form of vertical flexural vibration of the hull girder (taken from: F.H.Todd – Ship Hull Vibration, London 1961).

INFLUENCE OF REGULAR WAVE ON THE RESONANCE VIBRATION OF THE HULL GIRDER

The waves acting on the hull always cause more or less intensive vibrations, however in particular conditions when time of sequence of such pulses is regular and near or equal the period of hull free vibration, the amplitudes of such vibration can reach the values hazardous for the hull structure. The period of encounter for steady conditions is determined by the formula:

$$T_e = \frac{\lambda}{c - v \cos \beta}$$

where:

λ – wave length (m)

c – wave velocity (m/s)

v – ship speed (m/s)

β – heading angle against wave (degrees)

$\beta = 0^\circ$ – wave from stern (following sea)

$\beta = 180^\circ$ – wave from bow (head sea)

The values of λ and c are taken from the hydrographic data concerning the navigation route on which the ship will most frequently navigate.

In some cases the diagram for swelling sea, quoted in Fig. Z 5-1, may be used.

References:

1. J. Dudziak – Okręt na fali. (Ship on wave). Gdańsk 1980. Wyd. Morskie
2. Kenneth C. Barnaby – Basic Naval Architecture, Hutchinson. London 1969.
3. M. K. Ochi – Wave Statistics for the Design of Ships and Ocean Structures. Trans. SNAME Vol. 86, 1978.

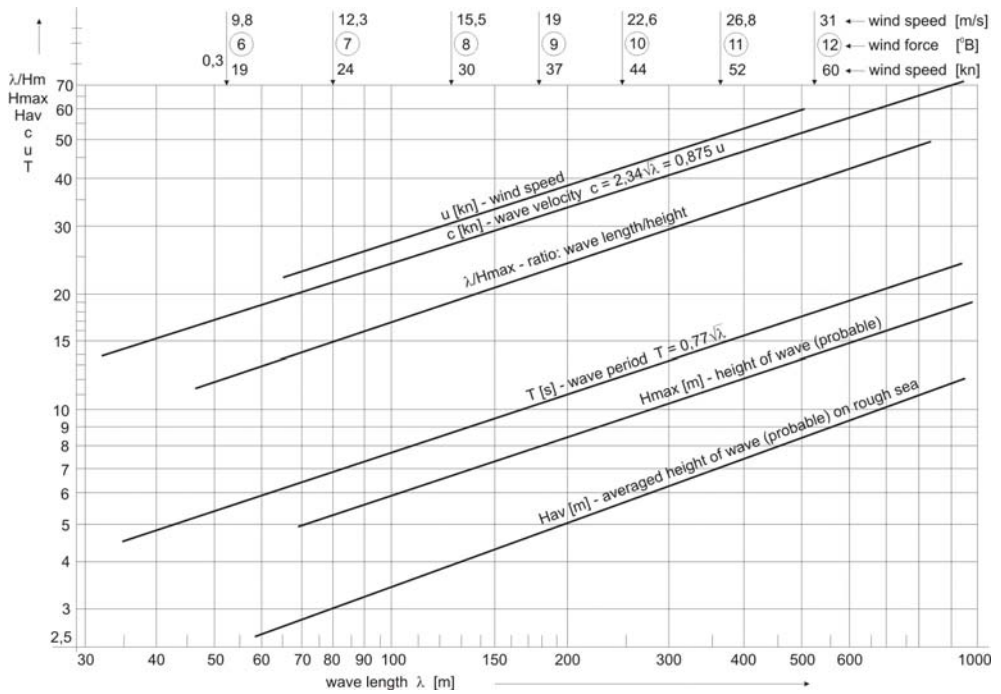


Fig. Z 5-1

An approach to set out the parameters of the rough sea

When using the diagram, the effects of both fetch and duration must be kept in mind; fetch of at least 300 naut. miles, duration a few “blowing” days (the so called “Zimmerman’s diagram”).

