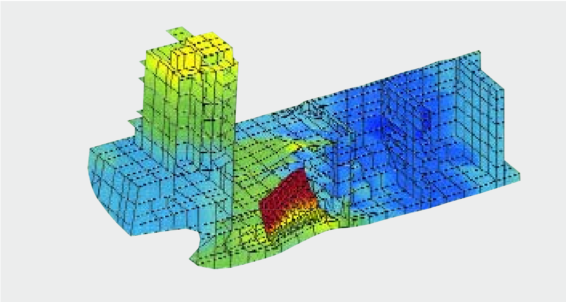
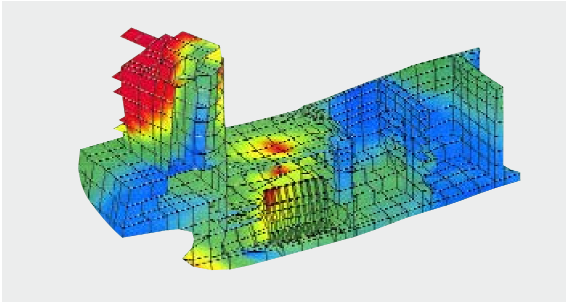
GL Technology

Ship Vibration

Ship Vibration

by

Iwer Asmussen / Wolfgang Menzel / Holger Mumm

Germanischer Lloyd

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This issue of GL's "Technology" contains the revision of an article which originally appeared in the German "Handbuch der Werften", published in 1998 by Schiffahrtsverlag "Hansa" C. Schroedter & Co., Hamburg.

The paper presents the "state of the art" of calculation and measurement techniques in the field of ship vibrations. In this respect, emphasis is put on the description of general procedures. Theoretical background is only explained when necessary for the comprehension of physical concepts. Specifically addressed are engineers/ inspectors at shipyards, shipping companies and consulting offices. The goal is to improve communication between specialists performing vibrational investigations and engineers concerned with the design and operation of ships.

Contents

1. Introduction

Despite considerable progress in the theoretical and experimental treatment of ship vibrations, questions about the accuracy of analysis methods for predicting the vibration behaviour as well as for solving vibration problems on completed ships are as topical as ever.

The aim of this article is to describe the “state of the art” in computational and measurement techniques. Here, the main emphasis is placed on the description of general approaches. Theoretical backgrounds are explained only if this is necessary for an understanding of physical situations. This paper is, therefore, aimed in particular at engineers and inspectors of shipyards, shipping companies and consulting offices, in the expectation that improved communication will be achieved between vibration specialists and engineers responsible for design and operation of ships.

In this context, “ship vibrations” consist exclusively of elastic vibrations of the ship’s hull and/or its parts. These vibrations can impair well-being, efficiency and the health of people on board, can cause damage to the ship and its cargo, and – in especially serious cases – can endanger the safety of the vessel.

The paper is structured as follows: after a discussion of questions concerning the building specification and standards for assessment of ship vibrations, analysis methods for calculation of free vibrations are dealt with. Here, various aspects of the determination of natural frequencies for simple components, large subsystems and entire ships are described.

After that, aspects of the calculation and assessment of the forced vibration level are dealt with, since in many cases a final evaluation of vibration questions in the design stage cannot be made with adequate certainty solely by comparing natural frequencies with main excitation frequencies.

Finally, this is followed by general remarks about the state of the art for experimental investigations and by a description of some vibration problems experienced on completed ships. The measurement procedure for diagnosis and actions taken to solve these problems are described in detail.

Here, it must also be pointed out that the subject of ship vibrations certainly cannot be dealt with completely and conclusively. It is our opinion that highly specialised questions – concerning, for example, elastic mounting of engines, sloshing phenomena in tanks or torsional vibrations of shafts – nevertheless lie outside the scope of this article, important as such questions undoubtedly are.

2. Standards for Assessment

In recent years, it has become standard practice to regulate vibration aspects for a newbuilding on a contractual basis. In the newbuilding contract, limit values that must not be exceeded during operation of the ship are defined as being part of the specification. The shipyard thus bears the responsibility for ensuring that limits agreed on with the shipping company are not exceeded or – if they are – for taking action with the aim of reducing the vibration level to the permissible value.

At the preliminary design stage or during the structural design phase, the shipyard will carry out adequate analyses or will have them performed by an independent consultant. Amongst other things, the scope of theoretical investigations regarded as neces-sary in a given case depends on the agreed limit values, the type of ship, the propulsion plant, and so on.

There are essentially three areas which are often included in the building specification to define vibration limits:

- Effect of vibrations on human beings
- Structural vibrations
- Vibrations of engines and equipment items

In the following, a few important standards that are often used to define limit values are dealt with briefly.

2.1 Effect of Vibrations on Human Beings

With regard to the effect of vibrations on human beings, it shoul basically be noted that existing standards are aimed solely at ensuring comfort and well-being. The word "habitability" is often used in this connection. If the recommended limits are not exceeded, damage to health is unlikely to occur.

2.1.1 ISO 6954

Internationally, the standard ISO 6954 (edition 1984) gained general acceptance for the evaluation of human exposure to vibrations [1]. An important feature of this standard was that, for the purposes of assessment, peak values of amplitudes had to be considered individually for each excitation frequency. Since the periodic excitation forces of the propulsion plant – especially of the propeller – are subject to a certain degree of variation, the vibration values, too, are measured with the corresponding variance. This fact led to problems in the assessment because the standard did not clearly define how a "mean" peak value (maximum repetitive value) was to be formed from values of differing magnitude determined over the duration of the measurement – constituting the well-known problem with the "crest factor".

This evaluation procedure contradicted the principles stipulated in the revised edition of ISO 2631-1 [2]. For this reason, too, a revision of ISO 6954 was initiated.

The new ISO standard was released in December 2000 [3].
It is now harmonised with the principles of ISO 2631-1 and integrates substantial improvements.

For assessment, a single value over the frequency range from 1 to 80 Hz is formed. As frequency weighting curve, the „combined curve" of ISO 2631-2 [4] is used, see Fig. 1.

The final vibration criterion, the so-called "overall frequency weighted r.m.s. value", is no longer a peak value. Hence, the term "crest factor" is no longer necessary and is thus completely cancelled. Three classifications for different kinds of spaces are combined with recommended vibration limits, now giving additional orientation to contractual partners, see Table 1.

NOTE: For guidance, Classification A can be passenger cabins,

Table 1 Overall frequency-weighted r.m.s. values from 1 Hz to 80 Hz given as guidelines for the habitability of different areas on a ship [3]

	Area classification					
	A		B		C	
	mm/s^2	mm/s	mm/s^2	mm/s	mm/s^2	mm/s
Values above which adverse comments are probable	143	4	214	6	286	8
Values below which adverse comments are not probable	71.5	2	107	3	143	4
NOTE The zone between upper and lower values reflects the shipboard vibration environment commonly experienced and accepted.						

Classification B crew accommodation areas, and Classification C working areas.

However, it is still up to owner and yard to exactly define the vibration comfort on board "their" vessel. Nevertheless, the new ISO 6954 is clearer and avoids misunderstandings due to deletion of the crest factor. Furthermore, it better reflects the human sensitivity by taking into account the entire spectrum from 1 to 80 Hz.

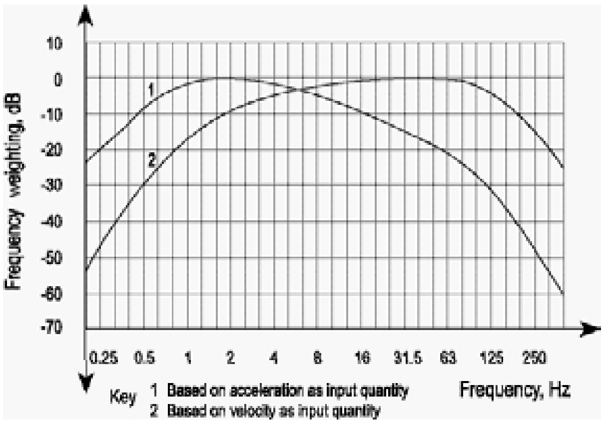


Fig.1: Frequency weighting curve, "combined curve" of ISO 2361-2

2.1.2 Class Notations

In terms of comfort on board, the rise of the cruise market in the last few years led to advanced developments by Classifi-cation Societies in this field. Notations of comfort, obviously, needed particular attention for passenger ships. The main objective is to support the owner/yard when detailing a newbuilding specifi-cation with regard to vibration and noise. Therefore, Classification Societies, with initiative from cruise ship owners, began to introduce vibration and noise as voluntary class notations.

The respective GL class notation is called "Harmony Class" [5]. It is focused on noise and vibration criteria on-board passenger vessels in a first step and will be followed by additional criteria for other types of ships. The comfort is scaled according to harmony criteria numbers, hcn 1 to 5, where 1 represents an extraordinary comfort (most ambitious level). The Rules do not only comprise limits and assessment procedures for the normal (seagoing) service con-dition, but account for thruster operation and harbour mode as well. Moreover, "acoustic privacy" is introduced as an additional noise criterion, reflecting both sound insulation and impact sound insu-lation of cabins to adjacent spaces.

Germanischer Lloyd was the first Classification Society to base the vibration part on the new ISO 6954 standard.

For illustration, Table 2 displays the vibration limits for passenger spaces.

The class notation requires a detailed documentation of plans and drawings to be submitted by the building yard. On this basis the Survey Programs, describing the extent of vibration (and noise) measurements for different criteria and operation modes, are checked and finally approved. The measurements cover a variable but relatively high percentage of the various kinds of spaces and areas of the ship.

The measurements of each space investigated are documented in the Survey Report and finally condensed to an hcn number, that is – as the final result – certified in the class notation.

Generally, the Rules are detailed, leading the user through different technical aspects of noise and vibration on-board passenger ships. Hints to potentially critical areas are given. A separate section is dedicated to the different theoretical analyses (FEM-based, for instance), recommended to achieve the hcn number desired.

Table 2 Vibration limits, passenger spaces

Vibration limits	Sea mode					Harbour operation					Thruster operation [1])				
	hcn					hcn					hcn				
	1	2	3	4	5	1	2	3	4	5	1	2	3	4	5
Indoor spaces forward of frame B															
First-class cabins	0.8	1.2	1.6	2.0	2.4	-	-	-	-	-	1.6	2.0	2.4	2.8	3.2
Standard cabins	1.2	1.7	2.2	2.7	3.2	-	-	-	-	-	2.0	2.4	2.8	3.2	3.6
Public spaces, *short* exposure time	2.0	2.5	3.0	3.5	4.0	-	-	-	-	-	-	-	-	-	-
Public spaces, *long* exposure time	1.4	1.9	2.4	2.9	3.4	-	-	-	-	-	-	-	-	-	-
Outdoor spaces forward of frame B															
Open deck recreation	2.0	2.4	2.8	3.2	3.6	-	-	-	-	-	-	-	-	-	-
Open deck recreation, overhangs	2.2	2.6	3.0	3.4	3.8	-	-	-	-	-	-	-	-	-	-

[1]) Thrusters operating at not less than 70% full load

2.2 Structural Vibrations

Even if the limits of human exposure to vibrations are not exceeded in the accommodation area of a ship, vibration problems can nevertheless occur in other areas in which these limit values do not apply. Typical examples are tank structures or other local components in the aftbody of the ship and in the engine room.

Therefore, in the case of resonance or near-resonance, considerable dynamic magnification relative to the edge supports is possible. The risk of damage resulting from inadequate fatigue strength is then particularly high. Because of the many factors that influence the fatigue strength, such as

- material
- structural details (stress concentrations)
- vibration mode
- welding processes applied
- production methods employed and
- environment (corrosive media)

the bandwidth for the possibility of occurrence of cracks is large, see Fig. 2.

This figure shows two limit curves, derived from a large number of measurements, that can be used as a guide in assessing the risk of cracks in local structures as a consequence of vibration. The amplitudes are peak values.

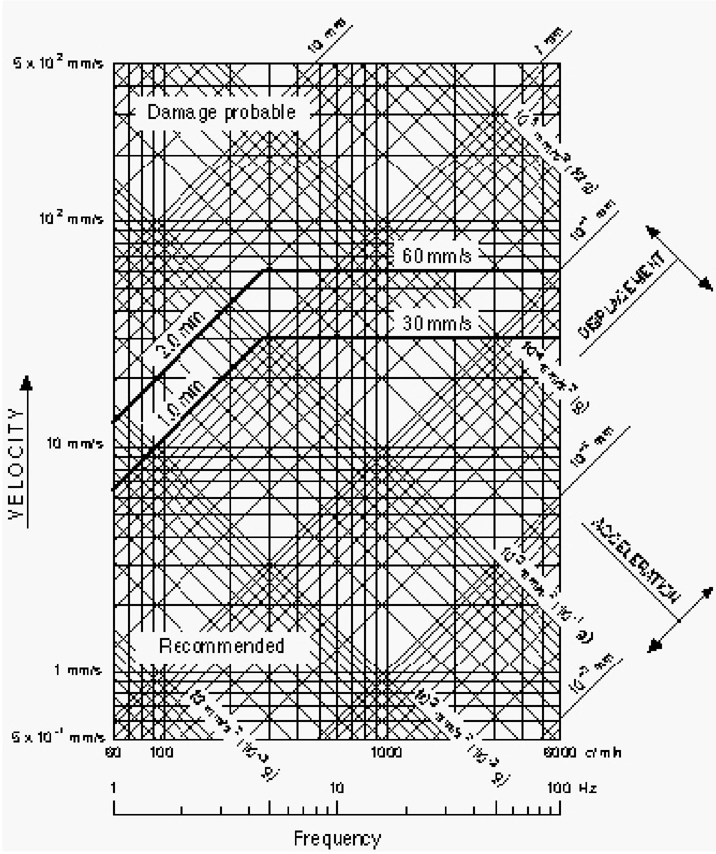


Fig. 2: **Assessment diagram for vibration of structures**

2.3 Engine and Equipment Vibrations

In past years, several standards dealing with engine vibrations were replaced. In this connection, the well-known standards VDI 2056 and 2063, as well as ISO 2372, 2373 and 3945, have been discontinued. The series ISO 7919 and 10816, [6] and [7], which also cover – among other things – the scope of the above-mentioned discontinued standards, were revised and are widely used today. These series are now also published in Germany as DIN ISO standards.

Germanischer Lloyd also published corresponding vibration limits in its Rules [8]. Here, values are quoted which, to avoid premature failure or malfunctions of components, must not be exceeded by engines, equipment items or peripheral devices. Fig. 3 shows the evaluation curves from [8].

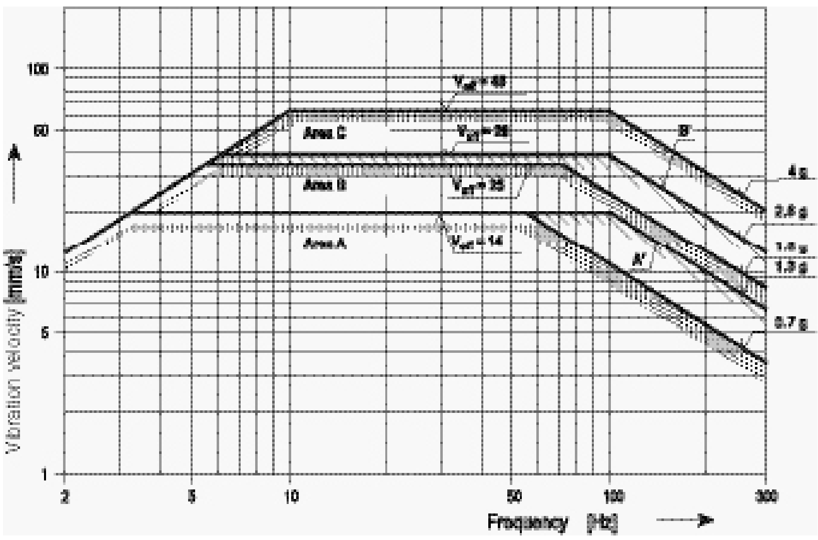


Fig. 3: **Assessment diagram for engine vibrations**

In general, limit curve A can be applied to assess vibration levels regarding machinery items. The criteria for use of curves A‘, B, B‘ and C are described in [8] and will not be repeated here. These criteria mainly involve reciprocating engines with peripheral devices connected to them.

In addition, limit curve B can also be used to assess equipment and components installed in steering gear rooms or bow thruster compartments.

Difficulties often occur in the assessment of components situated on masts. However, as a rough guide, it can be assumed that damage to these components can largely be prevented if the vibration levels remain within area A.

3. Calculation of Natural Vibrations

Because low-cost building and operation aspects of a ship increasingly influence the design, vibration problems occur more frequently. The following design trends contributed to this:

- Light weight construction and, therefore, low values of stiffness and mass (low impedance)
- Arrangement of living and working quarters in the vicinity of the propeller and main engine to optimise stowage space or to achieve the largest possible deck openings of container ships
- High propulsion power to achieve high service speed
- Small tip clearance of the propeller to increase efficiency by having a large propeller diameter
- Use of fuel-efficient slow-running main engines

On the other hand, the consistent application of labour legislation rules and higher demand for living comfort underline the need to minimise the vibration level.

The simplest way to avoid vibrations is to prevent resonance conditions. This procedure is successful as long as natural frequencies and excitation frequencies can be regarded as being independent of environmental conditions. In questions of ship technology, this prerequisite frequently remains unfulfilled. For example, different filling states change the natural frequency of tank structures. The overall hull of the ship takes on different natural modes and natural frequencies for different loading conditions, or there might be variable excitation frequencies for propulsion plants having a variable speed. In many cases, however, a resonance-free design of structures and equipment items is possible for all service conditions. A subcritical or supercritical design can be selected. As shown in Fig. 4, a subcritical design must ensure that, considering a certain safety margin, all natural frequencies of the system are higher than the highest significant excitation frequency.

The dynamic magnification factor depends not only on the safety margin between excitation frequency and natural frequency, but also on the damping coefficient of the system.

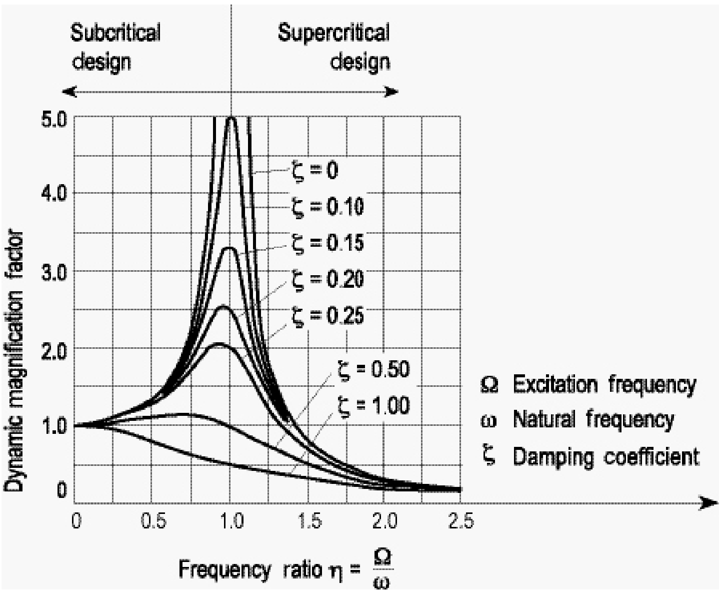


Fig. 4: Dynamic magnification factor for a single degree-of-freedom system

In Fig. 5, the vibration phenomena relevant in shipbuilding applications are plotted versus frequency. The frequency limits indicated are valid for standard designs and for normal ship types.

The transitions between ship motions, ship vibrations and ship acoustics are smooth. In the field of vibration, it is possible to distinguish between three different phenomena: global hull vibrations, vibrations of substructures and local vibrations.

In general, the higher the frequency, the greater the modal density, i.e., "the number of natural frequencies per Hertz". As a result, the system response in the higher frequency range is defined by the interaction of more natural modes than at low frequencies. In the transition to structure-borne noise, the mode density finally becomes so large that a frequency-selective analysis of the structure's dynamic behaviour requires an unacceptably large effort. One then has to make do with characteristic energy values averaged over frequency intervals (Statistical Energy Analysis, Noise-FEM, etc.). Today, of course, FEM is used to some extent in this frequency range, too. However, with the currently available power of computers, frequency-selective computation is limited to partial areas of particular interest, such as engine foundations. For example, an FE model intended for reliable computation of natural frequencies and natural modes of an engine foundation up to a frequency of about 200 Hz has about the same number of degrees of freedom as a complete hull model used to compute the natural vibrations up to 20 Hz.

3.1 Global Structures

Global vibrations in this context are vibrations of the ship's entire hull in the frequency range from about 0.5 to 10 Hz. Typical large substructures, such as the aft part of the ship, the deckhouse and the doublebottom, are coupled in a way that they cannot be considered isolated. Thanks to advances in computer technology, computation methods for determining global vibrations progressed rapidly during the past two decades. From today's point of view, classical approximation formulas or simple beam models for determining natural bending frequencies of a ship's hull are in many cases no longer adequate. For container ships with a high deck-opening ratio, e.g., for which coupled horizontal and torsional vibration modes play an important part, they do not offer the necessary degree of accuracy. In the past, one had to make do with beam models of a more complex type to cover shear and torsional stiffnesses of the ship's hull. However, in the meantime FE analyses using 3D models of the hull became the standard computational tool, as described in [9], for example.

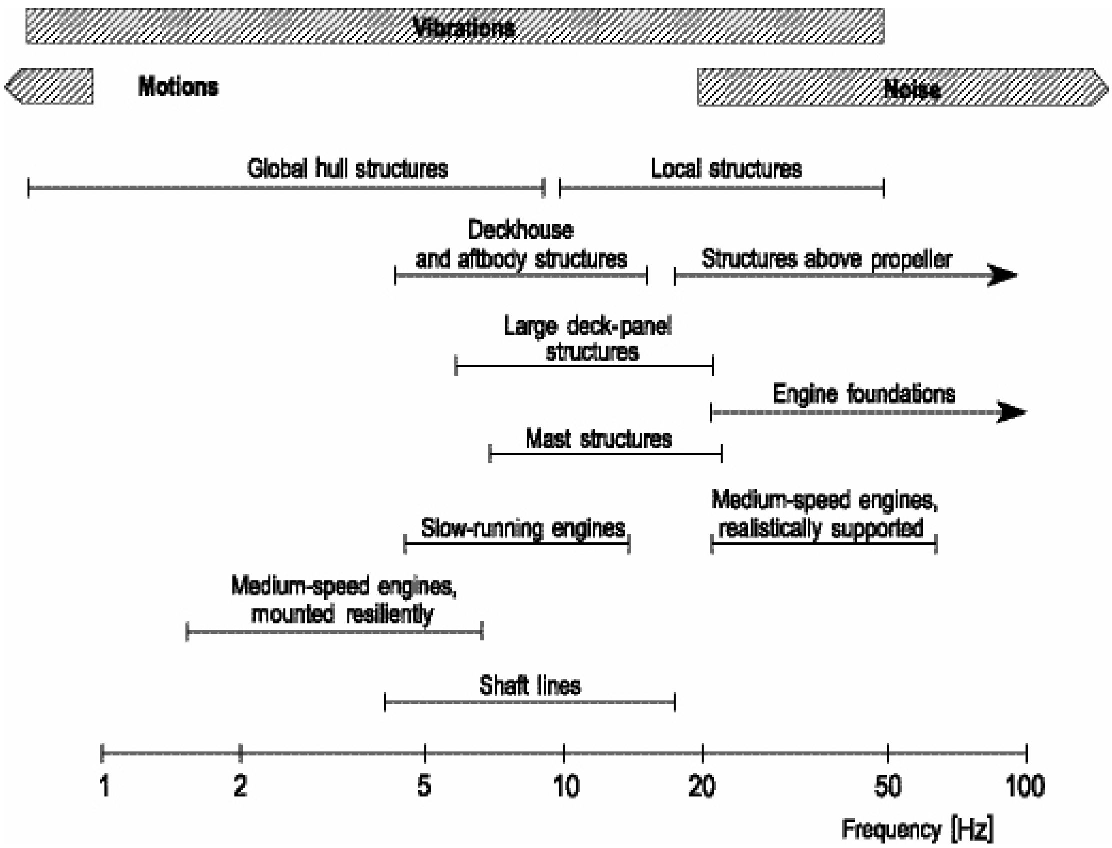


Fig. 5: Natural frequency ranges in shipbuilding applications

3.1.1 Modelling

The replication of a ship's structure in an FE model is generally the most laborious step of the analysis. For global vibrations, it turns out to be sufficient to represent primary structural components with the aid of plane stress elements. Bending stiffnesses of deck and wall girders are not covered by this type of modelling, since they are generally simulated by truss elements. Large web frames are taken into account by plane stress elements as well. For the sake of simplicity minor structural components lying outside the planes of the modelled sections are considered as additional element thicknesses or are ignored altogether. The division of the model is oriented relative to deck planes and to main longitudinal and transverse structures. The number of degrees of freedom is 20 to 40 thousand, yielding 50 to 150 natural vibration modes in the range up to 20 Hz. Three typical models are shown in Fig. 6, namely, a 700 TEU container ship, a smaller double-hull tanker, and a 4,500 TEU container ship.

