Tanker Structure Co-operative Forum

Vibration Information Paper

SUMMARY

The paper reviews the main sources and types of vibration in ships and provides details of their impacts on habitability, safety, and equipment integrity. Modern vibration analysis and prediction methodologies, industry acceptance criteria, and standardized measurement procedures are discussed, and practical solutions for mitigating vibration problems from the newbuilding stage through to operation are offered. The practical case studies included in the appendix illustrate a variety of vibration defects and demonstrate effective mitigation strategies.

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Contents

1 Introduction

It is recognized that vibration of ship structures can lead to structural damages and create a poor working environment for the crew. The aim of this paper is to provide a “layman’s” guide to vibration for superintendents, inspectors, staff naval architects, and others. outlining how to measure, analyze, and mitigate it.

1.1 Sources of Vibration

There are many vibration sources for a vessel, ranging from those due to the ship’s environmental factors, such as wind and waves, to man-made excitation sources, such as propeller and machinery. In this section we describe the most common sources of vibration and how they affect the vessel and its crew.

1.1.1 Propeller

The propeller is a common source of global and local vibration in a ship. Vibration from the propeller can be generated in two ways; either from pressure pulses (sometimes due to cavitation which can be also met over a large frequency bandwidth) impacting the underside of the ship’s hull or from thrust pulses transmitted through the propeller shaft.

1.1.1.1 Propeller Induced Pressure Pulses

In an ideal world the propeller would experience a homogenous flow throughout its radius and in all parts of the propeller disc. The presence of the ship’s hull in front of the propeller causes the flow into the propeller varies along the blade radius and as it rotates around the propeller disc, which is the virtual area swept out by the propeller blades. Figure 1-2 shows how the axial and tangential velocities of the water flow into the propeller, also known as the wake, vary along the propeller radii and sectors of the propeller disc. The colored contours indicate the axial velocity, with a value of 1.0 indicating an axial velocity equal to the ship’s speed (in this case 15 knots) and a value of 0 indicating no axial velocity. The dotted radius line represents the extent of the propeller disc.

The propeller blade’s design is varied along its radius to match the local characteristics of the wake at that blade radius. If the wake is constant at each radius of the propeller disc, then each radius of the propeller will always be correctly designed for the wake it experiences. However, if the wake is not constant then, as the propeller rotates, the wake experienced by the propeller at each radius will change resulting in the propeller not being correctly designed for some parts of the wake. Figure 1-2, represents a very good wake field with consistent axial velocity at each propeller radius and limited areas of low axial velocity (yellow and orange areas). However, even for this wake field you can see that the part of the propeller blade represented by the red circle (approximately 70% of blade radius) experiences very different wakes as it rotates around the propeller disc.

When the propeller enters a part of the wake with low axial velocity, cavitation can develop on the blade. As this cavitation moves downstream of the blade the increasing hydrostatic pressure causes the cavitation to collapse creating a pressure pulse, see Figure 1-3. This pressure pulse can impact the ship’s shell plating above the propeller and thus induce vibration. Even in the absence of cavitation the propeller can induce pressure pulses which can induce vibrations. Due to the variations in the wake field the pressure on each blade will vary as it rotates through the wake. In addition, if the clearances between the blade tip and the hull are too small the hull will experience a pressure pulse at each blade tip pass.

The excitation frequencies from the propeller are related to the frequency at which the blades pass through these areas of low velocity and are known as blade passing frequencies. Typically, the strongest propeller-induced vibrations in hull structures can be found at the first blade pass frequency, with vibration levels reducing for subsequent blade pass frequencies (i.e. second, third, fourth etc.).

The lowest excitation frequency is at the propeller RPM times the number of blades, see Figure 1-1.

$$\boldsymbol{Blade\ Pass\ Frequency} = \frac{\boldsymbol{Number\ of\ Blades * RPM}}{\boldsymbol{60}}$$

FIGURE 1-1 - FIRST BLADE PASSING FREQUENCY

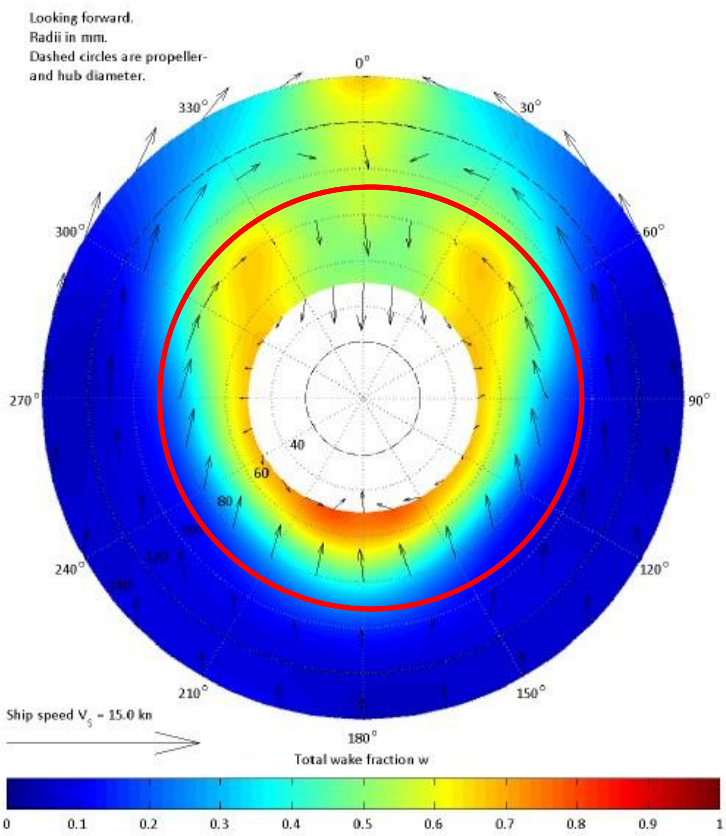


FIGURE 1-2 WAKE FIELD

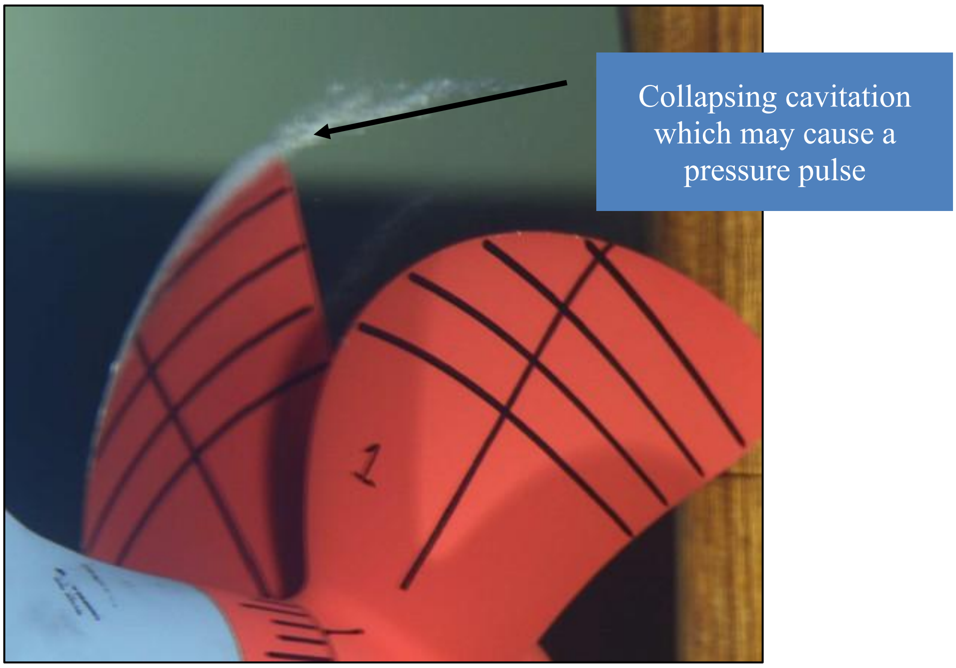


FIGURE 1-3 COLLAPSING CAVITATION

1.1.2 Machinery

There are several pieces of machinery within a ship that could be a source of vibration, for example main and auxiliary engines, emergency generators, turbines, turbos, compressors, etc.

1.1.2.1 Main Engine

The primary source of machinery vibration issues originates from the main engine. Vibration from the main engine typically comes in three forms and is shown pictorially in Figure 1-4.

i) External moment
ii) H moment
iii) X moment

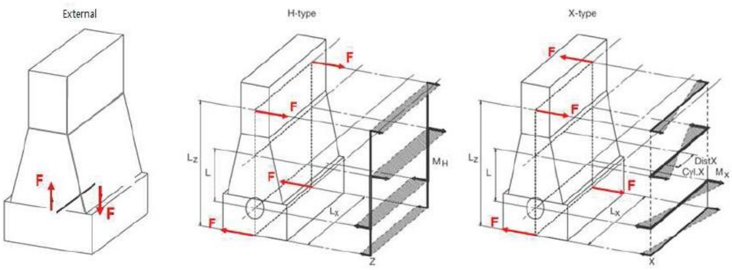


FIGURE 1-4 ENGINE VIBRATIONS

The external moment comes from imbalances in the mass forces of the various moving/rotating parts of the engine (crankshaft, conrods, piston heads, counterbalances, etc.). The H and X moments originate from the internal forces within the engine resulting from the expansion of the gas within the cylinders. The engine manufacturer will provide the frequencies and magnitude of the various orders of excitation for each of the three types listed above.

For a given engine design, both the unbalanced forces and the excitation frequencies will depend on the number of cylinders. A low number of cylinders gives a lower excitation frequency, which is more likely to be close to the natural frequency of the structure. This, combined with the higher imbalanced forces, increases the likelihood of experiencing vibration issues.

1.1.2.2 Other Machinery

Other equipment such as auxiliary engines, thrusters, anchoring equipment and pumps can be sources of local vibration. For example, the power packs (electric motors and hydraulic pumps) for hydraulically driven submerged cargo pumps can introduce high frequency vibrations. These vibrations may reach a significant distance away from the power pack location. High frequency vibrations from these power packs have been observed in on-deck piping that required additional bracing to prevent damage.