ASSESSMENT OF CLEARANCES BETWEEN THE BLADE TOP AND SHELL PLATING IN THE PROPELLER APERTURE

When there is a probability of occurrence of such cavitation that is hazardous as vibration-exciting forces on aft (i.e. the high-speed variability of the bubble volume on blades going by region of the rapid fall of inflow wake velocity), the minimum value of the upper clearance between blade top and shell plating may be assessed from the formula:

$$\left(\frac{c}{D}\right)_{min} \geq \frac{0.96}{\sqrt{(AB)^2 + A\left(\frac{15}{nD}\right)^2} - AB} - 0.05$$

where:

$$A = 0.01 \frac{Z^{\frac{3}{2}}}{\left(\frac{t}{D}\right)^{\frac{4}{3}}}$$

$$B = \frac{W_{Tmax} - W_T}{1 - W_T} \frac{1}{\sqrt{\delta_A}}$$

$$\delta_A = \frac{(h_a + 10.4)9.81}{0.5v_s^2(1 - W_T)^2}$$

- c – clearance between the outer top of the propeller blade and the hull plating above it, [m]
- D – propeller diameter, [m]
- n – propeller revolutions, [cps]
- Z – number of blades
- t – the maximum blade thickness on radius $0.7 R$, [m]
- h_a – draught of aft bearing axis regarding the influence of wave elevation at stern, [m]
- v_s – ship speed, [m/s]
- W_T – effective Taylor's wake
- W_{Tmax} – maximum wake peak at $0.9 R$ in the upper part of inflow

When there is a possibility of designing the propeller in such a way that it will be noncavitating in all blade positions over the disc area (particularly in the upper part of wake), the minimum clearance may be assessed from the formula:

$$\left(\frac{c}{D}\right)_{min} \geq \frac{0.64nD}{z^{\frac{3}{4}}} \left(\frac{t}{D}\right)^{\frac{2}{3}} - 0.05$$

In both of the above formulae the pressure limit on the aftbody shell has been assumed:

$$\Delta p_z = 800 \cdot 9.81 = 7848 \quad [\text{N/m}^2]$$

It is correct only for a typical feature aftbody construction.

References:

1. K. O. Holden – Excitation Forces and Afterbody Vibrations Induced by Marine Propeller Blade Cavitation.
Norwegian Maritime Research No 1/1979.
2. H. Jarzyna, T. Koronowicz – Analiza oraz ocena i propozycje zmian kryteriów określania dopuszczalnych sił wymuszających drgania kadłuba, stosowanych we wstępnej fazie projektowania statku. Gdańsk 1980.
(Analysis, estimation and improvement of criteria to evaluate the forces exciting ship vibration. Study for early stage of design, prepared on PRS order. Gdańsk, 1980)

ESTIMATION OF THE NON-UNIFORMITY OF AN INFLOW WAKE

The non-uniformity of the velocity field can be determined by difference of coefficients of the wake:

$$W_{max} - W_T$$

where:

W_{max} – maximum value of the nominal wake coefficient in the region $r/R = 0.9 \div 1.0$

W_T – effective coefficient of the wake.

The permissible value ($w_{max} - w_T$), from the point of view of cavitation and vibration forced by the propeller on the hull plating can be determined by the Holden's formula:

$$(W_{max} - W_T) \leq 0.02 \frac{\Delta p_{dop} \sqrt{h+10} \left(2 \frac{c}{D} + 0.1\right)^k}{(nD)^2 v_s f_2} \frac{\Delta J}{J} \quad (1)$$

where:

Δp_{dop} – permissible value of the pressure on the hull plating (Pa).