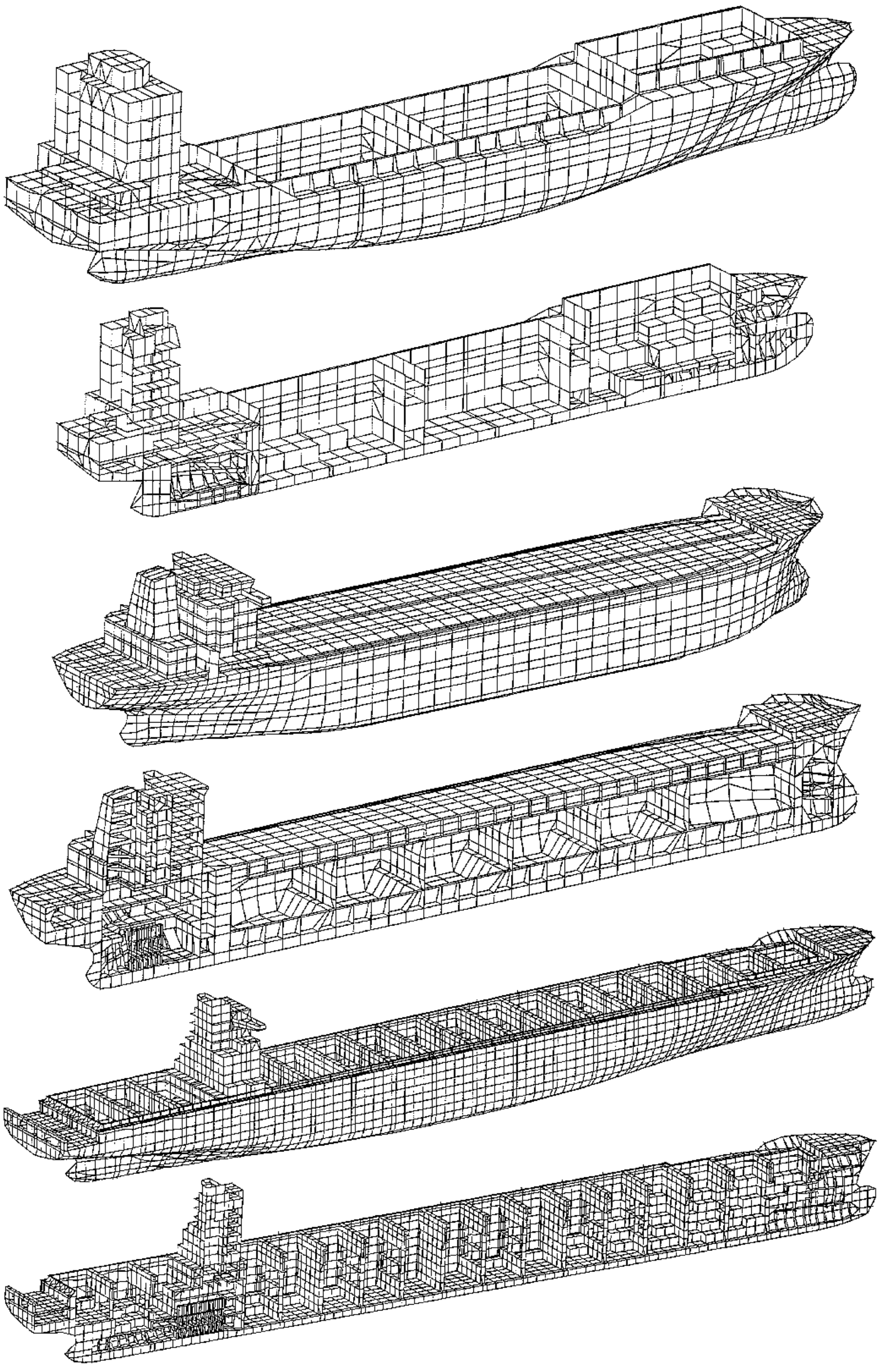


Fig. 6: FE models of various types of ships

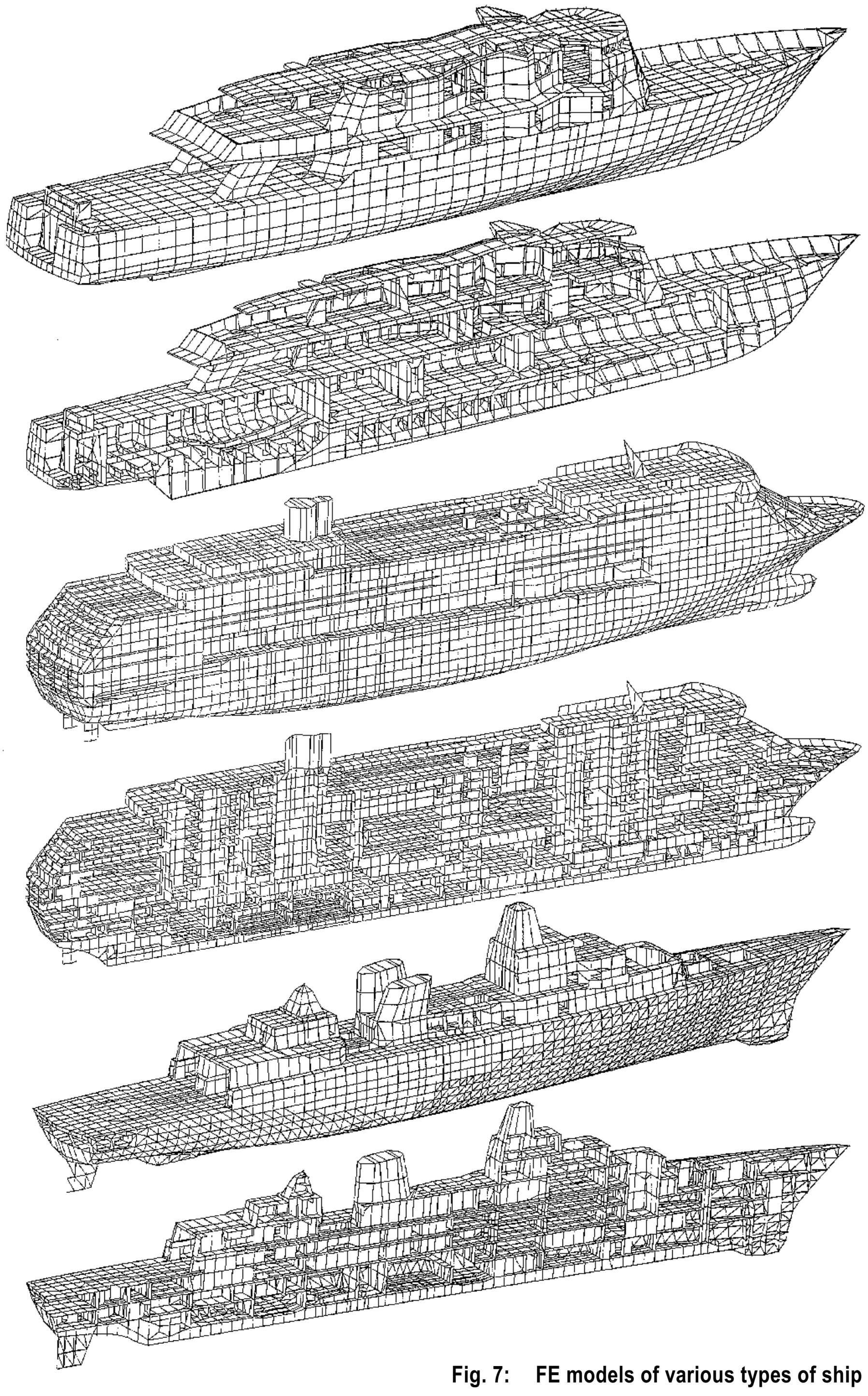


Fig. 7: FE models of various types of ship

In global vibration analyses, it is not necessary to model the middle and the forward part of the ship with the level of detail shown. However, the global models are mostly used for strength analyses, too, which require a more accurate modelling of the structure in these areas. If the bending stiffnesses of deck grillages are also to be included in the global model, the representation of transverse and longitudinal girders of decks is necessary, at least in the form of beam elements. Normally, these models possess 40 to 80 thousand degrees of freedom and have 300 to 500 natural frequencies in the range up to 20 Hz. An alternative for taking account of deck grillages in the form of beam elements is to model the webs of girders by means of plane stress elements and flanges by truss elements.

Fig. 7 shows three typical FE models of this kind in an overall view and a longitudinal section: a yacht approximately 60 m long, a 240 m passenger ship, and a frigate. As can be seen from the centre-line sections, webs of the deck grillages are modelled three-dimensionally in the case of the yacht only. For the other two much larger ships, this procedure would have led to unnecessarily large models.

In the computation of global vibrations of ships, it must be borne in mind that natural frequencies are highly dependent on the loading condition. From a draught variation of about ± 1.0 m upwards, it should be considered to take a further loading condition into ac-count. For cargo vessels, therefore, at least two or three mass distributions have to be considered. In contrast to strength analyses, no extreme cargo distributions should be selected, but rather homo-geneous ones typical for the expected ship operation.
The following masses must be taken into account:

- Ship structure
- Outfitting and equipment
- Tank filling
- Cargo
- Hydrodynamic masses

In FE techniques, a distinction is drawn between node masses and element masses. Node masses are concentrated at the respective nodal points of the FE model. This arrangement of masses is advisable for heavy parts of equipment whose centres of gravity are not automatically evident from the model geometry. The three types of masses last mentioned are likewise arranged as node masses. For the arrangement of structure masses, as well as for the "distributable" part of equipment masses, the existing geometric information of the FE model should be used (element masses).

The masses of tank contents are distributed over the nodes of the relevant tank structure, taking correct account of the centres of gravity. If nodes are available, the same applies to cargo masses. However, in many cases, for example for container masses, auxiliary structures must be provided to introduce masses into the FE model in a realistic manner. It must be ensured that these auxiliary structures do not unacceptably stiffen the ship's hull.

To determine hydrodynamic masses, separate computations must be performed. The procedures used are still often based on the method of Lewis [10], which involves a 2D theory derived for elongated, slim bodies. The associated set of potential-theory formulas is based on conformal mappings of a circular cross-section. The water flow in the ship's longitudinal direction is taken into account by correction factors that depend mainly on the length-to-width ratio, and also on the natural mode being considered. Because hydrodynamic masses have to be determined prior to the calculation of natural vibrations, the selection of correction factors should be co-ordinated with the expected frequency range of natural modes. Strictly speaking, it is possible to accurately determine only the natural frequency of the particular mode used as the basis to select correction factors.

The Lewis method offers the advantage that the hydrodynamic mass matrix to be used for the eigenvalue solution contains terms on the main diagonal only. Thus, the same numerically effective algorithms can be used for solving the eigenvalue problem as those used for problems involving only structural masses.

More comprehensive methods to calculate hydrodynamic inertia effects take account of the fact that the acceleration of a point on the wetted shell also causes changes in the hydrodynamic pressure at adjacent points. This coupling leads to the introduction of terms on the secondary diagonals of the mass matrix, which in turn leads to a considerably more effort-intensive calculation of eigenvalues. A calculation method that takes account of these couplings is described in [11]. Conversion into a practical computation method on the basis of a boundary value formulation is described in [12].

3.1.2 Calculation

If stiffness and mass matrices are known, natural vibration calculation can be performed. For this purpose, numerically effective approximation methods, such as the Ritz procedure, are used.
For the eigenvalue solver, starting vectors must be specified, the superimposition of which permits as accurate a representation as possible of expected vibration modes. However, only mode shapes can be calculated for which corresponding starting vectors have been specified.

As starting vectors the Lanczos method presented in [13] and [14], for instance, selects in an automated manner unit load cases that act in every degree of freedom of the system. This leads to the computation of all existing natural frequencies in the desired frequency interval. At present, the natural vibration analysis of a large global model takes several hours on a high-performance workstation.

To illustrate the situation, some typical fundamental natural vibration modes calculated for the previous FE models are shown in Fig. 8 and Fig. 9. In each case, the first torsional vibration mode and the second vertical bending vibration mode are presented together with the computed natural frequencies. Because of the large deck-opening ratio, the natural torsional frequencies for container ships are low. As a result of the comparatively short deckhouses there is no significant stiffening effect on the ship's hull.

For the other ship types presented, on the other hand, it can be assumed that the superstructures contribute considerably to hull stiffness.

Vibration modes of ship hulls lie in the lower frequency range. Because of the usual higher excitation frequencies their contribution to the vibration level is small. Nevertheless, knowledge of these vibration modes is important for validation purposes.

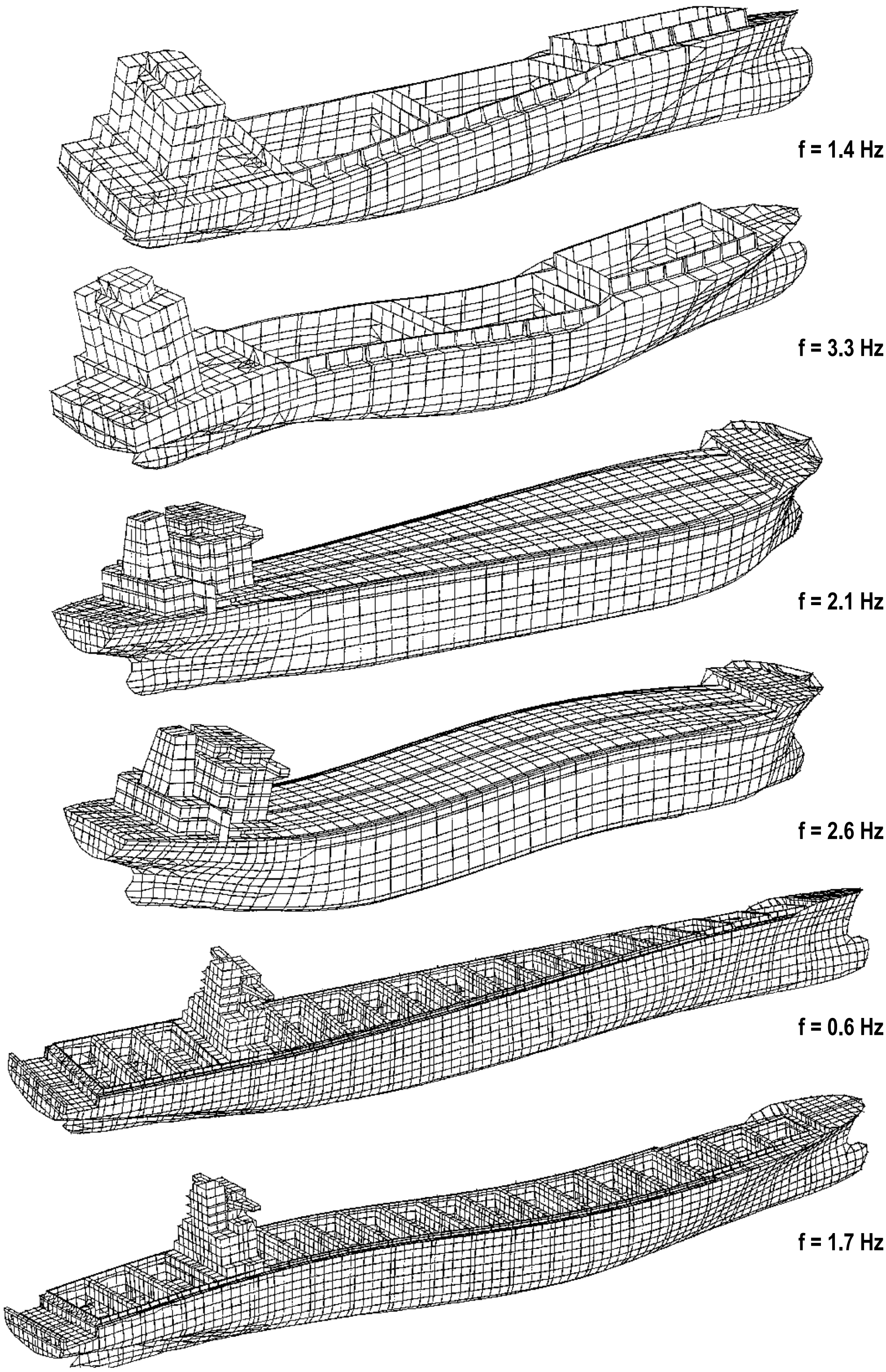


Fig. 8: Natural torsional and vertical bending modes of various ship types

3.2 Substructures

In the transition between global and local vibrations, vibrations of large subsystems, too, are of interest in practice. Here subsystems are structures or equipment items whose natural vibration characteristics can be regarded, for the sake of simplicity, as being independent of the vibration behaviour of the structure surrounding them – which is the case with a vibrating radar mast, for example. However, in the analysis of subsystems, the surrounding structure must not be ignored, because it defines the connecting stiffness, i.e. the supporting conditions.

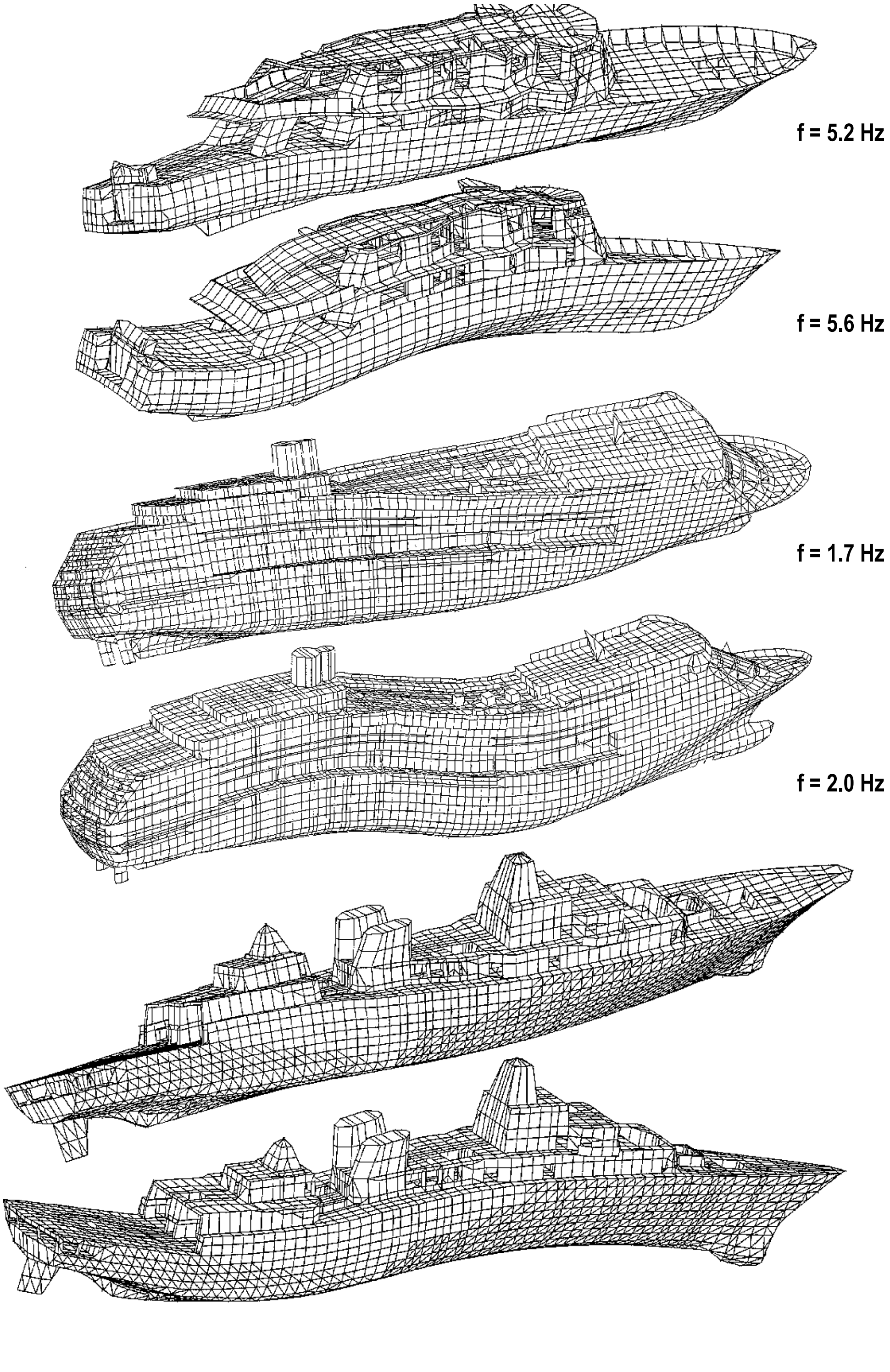


Fig. 9: Natural torsional and vertical bending modes of various ship types

3.2.1 Deckhouses

The aim of analyses of this type is the avoidance of resonance between fundamental vibration modes and main excitation frequencies. A typical example of a substructure is a deckhouse when considered as an isolated system. Fig. 10 shows such a model with the calculated fundamental vibration modes.

The longitudinal and transverse vibration modes, in particular, are significantly affected by the vertical stiffness in the supporting area. Therefore, an attempt must be made to incorporate, in a simplified manner, an appropriate part of the ship's hull in the region of the deckhouse into the model.