1.1.3 Environment

The environment in which the ship is operating can induce global and local vibrations from two sources: waves and wind. Wave forces can produce two forms of vibration, commonly named ‘springing’ and ‘whipping’.

1.1.3.1 Springing

Springing is the term given to the two-node vibration, shown in Figure 1-5, resulting from wave loads acting on the hull girder. It occurs when the encounter frequency of the waves is equal to, or a multiple of, the natural frequency for this vibration mode. The natural frequency of the hull girder decreases with length and therefore as ships get longer, waves of longer wavelength and more energy synchronize with the hull girder. However, the occurrence of springing can be removed relatively easily by changing the encounter frequency through a change of vessel speed or heading.

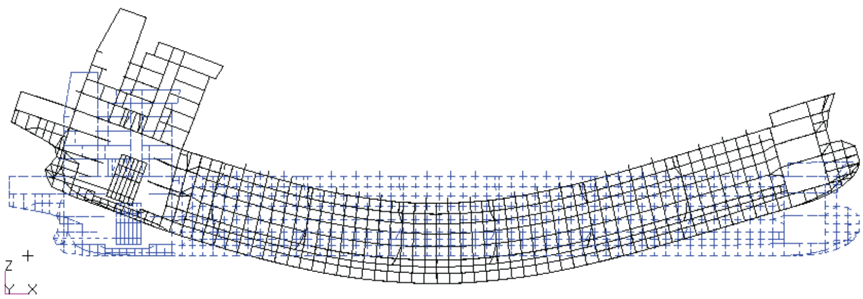


FIGURE 1-5 VIBRATION MODEL SHOWING SPRINGING

1.1.3.2 Whipping

Whipping is the term given to the rapid, transient excitation of the hull structure due to a slamming and/or bottom impact event at the ship's bow. These impacts create a high frequency vibration that can be large in magnitude but quickly fades away. Whipping is believed not to be a significant source of vibration for tanker hull forms.

1.1.3.3 Wind Induced

Wind may cause resonant vibration of posts, such as the foremast, due to vortex shedding excitation forces. This can occur at relatively low wind speeds if the vortex shedding occurs close to the natural frequencies of the post or mast.

1.2 Types of Vibration

1.2.1 Global Vibration

The term 'global vibration' primarily refers to the vibration of the hull girder. It can occur in several modes, characterized by different numbers of nodes and directions, such as vertical, horizontal, or torsional. Typically, global vibration results from excitation forces from the propeller and main engine. In addition to exciting the hull girder, these vibrations can also cause global vibration of the engine casing and accommodation areas. A ship's hull will respond to exciting forces and moments as a beam freely supported in water. Hull vibration modes are described by the plane of vibration and the number of nodes (points of no deflection) as shown in Figure 1-6, Figure 1-7 and Figure 1-8.

Frequencies are low, typically 1-10 Hz. This excitation may come from engine external forces or moments, propeller pressure impulses or, occasionally, by waves. Wave excitation frequencies are typically 1-2 Hz.

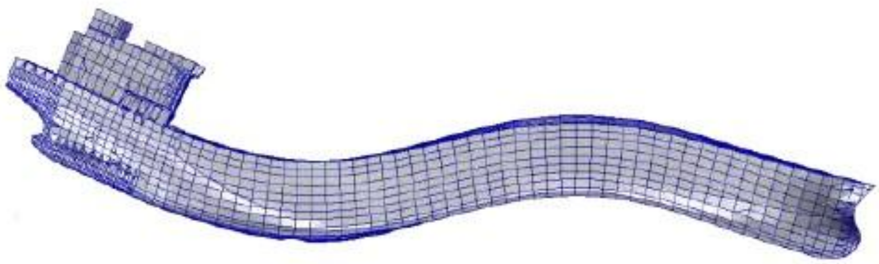


FIGURE 1-6 3 NODE VERTICAL MODE

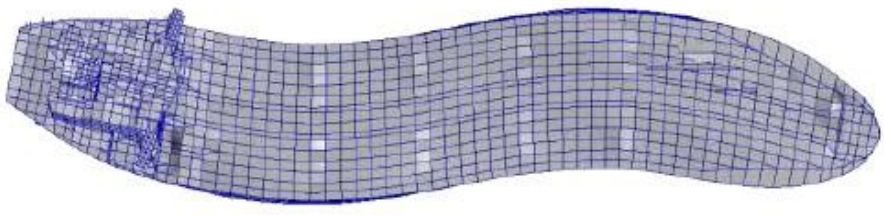


FIGURE 1-7 3 NODE HORIZONTAL MODE

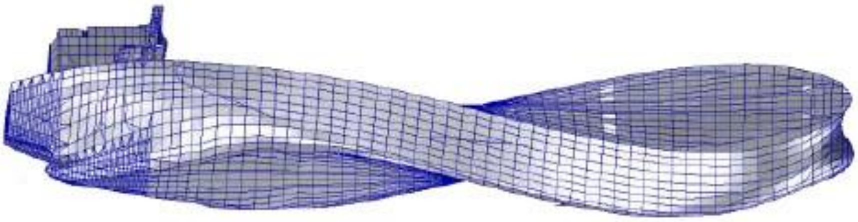


FIGURE 1-8 1 NODE TORSIONAL MODE

1.2.2 Local Vibration

The term local vibration defines vibration of local structures, for example individual deck plates, bulkhead panels, lifeboat davits, lighting posts, masts, cranes, pump towers (on LNG ships), etc. Local vibration can be excited by the propeller and main engines but also other machinery such as pumps, compressors, auxiliary engines, turbos, etc.

During design, the Shipyard may check resonance of the local structures with propeller and main engine excitation frequencies, using either finite element models or simple 2-D plate assessments (See Figure 1-9 and Figure 1-10).

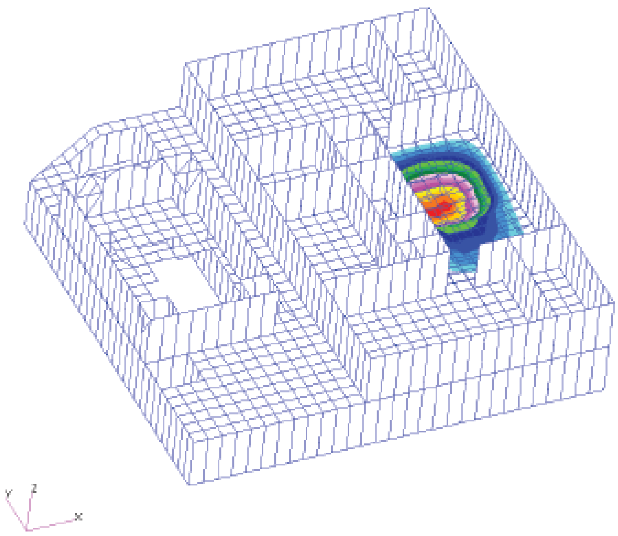


FIGURE 1-9 FE MODEL OF ACCOMMODATION BLOCK

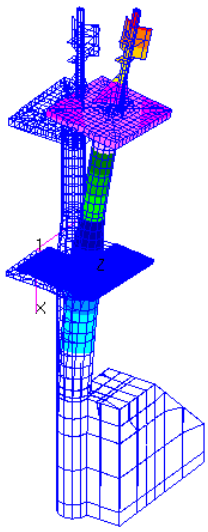


FIGURE 1-10 FE MODEL OF FORE MAST

1.3 Problems Associated with Vibrations

1.3.1 Habitability and Safety

Vessel safety is determined by the crew’s performance which is impacted by their working environment. When high levels of vibration are experienced this could lead to an inability to perform the intended work satisfactorily due to crew fatigue, lack of sleep etc.

1.3.2 Structural Issues

Where ship structures exhibit significant vibrations, structural damage can occur such as cracks on tank bulkheads, engine seats, pillar connections, cross ties, davit structures, etc. Classification Societies provide guidance (See Section 3.2) on the recommended maximum vibration velocities for structures to prevent damage. However, these recommendations are not mandatory and may not be applied by a Shipyard unless specified by the Shipowner.

1.3.3 Machinery Issues

Propeller shaft torsional vibration is not covered by this information paper as it is not associated with the hull structure. However, high levels of torsional vibration in the propeller shaft can mean that certain main engine RPMs must be avoided to prevent resonance of the propeller shaft. These RPMs are known as barred speed ranges and, if they lie in the operating speed range of the vessel, they must be transited as quickly as possible.

High levels of local vibration can have significant adverse effects on ship machinery. Electronics and printed circuit boards are particularly susceptible to failure from repeated mechanical loading due to vibration. Soldered components rattle loose and wire terminations shear. The resultant failure often comes with minimal warning and can have a swift operational impact. Equipment located in the bow compartments can be particularly exposed to excessive vibration during bow thruster and anchoring operations. Equipment mounted in these locations must often be protected from vibration or have internal components that are firmly secured. Figure 1-11 shows a speed log electronic enclosure that has suffered multiple failures due to vibration damage. A faulty speed log will detain a vessel in some jurisdictions.