The recommended value of pressure limit on the hull plating for standard hull structures is $\Delta p_{dop} = 4000$ Pa

h – draught [m]

n – propeller rotational speed [cps]

D – propeller diameter [m]

v_s – ship speed [m/s]

c – clearance between the outer top of the propeller blade and the hull plating

k – coefficient determined as follows:

$$k = 1.63 + 1.4 \frac{c}{D} \quad \text{for} \quad \frac{c}{D} \leq 0.45$$

$$k = 1.00 \quad \text{for} \quad \frac{c}{D} \geq 0.45$$

$$f_2 = \frac{(f P)_{0.95}}{(f P)_{0.8}}$$

ratio of products of the deflection ratio of products of the deflection of profile “ f ” and pitch “ P ” at radius $r/R = 0.95$ and at radius $r/R = 0.8$

The following values “ f_2 ” in the formula for W_{max} are assumed:

if $f_2 < 0.3$, $f_2 = 0.3$

if $f_2 > 0.8$, then $f_2 = 0.8$

$$J = \frac{v_s}{nD} \quad \text{–} \quad \text{advance ratio.}$$

The allowance ΔJ which enables determination of the difference ($W_{\max} - W_T$) can be calculated from the following algorithm:

$$1. \quad \left(\frac{A_e}{A_o}\right)_c = \frac{T}{1850n^2 D^4 \left[\frac{h+10}{nD^2} + 0.066 \right] \left[1.067 - 0.23 \left(\frac{P}{D}\right)_{0.8} \right]} \quad (2)$$

where:

T – thrust [N]

h – draught of propeller axis [m]

n – propeller rotational speed [cps]

D – propeller diameter [m]

$\left(\frac{P}{D}\right)_{0.8}$ – pitch coefficient on radius $r/R = 0.8$

$$2. \quad f_1 = \frac{\left(\frac{A_e}{A_o}\right)_c 2.13 D}{1_{0.9} z} \quad (3)$$

where:

$A_o = \frac{\pi D^2}{4}$, A_e – expanded area of the blades [m²]

z – number of blades

$1_{0.9}$ – length of profile on radius $r/R = 0.9$ [m]

$$3. \quad \Delta k_T = \frac{f_1 - 1.0}{f_1} \frac{T}{\rho D^4 n^2} \quad (4)$$

where:

T – thrust [N]

ρ – water density ($\rho = 1025 \cdot \left[\frac{Ns^2}{m^4} \right]$ for sea water)

n – propeller rotational speed [cps]

$$4. \quad \Delta J = \Delta k_T \frac{1}{\frac{\Delta k_T}{\Delta J}} = \frac{\Delta k_T}{0.06(2+z)} \quad (5)$$

where:

z – number of blades

If the value of the effective wake coefficient used in the formula (1) is unknown, such value can be determined by multiplying the mean nominal value W_n of the wake by 0.7.

Irrespective of the criterion mentioned above, the maximum W_{\max} value should not exceed 0.8. When this condition is not fulfilled the great pulsations of pressure, noise and cavitation erosion can be expected on the hull plating.

If $W_{\max} > 0.8$, the shape of the ship stern should be re-designed.

Reference: the same as in Appendix 6.