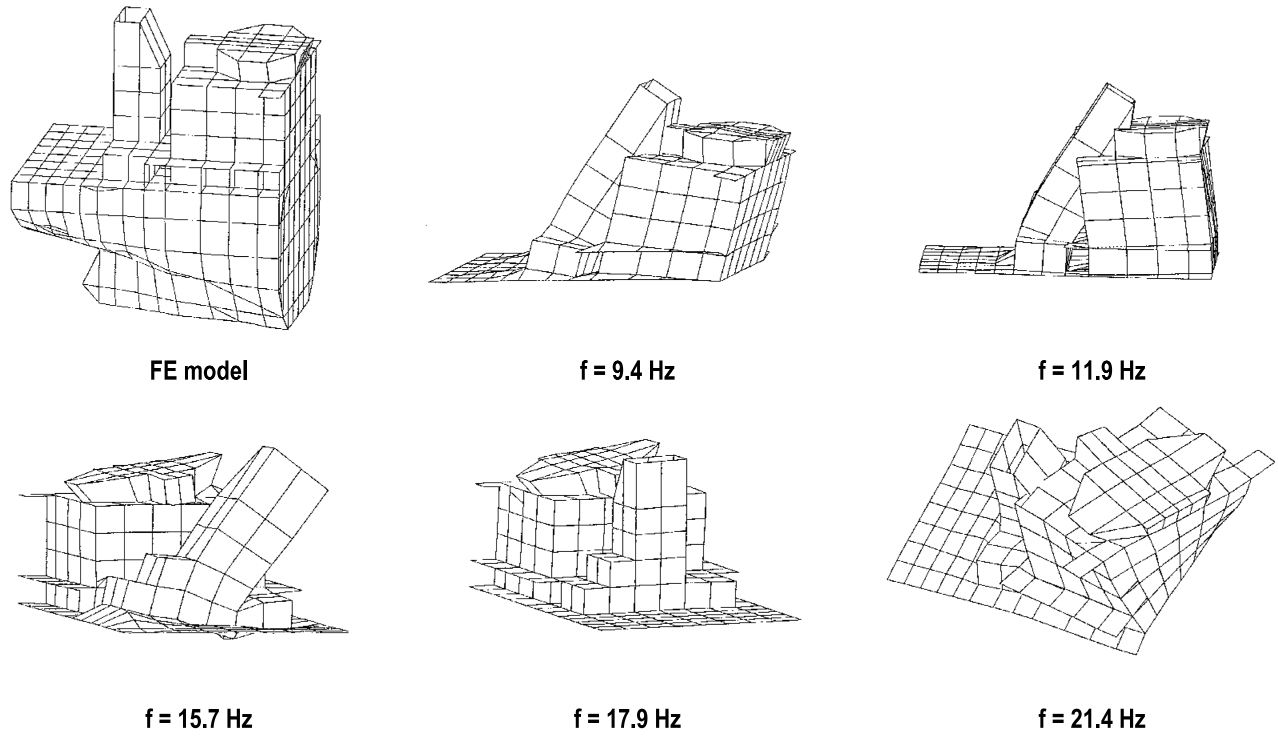


Fig. 10: Natural vibrations of a deckhouse

In this way, it is also possible to investigate the effect of design changes in the deckhouse foundation on the vibration behaviour. As can be seen from the natural vibration modes presented, the foundation is stiffly constructed. There are two coupled natural modes for longitudinal vibrations of the deckhouse and funnel. The low-frequency vibration is the in-phase vibration, whereas these subsystems in the following vibration mode vibrate in the anti-phase mode at 11.9 Hz. Because of the stiff foundation, the natural frequency is defined mainly by the shear stiffness of the deck-house. Global transverse vibration of the deckhouse does not exist in the frequency range considered. The supporting structure governs the vibrational behaviour of the funnel as well, leading to a natural frequency of 15.7 Hz for the transverse mode. Vibration of the upper region of the deckhouse occurs at 17.9 Hz. This natural frequency is defined mainly by the grillage stiffness of the bridge deck. The natural torsional vibration frequency is found to have a comparatively high value of 21.4 Hz because of the large external dimensions of the deckhouse.The design proves to be advantageous from the point of view of vibration because the basic recommendations had been adopted:

- Minimum possible height and maximum possible length and width of the deckhouse

- Stiffly designed foundations, especially the arrangement of bulkheads or wing bulkheads under the fore and aft bulkheads of the deckhouse (alternatively: support of longitudinal deckhouse walls on longitudinal bulkheads in the ship's hull)

- Maximising the longitudinal shear stiffness of the deckhouse by means of continuous longitudinal walls having as few and small cut-outs as possible

For container ships, in particular, the first two of these recommendations are often unachievable, since deckhouses are designed to be both short and tall to optimise stowage space. For the same reason, deckhouses are additionally often situated far aft, i.e. in the vicinity of the main sources of excitation. Thus, a risk of strong vibrations exists in many cases.

However, it is not possible to assess, on the basis of such models, whether resonance situations may lead to unacceptably high vibrations, since couplings with hull vibrations cannot be taken into account. Thus, for example, vertical vibrations of the aft part of a ship lead to longitudinal vibrations in the upper region of the deckhouse. These vibrations attain a significant level in many cases. This situation can only be investigated in a forced vibration analysis by taking account of stiffness and mass characteristics of the entire hull and by considering excitation forces realistically. It is not least due to this fact that an isolated consideration of deckhouses is increasingly giving way to complete global vibration analyses.

3.2.2 Masts

In the case of masts, there is a clear separation from the surrounding structure. Depending on the size and nature of the equipment fixed to a mast, four design principles can be distinguished:

- Simple masts whose cross-sections make them fairly flexible and which are stiffened by means of additional stays
- Welded tripod constructions
- Streamline-shaped masts with large, closed cross-sections and correspondingly high bending and torsional stiffness
- More complex beam type structures, which are mostly designed as latticework constructions made of tubular or MSH members

In the case of stayed masts, adequate stiffness of the connecting structural members on deck and of the corresponding foundation must be ensured. Stays should be provided with pre-tensioning devices and should form as small an angle as possible with the horizontal. In the case of both tripod and latticework designs, it turns out that the frequency depends not only on the height and the location of the centre of the mass, but also on the stiffness of the foundation at the footing. Mounting on deck areas supported by bulkheads is the best solution. Particularly in the case of masts mounted on the wheelhouse top, this requirement can be met only if communication between steel construction and equipment departments is well coordinated at an early stage.

In many cases, it is possible to position the mast on bulkheads of the stair casing or on pillars integrated in accommodation walls. Permissible vibrations are mostly defined by limit values for electronic equipment situated on mast platforms. These limits are not standardised, and at present they are mostly based on empirical values.

A mast vibrating in resonance can also act as a secondary source of excitation. As a result, deck coverings and partial walls can, in turn, experience excitation. This usually involves generation of noise.

Fig. 11 shows the possible extent of a computation model for a mast situated on a wheelhouse top. The FE model should continue at least down to a level one deck below the mounting deck. This is the only way to ensure that supporting conditions are taken into account realistically. Natural frequencies of longitudinal and transverse vibrations are 16.2 and 17.2 Hz, respectively. This means that a subcritical design with regard to a frequency twice that of the propeller blade frequency was ensured in this case.

As with any design aimed to avoid resonance, it is necessary to select – mainly in conjunction with distance from propeller and main engine – the order of excitation up to which there should be no resonances. In general, it turns out to be adequate to design fundamental vibration modes of the mast construction subcritically relative to twice the propeller frequency or to the ignition frequency.

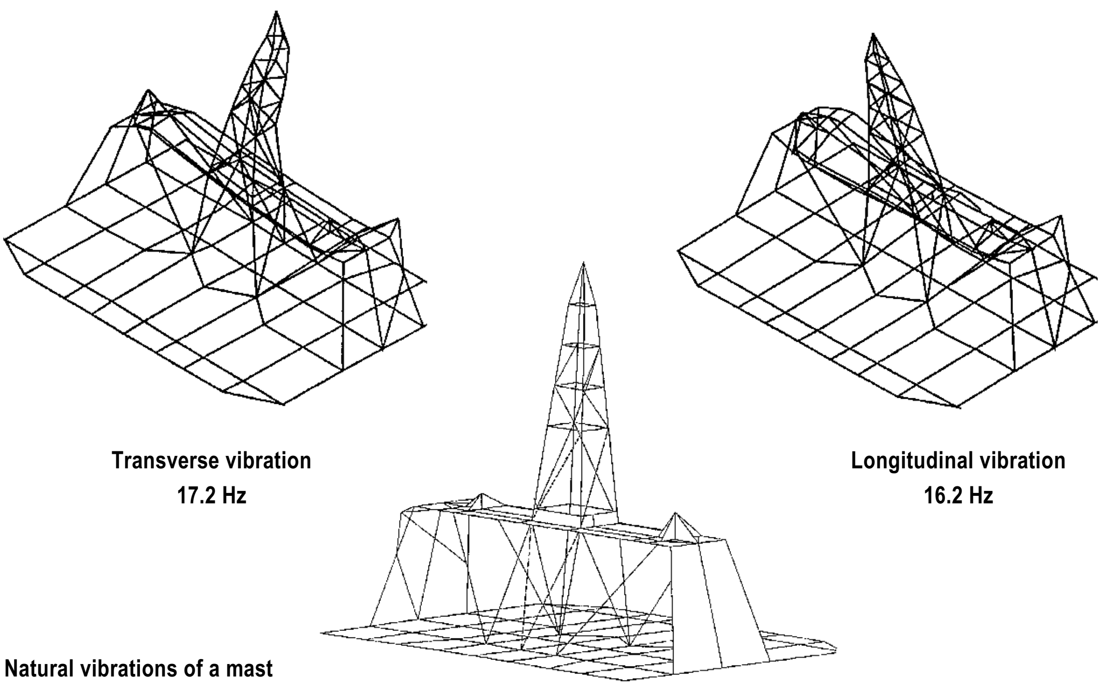


Fig. 11: Natural vibrations of a mast

3.2.3 Engine/Foundation Systems

Subsystems described so far refer to typical shipbuilding structures. In the following, the natural vibration of ships' main engines are described.

Fundamental natural frequencies of main engine vibrations depend on the distribution of stiffnesses and masses of the engine itself, but they are also determined to a large extent by the stiffness of adjoining structures. The effect of the doublebottom stiffness is more marked for slow-running engines than for medium-speed ones. Fig. 12 shows natural modes of a slow-running, rigidly mounted 7-cylinder engine, compared to those of the engine supported realistically in the ship. Furthermore, corresponding natural frequencies are given for an infinitely rigid engine structure supported on a realistic ship foundation. The global stiffness of the engine housing is represented in a simplified form by means of plane stress elements.

Fundamental vibration modes of housings – called "H", "X" and "L" modes – depend mainly on the doublebottom stiffness. Since doublebottom designs for slow-running main engines do not differ significantly, bands for the probable natural frequencies can be derived for engines having a certain number of cylinders, see [15].

For slow-running engines resonance situations can be experienced for all three fundamental modes, with typical combinations of number of cylinders and speed.

In the case of medium-speed engines this is true only for the H-type vibration mode, which might be in resonance with the ignition frequency. Corresponding computation models should contain at least the doublebottom structure in the engine room area and the structure up to the next deck. However, the engine housing, too, must be included in the model. Because the effect of the engine's frame stiffness is more marked for medium-speed than for slow-running engines, the engine structure must be simulated with greater accuracy – see also [16]. A computation model with a typical level of detail of engine and ship structure is presented in Fig. 13.
This shows the port half of the engine room area of a RoRo trailer ferry powered by two 7-cylinder, 4,400 kW main engines driving two propellers.

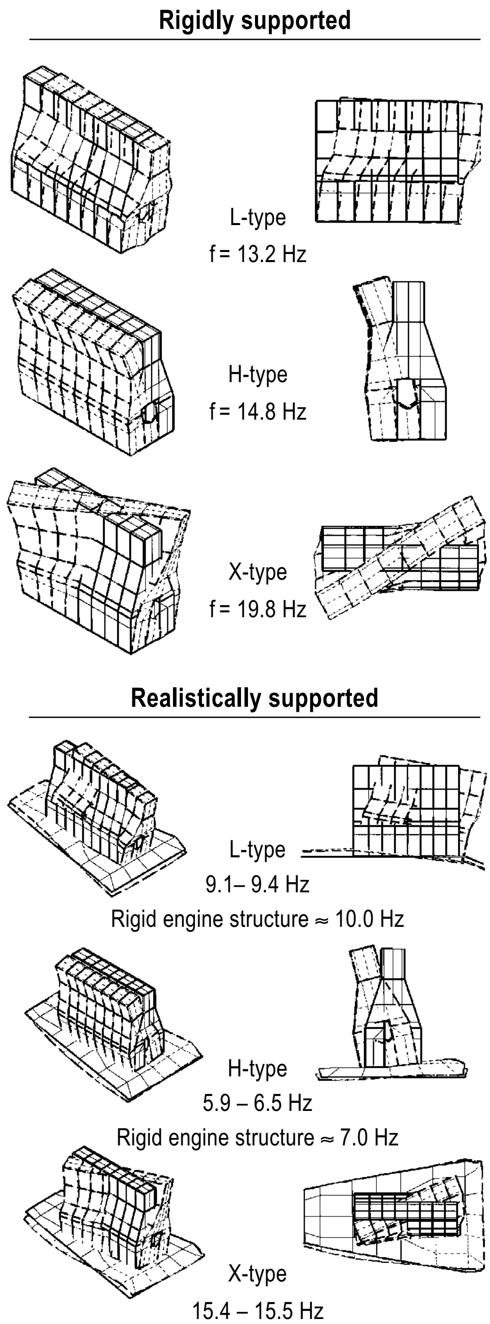


Fig. 12: Natural vibrations of slow-running main engines for various boundary conditions

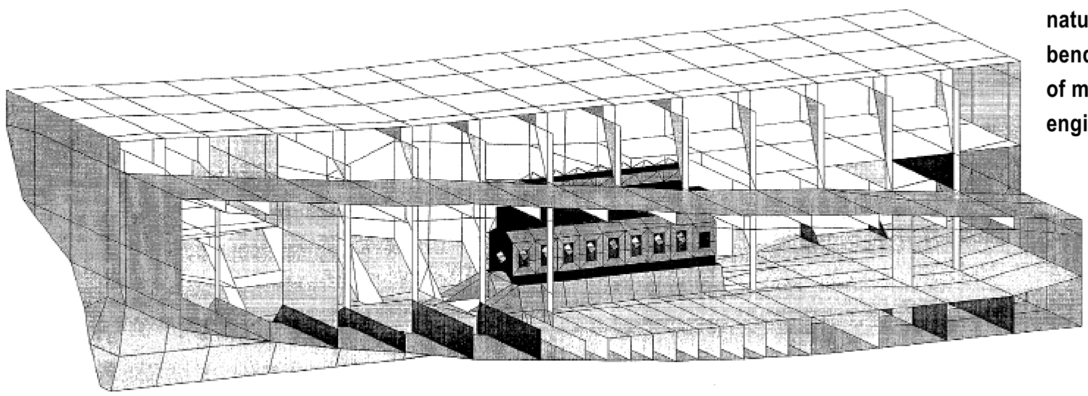


Fig. 13: Computation model for determining the natural transverse bending frequencies of medium-speed engines

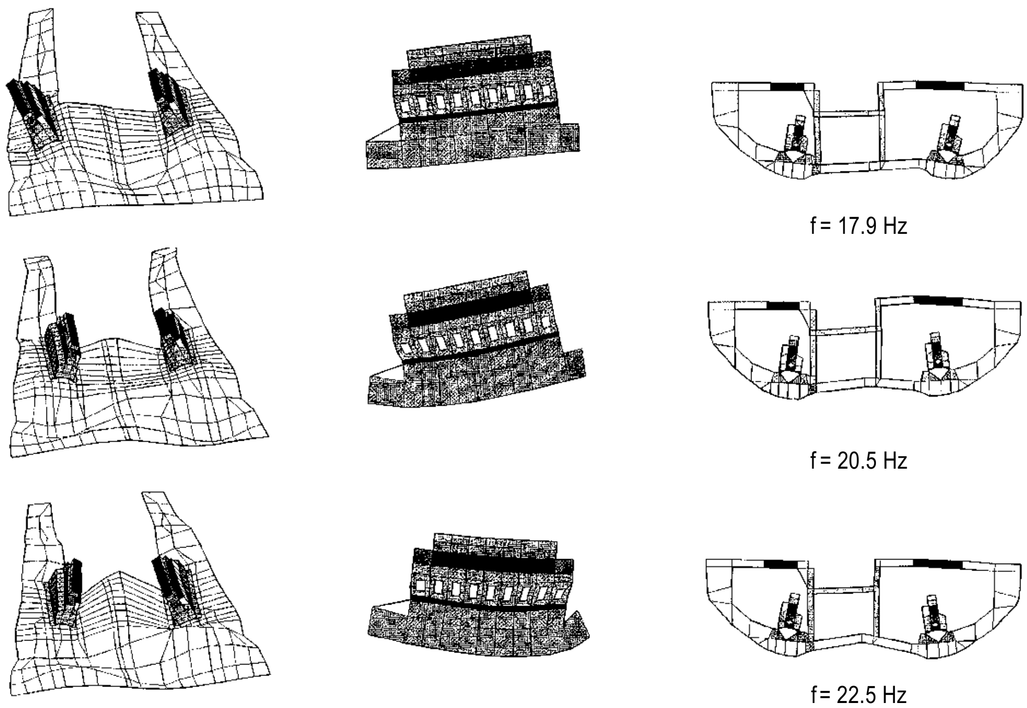


Fig. 14: H-type natural vibration modes of two 7-cylinder engines on a RoRo trailer ferry

Because the transverse members in the ship's aftbody, which tapers off in a catamaran-like manner, are not very stiff, the task was to check the risk of resonance between transverse modes of the engines and the ignition frequency. Because the H-moment also leads to vertical vibrations of the doublebottom, hydrodynamic masses act on the ship, which have to be considered. Large tank-fillings in the vicinity of the main engines are taken into account as well. For this example various natural frequencies were determined, reflecting coupled vibrations of the port and starboard engines.
Fig. 14 shows three corresponding vibration modes. Depending on coupling conditions of the port and starboard engines, H-type trans-verse vibration modes occur at 17.9, 20.5 and 22.5 Hz. The design was, therefore, supercritical relative to the ignition frequency of 30 Hz. Consequently, there was no need to install an elastic or semi-elastic mounting. Computations of X-type vibration modes of the main engines revealed frequencies in a band between 34 and 38 Hz, thus indicating an adequate safety margin to the ignition fre-quency as well.

3.2.4 Shaft Lines

Axial/Torsional Vibrations

As far as axial and torsional vibrations are concerned, it is often adequate to consider shaft lines isolated, i.e. independent of the surrounding structure of the ship. With regard to torsional vibrations, relevant requirements of the Classification Society have to be accounted for – see [8]. Axial vibrations are usually calculated by isolated models consisting of point masses, springs and damping elements.

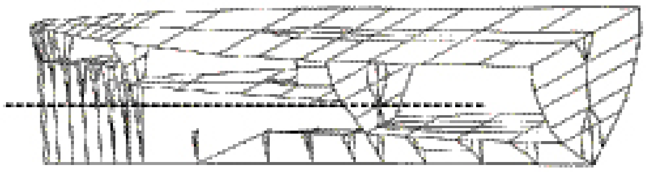
The same applies to the calculation of coupled torsional/axial vibrations. In practice, these turn out to be relevant only for shaft sys-tems driven by slow-running main engines.

A corresponding computation model includes both the entire shaft line and the crankshaft – see also [17]. If the axial/torsional vibration resonates with the thrust fluctuation of the propeller or with a radial force excitation of the main engine, comparatively strong axial force fluctuations can appear at the thrust bearing. These forces are further transmitted into the ship then acting as a secondary source of excitation. However, this will not be dealt with here.

Bending Vibrations

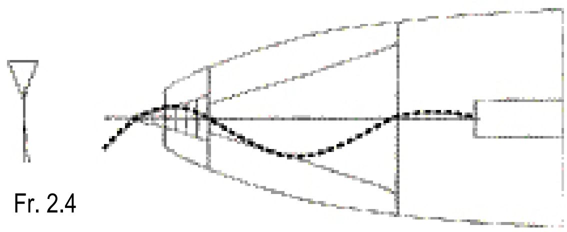
For the determination of natural frequencies of shaft bending vibrations, it is advisable to take account of the structure surrounding the shaft system. Detailed investigations should be considered for a shaft line design for which at least one of the following criteria apply:

- Soft structure in the vicinity of the stern tube bearing
- Guidance of the shaft in a shaft bossing, thus causing hydrodynamic masses to act
- Arrangement of shaft brackets, which themselves can have natural frequencies close to the propeller blade frequency
- Estimation of clearance between shaft and bearing shell as well as of dynamic bearing loads in a forced vibration analysis

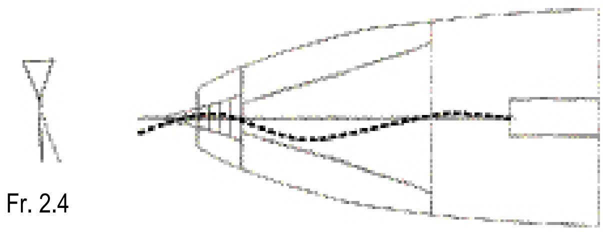


Shaft in aftbody

Shaft vertical bending vibration
(side view) at 11.0 Hz



Shaft transverse bending vibration at 8.6 Hz



Aftbody transverse vibration at 6.8 Hz

Fig. 15: Natural vibrations of a shaft system

An analysis example, where the first and last of these criteria apply, is given in Fig. 15. It involves a beam model of the shaft system integrated in a simple 3D model of the surrounding aftbody of a sailing vessel having a length of about 90 m. Distances between bearings are comparatively uniform in the range of 4.5 m. The oil film stiffness of slide bearings is an important parameter for the calculation. In most cases it turns out to be up to an order of magnitude smaller than the stiffness of the adjacent structure. As described in [18], the oil film stiffness depends on the shaft speed and the static loading of the bearings, among other things. It is also necessary to take account of the propeller's hydrodynamic mass moments, the magnitude of which can certainly equal the mass moments of the "dry" propeller.

Because of uncertainties in estimating the oil film stiffness and hydrodynamic masses, computation results concerning shaft bending vibrations always involve some degree of variance. Therefore, in many cases, it is advisable to perform parametric investigations varying the input data within the range of practical relevance. In the case described here, it turned out that the natural frequency of the vessel's basic aftbody vibration mode was lower than the fundamental natural frequency of the shaft system itself (6.8 as opposed to 8.6 Hz). It is obvious that shaft vibration modes couple with structural modes, since their natural frequencies are comparatively close together.

In the case concerned, the propeller shaft's vertical bending vi-bration mode, which is also shown, turned out to be the critical vibration mode, since its frequency was close to the propeller blade frequency (10.8 Hz). Although propulsion power was comparatively low, damage occurred in the aft stern tube bearing. Through an increase in the diameter of the propeller shaft, the relevant natural frequency was raised by about 4 Hz, resulting in an adequate safety margin relative to the propeller blade frequency.

3.3 Local Structures

Because of comparatively high natural frequencies of local ship structures, FE models for their calculation must be detailed. In particular, bending stiffnesses of local structures must be con-sidered as realistically as possible, in contrast to their representation in global computations. The aim of local vibration investigations is usually to limit vibration magnification relative to the global level. Thus, for example, vibration amplitudes at the centre of a deck grillage of an accommodation deck should not be much larger than at stiffly supported edges. This can be achieved only if freedom from resonance exists for all structural components of the deck.

In calculation practice, a distinction is drawn between vibrations of plate fields, stiffeners and panels (grillages) – see also Fig. 16. The amount of effort needed for the creation of FE models of such structures should not be underestimated. In spite of parameterised input possibilities and extensive graphic support, experience has shown that this type of analysis can hardly be carried out within the given time schedule. In addition, other important parameters of influence, such as rotational stiffnesses at plate field edges and effective mass distributions, also have to be taken into account here.

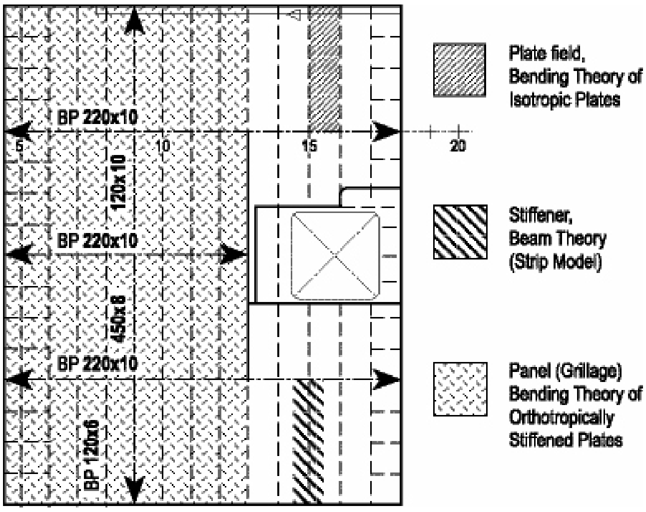


Fig. 16: Structural components in local vibration calculations

3.3.1 Calculation Methods

For practical calculation of natural frequencies of geometrically simple structures, it is most effective to use analytical approximation formulas, for example as mentioned in [19] and [20]. As long as assumptions for the derivation of these formulas are valid, results will be in good agreement with those achieved by more complex methods. The basic assumptions are:

- Freely rotatable, non-displaceable supporting conditions at edges
- Rectangular shape
- Regular arrangement of stiffeners
- No pillars or stanchions within the panel area
- Uniform distribution of added mass

If these preconditions are not fulfilled, the structures must be reproduced in an FE model. For these problems, the best cost/benefit ratio is certainly offered by beam grillage models. However, the effective width of the deck plating to be included in the section moment of inertia of beam elements depends on the vibration mode to be determined. Therefore, models of this quality are used only to determine basic vibration modes of deck panel structures.
If higher modes are to be included in the analysis, models with greater precision are required to simulate the stiffening effect in a three-dimensional form. In particular, webs of girders and stiffeners must be represented with the aid of membrane or shell elements and flanges by means of truss or beam elements.