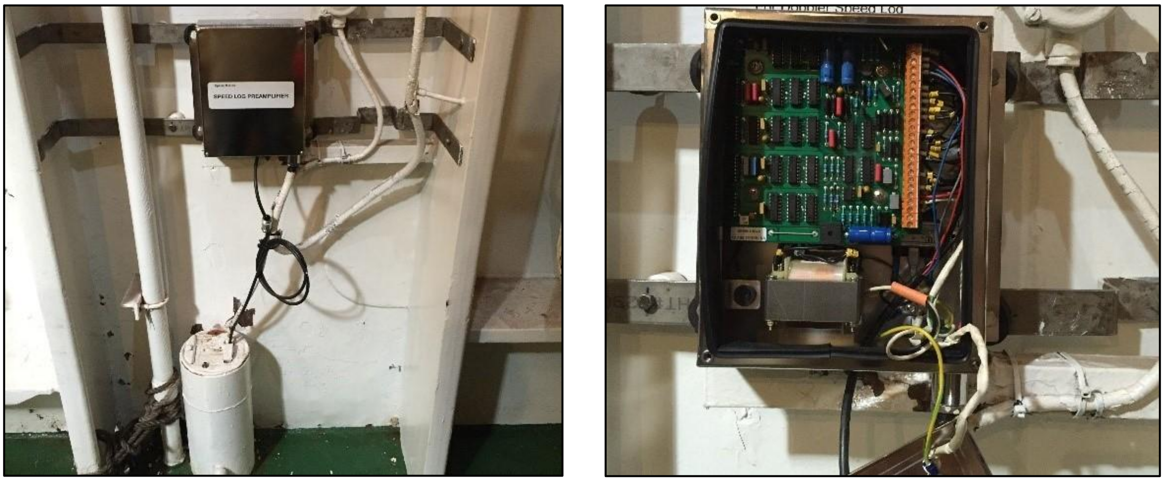


FIGURE 1-11 SPEED LOG CONTROL CABINET

2 Vibration Analysis and Prediction

Ship vibrations are divided into two main categories: global vibrations and local vibrations. Global vibrations as further described in Section 2.1.

There are two basic types of vibration analysis: modal analysis and forced response. Modal analysis yields natural frequencies and vibration mode shapes of a system. It is associated with solutions of the undamped, unforced system of equations i.e. the system is vibrating freely. The natural frequencies of the structure can be compared with known excitation frequencies. If the structural frequencies are close to the excitation frequencies (+/- 10%) then vibration may become a concern. This type of analysis doesn't give any indication of the actual response level but is used to identify potential problems or in "trouble shooting" if problems first arise.

In a forced response analysis, the actual cyclical excitation forces are applied to the model. This will give the actual response of the structure. Very often it is difficult to know the actual forces. For example, hydrodynamic forces due to the propeller blades passing the hull are difficult to calculate and are best determined by measurements. Therefore, in some analyses, a unit force is applied to assess the structure's sensitivity to vibrations.

2.1 Global Vibration

The investigation of hull girder vibration is fundamental to the identification of the possible causes of high shipboard vibration.

A finite element (FE) analysis of the global ship structures can be used to determine the natural frequencies of the hull girder and superstructures. Hull girder natural frequencies are dependent on the stiffness and mass of the structure and the virtual added mass of water. They vary with changes in draught, trim, and mass distribution. The number of propeller blades, RPM, and number of main engine cylinders can be chosen to avoid resonance between excitation frequencies and natural frequencies.

The objective of a global vibration analysis is to investigate the ship vibration performance at intended service conditions. Therefore, typical loading conditions, such as the full load and ballast conditions, will be the focus of the vibration analysis. In addition, it is often desirable to investigate the sea trial condition for the purpose of calibrating calculated numerical results with measurements. Typically, considering the analysis efforts, it is recommended that vibration analysis be performed for two selected conditions, either:

- Full load condition and sea trial condition, or,
- Full load condition and ballast condition.

The mass of the cargo and ballast is distributed in the structural model using mass elements. The corresponding added mass and the spring effects due to buoyancy are then calculated and added to the model.

Accurate weight distribution is an important factor in any vibration analysis. Results can change dramatically if the distribution is not properly represented. Generally, for the chosen loading condition, the Longitudinal Center of Gravity (LCG) from the FE model and the LCG from the Trim & Stability booklet is to be within 0.5% of the total ship length.

2.1.1 Methodology

In the past, beam models were often used to represent the global ship structure. This approach is limited to lower vibration modes because three dimensional effects become significant for higher modes.

With increasing computer power and use of large-scale finite element FE model, together with global stress analysis finite element models often being available, FE analysis is now the main vibration analysis method. A typical ship finite element model created for global vibration analysis is shown in Figure 2-1 below.

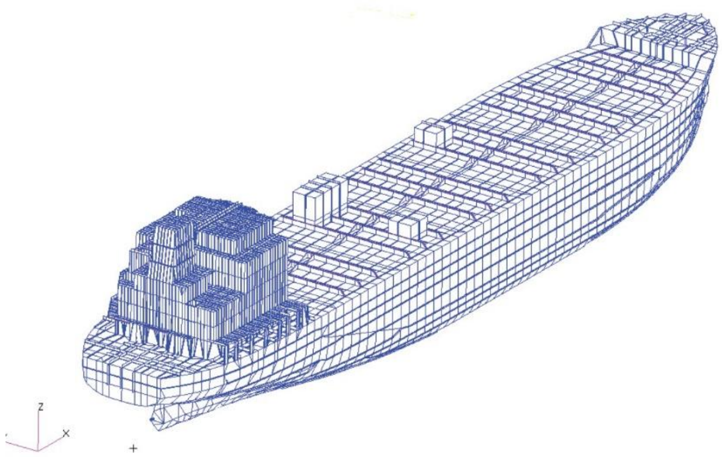


FIGURE 2-1 TYPICAL FEA MODEL CREATED TO ASSESS GLOBAL VIBRATIONS

To be able to predict dynamic responses accurately, it is necessary to know the mass, damping, stiffness and dynamic forces. Damping is the most problematical factor, particularly since any vibration problems usually occur in way of resonances, where responses are inversely proportional to damping. Vibration excitation forces from engines are normally well defined; those predicted for propellers are less certain. Mass is not accurately known because of the added mass of sea water that vibrates with a ship, and the outfit mass on local deck panels.

Interaction between hull structure and machinery/shaft line should also be considered. It is also recommended to model the shaft line (if long) and engine induced loads (usually provided by maker) together with the hull structural model in the predictive calculations.

Generally, a significantly coarser FE model is adequate for global vibration analysis than that which is required for stress analysis. Global FE models that were made for stress analysis purposes are usually of much finer mesh than is required for global vibration analysis. Such a model will have too many local vibrations modes/frequencies. To reduce the multitude of local modes being derived that are not desired or accurately represented the following can be used:

- Dynamic reduction methods (Guyan)
- Restrict eigenvalue extraction to a specified frequency range
- Modal synthesis

To represent the internal fluids in a finite element model, fluid finite elements could be used. Typically, the use of special "fluid" boundary elements may be satisfactory for modeling the vibration of tank boundaries, but not satisfactory when considering the tank contents for global ship vibration. The distribution of tank fluid masses at the tank boundaries is satisfactory in relation to vertical and horizontal ship vibration modes, but not for hull torsion modes because of the incorrect mass radius of gyration. However, hull torsion modes are usually not significant for tanker vibration. Using fluid finite elements should be satisfactory for small tanks when studying global vibrations.

Reliable estimates of natural frequencies which might be excited at the service speed can be made using a ship specific Campbell diagram, see Figure 2-2.

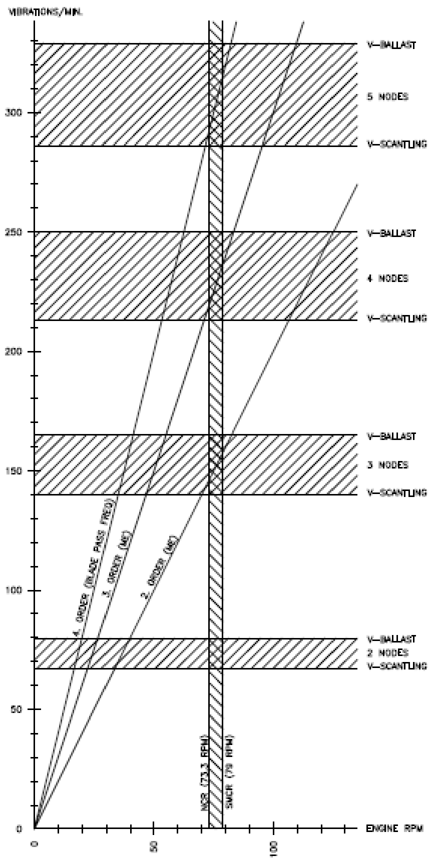


FIGURE 2-2 EXAMPLE CAMPBELL DIAGRAM

A Campbell diagram shows the relationship between excitation and response frequencies. The horizontal axis is the engine RPM and the vertical axis is the vibration frequency. The diagonal lines are the excitation frequencies, for different orders, which are proportional to the engine RPM. The horizontal bands show the natural frequency of the hull girder for different vibration modes. Where the excitation frequency intersects with the natural frequency of the hull girder there is a risk that vibrations will occur. This is most critical at the normal operational RPM of the engine, shown by the vertical shaded band. The intersections of the lines of excitation frequencies with natural frequencies indicate critical speeds, see Section 6.2.1 for further description and calculation example.

2.2 Local Vibration

Vibration results from external force induced cyclical movement of objects restrained by a restoring force - the classic mass/spring/damper system subject to a periodic force.

Damping is usually the most difficult property to determine. For large structures damping is usually a result of structural "friction" resulting from attached flooring/wall covers, etc. the welding joints, piping and other attachments and connections which rub and flex absorbing the vibration energy of the larger structure. Small machines may have rubber pads or shock absorbers for damping.

Machines will often have higher super-harmonics (e.g. number of cylinders times the RPM, crank shaft rotation) or lower sub-harmonics (e.g. half the rudder vortex wake frequency, main engine combustion forces).

5-cylinder low-speed diesels are known to have high unbalanced forces and therefore need to be considered carefully.

The larger the machine (e.g. main engine, prop) the more important at least the first or second super-harmonic. The main engine causes both vertical (M_{2V}) and horizontal (rocking guide M_{6H}) excitations. Table 2-1 shows typical excitation frequencies to consider - vibration cycles are measured in Hertz or cycles per second.

EXCITATION SOURCE	FREQUENCY [Hz]	HARMONICS
Main engine (M/E) RPM	1-2	2-4, 3-6
Propeller blades times M/E RPM	4-8	8-16
M/E cylinders guide force	5-16	
Auxiliary Diesel Generator (D/G) RPM	12-30	
D/G cylinders times D/G RPM	72-270	
Steam cargo pumps	12-25	
Inert gas fans RPM	30-60	
General sea water pumps	25-60	
vortex wake (air/water) 0.2x 10 m/s / 0.2 m	10	

TABLE 2-1 TYPICAL EXCITATION FREQUENCIES

Depending on the object being analyzed there may be a nearby motor which is typically running at super or sub harmonic of the ship’s power supply frequency (e.g. 120 or 15 Hz). For example, a local vibration of an air supply fan could be at 1200 RPM or 20 Hz. Measurement of the vibration spectra (a Y-axis excitation force versus an X-axis frequency) will often identify the source of the vibration.