**EXAMPLES OF FAULTY DESIGNS DUE TO GENERATION
AND SPREADING OF VIBRATION**

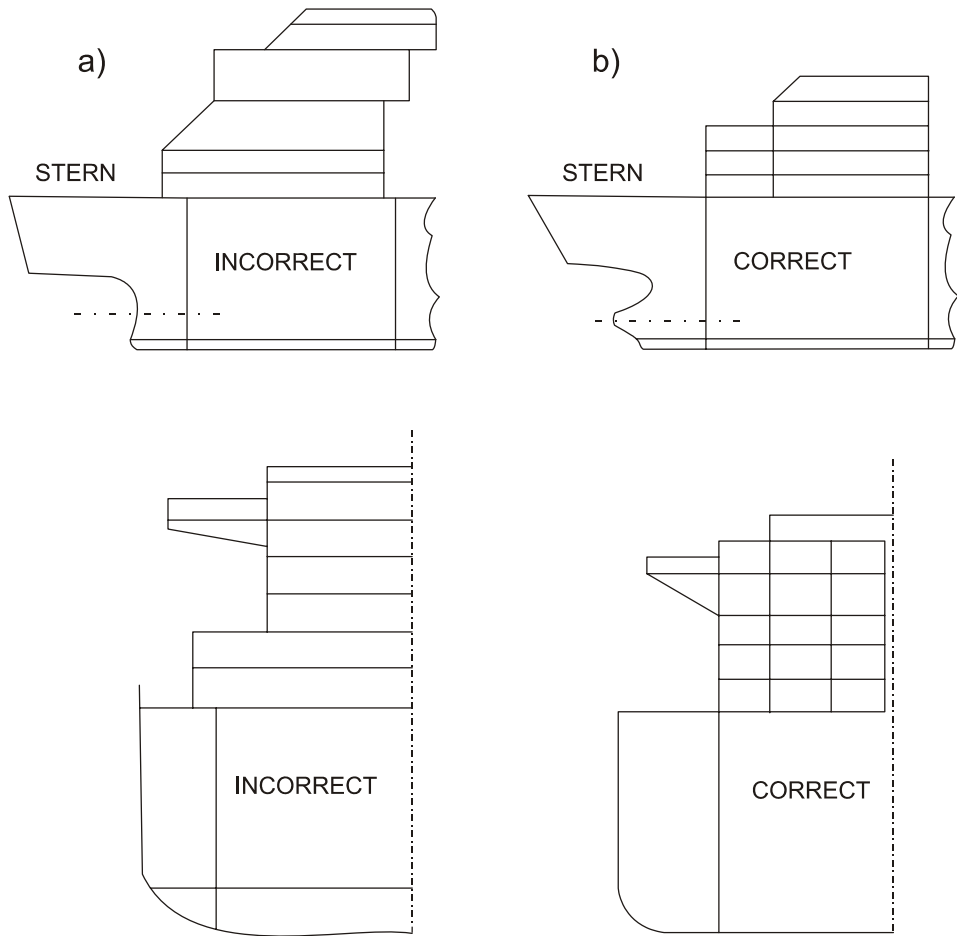


Fig. Z 8-1

The superstructure of version b) is low, compact, less flexible (because of longitudinal bulkheads, shorter bridge wings), and well fixed to the hull. The propeller aperture provides for a more uniform inflow to the propeller disc.