The distribution of effective masses is often impossible to specify accurately. However, as verified in a large number of local vibration analyses, it is recommended to take an effective added mass of 40 kg/m² into account for decks in living and working spaces. Assuming freely rotatable edge conditions, this leads to adequate scantlings of local structures from a vibration point of view.

For walls, an added mass of 20 kg/m² should be chosen. For tank walls, hydrodynamic masses of tank fillings have, of course, to be considered.

There are a number of other parameters that influence natural frequencies of local structures, such as:

- Curvature of the structure
- Residual stresses of welds or distortions – see [21]
- Vibration behaviour of adjacent structures

Taking account of these imponderables, it becomes clear that the main aim is often not to predict natural frequencies of local structures with a high degree of accuracy, but rather to ensure a "basic stiffness" throughout the structure. Even in the case of resonances of vibration modes with higher excitation orders, this "basic stiffness" will ensure a moderate level.

In this connection, the concept of "critical lengths" for the design of plate fields and stiffeners turns out to be useful. The plate field and stiffener lengths that must not be exceeded are specified for the designer, based on relevant excitation frequencies. Critical lengths of plate fields can be determined in an iterative process, considering frame spacing, plate thickness and added mass. For the calculation of maximum stiffener lengths, the profile type is also used.

3.3.2 Design Criteria

Normally, an attempt is made to achieve a subcritical design of all structural components relative to the main excitation frequencies. Only structures situated in the vicinity of main excitation sources (propeller, main engine, bow thruster) are considered. In a first step, plate thicknesses and dimensions of stiffeners and girders are determined in the preliminary design phase in accordance with relevant Classification Rules. In particular, for accommodation decks with higher added masses and tank structures on which hydrodynamic masses act, vibration-related aspects often necessitate stronger dimensioning compared to Rule requirements.

The following recommendations for minimum natural frequencies of local structures can be stated as a guide:

$f_{natural} > 1.2$ x	twice the propeller blade frequency or main engine ignition frequency in the ship's aftbody, engine room and deckhouse area
$f_{natural} > 1.1$ x	four times the propeller blade frequency for the ship's shell structure directly above the propeller

For a subcritical design, the assumption of simply supported edges is conservative, since each constraining effect increases the safety margin between natural frequency and excitation frequency. Freely rotatable support can normally be assumed for plate fields. This assumption can, however, lead to considerable overdimensioning of stiffeners and girders connected via brackets to adjacent structures. Such brackets cause a certain clamping effect that, in turn, leads to an increased stiffness. To compensate for this, bracket connections are often accounted for in the design process by taking about 70–50 % of the actual bracket length as "effective" in the analysis.

In most cases, it is sufficient to design natural frequencies of structural components subcritically up to about 35 Hz. Any further increase of natural frequencies requires an unjustifiable amount of effort. A supercritical design or a "design in frequency windows" should be chosen with regard to higher dominant excitation frequencies.

Dimensioning principles stated above are fairly easy to put into practice in the case of cargo vessels. However, for passenger ships the design of local structures with natural frequencies above 20% of twice the propeller blade frequency is generally impossible to realise for weight reasons. In such cases, structures are designed "in the window" between blade frequency and twice that frequency, provided main engines are mounted elastically. In these cases the design frequency band is consequently small. Therefore, a larger amount of computation effort is required to ensure that natural frequencies are calculated with the necessary degree of accuracy. More effort has then to be spent on modelling boundary conditions, specific structural features, effective masses, etc. Designs aiming at less than the single blade frequency are inadvisable for reasons of lack of stiffness.

However, there is a strong interaction between local vibrations of structures and ship's acoustics. This relationship is manifested by the fact that a ship whose local structures have been consistently designed in respect to vibration also gains acoustic advantages.

3.3.3 Inclusion in the Design Process

The earlier the stage at which vibration-related aspects are included in the design, the simpler and better the solutions are. If the shipyard has no experience with this matter, an external expert should be consulted after completion of the general arrangement plan and after the propulsion plant has been fixed, i.e. when dominant excitation frequencies are known. Even at this early stage, it is advisable to introduce concepts of the stiffening pattern for decks and tank walls in the ship's aftbody and deckhouse area. The intermeshing with other design questions is another reason for examining the design from a vibration point of view as early as possible. One typical example is the choice of web heights of deck girders in the accommodation area. In practice, web heights that can be implemented are limited by restricted deck heights and by the need for adequate space under the flange plane of the deck grillage for routing of piping and cable runs. In these areas, web heights of 250 to 400 mm are aimed at, although this can lead to comparatively soft panel structures if supporting walls are far away. An alternative is the selection of high-webbed girders (600–800 mm), which offer sufficient margin to route piping and cables through adequately large cut-outs in webs.

The close relationship between vibration related questions and other design targets is illustrated by the example of a container ship that exhibited large vibrations on the bridge deck. The equipment numeral of a ship according to applicable Classification Rules depends on the closed wind-drag area of the deckhouse. To keep the equipment numeral low, the shipyard decided to make a break in the deckhouse front and aft bulkhead in the space under the bridge deck, as this space was not needed for living purposes, and to replace the bulkhead by an open beam structure. Thus, on the one hand, the desired aim of reducing the equipment numeral and, consequently, saving money in the purchase of, for example the anchor gear was achieved. On the other hand, however, this design variant also resulted in a reduced shear stiffness which, in turn, led to a high vibration level on the bridge deck.

3.3.4 Case Study

The importance of freedom from resonance and of a certain degree of stiffness for structural components is demonstrated exemplarily on damaged freshwater tanks of a container ship. Because of operational requirements, the tanks were moved one deck level higher compared to the original design. This retrospective measure had evidently not been checked with regard to vibration aspects. The stiffening system of the aft tank-bulkhead and the longitudinal wall is sketched in Fig. 17. In the entire frequency range around nominal speed (110–130 r/min), severe vibrations of the tank structures occurred. Measurements revealed vibration velocities about 30–50 mm/s at the centre of plate fields and 15–30 mm/s at stiffeners. Asymmetrical stiffener profiles exhibited vibration velocities of up to 30 mm/s in their flange plan. Depending on measurement location, engine speed and filling level, excitation frequencies were either the propeller blade frequency, twice that frequency (≈ 10 and 20 Hz) or the ignition frequency of the main engine (U14 Hz). A rough check indicated that natural frequencies of all structural components of the tanks lay in the range between 10 and 20 Hz.

Thus, it was no wonder that cracks shown in the sketch occurred after a comparatively short period of operation. After raising natural frequencies of all local structures to about 24 Hz, the problem was solved.

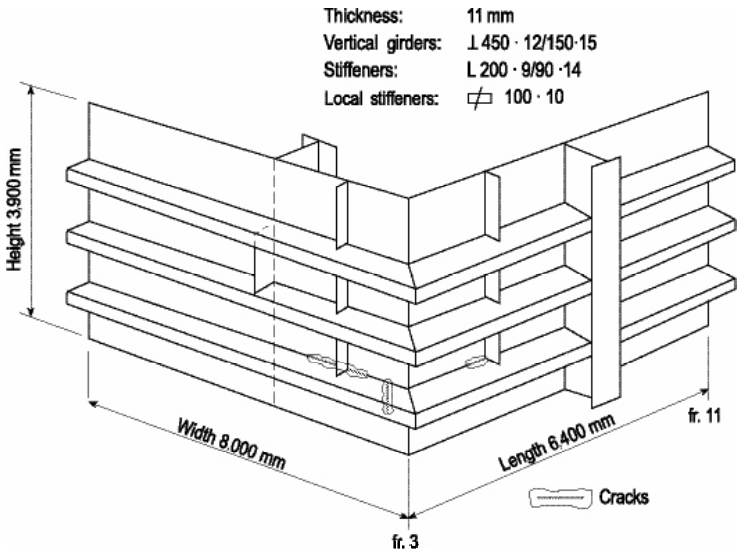


Fig. 17: Vibration damage to a freshwater tank in the aft part of a ship

4. Calculation of Forced Vibrations

The greater the mode density, the more difficult it becomes to apply the criterion of resonance avoidance. Computation of forced vibrations often turns out to be the only possibility of assessing on a rational basis the large number of natural frequencies. In addition to a realistic simulation of stiffness and mass characteristics of the structure, it is thus necessary to consider damping and excitation forces.

Ultimately, it must be proven during sea trials that maximum vibration velocities specified in the newbuilding contract are not exceeded. Therefore, a complete judgement of ship vibrations cannot be limited to an analysis of the free vibration problem, but must also give an insight into vibration amplitudes expected at critical points.

4.1 Computation Methods

A large number of FE programs provide various algorithms for solving the equation of motion of forced vibrations in MDOF ("Multiple Degree of Freedom") systems. Basically, a distinction has to be drawn between solutions in the time domain and those in the frequency domain. In ship structural applications, the solution in the time domain is confined to special cases, such as the analysis of the vibration decay of a ship's hull in the event of excitation by a slamming impact ("whipping"), for example.

Vibration questions in shipbuilding mainly involve types of excitation which are either harmonic or which are capable of being represented as a harmonic series and, therefore, can be distinguished by a characteristic frequency content.

Because of its extraordinarily high numerical effectiveness, the mode superposition method, [22] and [23], achieved great acceptance for calculation of the forced vibration level.

In this process, the first step is to determine natural vibrations of the structure in the frequency range of interest. Natural modes are then transformed and used as generalised, orthogonal coordinates. This procedure causes a decoupling of all degrees of freedom contained in the equation of motion. Due to this a reduction of the effort needed to solve the equation system is achieved. Thus, it is possible to compute the vibration level even for large systems over a wide range of frequencies at moderate cost.

4.2 Damping

Various physical mechanisms contribute to damping of vibrations on ships:

- Material damping
- Component damping, especially that produced by floor and deck coverings
- Cargo damping
- Hydrodynamic damping
- Mechanical damping (concentrated damping)

To characterise damping properties of a structure or a vibration absorber, a number of different parameters are used:

- Damping coefficient $b = \dfrac{\text{damping force}}{\text{vibration velocity}} \left[\dfrac{N}{m/s}\right]$
- Degree of damping ϑ or Lehr's damping coefficient D [-], where $\vartheta = \dfrac{b}{\sqrt{c \cdot m}}$ and c = stiffness, m = mass
- Logarithmic decrement Λ [-], where $\Lambda = \dfrac{2 \cdot \pi \cdot \vartheta}{\sqrt{1 - \vartheta^2}}$

In structural mechanics, Lehr's damping coefficient is normally used in the form of modal damping. It refers to individual natural vibration modes which, in the context of the mode superposition method, can each be thought of as an SDOF ("Single Degree of Freedom") system. Thus, in the case of MDOF systems, each natural vibration mode has a particular damping coefficient assigned to it. To illustrate the magnitude of damping, the logarithmic decrement can be calculated from the modal damping by means of the relation stated above. For a modal damping of 2%, for example, the value of Λ is 0.13. According to the definition of the logarithmic decrement, e^{Λ} corresponds to the amplitude ratio A_1/A_2 of two successive maxima in the vibration decay of a natural vibration mode excited by an impact force. From Λ = 0.13, it follows that A_1/A_2 = 1.13. Thus, the amplitude decreases by 13% with each vibration cycle.

Whereas material damping is easy to quantify (0.5–1.5%), component damping depends mainly on floor and deck coverings (4–10%). Cargo damping is heavily dependent on the nature of the cargo (container, fluid, bulk, etc.). Hydrodynamic damping is generally regarded as negligible in the frequency range of ship vibrations. Torsional and axial vibration damping devices of crankshafts as well as hydraulic units of transverse engine stays are examples of mechanical damping systems. In the literature, widely differing values are stated for damping characteristics in ship structural applications. There are clear descriptions in [24] and [25]. For the higher frequency range in particular, it turns out to be difficult to measure modal damping coefficients since the mode density is high and natural modes can, therefore, not be definitely identified by measurements. However, the damping grows with increasing fre-quency. For calculation of global vibrations, e.g. for container vessels, satisfactory results for the damping coefficient as a function of frequency were achieved with the approach outlined in Fig. 18.

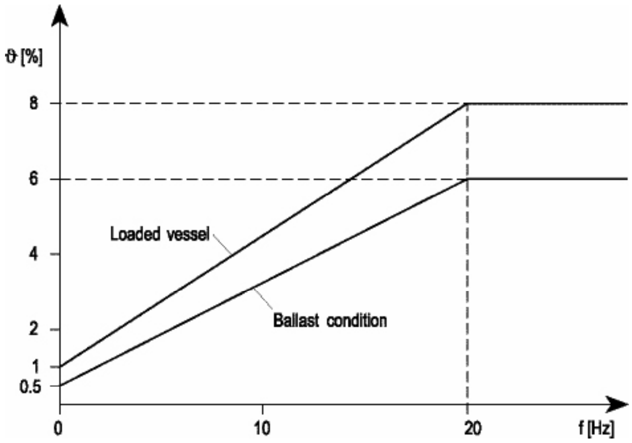


Fig. 18: Modal damping for global calculations of vibrations

4.3 Excitation Forces

In ship technology, with the exception of special problems (e.g. impact excitation), periodically varying excitation forces are of interest. If the excitation forces do not vary harmonically, they can usually be split into harmonic components (excitation orders) with the aid of a Fourier analysis.

Excitation forces are introduced into the ship's structure by a large number of machinery units:

- Main engines and auxiliary machinery: excitation frequencies are half and/or whole multiples of the frequencies of revolution
- Shaft machinery: excitation frequencies are equal to the frequency of revolution and, in the case of cardan shafts, also to twice that frequency
- Compressors: excitation frequencies are equal to the frequency of revolution and to twice that frequency
- Gearboxes: excitation frequencies are equal to frequencies of revolution and meshing
- Propellers: excitation frequencies are equal to the blade frequency and its multiples

In addition, there are some special types of vibration excitation, such as periodic flow-separation phenomenon at structural appendages or torque fluctuations in electric engines. In this paper, only the main sources of excitation will be dealt with, i.e. the excitation effects stemming from propeller and main engine.

4.3.1 Main Engine

Basically, the main engine of a ship introduces excitation forces into the foundation at all frequencies that are half and/or whole multiples (four-stroke and two-stroke engines, respectively) of its frequency of revolution. The waterfall diagrams shown in Fig. 19 depict the characteristic behaviour of various excitation orders during the start-up process of a six-cylinder two-stroke main engine. Measurement sensors were positioned on the engine bedplate as well as on the top plate of the innerbottom.

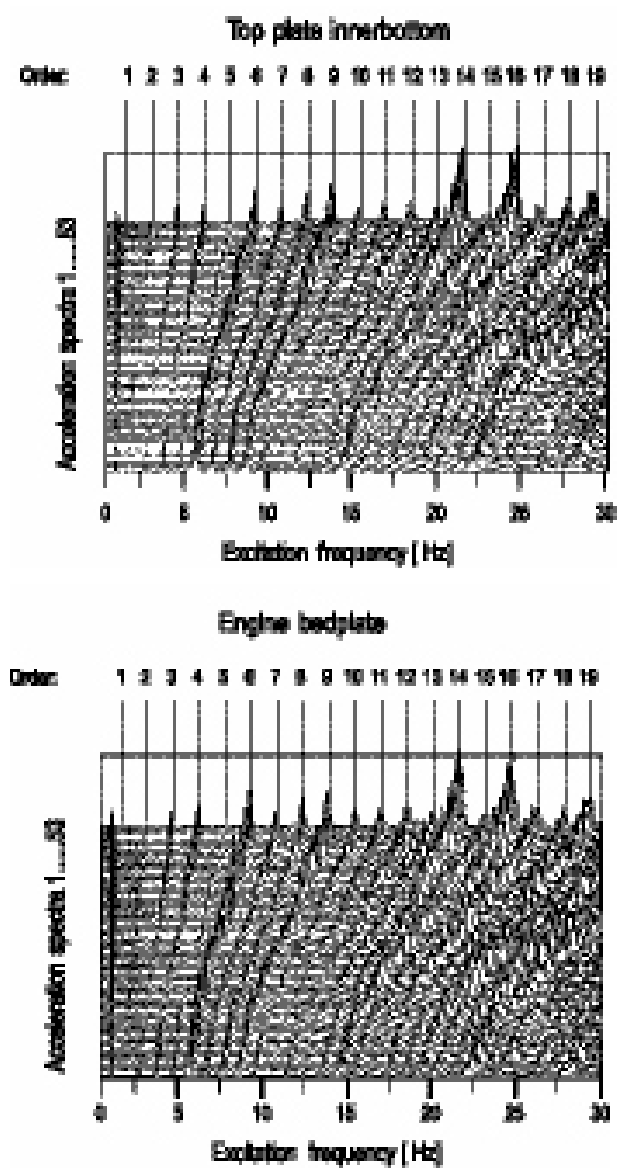


Fig. 19: Waterfall diagrams for a measuring point on the ship and for one on the engine

From the nearly identical behaviour of vertical accelerations at the measuring points on the engine's bedplate and on the top plate of the innerbottom, it was verified that they were stiffly connected to each other. The most significant exciters here are those of the 6th, 9th, 14th and 16th orders, but the other orders can also be recognised clearly. High vibration velocities can be expected only in those cases in which conspicuous excitation orders resonate with a natural vibration mode of the coupled system consisting of main engine and foundation.

Orders transmitting free forces or moments to the hull must in all cases be regarded as significant. Theoretically, internal orders of excitation do not transmit forces into the foundation, since the force components occurring in various cylinders – summed over all cylinders with phase relationships taken into account correctly – cancel one another. However, because the stiffness of the engine housing is finite, deformation-induced excitation forces, nevertheless, do penetrate to the outside. To estimate the part of the forces introduced into the foundation by internal orders of excitation, global stiffness characteristics of the engine housing must be taken into account. One possible approach is to integrate a simple FE model of the housing into the computation model of the ship and to simulate the forces directly at the place where they originate. In this way, it is possible not only to cover the proportion of the internal excitation forces that produces an effect externally, but also to take account of coupled natural vibration modes of foundation and engine housing (see Fig. 12). Normally, it is only in case of slow-running main engines that the ship's structure exhibits significant global vibrations caused by internal orders of excitation. Therefore, in computation practice, the engine structure of medium-speed and fast-running machines is not simulated for the purpose of considering excitation forces.

Forces in a single-cylinder engine

For the simulation of excitation forces generated by slow-running main engines of ships, forces occurring within one cylinder are taken as the starting point. As a result of combustion, gas forces are produced which cause the piston to perform translational movement. This movement is transformed by the driving gear into a rotational movement of the shaft. Thus, in addition to gas forces, both oscillating and rotating inertia forces are created.

Forces acting in a single-cylinder unit are illustrated in Fig. 20. For a given rotational speed, they can be calculated directly from the gas pressure characteristic by means of the formulas stated in [26]. The gas force acting on the piston in the vertical direction is obtained by multiplying the gas pressure with the piston area. To obtain the total vertical force Fz, the inertia force of the oscillating masses must be superimposed on the contribution made by the gas force.

Fz is transmitted via the piston rod, connecting rod and crankshaft into the main bearings where corresponding reaction forces act. The engine housing is thus subjected to a periodic change of compressive and tensile forces of considerable magnitude.

Owing to the oblique position of the connecting rod, a transverse force is created that affects the crosshead guide. Like the vertical force, it is transmitted via the engine housing into the main bearings where, in turn, the equilibrium forces act.

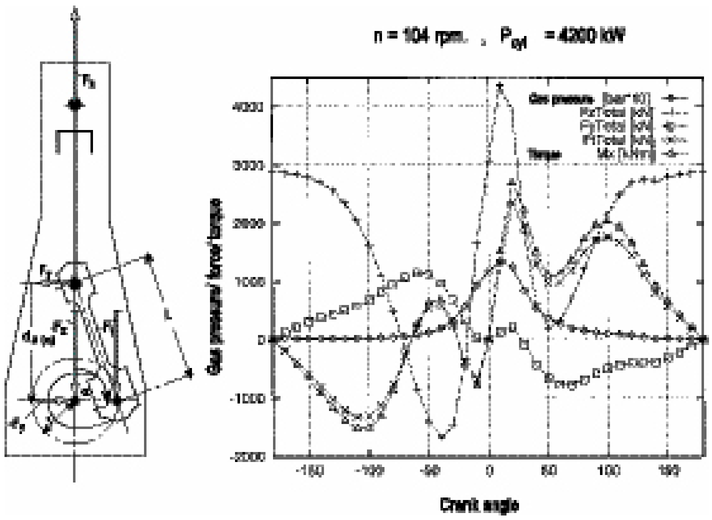


Fig. 20: Forces in the single-cylinder engine

Naturally, engine forces – formulated in Cartesian coordinates – can be converted to polar coordinates, so that the tangential and radial forces acting on the crank web pins are obtained.

The product of the transverse force and the current distance be-tween main bearing and crosshead gives the moment about the longitudinal axis of the engine. For reasons of equilibrium, this moment must equal (at ignition frequency) the torque generated by the tangential force.

The characteristic of various force components is shown in Fig. 20, taking an engine with a cylinder power of 4,200 kW and a revolution rate of 104 r/min as an example. The maximum vertical force is about 4,200 kN, whereas a force of about 1,100 kN acts in transverse direction.

In the next step, the force curves shown are transformed into the frequency domain by means of a Fourier analysis. If vibrations of the crankshaft or shaft line are to be considered, harmonic components of the tangential and radial forces are applied as sources of excitation. If, on the other hand, engine housing vibrations play the major role, vertical and transverse forces have to be considered.

The following table summarises harmonic components of vertical and transverse forces for the single-cylinder engine described above:

Table 3

Ord.	F_z [kN] From gas forces	F_z [kN] From osc. inertia forces	F_y [kN] Total force
1	2553	3575	508
2	1712	875	165
3	1133	0	37
4	736	31	144
5	414	0	101
6	237	0	54
7	133	0	37
8	63	0	23

The magnitude of excitation forces decreases with increasing order. Influence exerted by the oscillating masses exists for the first, second and fourth order of excitation.

The values quoted are valid for the revolution rate on which the calculation is based. For revolution rates varying linearly, both inertia forces and mean gas pressure change quadratically. However, this does not apply exactly to individual harmonic components of the gas pressure. In computation practice, this inaccuracy is tolerated in return for a considerable reduction of data input.

Application of the Total Excitation Forces

If forces acting in a single-cylinder unit are known for the individual orders, their phase relationship with other cylinder units can be calculated, considering the ignition sequence. In the next step, vertical and transverse forces are applied – with the correct phase relations – to relevant nodes of the FE model of the engine housing. Fig. 21 shows a schematical "snapshot" for a typical distribution of excitation forces of an engine housing. Vertical forces are assumed to be acting on the top of cylinder units and on main bearings, whereas transverse forces are acting at the centre of the crosshead guide and on main bearings. Another advantage of this procedure lies in the simple method of taking account of irregular ignition sequences and in the possibility of simulating ignition failures, for instance.