The excitation forces by themselves are often low and do not cause concern. However, if the forces are within +/- 10% of the resonant frequency of the vibrating object then the vibration will be amplified and may result in damage.

To determine what these resonant frequencies are and whether proposed mitigation measures might be effective, a model of the vibrating object is needed. The resonant, or natural, frequencies of an object depend on both mass/stiffness of the object and on its boundary conditions, i.e. how well it is clamped/welded to its surroundings or "free" from them.

Typically, the high energy vibration source frequencies are tabulated and, based on their expected influence distance, the structures within the influence region are identified and the designers should design the structure’s natural frequencies outside the excitation frequency range.

Complex boundary conditions, 3D geometry, curved and out-of-plane geometries can couple resonant modes in the various directions resulting in flexural-torsional vibrations. A ship’s longitudinal vibrations can end up causing vibrations in transverse structure. The excitation forces themselves often are multi-axial.

Despite practical complications, simple classical models for cantilever beam and plate can provide a framework in which to qualitatively understand many instances of local ship vibrations. The plate breadth or length is generally 10-50 times the thickness and thus the simple Bernoulli-Euler formulations (shear and rotary inertia ignored) are appropriate.

These simple models can also be used to test more complex models e.g. Finite Element (FE) models and analysis. The most important vibration characteristics can be obtained from simplifying the vibrating object to one of these classic models. A FE analysis of qualitative behavior can be checked against that of the classic model. It can also help in finding the appropriate FE model meshing. Anomalous behavior likely means the FE model and analysis is incomplete or incorrect.

The values of a model’s parameters and its boundary conditions may be either calculated or determined from experimental measurements of the vibrations using the hammer/impact test or a frequency sweep which will help to determine the resonant frequencies which can then be used with the model to determine its parameters and/or boundary conditions.

Steel properties are considered the same for all grades of steel although there may be a slight (5%<) increase in both Young Modulus and Poisson’s Ratio for steel with higher tensile strengths. Commonly accepted steel property values are specific gravity 7.85, Young’s modulus 210 GPa, and Poisson’s ratio 0.29.

For simple ship structures natural frequencies can be estimated using simple models such as a cantilever beam. Cantilever beams can be used to approximate frequencies of bridge wings, radar and light masts, and cranes/kingposts.

Any long beam, especially with a large end mass and weak intermediate supports, may be modeled as a cantilever. Generally, each natural frequency is a multiple of the first natural frequency. As an example, the natural frequencies and mode shapes can be calculated for a uniform cantilever beam using the formulas in Figure 2-3. The first 3 modes for the cantilever beam are shown in Figure 2-4.

Natural frequencies(f_n) and mode shapes(M_n). n=1,2,3,...

$$2\pi f_n = \beta^2 \left\{ \frac{EI}{\rho A L^4} \right\}^{0.5} \qquad \beta_n: \quad \cosh(\beta_n L)\cos(\beta_n L) = -1$$

$$\beta_1 = \frac{1.875}{L} \qquad \beta_2 = \frac{4.694}{L} \qquad \beta_3 = \frac{7.855}{L}$$

$$R_n = \frac{\cosh(\beta_n L) + \cos(\beta_n L)}{\sinh(\beta_n L) + \sin(\beta_n L)}$$

$$M_n = (\cosh(\beta_n x) - \cos(\beta_n x)) - R_n(\sinh(\beta_n x) - \sin(\beta_n x))$$

FIGURE 2-3 NATURAL FREQUENCY AND MODE SHAPES OF CANTILEVER BEAM

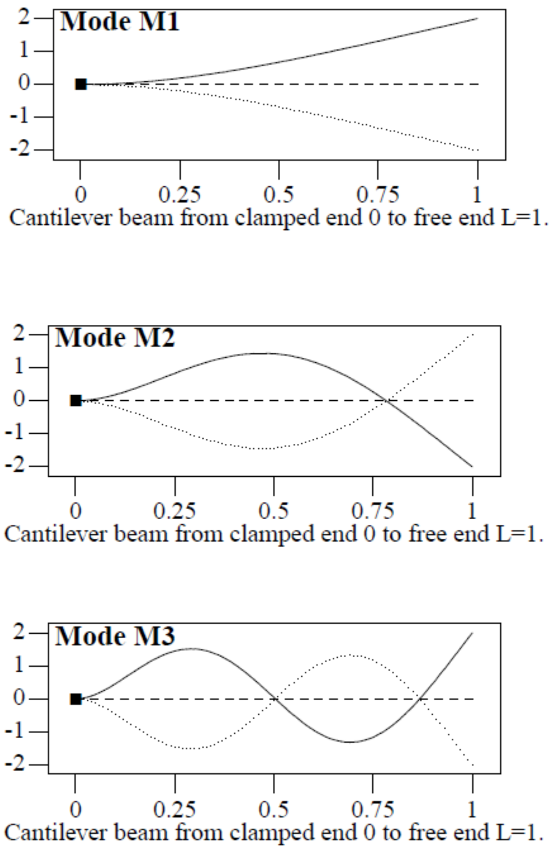


FIGURE 2-4 FIRST 3 MODES OF VIBRATION

For plates such as accommodation decks, working platforms, tank wall plates, and large machine platforms such as a diesel generator platform, any large spanning plate that is not supported by or stiffened by an intermediate structure can become a resonant "drum".

Plates are more complex than beams and have natural frequencies in both plane dimensions - for extensive literature see Chladni Plates (Figure 2-5). Their natural frequencies are neither simple multiples of a fundamental frequency nor are their mode shapes simple. The example presented in Figure 2-5 shows many node lines for the lower modes of a free edged square plate as determined experimentally by Chladni (1787, Entdeckungen uber die Theorie des Klanges).

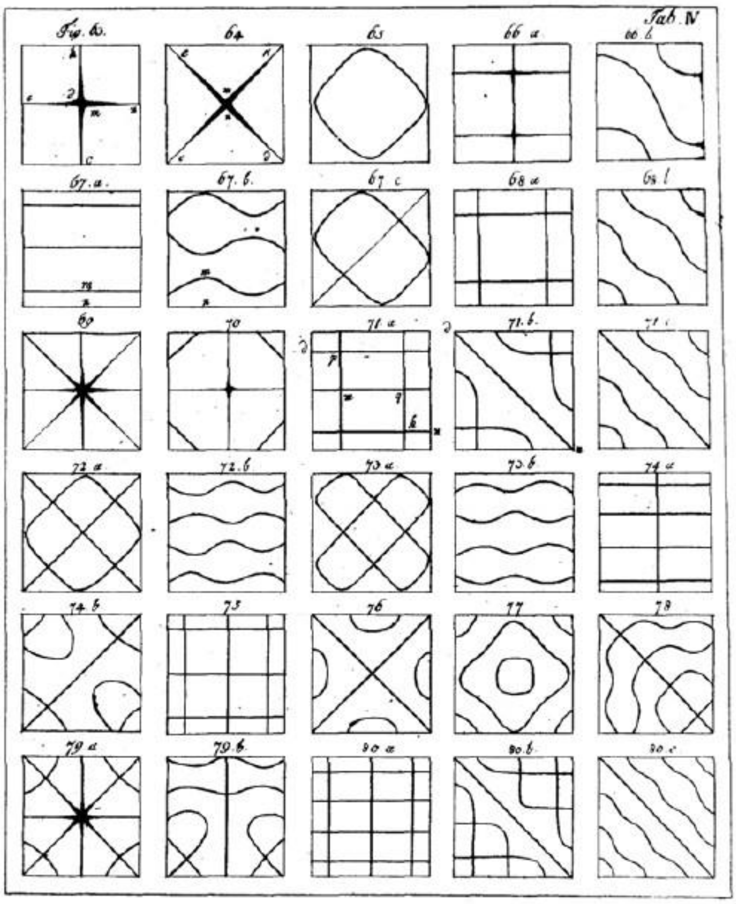


FIGURE 2-5 CHADLI PATTERNS

Because of the complexity of estimating vibration of even simple plates the shipyards with design departments or design houses should use detailed FE models of large ship structures such as: all accommodation decks, engine room decks/platforms, bridge and compass decks, bridge wings, large masts/cranes, and the larger tank bulkheads and stringer platforms.

The details and model mesh size should be smaller than any structure that is likely to be removed, added or significantly changed including large openings. Plate thickness will not determine the mesh size for vibration analysis.

Typically for an accommodation deck or engine room platform the mesh size would be 0.10 to 0.25 times the transverse frame spacing and 0.25 to 1.00 times the longitudinal members spacing but with additional meshing at reinforcements or high mass density areas. Vertically the mesh size should be 0.25 times the inter-deck spacing although a smaller mesh size would be needed to model vertical/accordion-like motion.

The lower (e.g. 1st, 2nd, 3rd) natural frequencies of these structures should be calculated and then compared with the likely excitation frequencies. Structures with natural frequencies too close (typically +/-10% can be used for simplified checks) to those frequencies of the excitation forces within the zone of influence should be modified (generally stiffened or strengthened) to "move" the natural frequencies.

2.2.1 Procedural Checks

The example below is shown to illustrate the conceptual procedure that should be considered in the design and construction of the ship. It is always easier to make changes earlier in the design process, but some changes can be made during fabrication. Modifications after construction are difficult and generally not satisfactory.

Step 1: Excitation Frequencies

We will use the standard +/- 10% as the typical range within which the excitation frequency's influence normally extends, see Table 2-2.