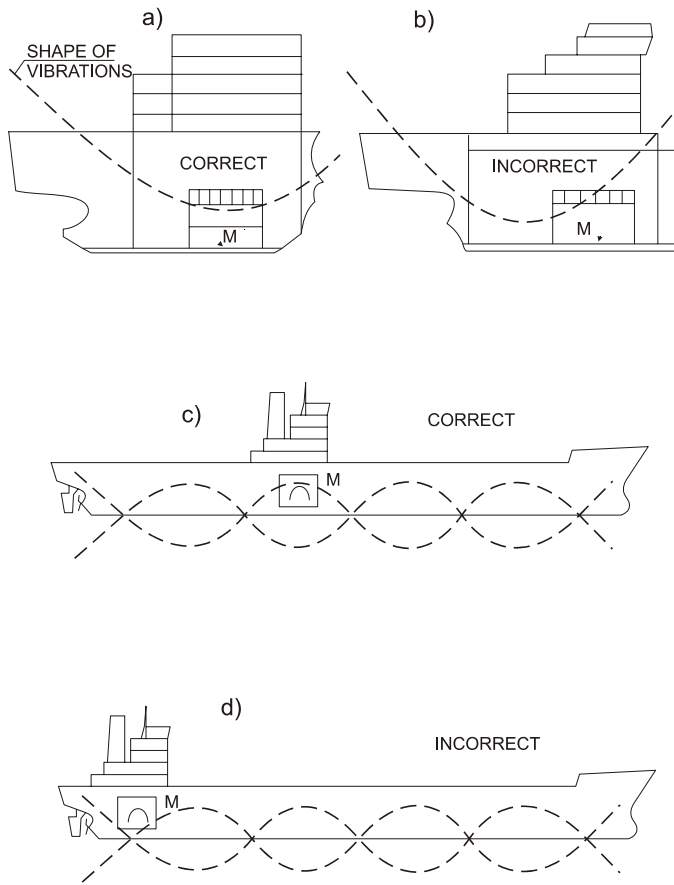


Fig. Z 8-2

Incorrect location of the superstructure (b, d) and main engine with regard to modes of hull-girder vibration. This is particularly important when the unbalanced moments, and exciting moments produced by main engine are significant.

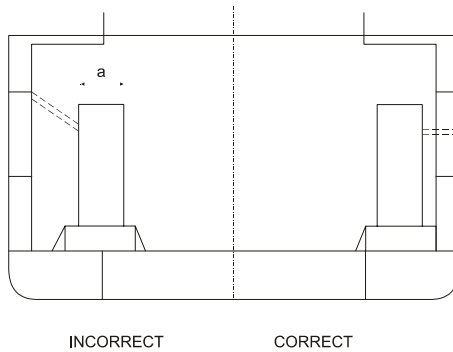


Fig. Z 8-3

Incorrect and correct stiffening of tank

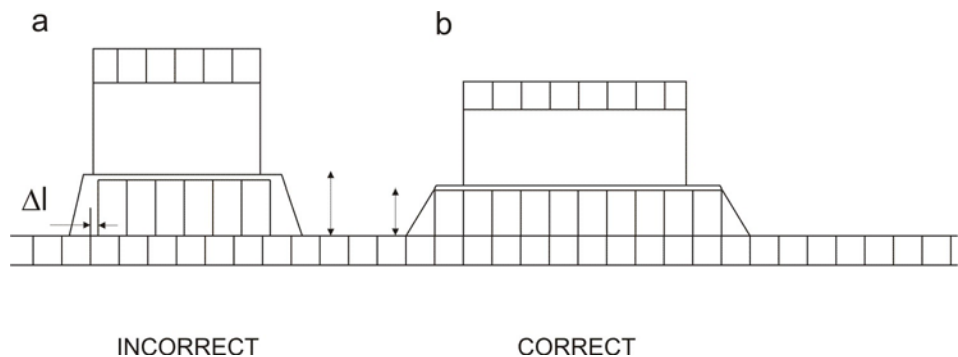


Fig. Z 8-4

Foundation bed of the auxiliary generating set should be possibly the lowest and correctly fastened to the bottom framing (stiffening)

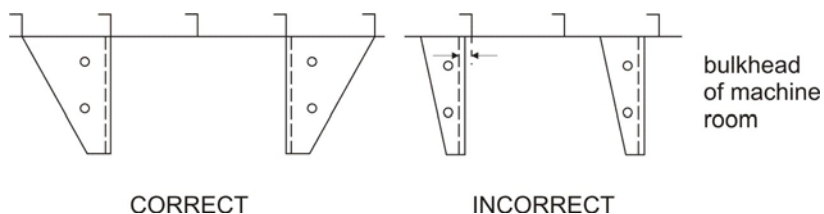


Fig. Z 8-5

Brackets fitted to bulkheads, walls and other elements of the hull to support the equipment, should be properly connected to their stiffeners to avoid the membrane effect.