To judge whether excitation forces of the individual order are significant, forces acting in their planes of effect are added over the length of the engine, with the phase relationships being taken into account correctly. Let $Fy_{i,k}$ and $Fz_{i,k}$ be the complex force amplitudes of order "i" acting on cylinder "k", let "dx_k" be defined as the distance of cylinder "k" from the centre of the engine, and let "dz_k" be defined as the distance between main bearings and crosshead guide. Thus, the following quantities can be defined for individual orders "i" of excitation; see also [27].

The subscripts “Osc”, “Rot” and “Gas” indicate the physical cause of force effects (oscillating/rotating masses and gas forces).

- Vertical moment of inertia (external) $$M_{V_i} = \sum_{k=1}^{nk} F_{z-Osc\;i,k} \cdot dx_k$$
- Horizontal moment of inertia (external) $$M_{H_i} = \sum_{k=1}^{nk} F_{y-Rot\;i,k} \cdot dx_k$$
- H-type moment (external) $$M_{D_i} = \sum_{k=1}^{nk} F_{y-Gas\;i,k} \cdot dz_k$$
- X-type moment (internal) $$M_{X_i} = \sum_{k=1}^{nk} F_{y-Gas\;i,k} \cdot dx_k$$
- Pitching moment (internal) $$M_{P_i} = \sum_{k=1}^{nk} F_{z-Gas\;i,k} \cdot dx_k$$

Except for the pitching moment, all excitation parameters can be taken from the engine manufacturers’ standard catalogues. However, the danger of pronounced vibrations cannot be estimated on the basis of these values alone. It is also necessary to consider whether the shape of the force distribution also corresponds to a coupled engine and foundation vibration mode. Vertical moments of inertia and pitching moments mainly excite bending vibrations of the doublebottom in conjunction with an L-type vibration mode of the main engine. Horizontal moments of inertia and X-type moments cause torsional vibrations of the engine housing about the vertical axis. Although excitations mentioned last can cause high vibrations in the transverse direction at the top of the engine, they are normally of minor interest as far as excitation of the foundation is concerned. The recommendations of the manufac-turers to provide engines with a large number of cylinders with transverse bracings are mainly aimed at avoiding resonance with the X-type natural vibration modes. However, installation of trans-verse stays also causes an increase in the H-type natural frequency of the main engine. In the case of engines with six or seven cylinders and a revolution rate of about 100 r/min, transverse bracings may, in turn, result in unfavourable resonance situations with the H-type moment acting at ignition frequency.

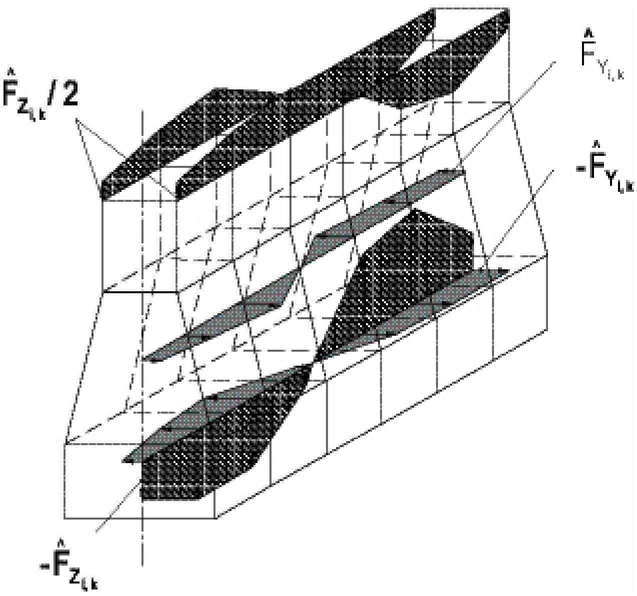


Fig. 21: Typical excitation force distribution over the engine frame of a ship’s slow-running main engine

The choice of number of cylinders for a ship’s main engine is not based primarily on vibration aspects. However, the following table provides an indication for slow-running engines having 5 to 9 cylinders and the usual ignition sequences, which orders of excitation may have a significant effect on global ship vibrations.

Table 4

Number of cylinders	Order of excitation								
	1st	2nd	3rd	4th	5th	6th	7th	8th	9th
5	⊗	⊗	⊗	–	⊗	–	⊗	–	–
6	–	⊗	⊗	⊗	–	⊗	–	–	⊗
7	⊗	⊗	⊗	⊗	–	–	⊗	–	–
8	–	–	⊗	⊗	⊗	–	–	⊗	–
9	⊗	⊗	⊗	⊗	⊗	⊗	–	–	⊗
⊗ Influence exists – Negligible influence									

Secondary sources of excitation, especially axial force fluctuations that result from vibrations of the crankshaft and shaft line, are not taken into account in this table, since no general statements can be made in this context.

4.3.2 Propeller

From the propeller, excitation forces are transmitted into the ship via the shaft line and in the form of pressure pulses acting on the ship’s shell. Whereas shaft line forces are the most significant factor for vibrations of shaft lines, the predominant factor for vibrations of ship structures are pressure fluctuations.

Shaft Line Forces

Fluctuating shaft line forces result from the non-uniform wake. The creation of these forces can briefly be described as follows: the relative velocity between the individual profile section of the pro-peller blade and the water depends on the superposition of the ship's speed and the peripheral velocity at the profile section under consideration. As a simplification, the influence of the wake can be considered as the change in the angle of attack at the profile section, this change being proportional to the inflow-speed variation. During each revolution, both the thrust and the tangential force on the individual blade behave irregularly. Since these forces act eccentrically at about 0.7 R, periodically fluctuating moments also occur. To obtain values for the overall forces and moments at the propeller, individual blade effects are superimposed, with the phase being taken into account correctly. The thrust fluctuation can be up to about 10% of the mean thrust, but it is usually between 2 and 4%. The force fluctuations in transverse and vertical directions are between about 1 and 2% of the mean thrust. In most cases, the moment fluctuation about the transverse axis is predominant, compared to fluctuations of the moment about the vertical axis and of the torque (5–20% of the mean torque, compared to 1–10%). As can be seen from the schematic diagram of the overall excitation in Fig. 22, forces may excite various modes of vibration. For computations of axial and torsional vibrations of the shaft line, fluctuations in thrust and torque must be taken into account. Bending vibrations of the shaft are influenced by transverse forces in horizontal and vertical directions as well as by bending moments about the corresponding axes.

In contrast to pressure fluctuations transmitted into the hull via the shell, fluctuating forces of the shaft line are only slightly affected by cavitation phenomena. Therefore, a determination of the extent of cavitation is not necessary for the computation of shaft line forces. The simple computation methods are based on quasi-static con-siderations that determine thrust and tangential forces at the individual blade directly from the wake induced variation of thrust and moment coefficients throughout one revolution – see [28], for example. The quasi-static manner of consideration proves adequate only as long as the cord length of the blade profile can be regarded small compared to the wavelength of the flow disturbance. In practical applications, this condition is fulfilled only for the blade frequency. The phase relationship of the excitation forces cannot be considered reliably by quasi-stationary methods. For this purpose, as well as for the computation of the excitation effects for multiples of the blade frequency, unsteady methods must be used. Because of the computing powers available today, lifting-surface methods have won out in practice against procedures that simulate the lift effect of the propeller blade by means of a lifting line, e.g. the method described in [29].

In this connection the computation effort needed and the available input data and deadline-related constraints should be well coordinated within the framework of the overall analysis. When the shaft system is being designed, the geometric data of the propeller are usually not available with the degree of detail necessary for the computation of excitation forces by a lifting-surface method.

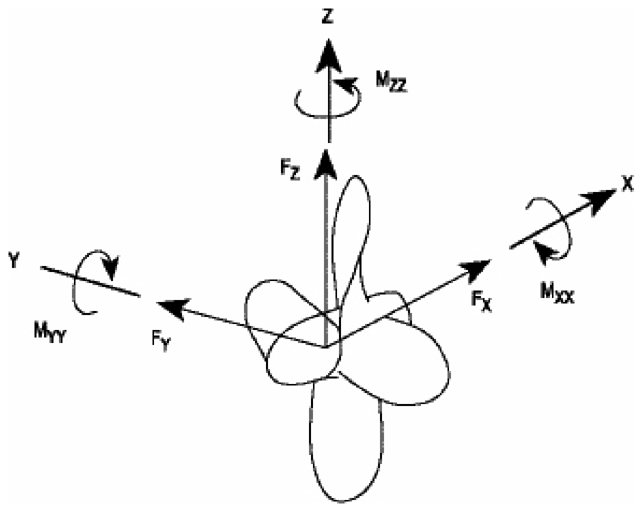


Fig. 22: Overall forces and moments at the propeller

Pressure Fluctuations

Regarding excitation of ship vibrations, pressure fluctuations are more significant than shaft line forces. In merchant ships, for which a certain degree of propeller cavitation is generally tolerated for the sake of optimising the propeller efficiency, about 10% of propeller- induced vibration velocities are caused by shaft line forces, whereas approximately 90% are due to pressure fluctuations. In the design of vessels having propellers with weak cavitation, this ratio is reversed, while at the same time the absolute excitation level is much lower.

Pressure fluctuations acting on the shell are a result of several physical causes:

- Displacement effect (thickness effect) of the rotating propeller. This effect is independent of the wake field, and its contribution to the overall pressure amplitude for the propeller of a merchant ship is about 10 to 30%.

- Portion resulting from or induced by the pressure difference between the back and the face of the blade. This effect, too, occurs independently of the wake field and contributes up to about 10% to the overall pressure amplitude.

- Displacement effect of the fluctuating cavitation layer that typically forms when the propeller blade is moving through the wake peak in the region of the outer radii. For the propeller of a merchant ship, the contribution of this effect to the overall pressure amplitude is approximately 60 to 90%.

Pressure pulses on the shell are also caused by the induction and displacement effect of the propeller tip vortex and the collapse of the individual cavity bubbles. Whereas the former process mainly has an effect in the frequency range corresponding to the higher harmonics of the propeller blade (see [30], for example), the latter phenomenon mainly influences the excitation characteristics in the noise frequency range. Both processes are excitation phenomena that, at present, are scarcely amenable to methods of calculation and should – if necessary – be investigated in a cavitation tunnel. In the following, these special aspects will not be dealt with.

From the above-mentioned contributions to the overall pressure amplitude, it can be concluded that high excitation forces can be expected only in case of cavitating propellers. As described in [31], for example, pressure pulses caused by a fluctuating cavitation volume V at a point situated at a distance r can be approximated by the following formula:

$$p = \frac{\rho}{4\pi} \cdot \frac{1}{r} \cdot \frac{\partial^2 V}{\partial t^2} \quad \text{where } \rho = \text{fluid density.}$$

The pressure amplitude is thus proportional to the acceleration of the volume of the cavitation layer. The formula is applicable for the free field and does not take account of the obstructing effect of the ship's hull. The magnitude of this effect depends on the geometry of the ship's hull and must be determined separately.

To derive the pressure amplitude from the formula stated above, the curve of the cavitation volume versus propeller blade position must be known. Principally, this curve can be estimated by calculation, by model experiment or by full-scale observation. However, the quantity to be used is the second derivation of the cavitation volume curve, i.e. the pressure amplitude is governed by the curvature of the volume curve.

Calculation of the volume curve requires knowledge of the pressure distribution on the propeller blade, both in the radial and circumferential direction. The flow condition in the tip region of the propeller blades has a particularly strong influence on cavitation phenomena. The flow condition at the blade tips is additionally complicated by the formation and subsequent detaching of tip vortices. Computation programs for the prediction of cavitation volumes are correspondingly complex.

To avoid strong cavitation-induced pressure amplitudes, the volume curve should exhibit the smallest possible curvatures. This can be achieved by influencing the wake (minimising wake peaks) and by a suitable choice of propeller geometry. However, improved cavitation characteristics must normally be "paid for" by reductions in efficiency. Selection of a larger area ratio Ae/A0 and reduction of the propeller tip loading by selection of smaller pitch and camber at the outer radii are the most effective measures. In addition, by means of skew, a situation can be achieved where the individual profile sections of a propeller blade are not all subjected to their maximum loading at the same time, but instead the volume curve is rendered uniform by the offset in the circumferential direction. Some concepts tolerate comparatively severe cavitation phenomena, and are aimed at making the growth and collapse of the cavitation layer as slow as possible; see [32], for example.

The formula stated above characterises the principal mechanism of creation of pressure fluctuations, but it is not suitable for the actual prediction. The corresponding computation methods can be divided into empirical, semi-empirical and numerical procedures:

The method presented in [33] is a purely empirical procedure where the pressure amplitude is determined from a small amount of geometric data related to the propeller as well as from the wake. The formulas given therein are based on regression analyses of data determined with a large number of full-scale measurements. However, propellers having a high skew are not covered by these statistics. For high-skew propellers, an appropriate correction can be made in accordance with [34], for example.

One semi-empirical method is the Quasi-Continuous Method, [35] and [36], developed at the Hamburg Ship Model Basin (HSVA). Here the determination of the expected cavitation volume curve is performed with the aid of a numerical approximation process based on vortex distribution. The following step, namely the calculation of the pressure amplitude from the cavitation volume for a given geometry of the ship's shell, is carried out by means of empirical formulas. With software based on [29], the latter calculation step, too, is performed using a numerical approximation method.

A comparison of results obtained from empirical and semi-empirical methods with full-scale measurements is dealt with in [37]. An evaluation of the two methods is described in [38].

If the aim is solely to determine the pressure fluctuations for a standard vibration analysis, empirical methods are often still preferable. More advanced methods should be reserved for novel designs where a higher computational effort is justified.

Normally, pressure fluctuations decrease for higher blade harmonics. If this is not the case, unusual characteristics of the propeller may be the cause. Exceptionally, the propeller can also generate excitation effects with frequencies differing from multiples of the blade frequencies. This generally indicates transient phenomena occurring in the ship's wake.

The number of propeller blades does not have any marked effect on the magnitude of pressure fluctuations.

To obtain overall excitation forces, pressure fluctuations must be integrated over the immersed part of the aftbody shell, taking account of phase relations. In this connection, two differences are pointed out between pressure fluctuations induced by propeller blade thickness and those induced by cavitation:

- The thickness-induced contribution decreases much faster with increasing distance from the propeller than the cavitation- induced contribution (approximately in proportion to $1/r^{2.5}$ as opposed to $1/r$)

- In contrast to thickness-induced amplitudes, phase relationships of the cavitation-induced pressure fluctuations change only insignificantly with increasing distance from the propeller, i.e. fluctuations are almost in-phase throughout the entire region affected

Superimposition of the two contributions is shown schematically in Fig. 23. As a result of the differences mentioned, the cavitation effect on integrated overall forces is even more predominant than is already the case due to the significant influence on pressure fluctuations.

As known from experience, the pressure amplitude above the propeller alone is not adequate to characterise the excitation behaviour of a propeller. Therefore, no generally valid limits can be stated for pressure fluctuation amplitudes. These amplitudes depend not only on technical constraints (achievable tip clearance of the propeller, power to be transmitted, etc.), but also on the geometry-dependent compromise between efficiency and pressure fluctuation. Nevertheless, pressure amplitudes at blade frequency of 1 to 2, 2 to 8 and over 8 kPa at a point directly above the propeller can be categorised as “low”, “medium” and “high”, respectively. Total vertical force fluctuations at blade frequency, integrated from pressure fluctuations, range from about 10 kN for a special-purpose ship to 1,000 kN for a high-performance container vessel. For usual ship types and sizes, corresponding values lie in be-tween 100 and 300 kN. Whether these considerable excitation forces result in high vibrations depends on dynamic characteristics of the ship's structure, and can only be judged rationally on the basis of a forced vibration analysis.

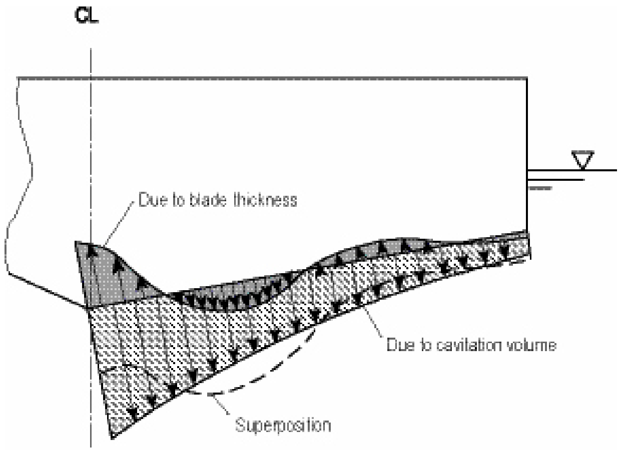


Fig. 23: Propeller-induced pressure distribution at the frame cross section

4.4 Evaluation and Assessment

A complete investigation of the vibration behaviour should involve not only an examination of the local structures for the danger of resonance (see 3.3), but also a prediction of the vibration level. Predicted amplitudes can then be compared with limit values specified for the ship concerned. Furthermore, a number of design alternatives can be checked, using results of forced vibration analyses for various variants. Decisions to be taken include the following:

- How many propeller blades are to be recommended?

- Are mass moment balancers necessary to achieve the agreed vibration level?

- Do engine supports (transverse or longitudinal) improve or worsen the vibration behaviour?

- Would it be advisable to provide a damping tank?

- Would structural modifications be advisable?

For large series of ships, for which the cost of calculations is almost negligible compared to potential savings, variant calculations are to be recommended as part of the evaluation procedure.

For optimisation of a design from the vibration viewpoint, evaluation methods that reflect the spatial distribution of vibration velocities for individual excitation frequencies proved to be helpful. In the following, some possibilities for a corresponding evaluation of calculation results of forced vibrations are described.

4.4.1 Vibration Velocity (Response) Spectra

It is recommended to perform the calculation of forced vibrations separately for relevant orders and sources of excitation, because the vibration response determined can then be attributed to a definite cause. Furthermore, a judgement can be conducted directly according to the old ISO 6954 standard as described in section 2.1, and the determination of overall frequency weighted r.m.s. values according to the new standard can be derived from amplitudes calculated for individual excitation orders. Since natural frequencies vary for different loading conditions and also because, in many cases, a particular revolution rate has not been specified, investigations should be carried out over a large frequency range. Therefore, calculation of amplitude spectra has proven its worth in practice.

A determination of the vibration response over the total frequency range for all nodal points of the FE model is not possible in prac-tice, nor is it necessary. This form of evaluation is generally per-formed for a maximum of about 50 representative points of the ship's structure.

Fig. 24 shows two typical vibration velocity spectra. Here, predicted vibration velocities at a stiffly supported (global) point on the bridge deck of a container ship is shown. The upper diagram applies to excitation by the 2nd order of the main engine (external vertical mass moment) and the lower diagram refers to the 7th order fluctuation of torque (ignition frequency). Vibration velocities (peak values) in longitudinal (x), and transverse (y) directions are plotted logarithmically for a range surrounding the nominal revolution rate. The latter is indicated by a vertical line in the diagrams. The lower and upper limit lines as per the old ISO 6954 are highlighted by horizontal lines. Vibration velocities determined during full-scale measurement are likewise shown (curves with markings).

The curve characteristic of the 2nd order excitation differs greatly from that of the 7th. Whereas for the 2nd order there are definite maxima, the curve for the 7th order is more balanced. Both, calculation and measurement, indicate the larger mode density in the higher frequency range. In contrast to the 7th order, definite natural frequencies can be assigned to the 2nd order curve. In this case, vertical bending vibrations of the ship's hull at 2.1 and 3.0 Hz can be identified. Calculated natural frequencies are about 5% higher than measured ones. This may be due to inaccuracies in the calculation, or to the use of slightly different draughts in the calculation and measurement condition, respectively.

One principal advantage of this kind of diagram is that trends on the amplitude level for varied natural and excitation frequencies are illustrated.

For this kind of presentation divisions on the frequency axis can be selected freely. Normally, vibration velocities are calculated for about 200 excitation frequencies within the selected frequency interval to obtain sufficient resolution of the curve shape.

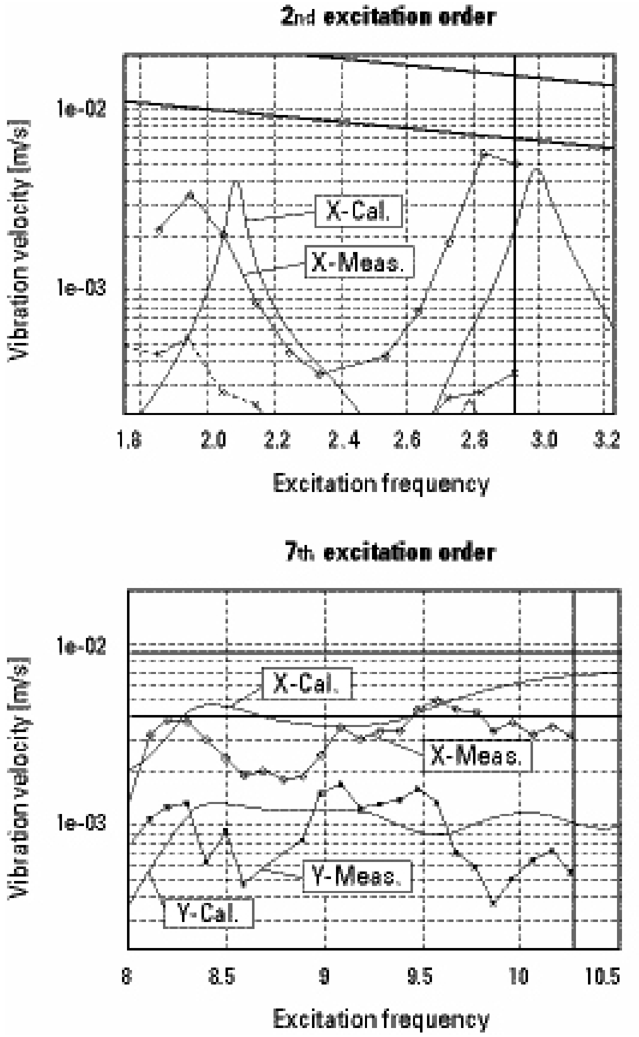


Fig. 24: Velocity spectra for excitation by the 2nd and 7th orders of main engine excitation

Excitation forces are generally determined for a reference rate of revolution, e.g. the nominal revolution rate or the revolution rate during the acceptance measurements. An assumption is then made about the dependence of excitation forces on the frequency. As already explained in 4.3.1 in the case of main engines, it is assumed for the sake of simplicity that excitation forces are proportional to the square of the revolution rate. Especially for the evaluation of amplitudes determined for excitation frequencies differing more markedly from the reference revolution rate, assumptions about the dependence should be chosen with care.

If the mode superposition method was used for the calculation of forced vibrations, it is possible to state, for each excitation order and location, which natural vibration modes make a significant contribution to the system's response. On this basis most effective meas-ures for detuning these modes can be specified.

4.4.2 Velocity Distributions

It is not possible to derive the spatial distribution of velocities from the response spectra predicted for a few selected points of the ship's structure. In particular, only a limited statement can be made about local magnifications of vibrations from these spectra.

Especially for passenger and naval vessels with complex spatial structures, a diagram as presented in Fig. 25, showing the velocity distributions over a deck, turned out to be useful. This particular example involves the aft region of the main deck of a naval vessel. The length of the arrows indicates the vertical velocity at the node concerned. All revolution rates that have a predominant effect on ship operation should be investigated. In this case, there are two such states, indicated by the left and right arrows, respectively.