EXCITATION	**FREQUENCY** [RPM / Hz]	**-10% Hz**	**+10% Hz**
RPM at MCR	90 / 1.5 Hz	1.4	1.7
PROP1, 4Blades x RPM	4 x 90 / 6.0	5.4	6.6
PROP2, 4blades x RPM x 2	4 x 90 x 2 / 12.0	10.8	13.2
M/E Rocking, M6H	90 x 6 / 9.0	8.0	10.0

TABLE 2-2 TYPICAL EXCITATION FREQUENCIES

Step 2: Natural Frequencies

Table 2-3 shows a short sample of typical values for a mid-size tanker. They are given to illustrate the procedure; they are not actual data.

STRUCTURE	**NATURAL FREQ.** [Hz]
Bridge Wing	25
Accommodation Deck	17
Engine Room Platform Deck	15
Tank bulkhead	12
Radar Mast	9
Free Fall Lifeboat Davit	5

TABLE 2-3 TYPICAL NATURAL FREQUENCIES

Step 3: Cause for Concerns

Now look at each of the above "Step 2" structures (see Table 2-3) and check if its natural frequency is within +/- 10% of any of the excitation frequencies (see Table 2-2). Table 2-4 presents the identified structures of concerns.

STRUCTURE	NAT. FREQ. [Hz]	EXC. FREQ. [Hz]	EXCITER
Tank bulkhead	12	10.8 - 13.2	PROP2
Radar Mast	9	9.0 – 11.0 10.8 - 13.2	M6H PROP2
Free Fall Lifeboat Davit	5	5.4 – 6.6 10.8 - 13.2	PROP1 PROP2

TABLE 2-4 COMPARISION OF EXCITATION FREQUENCIES

It is noted that the “Free Fall Lifeboat Davit” is stated in Table 2-4 as a structure of concern even though the natural frequency is not within any of the excitation frequency upper and lower limits. The reason is that the excitation frequency upper and lower limits were stated in Table 2-2 for the main engine MCR. When the main engine and propeller run at lower RPMs than MCR, the excitation frequency band will also decrease meaning at some RPM the natural frequency of the Free Fall Lifeboat will be excited.

It is generally expected that the Main Engine and the propeller wake passing frequency at the top and bottom of the propeller are the two biggest excitation forces on a ship and the above are typical. The structures indicated in Step 2 can generally be arranged in two basic groups: decks/platforms and cantilevers (e.g. radar mast, davits, etc.). The methods used to solve vibration issues of structures within these two groups are likely to be different.

Step 4: Decks/Platforms

By the time the vibration analysis is performed the basic design is "fixed" and overall changes in frame spacing or the number of longitudinal members would be challenging. The investigation would focus on the vibration hot spots for these decks and platforms. The mode shapes associated with the hot spot natural frequencies in these decks and platforms, which are part of the FE analysis output, will be used by designers to generate candidate changes for the problem areas. Several candidate reinforcements can be generated and analyzed to see which ones are effective and then from those to select the one that yields the best improvement for the extra steel and fabrication difficulties. Additional plate thickness, sub-frames, girders or local tertiary structures may work but it may only locally move the hot spot rather than shifting away the resonant frequency. The FE model must be run again to confirm that the problem has been fixed and not just moved to another location.

Step 5: Masts/Davits

Although conceptually the way of solving mast vibration hot spots is the same as that used in Step 4 for Decks/Platforms, the fixes are generally easier and more localized. Side effects are less and often the need to be strictly optimal is less important than ensuring easy future maintenance. Typical improvements are increasing the triangular foundation brackets (or adding another), stiffening the cantilever with either internal/external flat bar or, in rare cases, reducing the natural frequency (a less stiff structure). Classical cantilever models are effective in understanding the problem and generating candidate solutions. The FE model is used to test and confirm the solutions.

Models may be used to determine the nature of vibrations and what measures may mitigate it passively (e.g. adding stiffeners, welding free edges/support members, damping isolators) or actively (e.g. vibration motion compensators). It should be noted that cables or wire stays, common on masts, usually do not solve vibration problems because of their inherent lack of rigidity and low tension compared to a piece of steel.

(1) Typical passive vibration mitigation measures are to stiffen the structure by adding either more stiffeners, increasing the thicknesses, or making the structure more rigid (more welding of the boundaries). In unusual cases making the structure more flexible (usually with damping) can be the solution. Damping, typically used for motor-induced vibration, usually means vibrating clamps, rubber pads between a motor and its foundation, etc. It attempts to absorb and dissipate the vibration's energy typically into a "rubber" sandwich although hydraulic shock absorbers or friction slides are typically used in main engine support stays.

(2) A common active vibration compensator consists of eccentrically mounted weights attached at adjustable distances on motor driven rotating shafts. Usually called motion compensators and attached to the object or a connected structural member, these devices create a vibration which is tuned (rotating speed and weight/distance) to be exactly out of phase with the vibration. The original vibration and the motion compensator's vibration cancel each other out. The motion compensator may require sophisticated sensing and vibration adjustments to cover the vibration range.

Another measure is a hydraulic fluid compensator which hardens or softens the foundation or in the case of axial loads, the compliance of the thrust bearing using fluid pressure. The load bearing compensator is tuned to be "out of phase" with the fluctuating axial load and thus actively damp the system. Passive and active compensators are available to reduce torsional vibrations.

2.3 Cavitation Testing

Section 1.1.1.1 outlined how cavitation pressure pulses can be created which could lead to damaging and/or uncomfortable levels of vibration within the ship. Cavitation tunnels can be used to determine whether these levels of cavitation will occur.

Figure 2-6 shows the formation and collapse of the cavitation which can lead to harmful pressure pulses. The cavitation tunnel is used to simulate the conditions of the actual propeller in terms of water pressure and velocity. For the larger cavitation tunnels, the same ship model as used for the resistance and propulsion measurement in the towing tank is used in the cavitation tunnel, see Figure 2-7.

High speed cameras and/or stroboscopic lighting are used to capture how the cavitation forms and collapses at various positions of the propeller. The severity of cavitation is assessed during observations by experienced technicians, in addition pressure pulse measurements can be made. These measurements involve pressure transducers being fitted to the model hull, above the propeller. The pressure pulses measured by these transducers are correlated to full scale pressure pulses which can then be used to determine the probability of problematic levels of vibration being induced in the ship's structure.

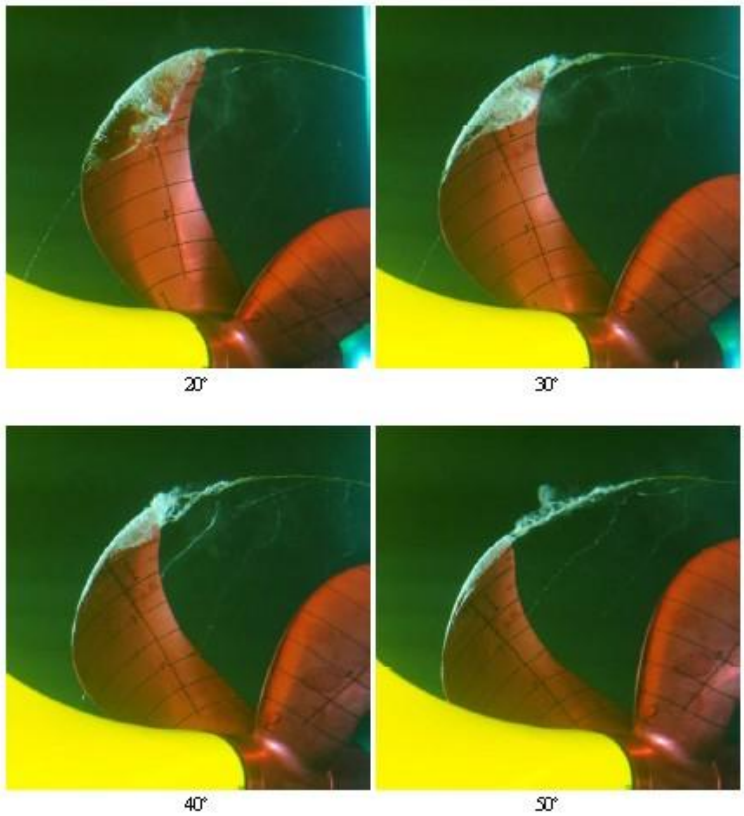


FIGURE 2-6 FORMING AND COLLAPSING OF CAVITATION

FIGURE 2-7 FULL SHIP MODEL IN CAVITATION TANK

If the cavitation tunnel testing indicates that problematic levels of cavitation may occur, then further testing with some of the flow improvement devices described in Section 5 can be performed.

3 Acceptance Criteria and Industry Standards

3.1 Class Notations

3.1.1 General

Each Classification Society has its own requirements and guidelines for vibration evaluation on board ships. Table 3-1 summarizes the scope of different Class Notations and guidelines.

Classification Society		Habitability / Comfort Notation	Guidelines for vibration analysis	Structural and machinery vibrations notation
Nippon Kaiji Kyokai	NK	x	x	x(machinery)
Lloyd's Register	LR	x	x	x
American Bureau of Shipping	ABS	x	x	x
DNV	DNV	x	x	x
Bureau Veritas	BV	x	x	
Korean Register	KR	x		
China Classification Society	CCS	x	x	
Registro Italiano Navale	RINA	x		

TABLE 3-1 CLASS NOTATIONS AND GUIDELINES

3.1.2 Habitability / Comfort

Before the Classification Societies' comfort notations were established, some international standards were already used to define recommended noise and vibration levels criteria. These standards gave the fundamentals of assessing the levels of noise and vibration on board ships.

For over a decade, Classification Societies have included specific notations in their rules to assess onboard noise and vibration. Shipowners and shipyards often find it beneficial to use these standards as objective criteria for contracts.These notations are meant to limit the noise and vibration onboard the ships to a comfortable level

Typically, two different sets of criteria are used for the comfort of the passengers and for the comfort of the crew.

Classification Societies generally base their Comfort Notations on international noise and vibration standards. These notations are typically adapted from the current editions in force, with specific criteria explicitly referencing a particular edition.

3.1.3 Structures & Machinery

Vibration assessment for local structures and machinery is typically not covered by Class Notations and needs to be specifically addressed in the Newbuilding Specification.