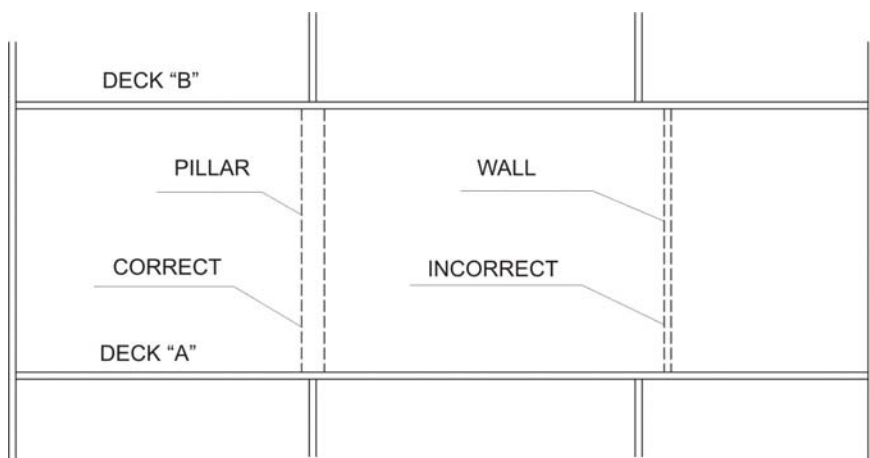


Fig. Z 8-6

Correct and incorrect stiffenings of the deck

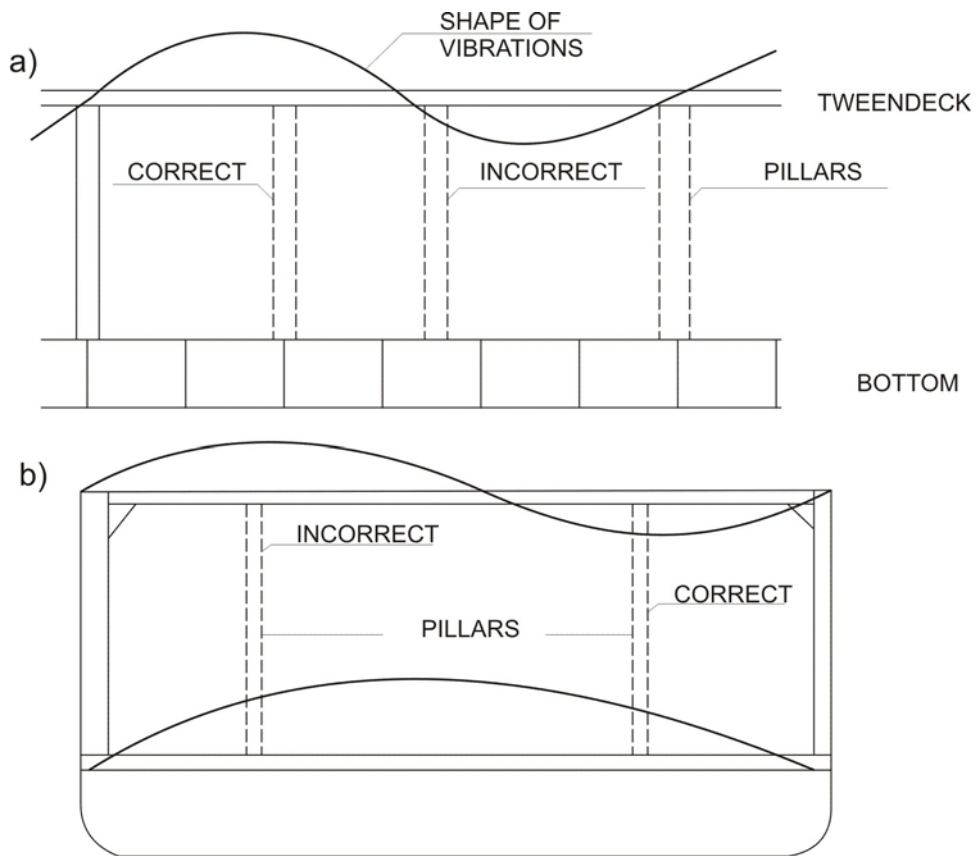


Fig. Z 8-7

Correct and incorrect stiffenings of:
 a) vibrating tweendeck longitudinal
 b) vibrating web frame

References:

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 (W. Ojak – How to control and approve the ship in order to have vibration off. Gdańsk, 1979)

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