In the example shown, the vibrations are excited by the propellers. Torsional vibrations of the aft part of the ship can clearly be identified. The vibration level decreases from aft forwards. Because deck grillages are taken into account in the FE model with the aid of beam elements, it is also possible to identify local increases of vibration. The outer starboard panel behind the fourth transverse bulkhead, for instance, is obviously in resonance in case of Operating Condition 1. Because of the high stiffness of the structure and the large distance from the source of excitation, the panel is, nevertheless, uncritical in spite of the resonance situation. In cases of this kind, there is some elbow room for the decision as to whether further stiffening measures are advisable. The shipyard can benefit from such diagrams, especially when trying to balance between the achievable improvement of the vibration behaviour and the effort required for such an improvement.

4.4.3 Mode of the Forced Vibration

Another clear illustration is a plot showing the spatial shape of forced vibrations. This kind of evaluation makes it possible to draw conclusions about the interaction between excitation forces and natural vibration modes. Fig. 26 shows forced vibration modes for the excitation of the ship's hull by various orders of a seven-cylinder engine.

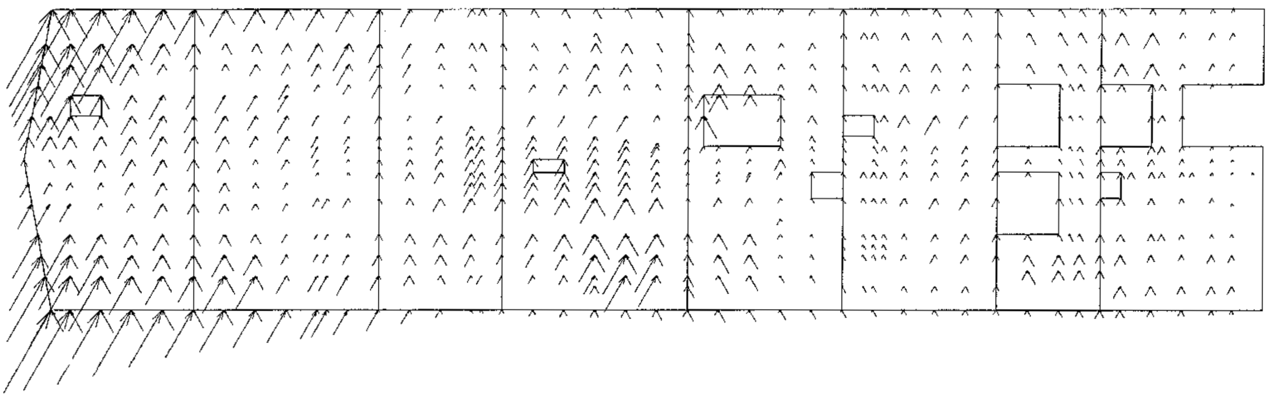
The 1st order (1.5 Hz) excites the fundamental torsional vibration mode of the ship's hull, whereas the vertical second-order mass moment (3.0 Hz) causes four-node vertical bending vibrations of the ship's hull. There is no recognisable increase in the longitudinal vibration of the engine.

At the 3rd order (4.5 Hz), it is mainly the X-type bending moment of the engine that takes effect. Because this is an internal moment and the stiffness of the engine housing is adequately high, practically no vibrations are transmitted into the foundation. Highest vibration velocities occur in the transverse direction at the top of the engine.

The 4th order (6.0 Hz) mainly excites longitudinal vibration of the engine, combined with longitudinal vibration of the deckhouse. The excitation is caused by the external fourth-order mass moment and by the pitching moment, which is likewise conspicuous at this order. As explained in [27], resonance with the L-type vibration of the housing can lead to a considerable increase of excitation forces. In this case, too, a definite increase can be seen in the vibration of the housing compared to the doublebottom. When reaching the aftmost hold bulkhead, the vibrations largely decayed.

5th and 6th orders of excitation cause a similar vibration mode as the 3rd order and are, therefore, not shown here.

The 7th order of excitation (10.5 Hz ignition frequency) evidently causes H-type transverse bending vibrations of the engine housing. Because the fundamental natural torsional frequency of the deck-



Left arrow: : Vibration velocity (vertical)
Operating Condition 1

Right arrow: : Vibration velocity (vertical)
Operating Condition 2

Fig. 25: Vibration velocity distribution in the deck area

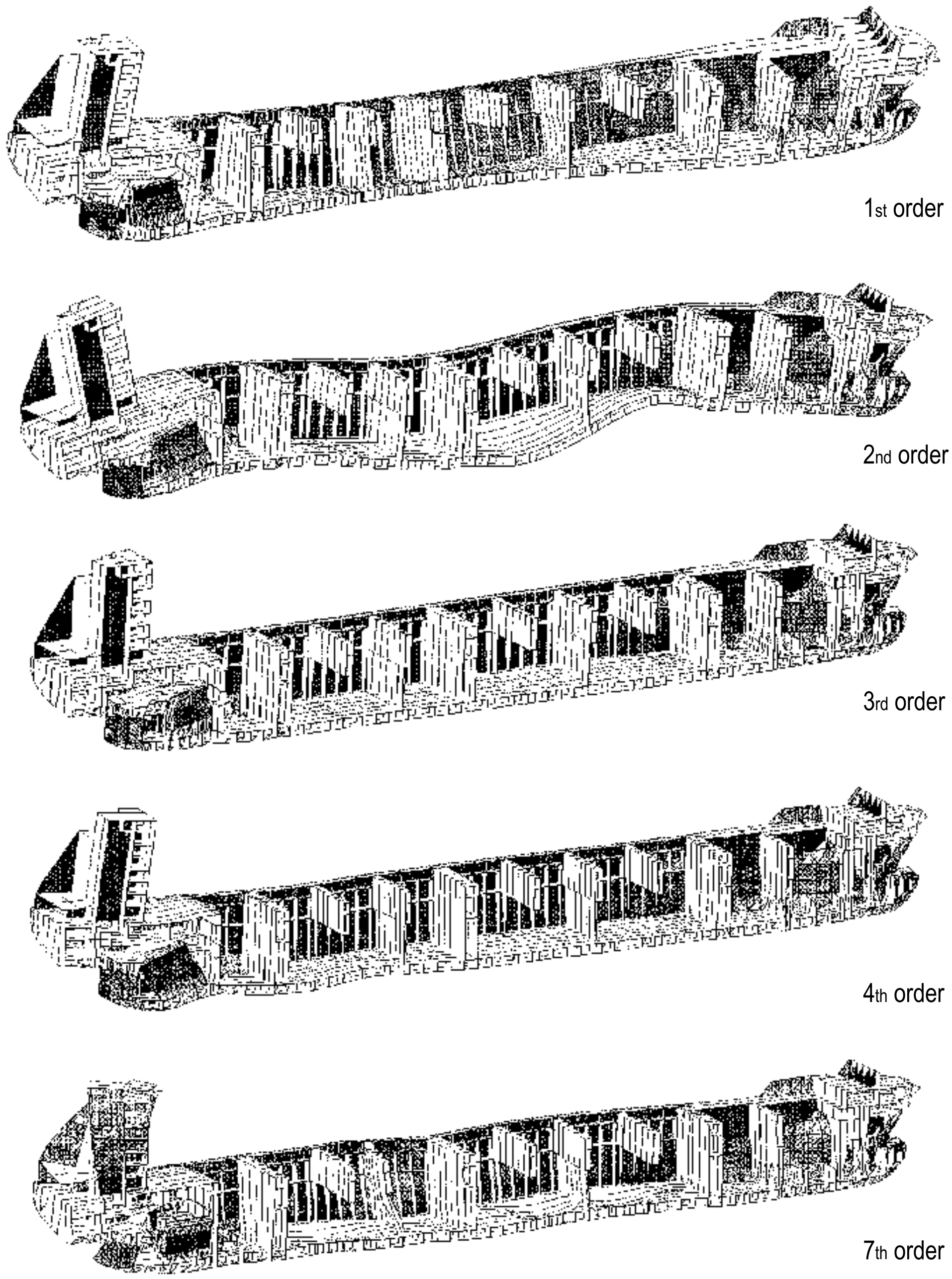


Fig. 26: Vibration modes in the case of excitation by vibration orders of the main engine

house is close to the excitation frequency, the bridge deck reveals high amplitudes. Deckhouse and engine vibrations couple in a complex mode, which is almost impossible to detect with simpler methods of calculation and evaluation.

A computer animation of forced vibration modes provides further important information and indicates potential improvements.

Naturally, a large number of evaluation algorithms can be used. However, because the calculation of forced vibrations of ships requires a considerable amount of numerical effort, the challenge here, too, is to find a reasonable compromise between cost and benefit.

5. Measurements

In parallel with the progress being made in the field of vibration prediction with mathematical models (FEM), which are becoming more and more detailed and hence require an increasing amount of effort, a growth in the use of experimental investigations is also evident.

One of the main reasons for this is undoubtedly the general progress being made in electronics.This has led not only to devices that are now portable and comparatively user-friendly (of the “plug & play” variety), but also to computing powers of PCs or workstations that, together with efficient software, permit the configuration and evalu-ation of even the most demanding and extensive experimental investigations. Thus, many investigations that even ten years ago were the domain of research can now be carried out within a reasonable time and almost as a matter of routine.

Specially worth mentioning are “pre-triggering” and “post-triggering”, by means of which even rare events can be measured automatically in a purposeful manner with almost all multichannel measurement systems. For example, flow-induced excitation forces transmitted into the ship’s structure via appendages, such as fins, nozzles or rudders, are difficult to identify since their occurrence often depends on specific, but initially unknown, conditions. In such cases, this triggering possibility can be extremely useful. The dis-advantage of processing the huge and bewildering amount of measuring results collected over months can thus be avoided by pre-selecting relevant data.

Further reasons for an increasing use of experimental investigations in ship technology include the general trend towards lower structural weight combined with increasing propulsive power. Moreover, production optimisation and the trend towards uncon-ventional designs and new hull forms require an increasing demand of measurement. In this connection, conventional ratios (e.g., length to beam, beam to draught, etc.) are disappearing more and more from the design process. Specific excitation phenomena may, for example, occur for a novel design as a consequence of poor inflow to the propeller, or because air is getting underneath the ship at its forward shoulder, resulting in unpredictable vibrations. Corresponding problems can only be solved by a well-planned measurement campaign.

A brief overview of sensors and measurement systems used for vibration analyses is given below. This is followed by an introduction to the procedure of various measurements frequently applied in shipbuilding and ship operation. Some typical methods to evaluate measurement data and to assess results are discussed next.

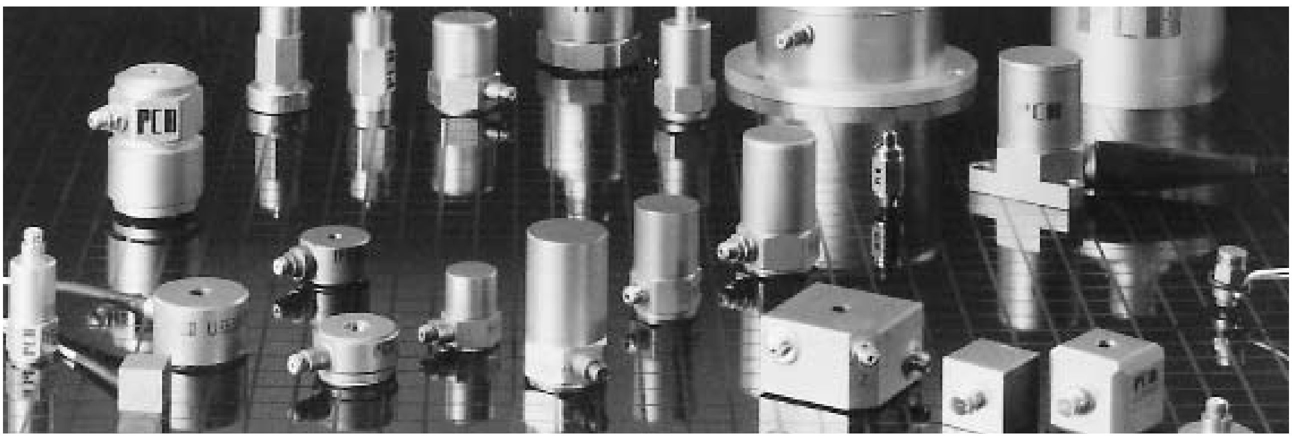
Finally, six specific examples from practical work (troubleshooting) indicate how special vibration problems on ships can either be avoided or solved at comparatively low cost. The problems selected are technically simple, but in some cases they had serious economical effects. The individual problem and the way it was satisfactorily solved on the basis of measurements can be clearly understood from the presented graphic presentations.

5.1 Sensors

In any application, accelerometers are essential. Every manufacturer offers a wide spectrum of standard sensors constituting an ab-solutely vast range to choose from.

For structures typical in shipbuilding and marine engineering, only the frequency range up to about 300 Hz is of interest from a vibrational point of view. The maximum acceleration values are generally less than 1 G, and in most cases they are distinctly lower.

Popular sensors consist, on the one hand, of seismic types. As spring-mass systems they also measure statically (0 Hz) and function as inclinometers. On the other hand, piezo-electric sensors are widely used. The latter offer advantages in the high frequency



range. Depending on the design, the upper frequency limit ranges to about 10 kHz or even higher and thus can also cover the structure-borne noise range. However, because of the physical principle on which they are based, they are unsuitable for measurement of ship motions below 1 Hz.

The sensitivity of both types of sensor is adequate for the measurement of mechanical vibrations. To name an order of magnitude: 5 mm/s^2 (about 0.0005 G) can be measured without any problem. At a frequency of 10 Hz, this corresponds to a vibration velocity of approximately 0.08 mm/s and is, therefore, at the limit of the human perception threshold.

Sensors for the direct measurement of vibration velocity, which is predominantly the quantity to be assessed, play a part in special cases only.

For measurements of propeller pressure fluctuations, requirements to be met by sensors with regard to pressure range, dynamic range and sensitivity are likewise comparatively easy to fulfil nowadays. A greater ruggedness to cope with possible influences in the vicinity of a propeller (such as sand or unfavourable cavitation effects) to re-liably withstand prolonged investigations would, however, be bene-ficial. As measurement cells, not only piezo-electric elements but also strain gauges are used.

To detect causes of special vibration problems, it is occasionally necessary to measure various operational parameters (e.g., of the engine). This requires, consequently, the use of a wide variety of sensors.

5.2 Measurement Systems

The production lines range from handy mobile (i.e., battery-powered) and easy-to-operate single-channel compact devices up to PC-controlled multi channel measurement systems. These advanced systems, equipped with 32 or 64 channels and extensive triggering features, represent the upper end of the scale. In view of the wide variety of devices available, a reasonably rational decision in favour of a specific system is possible only if the measurement task is clearly defined. However, experience has shown that clear definition is difficult since the decision has also to be made for unknown future tasks. There is no such thing as a practical device for all possible applications. On the other hand, a higher-performance and more compact generation of the product will generally appear on the market after a few years.

For the widespread need to determine the vibration level at various places on the ship and to clearly identify the main source of exci-tation, it is sufficient to use single or dual-channel frequency an-alysers in hand-held format capable of showing the measured spectrum on a small display and to store it. Unfortunately, they do not yet feature evaluation procedures as per the new ISO 6954. Also, for reasons of memory capacity, the time signal is mostly not available.

1- to 4-channel systems consisting of sensors, amplifiers and a DAT recorder can still be classed as “mobile”, and they have the advantage of being able to record time signals simultaneously in a practically unlimited manner.

DAT cassettes, with up to 2 hours capacity, are a worthwhile stor-age medium. Evaluation takes place later with a PC using appropriate software. If data quantities are small, laptops can be used instead of a DAT recorder. Measurement data are then present in digitised form on the hard disk at a defined sampling rate. In the conversion of analog to digital data, possible aliasing effects must be taken into account. These effects can generally be ruled out if analog signals are suitably filtered.

At the next level, there are 8- to 16-channel units, which require a place in a protected environment and a 220 V power supply. These units can certainly not be classed as “mobile”. In the simplest case, the measurement chain corresponds to a 1- to 4-channel system. The analog amplifier output (generally with a maximum of 2, 5 or 10 V) make it possible to connect other devices, such as 2-channel analysers or thermal printers for checking and observing the measurement. Equipment with this scope is typical for investigations of the global vibration behaviour, e.g. of the aft part of the ship including the deckhouse as part of the troubleshooting process.

The range comprising 16 or more channels is the domain of complete data acquisition systems, “front-end” units as they are called, containing amplifier boards often integrated in the housing. The configuration of measurement channels and other setting activities, such as those for trigger functions and for controlling the measurement procedure, take place via the interface connection to the PC or laptop. Depending on the requirements, extensive meas-urement data are stored on the PC hard disk or via external drives, e.g., on hard disks, DAT cassettes or MO (magneto-optical) disks. These systems make it possible to program a completely auto-matic measurement procedure – for example, for long-duration measurements – and are mentioned here only for the sake of completeness. For vibration investigations on board ships they are necessary only in exceptional cases, and then because of their trigger functions, and hence the possibility of automatically measuring vibration phenomena that rariley occur.

5.3 Measurement Procedures

Depending on the kind of problem and the measurement effort to be expended, the vibration measurement procedure must be adapted to suit the given operational constraints. Basically, a distinction can be drawn between measurements for various excitation types.

5.3.1 Impact Method

The purpose of these measurements is to determine natural frequencies of particular structural components or equipment items. They are mainly used to check the design of plates, panels and stiffeners of superstructure decks and tank walls in the engine room area before the ship is completed. If measured natural frequencies indicate a danger of resonance with main excitation orders (propeller and engine), changes at this time are still com-paratively cheap for the shipyard. In case of passenger ships, for example, a mistake here would be particularly critical. This method is also used when vibration problems occur, but the necessary conditions cannot be realised on a sea voyage in the near future.
A further field of application is the investigation of structures and appendages situated below the waterline. In this case, results have to be corrected in accordance with hydrodynamic masses.

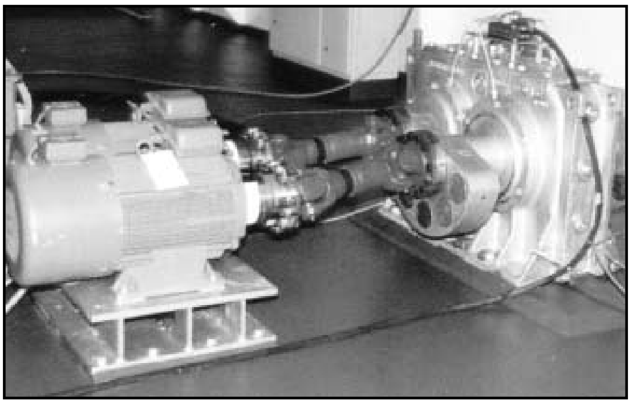
The structure concerned, on which generally two to eight accelerometers have been attached beforehand by means of magnets, is struck non-rhythmically with an impact hammer. The hammer has a suitable rubber buffer at its impact surface and is additionally equipped with an accelerometer for measurement of the striking force (the weight of the hammer being known). As a result of the impact, the component is deflected locally and performs decaying vibrations at its natural frequencies. Coherence and transfer functions, continuously monitored on an FFT analyser, indicate when the measurement can be discontinued. The coherence should be near unity over the frequency range of interest, and both functions should remain stable, i.e., they should not change when further impacts take place.

In the case of a newbuilding, for example, these investigations require a work pause during construction which is not always easy to arrange. Therefore, they are often possible only on weekends.

5.3.2 Exciter Test

In this case, too, the aim is mainly to determine natural frequencies. The fixing of the exciter foundation requires a certain amount of construction work. In addition, it must be estimated beforehand whether the frequency range and the excitation force are adequate for the vibration problem, on the one hand, and the guarantee that damage to the structure is avoided, on the other.

In individual cases, for example, when a high amplitude level was predicted, an attempt is made at an early stage to simulate ex-citation characteristics of the engine or propeller by means of an unbalance exciter test. The expected vibration level is then extrapolated from the measured level, taking excitation forces of the engine or the propeller into account.
Small unbalance exciters (F_{max} < 100 kN) are highly suitable for the investigation of appendages, larger panels, individual decks, or foundations of larger items of equipment. More powerful exciters require extensive installation effort because of their great weight and their dimensions and are, therefore, used in exceptional cases only.



5.3.3 Measurements During Ship Operation

Here, the ship's own vibration sources provide the excitation forces, i.e. various orders of the propeller and engine.

In rare cases, the vibration level is caused by slamming-induced impacts or by special flow phenomena.

Measurements at Rated Output of the Propulsion Plant

This measurement procedure, which reflects the most important operating condition of a ship, is applied to compare the measured vibration level with permissible values stipulated in the building specification.

This requires an adequate water depth (four to five times the draught to eliminate shallow water effects), small rudder angles, moderate wave heights, and the absence of violent ship motions.

For this purpose, it is sufficient to check or record about 20 measurement points distributed over superstructure decks and workshops, using a hand-held or a 1- to 3-channel device. The selection should cover various sizes of panels and positions on the deck for measurements in the vertical direction. For upper superstructure decks transverse and longitudinal directions should be considered as well. Only then can an informative overall picture be obtained. Furthermore, engines and peripheral devices are subject to limits that must not be exceeded if damage is to be avoided. Special attention must be paid to turbochargers.

If limits are not exceeded and the vessel's master and/or the owner indicate that they are satisfied, then the task has been completed with a minimum of effort.

However, if problems exist, these measurements are generally not sufficient to clarify causes, since they do not provide information either on natural frequencies or about vibration modes. Therefore, it is scarcely possible to develop a detailed diagnosis with the aim of working out effective remedies.

Measurements at Variable Revolution Rates

The simultaneous acquisition of 8 to 16 acceleration signals, possibly together with other measurement quantities, permits not only the determination of relevant vibration modes:
when combined with a run-up manoeuvre of the propulsion plant, e.g., in the range 50–100 % of the nominal revolution rate, results also reveal relevant resonance points showing corresponding natural frequencies.

For example four stiffly supported measurement points on the bridge deck of merchant vessels (two longitudinal, one transverse, one vertical) usually reflect with sufficient clarity global vibration modes of the deckhouse (longitudinal, transverse, torsion about the vertical axis). Combined with sensors in the vertical direction at the forward and aft footing of the deckhouse and at the transom, relevant natural modes and natural frequencies of the whole elastic system consisting of deckhouse and aftbody can be determined.

Accelerometers can easily be moved from one position to another. To determine the hull's basic vertical natural modes and frequencies, it is sufficient, for example, to move the four sensors placed on the bridge deck to one side of the main deck.

If the propeller has been identified as the cause of a vibration problem, it is only possible by means of pressure pulse measurements to determine whether induced pressure fluctuations are unusually high. In the case of controllable pitch propellers, not only the revolution rate but also the pitch should be varied in the range of 50–100 % to assess cavitation phenomena.

Measurement durations of up to 5 minutes for quasi-steady state conditions (constant speed and propeller pitch) make it possible to distinguish between excitation frequencies, even if they lie close together. On the other hand, in case of speed-up manoeuvres – either continuous or in small steps of, say, 3 or 5 revolutions – a total duration of 20 to 40 minutes must be expected if no resonance points in the relevant range of revolution are to remain undetected.

The measurements sketched represent the scope typically required for troubleshooting. They are, however, also appropriate if shipyards or owners attempt to reduce the vibration level or if they want to find out why an engine top bracing, for example, has failed to produce the hoped-for success. The measurement data then permit an identification – possibly supported by calculations – of cost effective measures.

Basically it can be stated that local problems ("calming" of panels, bulkheads, devices, or smaller equipment items) can be solved with little effort. Unacceptable global vibrations, on the other hand, require considerable modifications of the design or of the propulsion plant – extending in extreme cases up to changes of the revolution rate or of the number of propeller blades, for example.

5.4 Evaluation and Assessment

For various possibilities of data evaluation, it is necessary to have measurement data available as time signals. These signals are present either in digital form in bulk memories, such as hard disk, MO disk, DAT tape or CD, or in analog form if magnetic tape units are used. An alternative is offered by popular DAT recorders, which behave like analog devices (voltage output), but have finite data rates.