However, most Class Societies offer services for vibration measurement and assessment outside of the scope of a Class Notation.

3.2 Classification Society Guidance

Classification Societies do have optional Class Notations in relation to vibration and noise standards.

The following notations are offered by different Classification Societies depending on the level of noise and vibration levels requested.

3.2.1 Lloyd's Register

CAC 1, CAC 2, CAC 3

CAC is Crew Accommodation Comfort. This notation indicates that the crew accommodation and work areas meet the acceptance criteria. The numerals 1,2 and 3 indicate the acceptance criteria to which the noise and vibration levels have been assessed.

VIBM (H)

VIBM (M)

VIBM (H,M)

This ShipRight notation VBM() will be assigned when measurement of vibration has been carried out and verified in accordance with the relevant ShipRight procedures.

Where a vessel has had its hull vibration measured and certified in accordance with LR procedure, the **VIBM(H)** notation will be assigned.

Where a vessel has had its machinery vibration measured and certified in accordance with LR procedure, the **VIBM(M)** notation will be assigned.

Where a vessel has had both hull and vibration measured and certified in accordance with LR procedure, the **VIBM(H,M)** notation will be assigned.

3.2.2 DNV

COMF (V-crn)

The comfort rating number, crn, indicates the comfort rating of noise and vibration for the vessel. The crn may have the value 1, 2, or3, where 1 represents the highest comfort level.

The additional Class Notation **VIBR** is applicable to machinery, components and equipment. **VIBR** is also applicable to the structure in compartments where machinery, components and equipment are situated close to the propeller(s).

The **VIBR** notation is based on measurements on board and given criteria for maximum vibration velocity. The purpose of the notation is to reduce the risk of failure in machinery, components and structures on board ships, caused by excessive vibration.

3.2.3 BV

COMF-VIB-Crew

COMF-VIB-Pax

COMFNOISE-Crew

COMF-NOISE-Pax

The Class Notations **COMF-NOISE-Pax**, **COMFNOISE-Crew**, followed by grade **1**, **2** or **3** and **COMF-VIB-Pax** and **COMF-VIB-Crew**, followed by grade **1**, **2** or **3** or **1PK**, **2PK** or **3PK** are assigned to ships satisfying levels of noise / vibration defined in the BV Rules for passenger or crew area, as applicable.

3.2.4 CCS

G-ECO (VIB1),G-ECO (VIB2),G-ECO (VIB3)

The VIBx sub-element notation may be assigned to ships complying with the relevant requirements upon measurement, where x represents the comfort grades 1, 2, 3, where “1” means an acceptable grade and “3” means the highest grade.

Technical requirements for assignment of the notation of sub-element of comfort onboard

(vibration) are as follows:

(1) Vibration measurements are to be carried out in accordance with the requirements of Appendix 1 to the Rules.

(2) If, for all the compartments or spaces, vibration levels are lower than or equal to those corresponding to a given comfort grade, then the granted grade is that grade.

(3) Measured vibration levels slightly greater than those specified in the comfort criteria may be accepted. Not more than 20 percent of measuring points may exceed the relevant vibration criteria by 0.3 mm/s.

(4) The maximum allowable vibration levels for passenger spaces are given in “RULES FOR GREEN-ECO SHIPS”

3.2.5 ABS

HAB, HAB+, HAB++

This notation is assigned to vessels, which comply with the minimum criteria for accommodation area design, whole-body vibration (separate criteria for accommodation areas and work spaces), noise, indoor climate, and lighting. The “+” sign indicates the level of stringent requirements applied.

ABS offer a ship vibration analysis procedure guide, Ship Vibration Guidance Notes, that defines procedures and criteria for ship vibration analysis with the aim of designing vessels free of vibration. The guide is not a requirement for Classification and no notation is offered. Additionally, ABS published the Guide for Vibration of Machinery Equipment and Related Structures. This Guide provides vibration measurement requirements and acceptance criteria for vibration levels of local structures, machinery, and equipment for a vessel to obtain the optional classification notation **VIB-M**.

3.2.6 Class NK

NVC•A, NVC•B, NVC•B’, NVC•C

NVC•A+, NVC•B+, NVC•C+

MVA

The notation "Noise and Vibration Comfort" (abbreviated to NVC) is affixed to the classification characters of ships measuring and evaluating the noise and vibrations found in accommodation space as per the Guidelines. The additional notation “Mechanical Vibration Awareness” (abbreviated to MVA) is affixed to ships which measure and evaluate the vibrations found in machinery installed in engine rooms.

In general, there is not a direct equivalency between the notations offered by different Class Societies as the acceptance criteria and scope differ.

3.2.7 Korean Register

NVH-N1

NVH-N2

NVH-N3

NVH-V1

NVH-V2

NVH-V3

The NVH (Noise, Vibration, and Habitability) notation is granted to ships upon request, provided they meet the noise and vibration criteria specified in the KR Guidance for Noise and Vibration. The notation aims to enhance onboard comfort by assessing and controlling noise and vibration levels in various spaces.

KR offers three different noise categories (NVH-N1, NVH-N2, NVH-N3) and vibration categories (NVH-V1, NVH-V2, NVH-V3), ranked from the lowest to the highest standards.

KR’s guidance includes specific procedures and criteria for measuring and evaluating noise and vibration levels in ships, with the goal of ensuring a comfortable and safe environment for the crew.

3.2.8 RINA

COMF-NOISE

COMF-NOISE (DP)

COMF-NOISE (MM)

COMF-VIB

COMF-VIB (DP)

COMF-VIB (MM)

The notations are completed by letter A, B or C which represents the merit level achieved for the assignment of the notation, the merit A corresponding to the lowest level of noise.

These Class Notations are assigned when the ship achieves the relevant merit level either in normal seagoing conditions or in dynamic positioning conditions (DP) or in maneuvering mode conditions (MM). When the merit levels achieved for the passenger spaces (if any) and the crew spaces are different, the notation is completed by the suffix PAX, for passenger spaces, and CREW, for crew spaces.

In general, there is not a direct equivalency between the notations offered by different Class Societies as the acceptance criteria and scope differ.

3.3 Design and Acceptance Criteria

3.3.1 General

The Shipowner should specify the operating condition(s) for the vibration assessment and the associated level of criteria (for instance the level of Class Notation). More details can be found in Section 6.

Based on the specified criteria, the vibrations are to be assessed in two steps.

1) At the design stage where free and forced vibration analysis are conducted with the aim of avoiding potential vibrations that could generate costly repairs after construction.
2) After construction vibration measurements are carried out during sea trials.

Criteria for assessing the acceptability of the vibration levels from vibration measurements on sea trials have matured substantially in recent years. By incorporating the international standards and industry practices, the vibration acceptance criteria covering three areas are:

i) Vibration limits for crew and Passengers

ii) Vibration limits for local structures

iii) Vibration limits for machinery

3.3.2 Vibration Limits for Crew and Passengers

Vibration acceptance criteria are usually defined in the Newbuilding Specification. The actual vibration acceptance criteria are to be determined as appropriate based on the Newbuilding Specification and manufacturer's specifications.

Class Societies provide acceptance criteria based on the evaluation of mechanically induced vibrations.

Generally, the criteria is based on BS 6841 and ISO 2631.

3.3.3 Vibration Limits for Local Structures

Excessive ship vibration is to be avoided in order to reduce the risk of structural damage on the local structures. Structural damage such as fatigue cracking due to excessive vibration may occur on local structures, including but not limited to engine foundation structures, engine stays, steering gear rooms, tank structures, funnels, and masts.

The structural damage due to excessive vibration varies according to the local structural detail, actual stress level, local stress concentration and material properties of the local structures. Therefore, the vibration limits for local structures are to be used as a reference to reduce the risk of structural damage due to excessive vibration during the normal operating conditions.

Vibration limits can vary for local structures based on their configurations and details. For slender or highly flexible structures like masts, the actual stress level due to vibration is generally low, making the typical vibration limits of 1-2 mm potentially too conservative. Conversely, for local stiffened panels with fixed ends, these same limits may be less conservative. Therefore, the application of vibration limits for specific local structures can differ depending on the Newbuilding Specification agreed upon by the Shipyard and Shipowner.

3.3.4 Vibration Limits for Machinery

The main propulsion machinery often experiences severe vibrations due to excitation from the propellers, with particular concern for longitudinal vibrations at the propeller blade rate frequency. Vibrations can also be excited at the cylinder firing frequency. Table 3-2 provides general guidance on the vibration limits for the main propulsion machinery.

Propulsion Machinery	**Limits (rms)**
Thrust Bearing and Bull Gear Hub	5 mm/s
Other Propulsion Machinery Components	13 mm/s
Stern Tube and Line Shaft Bearing	7 mm/s
Diesel Engine at Bearing	13 mm/s
Slow & Medium Speed Diesel Engine on Engine Top (> 1000 HP)	18 mm/s
High Speed Diesel Engine on Engine Top (< 1000 HP)	13 mm/s

TABLE 3-2 VIBRATION LIMITS FOR MAIN PROPULSION MACHINERY

The vibration criteria of machinery and equipment are to be provided by the manufacturers. Otherwise, when the data on the vibration criteria are not available, the following criteria are recommended as a reference in terms of overall rms value (nominally from 1 to 1000 Hz) for normal operating conditions:

1) For reciprocating machinery, the vibration in all directions is to be less than 10 mm/sec rms at bearings
2) For rotating machinery, the vibration in all directions is to be less than 9 mm/sec rms at bearings

The machinery includes but is not limited to generators, motors, centrifugal pumps, compressors, turbochargers, blowers and fans. The application of vibration limits may vary depending on specific type, size, configuration, and mounting of the machinery.

The supporting structure is to be designed to withstand these vibration criteria. The equipment manufacturer may provide guidelines for adequate structural arrangements for the equipment foundations.

3.4 ISO Standards

Figure 3-1 summarizes the applicable ISO standards regarding vibrations.

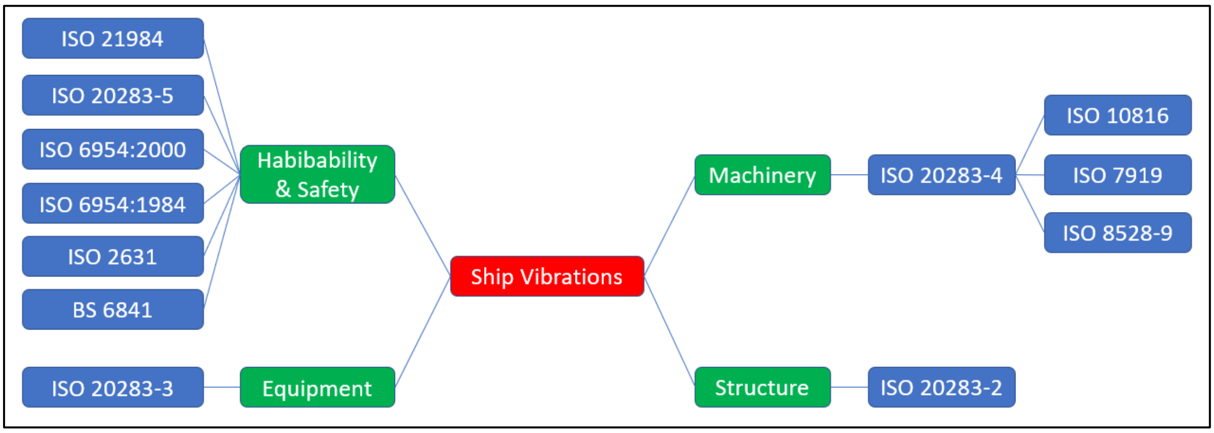


FIGURE 3-1 APPLICABLE ISO STANDARDS FOR SHIPS

An overview of each of these standards can be found in Section 7.

4 Vibration Measurements

4.1 Measurement Equipment and Procedures

A vibration measurement procedure is to be developed by the Shipyard and delivered before any measurements are carried out. This procedure should include at least the following:

1) Preliminary number and location of measurement points
2) Type of measuring equipment
3) Evaluation method
4) Operating conditions
5) Acceptance criteria

Measurements made during sea trial should be compared with the predicted levels and are to be lower than the contractual values specified by the Shipowner.

The following elements are based on:

- Standards ISO 6954:2000, ISO 20283-5 and ISO 21984 for habitability (ISO 6954:1984 was not considered)

- ISO 20283-2 for structure

- ISO 20283-4 for machinery

4.1.1 Measuring Equipment

Measurements can be performed using different types of measuring and recording equipment, e.g. instruments of digital, spectral or time-based type.

For structural vibrations, the use of multi-channel equipment is recommended. If this is not possible, at least two-channel equipment shall be used with one channel reserved as reference channel. The vibration transducers and the signal processing equipment shall be capable of measurement from 1 Hz to 80 Hz with a magnitude accuracy for the entire system of at least ± 5 % and a frequency resolution of at least 0.125 Hz.

For machinery vibrations, as a general guideline, an upper frequency limit of 1,000 Hz is normally used and is enough.

For acceleration measurements on gearboxes, higher frequencies, depending on the excitation forces (gear mesh frequencies), can be appropriate.

The instrument shall be capable of presenting the result while the measurements are carried out. Additional measurements may be needed in case the specified vibration acceptance criteria are exceeded.

For each instrument, proper calibration must be demonstrated before any measurements are taken.

4.1.2 Measuring Locations and Directions

4.1.2.1 Habitability

Measuring locations shall be selected on the decks of occupied spaces in enough quantity in order to characterize satisfactorily the vibration of the ship with respect to habitability. The transducer orientation shall correspond to the three translational axes of the ship: longitudinal, transversal and vertical. Vibration transducers shall preferably be located at positions where work is carried out. In work spaces (e.g. on the bridge, etc.) measurement transducers are to be specially placed at typical duty stations of the crew. In cabins, transducer shall be positioned in the center of cabins or where crew members stay for prolonged periods of time. Vibration transducers shall be located and attached properly to the floor surface such that the vibration at the interface between the person and the source of vibration is adequately captured.

Actual measurement location(s) in each space is to be determined during the measurement

4.1.2.2 Structure

The following locations are generally considered for vibration measurements:

- Superstructure - wheelhouse, bridge wing, aft end, centerline at the front of the deck house at main deck level
- Main engine and thrust bearing
- Local structures depending on evidence of severe local vibrations during sea trials
- Local machinery and equipment depending on evidence of severe local vibrations during sea trials
- Shell near the propeller

The focus is on the determination of the global operational deflection shapes, the indication of important natural vibration modes and on the identification of the dominant vibration excitation mechanisms. Consequently, measurement positions, shall reflect the ship's deflection shape and the energy and frequency content of the main vibration excitation sources, as normally represented by propeller and main engine.

Typical locations for measurements are given in the Table 4-1.

	Location	Measurement direction
1	Stern, port	Transverse
2	Stern, port	Vertical
3	Navigation bridge deck forward, port	Longitudinal
4	Navigation bridge deck forward, port	Transverse
5	Navigation bridge deck forward, starboard	Longitudinal
6	Navigation bridge deck forward, port	Vertical
7	Superstructure fore, foundation, centre line	Vertical
8	Main engine top, aft cylinder frame	Transverse
9	Main engine top, fore cylinder frame	Transverse
10	Main engine top, fore cylinder frame	Longitudinal
11	Main mast top	Longitudinal
12	Main mast top	Transverse

TABLE 4-1 TYPICAL GLOBAL MEASUREMENT POSITIONS

4.1.2.3 Machinery

Typical measurement points and definitions of nomenclature for the vibration are given in ISO 10816-1 and ISO 10816-6.

Actual measuring positions and directions to be decided on site.

4.1.3 Measurement Conditions

The conditions should be as specified in ISO 20283 to ensure reliable measurements are achieved. The conditions include:

1. Free-route test on a straight course
2. Maneuvers such as hard turn port and hard turn starboard
3. Constant representative engine output
4. Sea state 3 or less; for large ships greater than 10,000 tonnes
5. Sea state 2 or less for ships under 10,000 tonnes
6. Full immersion of the propeller
7. Water depth not less than five times the draught of the ship
8. In addition, the measurement conditions shall also be in accordance with ISO requirements and Shipowner’s requirements.
9. Machinery and equipment should be measured throughout their operational range

In order to cover the ship operational profile, several loading conditions/speed may be required.

4.1.4 Measurement Procedure

Measurement procedure should be in accordance with the specified standard.

4.2 Exciter Tests

A vibration exciter is a force generator that provides vibration, shock, or modal excitation for testing and analysis of the vibration characteristics of the test object. Exciters are designed to produce a given range of harmonic or time dependent excitation force and displacement through a given range of frequencies. These machines can be mechanical, electro-hydraulic or electro-dynamic in nature. Vibration exciters are used for development, simulation, production, studying the effects of vibration, and simulate the shock or vibration conditions in the structure.

The exciter is placed in some area and accelerometers are placed in the area that needs to be studied. This can be a useful tool to study vibrations that cannot be measured during normal operation or possibly before rotating equipment is installed, to verify a FEM analysis where for a specific frequency or when the damping is unknown.

Another method is an impact test. Usually a rather large hammer with a sensor measuring the applied force is used. By creating an impact to the structure, one can measure how far and severe the resonances become.

An impact test can excite a high number of natural frequencies, depending on the frequency content of the force, the properties of the structure, dispersive or non-dispersive waves, wave speed, boundaries, etc.

These types of methods are not so common and only used in complicated situations by expert companies.

5 Resolving Vibration Problems

If structural vibrations are first identified as a problem, there are basically two approaches to eliminating them. The first is to identify the source of vibration and either eliminate it or reduce it as far as practical. The second is to modify the affected structure. This usually involves changing the stiffness of the structure to change its natural frequency so that it is far enough away from the excitation frequency and resonance is eliminated.

5.1 Hydrodynamic Modifications

In most cases the most effective way of solving a vibration problem is to remove the source of the vibration. Where the vibration source is the propeller this often means either improving the flow into the propeller or reducing the strength of the pressure pulses produced by the propeller. As explained in section 1.1.1.1, vibration from the propeller is often due to cavitation resulting from the non-homogeneous flow into the propeller. The following sections outline the various methods that can be used to improve either the flow into the propeller or the way the propeller reacts to this non-homogeneous flow.

It should be noted that many of these steps will cause a change in propulsion efficiency.

5.1.1 Modified Propeller

Propeller design is a complex balance between efficiency and cavitation. A low blade area propeller is efficient because it produces the required thrust with lower drag, but the associated high pressures developed on the blade can lead to the formation of high levels of cavitation on the propeller and thus vibration of the hull structure. If the source of vibration has been identified as the propeller, one method to resolve the problem is to redesign the propeller, checking the improvement compared with the original design using cavitation tunnel testing with high speed cameras and/or pressure pulse transducers.

Figure 5-1 shows the original propeller (left) and the redesigned propeller (right) resulting in less cavitation being formed and a weaker collapse downstream of the propeller. This generates weaker pressure pulses and therefore reduced vibration of the hull structure.