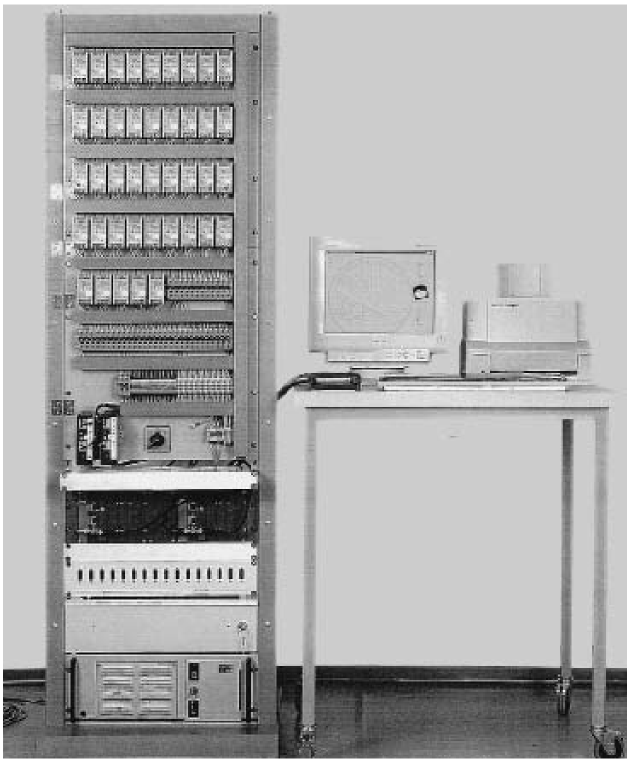
The sensors and the sampling rate define the highest frequency still contained in the signal. Frequencies cut off by the digitisation process are irretrievably lost unless the original signals are stored on magnetic or DAT tape.

In case of largely unclear vibration problems, both the sampling rate (the highest signal frequency) and the measurement duration (frequency resolution) should be chosen generously.

Theoretical background of various evaluation methods is dealt with in standard works, e.g. [39] and [40].

5.4.1 Time Domain

Assessment of the time signal itself is often neglected in favour of powerful statistical procedures and compact analysis methods in the frequency domain. The reason is that, even with the insight of an experienced engineer, it is still comparatively time-consuming to go through all time series and to check whether they meet expectations.



However, evaluation of some typical time series is advisable. Time signals provide valuable clues for an understanding of the vibration problem – clues which other methods are unable to supply.

The shape, first of all, makes clear whether time signals are periodic, harmonic or stochastic (random), steady-state or transient. Several signals recorded simultaneously indicate phases and amplitude relationships when grouped below each other or arranged in a common mesh.

A time plot, covering the entire measurement duration, immediately provides information as to whether steady-state conditions exist or time intervals are occurring with widely differing amplitude levels. For example, unpleasant beating effects become clear immediately.

Typically, several orders of excitation determine the vibration level on board. Furthermore, low-frequency motions of the ship are reflected in vibration signals where specific sensors are used. Additionally, trim and list change the mean value. The time signal can, therefore, become complicated and hence scarcely capable of interpretation, especially as a result of many components having different amplitudes and frequencies.

However, if the time signal of a particular frequency is of interest, e.g. of propeller blade frequency, filtering can be used as an aid. By elimination of the unwanted frequency ranges, a time signal is extracted that then represents the vibration created by the propeller blade frequency only.

The usual evaluation software basically makes it possible to per-form differentiation and integration in the time domain. For assessment, the vibration velocity signal is often desirable. However, integration of the acceleration signal causes difficulties, even in the case of measurement intervals in minutes range. The reason for this lies in the constraints of the electronics of sensors and other devices. Even if the mean value (and possibly also its trend) is eliminated beforehand, the result often remains unsatisfactory. Only time-consuming piece-by-piece integration can then lead to success.

5.4.2 Frequency Domain

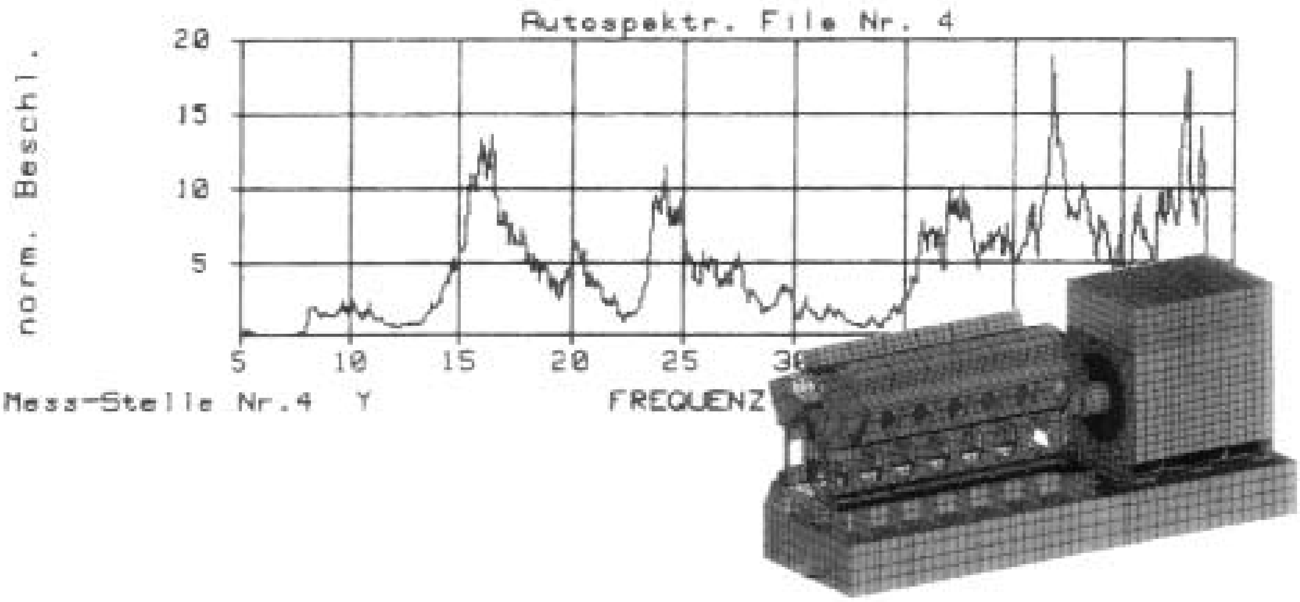
Spectral analysis based on the Fast Fourier Transformation (FFT) is by far the most powerful tool for assessment of vibrations. This method identifies main sources of excitation immediately on board. Multichannel measurements permit the determination of complicated vibration modes and associated amplitudes.

Even though vibration signals are often stochastic and transient in their nature, harmonic analysis is, nevertheless, generally successful. If the quantity of data is large enough, occasional faults such as signal gaps or the occurrence of peaks only have a slight falsifying effect on the result. However, as far as amplitudes are concerned, caution is advisable because FFT parameters, such as windows, block sizes and overlap, can have a significant effect.

Amplitude Spectra

In the selected band, spectra reveal the amplitude level corresponding to each frequency, and thus make it possible to identify main excitation sources.

Of course, vibration standards for comfort, structures, engines and electronic equipment do not only differ from each other in permis-sible amplitudes, they also require widely differing evaluation methods. These spectra likewise form the basis for assessment according to various technical standards – see also Chapter 2.



Waterfall Diagrams

"Waterfall" diagrams (three-dimensional spectra) additionally indicate the change of amplitude and frequency versus time. Over a period of 20 to 40 minutes, for example, the main parameter is varied either as uniformly as possible or in steps. This parameter is principally the speed of the main engine and hence of the propeller, but it can also be the propeller pitch while the speed remains constant.

In this connection, it must not be forgotten that this kind of speed-up manoeuvre also affects quantities influencing the vibration behaviour. Naturally, relevant excitation forces and ship's speed increase during this speed-up process.

As described in some of the practical examples, waterfall diagrams conspicuously reveal existing resonance points. In addition, amplitude curves varying with the engine speed can be distinguished from those in which the frequency remains completely invariable. The former represents forced vibrations and can be assigned directly to the nth order of the engine or to the propeller excitation. On the other hand, natural vibration modes are independent of the revolution rate. Lower natural frequencies of the hull's bending vibrations (vibrations with two to four nodes) are generally revealed. This can also apply to larger subsystems, such as a deckhouse or a radar mast.

Contrary to common assumptions, excitation of a hull's natural vibration modes does not require a "suitable" sea state in the sense of a particular wave encounter frequency or particular pitching motions. For large ships, wave heights of, e.g., 0.5 m are sufficient to excite these natural vibration modes to an extent that amplitudes and corresponding natural frequencies can be measured.

Order Analysis

This term refers to the relationship between amplitude and excitation frequency for a particular order of excitation.
In a waterfall diagram it corresponds to the mountain ridge.
The two-dimensional presentation shows the variation of amplitude as a function of frequency in a convenient manner.

Modal Analysis

This is generally regarded as a tool to determine vibration modes (free and forced) of complex structures by means of a large number of measurement signals. A necessary prerequisite is simultaneous acquisition of signals, so that phase relationships between individual measurement points can be considered.

In the event of vibration problems, knowledge of the causal vibration mode is crucially important because it is only on this basis that effective countermeasures can be worked out. Thus, for example, it would be useless to provide a strongly vibrating deck panel with supports situated at nodes of the vibration mode causing the disturbance. Corrective measures are generally aimed at connecting structural points characterised by large relative motions. Depending on mass and stiffness relationships, coupling of this kind also changes the vibration behaviour of the adjacent part of the structure, which might result in undesirable effects.

In practice, detecting global vibration modes requires much effort. Even with a 32-channel measuring equipment, the spatial vibration mode can be measured for only limited areas of the hull. It is, therefore, important to proceed with a specific purpose in mind during measurements. Knowledge of vibration modes acquired from FEM computations, for example, can be helpful.

In most cases, specific excitation arrangements to determine a hull's natural vibration modes require great effort (anchor-drop test, unbalance exciter units, stopping manoeuvres, impact systems, hydropulsers, shock tests, etc.). For this reason, among others, the use of modal analysis in practice is mostly confined to parts of structures or individual items of equipment.

Today, program packages for modal analysis not only offer the possibility of displaying detected natural modes as animations on a PC screen, but are also able – in conjunction with FEM programs – to simulate effects of changes in the structural model (stiffness, damping, mass characteristics) on results.

5.5 Practical Applications

Example 1

The first unit of a new series of container ships was investigated because of significant vibration problems on a loading voyage. In addition to severe vibration problems of some local components on the radar mast, high global longitudinal vibrations of the deckhouse, in particular, with values of up to 12 mm/s, were unacceptable. The port side exhibited higher vibration velocities. These were due to asymmetries of the deckhouse, resulting in torsional vibrations about the vertical axis.

In addition to resonance of the mast at propeller blade frequency, the main reason for the high level turned out to be resonance of the four-node vibration mode of the ship's hull with the engine's 2nd-order excitation. It was found that the shipyard installed a compensator, driven at twice the revolution rate, at only one end of the 6-cylinder engine. A one-sided configuration of this kind gener -ally produces incomplete compensation. Because the phase relationship of the compensator force is fixed, an increase of vibration is even possible to occur in the worst case, depending on the position of the vibration node relative to the balancer force. If the balancer acts at the node, no change occurs in the vibration level.

Depending on the loading condition, the position of the relevant vibration node can move considerably. The installation of a balancer acting at one end only, as a means of compensating a free 2nd-order mass moment, must therefore, be regarded as critical.

The shipyard initially hoped to reduce deckhouse and mast vibrations to an acceptable level by carrying out various structural alter-ations (stiffening of the radar mast, changing of the deckhouse asymmetry). However, it turned out that significant vibration problems still existed and that the limit curves specified were not being complied with.

Evaluation of the first course of remedial action revealed the following situation:

In the deckhouse (at the height of the bridge deck) there was a level of about 9 mm/s due to the engine's 2nd-order excitation. The radar mast reached amplitudes of more than 30 mm/s, caused by the propeller blade frequency (4th order). The vibration behaviour in ballast condition was found to be as unfavourable as in loaded condition.

The highest level of longitudinal vibration of the deckhouse occurred at engine speeds of about 115 r/min. This was particularly critical because it was the service speed with the shaft generator switched on. The radar mast, on the other hand, reached the largest amplitudes at a maximum revolution rate of 140 r/min.

An evaluation of the 2nd-order excitation at 115 r/min, corresponding to 3.83 Hz, exhibited proximity of resonance to the 4-node vi-bration of the ship's hull in ballast condition, whereas in the loaded condition resonance with the 5-node vibration occurred at about 126 r/min. In the waterfall diagram of Fig. 27, the resonance for the longitudinal vibration of the deckhouse is clearly evident for ballast condition.

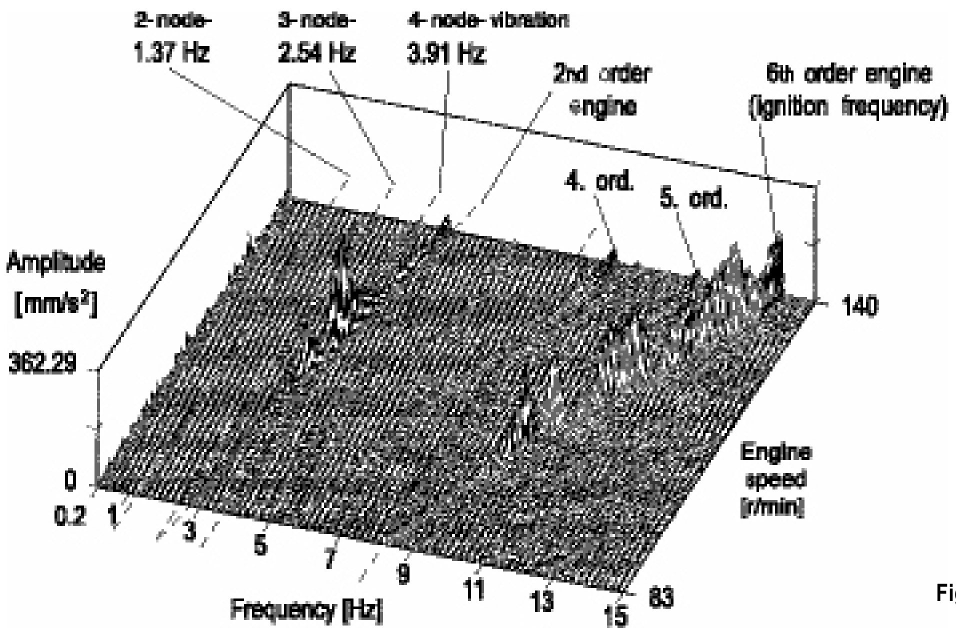


Fig. 27: Waterfall diagram for the longitudinal acceleration of the bridge deck

The natural frequency of the radar mast at 9 Hz exhibited proximity of resonance to the propeller blade frequency at almost full engine speed (9.3 Hz). Detuning of the natural frequency was achievable only by difficult-to-implement reinforcements of the mast foundation or by means of bracings or stays extending, e.g., to the forward edge of the wheelhouse deck. It was not expected that further stiffening of the mast structure itself would produce any significant effect.

The vibration behaviour of the mast in the longitudinal direction is illustrated in Fig. 28.

Sister ships were finally fitted with balancers at both ends of the engine as standard equipment. As a result, a complete balancing of the 2nd order free mass moment was achieved. The mast vibrations were greatly reduced by increasing the natural frequency to about 11 Hz.

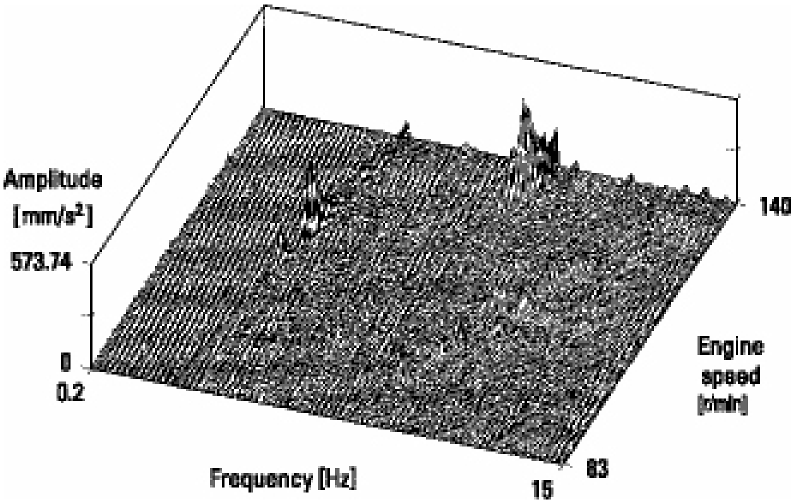


Fig. 28: Waterfall diagram for the longitudinal vibration of the radar mast

Example 2

The first voyages after commissioning of a container ship led to damage of deckhouse equipment, radar mast, crane boom, and equipment parts in the engine room.

The crew reported extreme vibrations in the deckhouse area in bad weather. It was, therefore, agreed to perform an investigation of the vibration behaviour for two sea conditions: one in a sea area as calm as possible; the other in rough waters.

Fig. 29 shows measured amplitude spectra of deckhouse vibrations in the longitudinal direction in calm as well as in rough seas.

The frequencies of 1.35 Hz and 2.60 Hz can be assigned to global hull vibrations. However, the amplitudes shown were not excited by the propeller or engine, but by the seaway. Considerable slamming impacts occurred in the forward part of the ship. Time signals from various measurement points are shown in Fig. 30, and excerpts from these curves are given in Fig. 31.

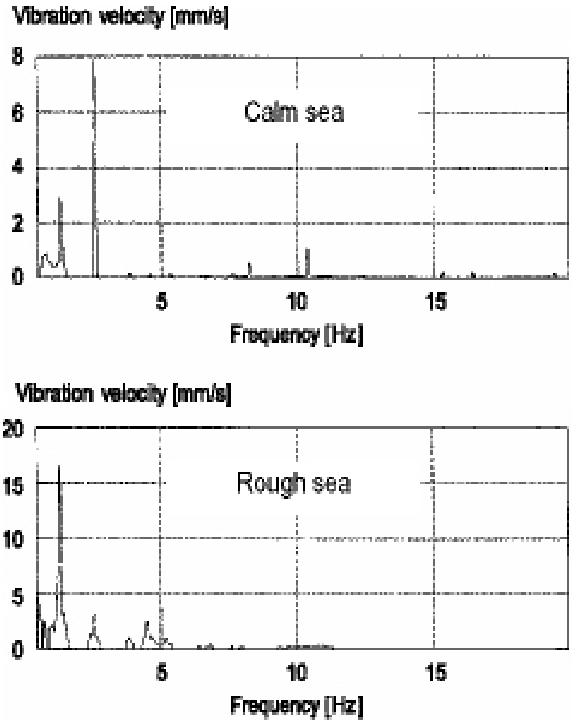


Fig. 29: Amplitude spectra of vibration velocity, bridge deck, longitudinal direction

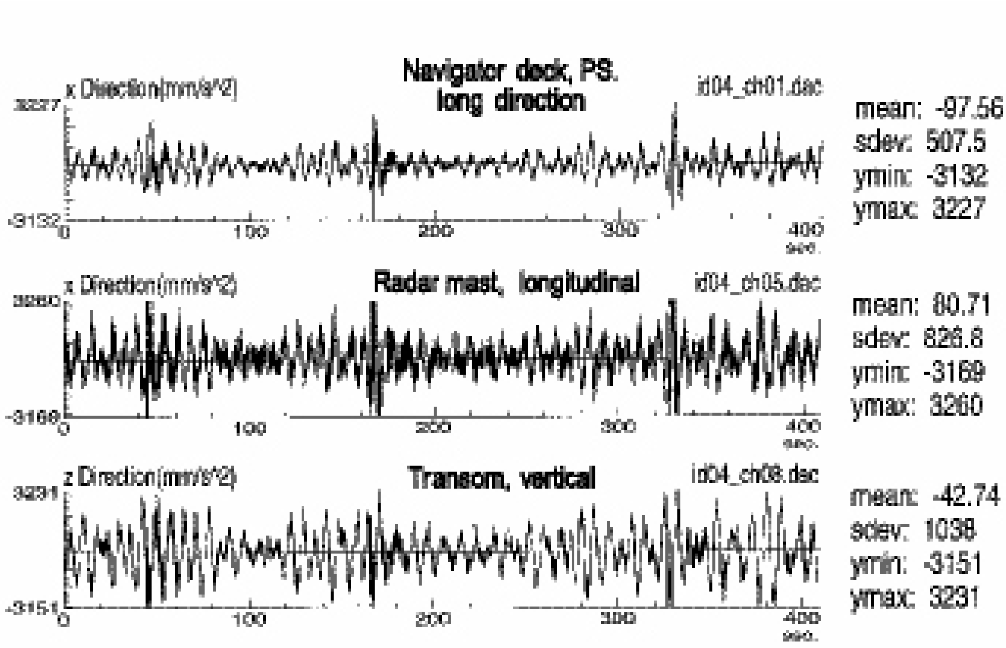


Fig. 30: Rough sea, three slamming impacts

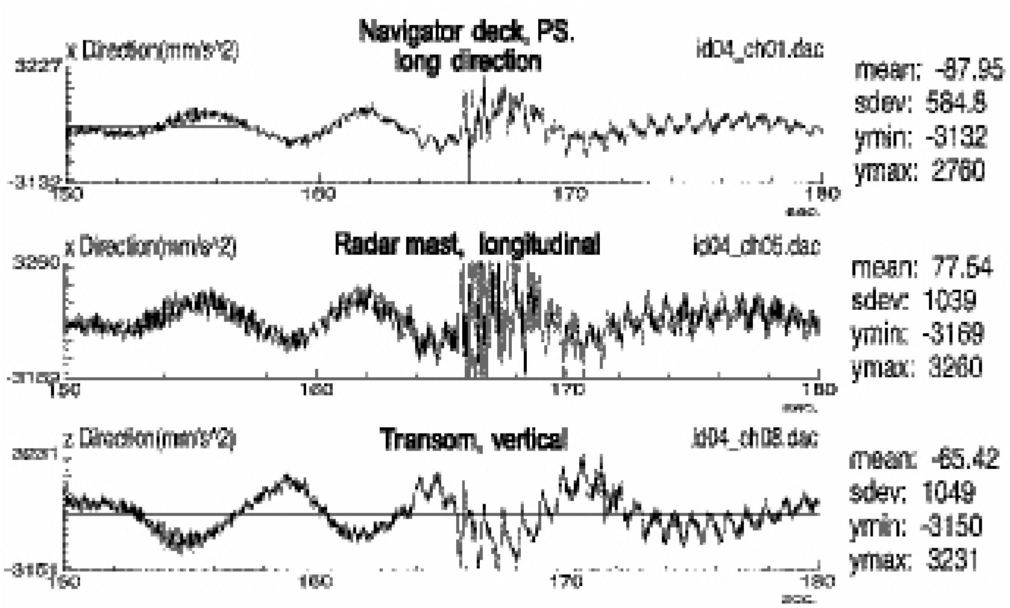


Fig. 31: Excerpts of above time signals

As a conclusion of the investigation, it was found that the damage was caused by operating the vessel almost non-stop in bad weather during the first few voyages at shallow forward draught. This resulted in slamming impacts, which occurred more frequently and even more severely than measured. In addition, definite structural deficiencies were also found in the detailed design.

However, the vibration behaviour caused by the propulsion plant was satisfactory, both for the comfort of the crew and for the integrity of the structure, machinery and electronic equipment (radar mast). The investigation underlined that severe slamming impacts extending over a long period of time must definitely be avoided.

Example 3

The starting point for these measurement-related investigations consisted of cracks that occurred on the main deck at the aft edge of the deckhouse just shortly after commissioning of the ship. The ship operator was concerned that vibrations in this area were the cause of the damage and that other areas of the structure might similarly be found to be damaged later on.

Therefore, measurements covered the investigation of the vibration behaviour of the entire deckhouse as well as of its possibly inadequate incorporation in the hull structure.

Signals from 14 accelerometers distributed over the main deck and the deckhouse region were recorded simultaneously for each of the following manoeuvres:

- Nominal speed, n_p : 204 r/min
- Run-up of main engine, n_p : 160-204 r/min
- Reversal from full speed ahead to full speed astern
- Anchor-dropping manoeuvre, propulsion plant not operating

Manoeuvre 1 gives the vibration level at nominal speed and thus presents the main part of the vibration concerning fatigue strength of welded joints in question. Manoeuvre 2 makes it possible to determine resonance points and to estimate associated amplitudes. Manoeuvres 3 and 4 were intended, in particular, to reveal lower natural frequencies of the ship's hull and corresponding vibration modes with regard to the possibility of inadequate mounting of the deckhouse.

Evaluation of measurement results revealed the following overall picture:

Amplitudes at various rigid points of the ship were small and gave no cause for complaint. However, the aft bulkhead of the deckhouse, and thus the detected cracks, were situated close to the aft node of the vertical 2-node vibration mode and were, therefore, in the unfavourable region of high alternating stresses.

Evaluation of the speed-up manoeuvre led to a surprising aspect that can be seen in Fig. 32, showing the longitudinal vibration of the deckhouse.

The deckhouse turned out to vibrate completely isolated from the hull at a frequency of 10 Hz. Amplitudes remained almost constant over the entire speed range, briefly rising only in case of resonance with the propeller blade frequency. This isolated vibration behaviour of the deckhouse is unusual.

The seaway is the only possible source of excitation, exciting the 2- and 3-node vertical hull vibrations of the ship, which in turn act as a source of excitation at the footing of the deckhouse.

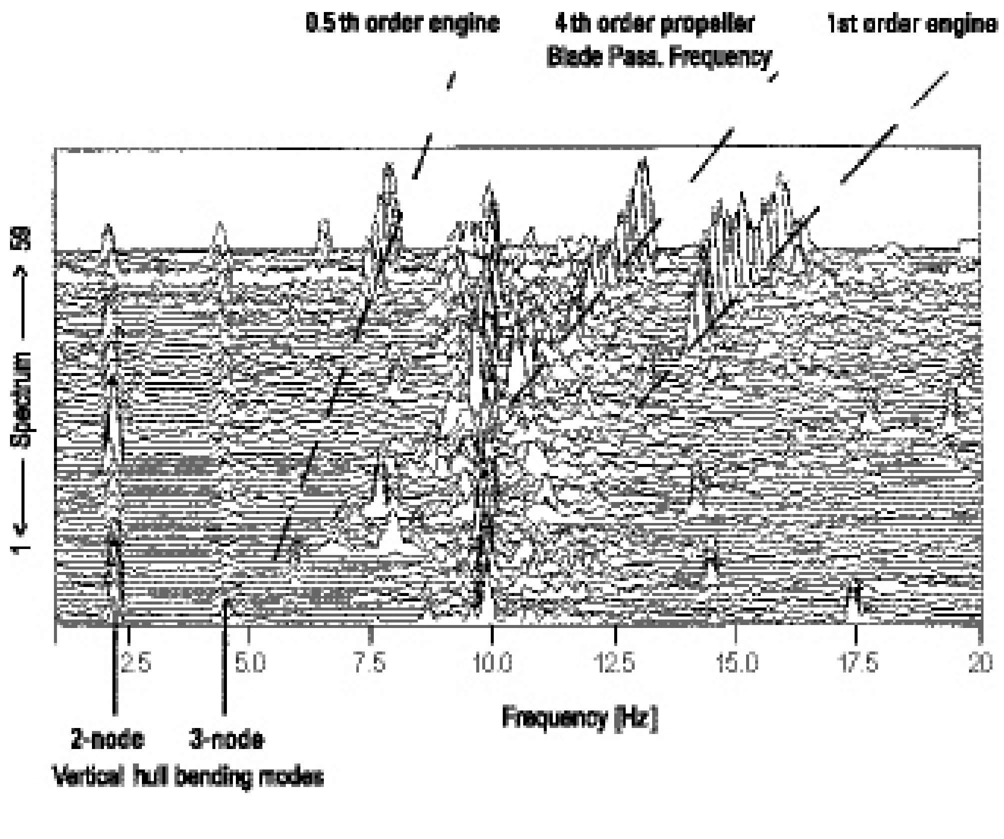


Fig. 32: Waterfall diagram for the longitudinal acceleration on the bridge deck

This vibration behaviour supported the presumption of comparatively poor vertical connection of the deckhouse, leading to significant amplitudes in rough seas.

In reconstructing the manufacturing process (mounting of the deckhouse on the main deck), various deficiencies and inaccurate fits were found in the region of the aft bulkhead of the deckhouse. These had to be regarded as having contributed to the cracks.

It was, therefore, recommended that, after completion of repairs, these points should be examined for new cracks after every period of rough weather.

Corresponding deficiencies of production did not occur on the sister ships. No further cases of relevant vibration damage came to light.

Example 4

The first unit of a series of container ships exhibited an unsatisfactorily high level of longitudinal vibration of the deckhouse in the vicinity of nominal propeller speed (100 r/min).

So as not to jeopardise the success of this ship type by an unfavourable impression of its vibration behaviour, the shipyard decided to per-form experimental investigations for three different variants, both of a hydrodynamic and of a structural nature.

- Initial situation — Variant A
- Schneekluth nozzle — Variant B
- Connection of funnel to deckhouse (incl. B) — Variant C
- Damping tank (incl. B and C) — Variant D
  (Additionally, variant D differed from other variants in that its deckhouse was 2 m taller.)

The main cause of the high vibration level was the excitation of the 4th order, namely, the propeller blade frequency.

For variant C, Fig. 33 shows the waterfall diagram of a speed-up manoeuvre for longitudinal vibrations at the top of the deckhouse. Important orders and significant amplitude changes as a function of revolution rate are recognisable.

Comparison of the four variants concerning longitudinal vibration of the deckhouse is shown in Fig. 34 as an order analysis for the propeller blade frequency.

Through use of the Schneekluth nozzle (variant B), the initial situation was somewhat improved. The aim of the funnel connec-tion, namely, to raise the relevant natural frequency of the deck-house to a value above that of the 4th-order excitation at nominal speed, was achieved.

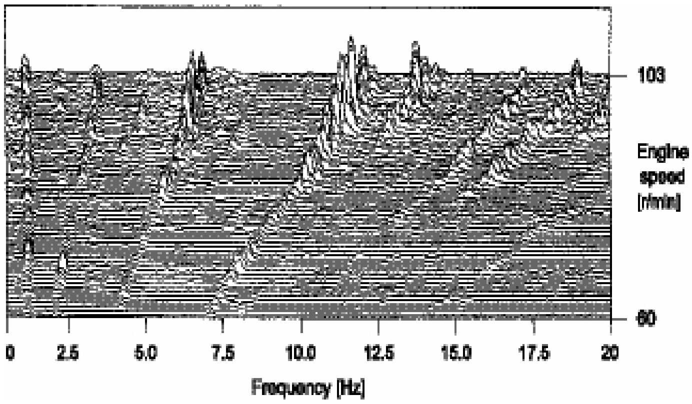


Fig. 33: Waterfall diagram for the longitudinal acceleration of the bridge deck

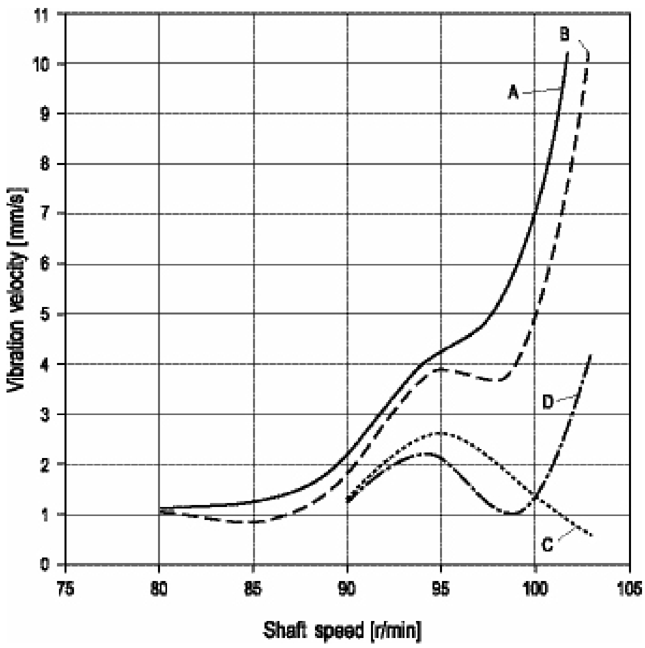


Fig. 34: Order analysis of the propeller blade frequency

In the speed range up to 100 r/min, variant D shows a further reduction in the vibration level, but above that speed there is again a steep rise (i.e. vicinity of resonance) up to 103 r/min. The amplitude reduction below speeds of 100 r/min is due to the damping tank, whereas the steep rise for this variant is attributable to the taller deckhouse. As a result of the increased height of the deckhouse, the natural frequency has fallen again. Due to this, stiffness effect of the funnel connection was compensated to a certain degree.

The investigation firstly underlined that extensive measurements above the standard scope – possibly in conjunction with theoretical analyses (FE computations) – can contribute significantly towards optimisation of the vibration behaviour. Secondly, it turned out to be advantageous for these measurements to exceed the nominal speed range as far as possible, so that a relevant danger of resonance could be detected in this range.

Example 5

For a diesel generator unit installed on an elastically mounted base-frame, it had to be demonstrated experimentally that there was no danger of resonance between a fundamental vibration mode of the frame and the ignition frequency of 25 Hz.

In a first step, the diesel generator unit and the base-frame were investigated with the aid of appropriate theoretical analyses.

By means of various calculations in which the structure was varied, recommendations for the final design were ultimately obtained.

For this kind of experimental investigation, mechanical unbalance exciters are particularly suitable, since they generate a defined harmonic force. Furthermore, by the use of different masses, this excitation force – which increases quadratically with the revolution rate – can be varied within certain limits. Naturally, it must be ensured beforehand that the exciter's range of force and frequency is appropriate for the vibration question concerned.

Because vertical and horizontal vibration modes were important here, the exciter unit was mounted in such a way that both horizontal and vertical forces acted on the base-frame at points above the elastic mounting (Fig. 35).

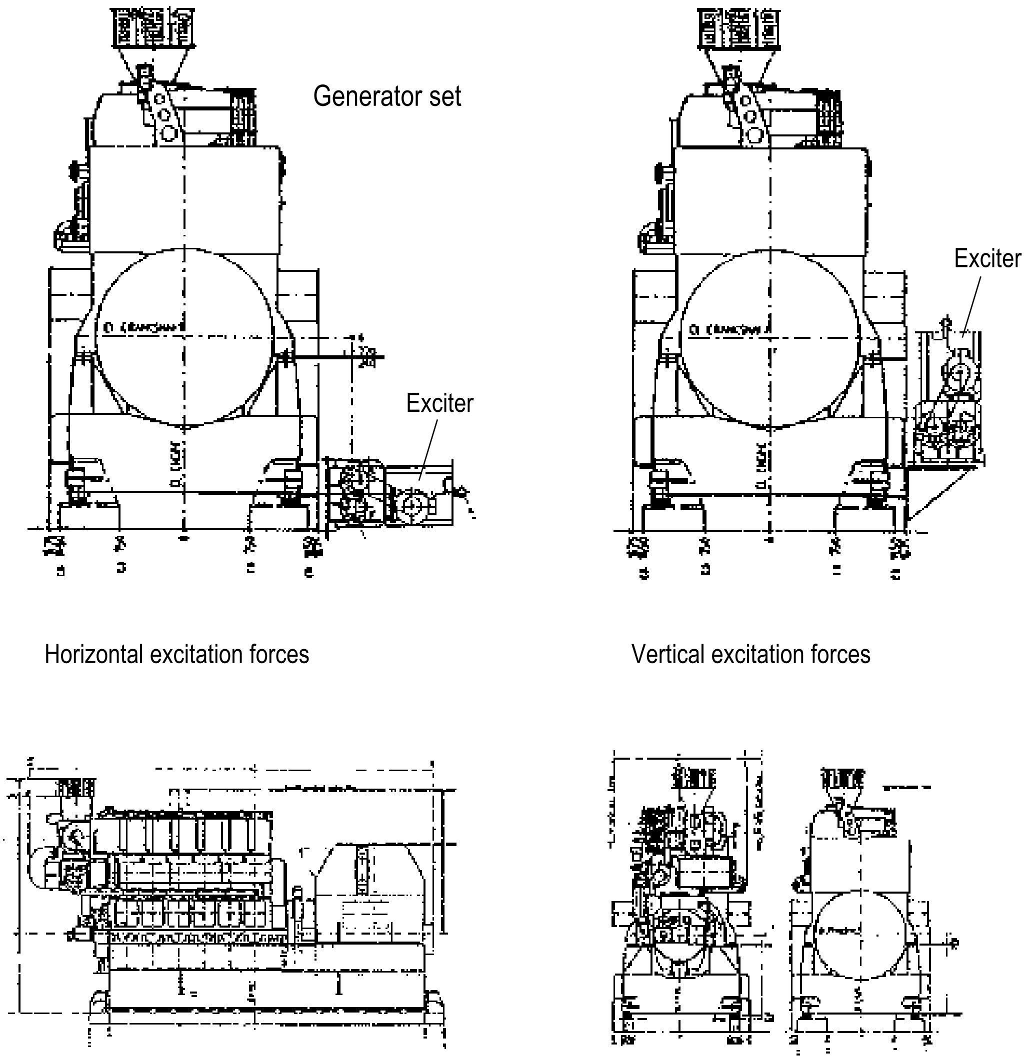


Fig. 35: Arrangement of the exciter

In each case, signals of seven accelerometers and a load cell between the exciter and the frame structure were recorded simultaneously, while the exciter was slowly passing through the frequency range from about 10 Hz to 50 Hz. In all cases, the measuring direction of the accelerometers corresponded to the force direction of the exciter.

During each measurement, coherence and transfer functions of the load cell as well as of one significant acceleration signal were checked with the aid of a frequency analyser to monitor the statistical dependence of the two signals.

The main results are summarised in the following Table 5:

Table 5

Vibration mode	Natural frequency [Hz]
Rigid body, about the transverse axis	10.5
Rigid body, about the longitudinal axis	13.4
Rigid body, vertical	12.1
Frame, vertical bending	31.8
Frame, horizontal bending	41.5
Frame, torsion	46.2

Natural frequencies of the rigid body modes were only of secondary interest here. Frequencies below 10 Hz could not be generated by the exciter.

As an example, the result of the modal analysis of the vertical frame bending mode is shown in Fig. 36.

FREQ.
(Hz)
31.79

Vertical bending

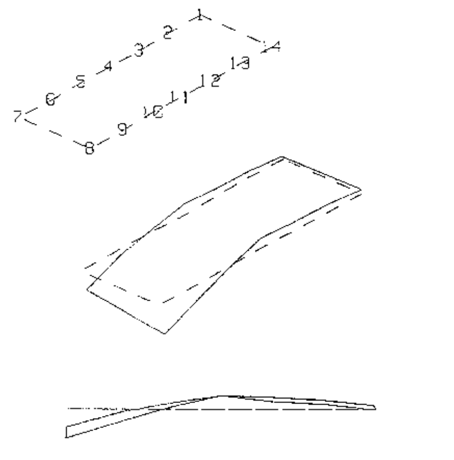


Fig. 36: Modal analysis

There was, consequently, no danger of resonance with the ignition frequency of the engine (25 Hz). The results determined theoretically were thus confirmed.

The procedure for determining natural frequencies by means of a mechanical unbalance exciter is basically suitable for a large number of elastic systems, for example subsystems such as rudders, shaft bossings, propeller nozzles, propellers, turbochargers and so on, but also for plane structures such as panels. The disadvantage is the comparatively large effort needed for mounting the exciter on the structure involved and the exciter's limited range of force and frequency.

Example 6

Strong vibrations in the aft part and in the deckhouse of a small container ship came to light during a first sea trial and were attributable to the propeller as source of excitation.

To investigate the behaviour of the propeller, including the magnitude of pressure fluctuations, extensive measurements were performed during a second sea trial. In addition to pressure pulses acting on the ship's shell in the vicinity of the propeller, parallel measurements of shaft power and mechanical vibrations at various locations were performed to obtain a comprehensive picture of cause and effect.

Of crucial importance was the question how the ship's vibration behaviour would change as a result of the greater draught during the second sea trial. Greater draughts often reduce the vibration level, and the shipping company can often accept the worse behaviour at ballast draught in case this condition plays no significant role in the future lifetime.

The following operating conditions were investigated during the second sea trial:

- Speed-up n = 130 to 174 r/min, propeller pitch 100%
- Constant speed n = 174 r/min, propeller pitch 10-100%
- Constant speed n = 174 r/min, propeller pitch 100%
- Constant speed n = 174 r/min, propeller pitch 70-100% (with lower draught, 1st sea trial)

The waterfall diagram of pressure pulses (Fig. 37) shows the dominance of the propeller blade frequency during the speed-up process, which was performed in steps of 5 rpm.

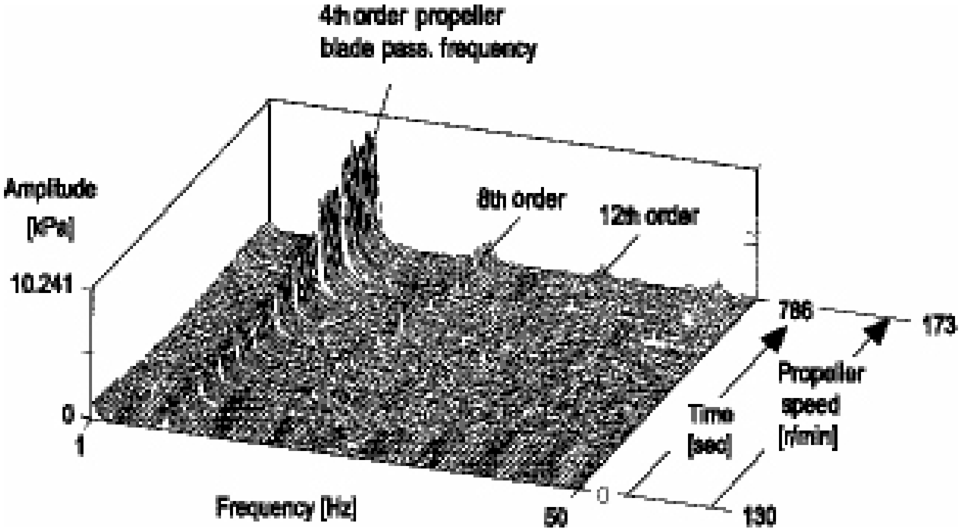


Fig. 37: Waterfall diagram of the propeller pressure fluctuations

Pressure pulses of three measurement points P1, P2 and P3 as a function of power exhibit a steeper rise from about 4,700 kW power, or 160 rpm, upwards (Fig. 38).

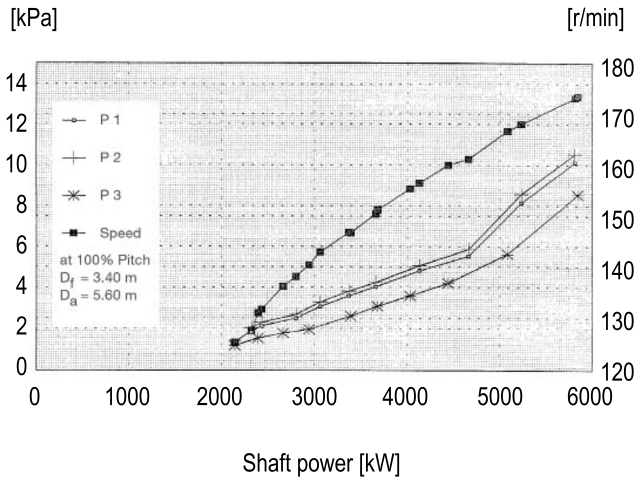
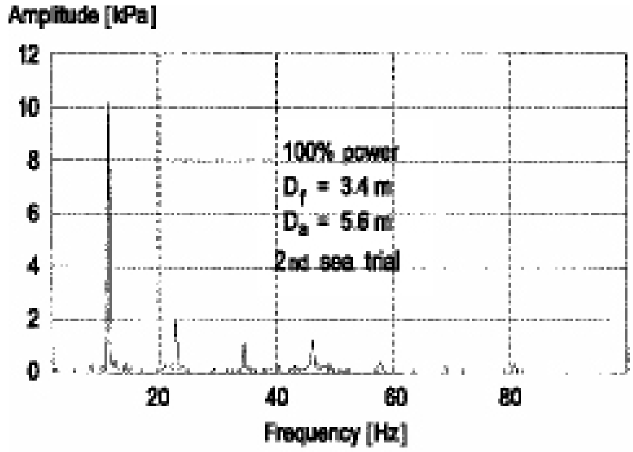


Fig. 38: Pressure fluctuations as a function of power



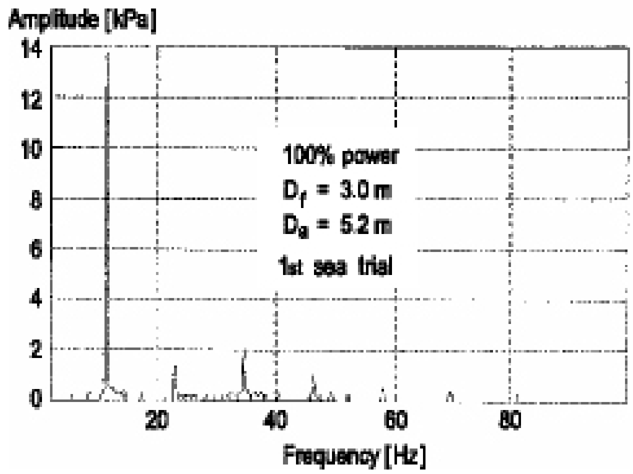


Fig. 39: Pressure fluctuations of P1 for the two draughts

The effect of the draught on pressure pulses at propeller blade frequency is pronounced. Amplitudes increase from values which are already high (10 kPa), by almost 40% to 14 kPa for the lower draught investigated (Fig. 39).

Discussions about the propeller design revealed in retrospect that the propeller was adequately designed for the power of 6,000 kW from a strength point of view. However, a reduced power of 4,500 kW was used as basis for the hydrodynamic design.

This example shows the great extent to which pressure fluctuations depend on the design point and how a ship's vibration level can be influenced by an inadequate propeller design.

In this case, an acceptable vibration level was achieved by a new propeller.

6. Conclusions

The foregoing remarks show how questions regarding ship vibrations can be dealt with comprehensively from a contractual, theoretical and experimental point of view. The procedures outlined can be applied to treat vibration questions on a rational basis at the design stage. They can, furthermore, also be used to solve vibration problems on ships already in service.

The paper documents the “state of the art” in the field of ship vibration technology. It is intended to be used as a manual in the daily work at shipyards, for inspectors of shipping companies, in engineering offices, and so on. By presenting a wide range of knowledge in this field, the paper contributes to prevent vibration problems on newbuildings as well as to find the most cost-effective solutions for vibration problems occurring on ships already in service from a well-planned measurement action.

The subjects dealt with certainly do not cover the entire field of vibration technology. The treatment of further topics, such as

- torsional vibrations of shafting systems
- elastic mounting of engines and equipment items
- sloshing
- slamming (whipping and springing)
- shock

are beyond the scope of this document.

Details of these subjects may be found in the literature mentioned, although this list does not claim to be complete. It is recommended that interested parties keep themselves informed of international activities, in particular, such as the “International Ship & Offshore Structures Congress”, ISSC.

Finally, we thank all those shipyards and shipping companies that kindly gave us permission to present some results, including the FE models.

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