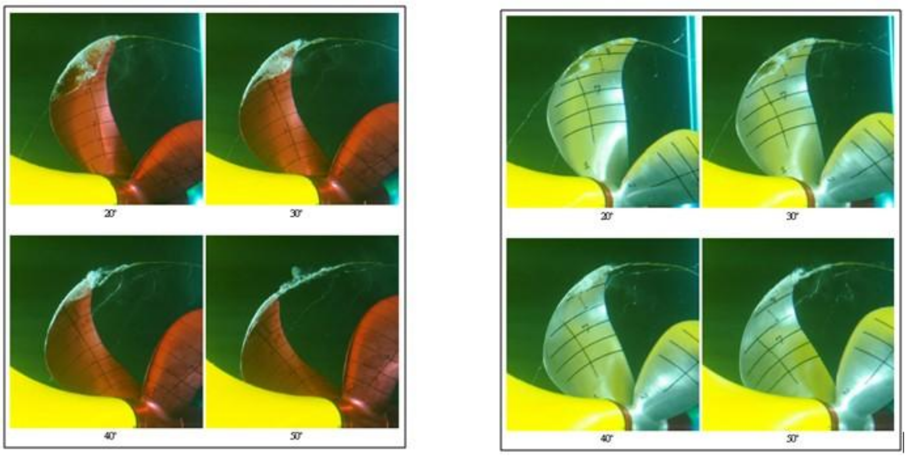


FIGURE 5-1 PROPELLER BEFORE AND AFTER REDESIGN

5.1.2 Flow Devices

There are several flow devices available, aimed at improving the flow into the propeller equalizing the wake field and thus reducing cavitation and pressure pulses. Some of these devices also improve the efficiency of the propeller and thus reduce fuel consumption. These devices include ducts, vanes, fins and combinations of these, located upstream of the propeller. Typically, they redirect the flow into the propeller so that it is more homogeneous, meaning the propeller is operating in the flow for which it was designed more of the time. These devices are usually designed using Computational Fluid Dynamics (CFD) and modelled in cavitation or wind tunnels to ensure they are producing the desired effect on the performance of the propeller. Some of these devices can result in increased resistance.

Another effect which sometimes causes structural vibrations is vortex shedding at the sea chest gratings. Certain combinations of grating geometry and flow rate over the grating can cause vortices to form. The vortices can cause vibrations in the gratings or “pumping” of the water in the sea chest. This can lead to cracking of the grating or the structure in the sea chest itself. Solutions can include mounting a round bar upstream of the grating which disturbs the flow over the grating eliminating the vortices. Changing the orientation of the grating can also be effective. In some cases, reducing the volume of the sea chest has also been effective.

The figures below, Figure 5-2, Figure 5-3, Figure 5-4 and Figure 5-5 show some examples of these devices and how they improve flow into the propeller.

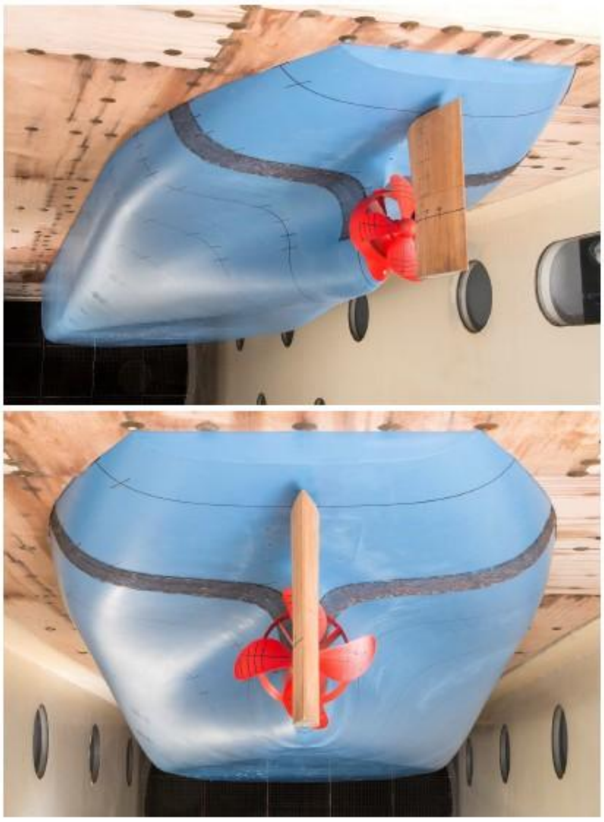


FIGURE 5-2 DUCT WITH INTERNAL FIN

FIGURE 5-3 FIN USED TO REDIRECT FLOW FROM CONDENSER DISCHARGE

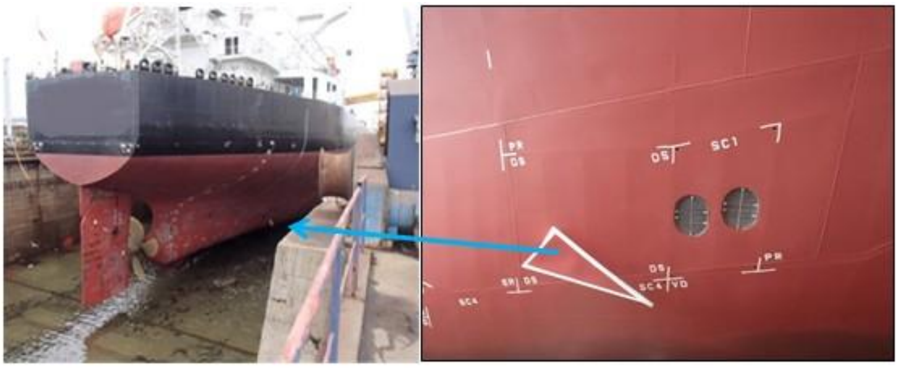


FIGURE 5-4 VORTEX GENERATOR

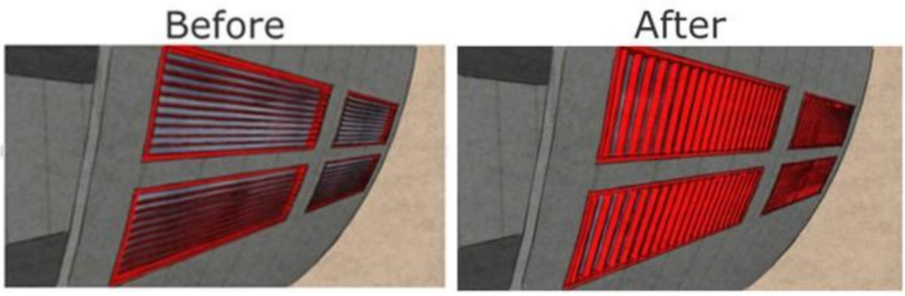


FIGURE 5-5 CHANGE OF ORIENTATION OF SEA CHEST GRID

5.2 Resilient Mountings

Resilient mountings are typically used with ships' generator sets and large compressors to limit the transmission of vibration forces from the machinery into the hull structure. They comprise of upper and lower sections, the upper section connected to the machine's seat and the lower section connected to the ship's structure. The upper and lower sections are connected to each other via a damping material, such as rubber.

FIGURE 5-6 ENGINE RESILIENT MOUNT

FIGURE 5-7 RESILIENT MOUNTINGS

5.3 Top Bracings

Top bracings come in mechanical and hydraulic versions where the hydraulic is most expensive and preferred. The function of the top bracing is to transfer main engine external transverse moment and forces to the hull structure and increase the natural frequency of the main engine-hull dynamic system to occur above the dominating excitation order at the engine rotational speed.

Figure 5-8 below shows a graphical presentation of the effect of the top bracing. The top bracing is capable of “de-tuning” the system by increasing the natural frequency. By using top bracings, the red color vibration amplification curve will be transferred to the blue color identical curve. By introducing hydraulic top bracing, the top of the red color curve will additionally be cut off at a lower resonance level as the hydraulic top bracing also has a damping effect.

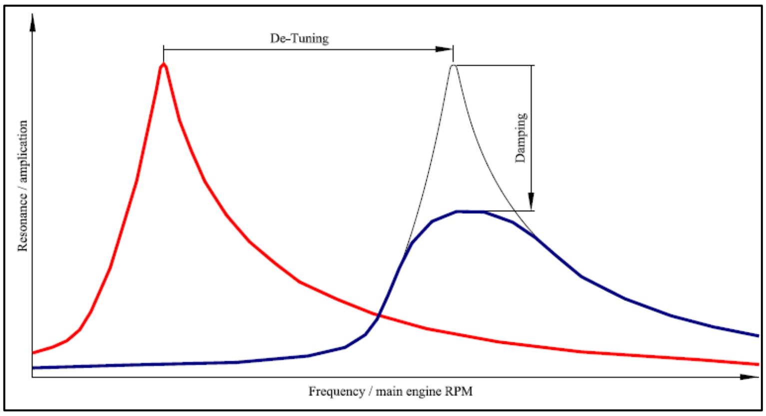


FIGURE 5-8 DE-TUNING AND DAMPING OF MAIN ENGINE

The more expensive hydraulic top bracing also has the advantage that it can be engaged/disengaged to couple/de-couple depending on main engine RPM. For some cases the vibration pattern of the system with top bracings installed can worsen the vibration pattern. Figure 5-9 below shows the vibration pattern of a main engine with and without top bracing. The blue color curve is with top bracing engaged and red color curve is with hydraulic top bracing disengaged. By disengaging the top bracing at “90% NCR” the lowest resonance level of the 2 curves will be the result.

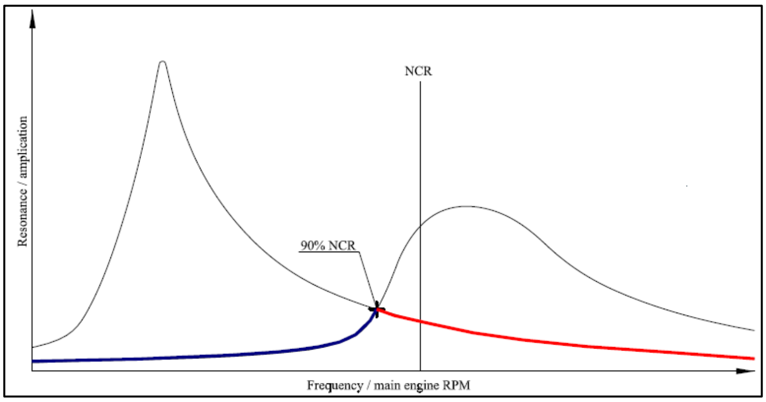


FIGURE 5-9 COUPLED/DECOUPLED HYDRAULIC TOP BRACING

The hydraulic top bracing also has the advantage that it can be adjusted according to the loading condition of the vessel whereas the mechanical is less flexible to move in the frictional contact point. When a vessel is loaded to a deeper draught the water pressure will deform the hull inwards and pre-set the top bracings with a deformation if no adjustment is possible as for the hydraulic top bracings.

Figure 5-10 below shows how the hydraulic bracing works. The left side of the bracing is connected to the engine and the right side to the hull structure. If the deflection force is above a pre-set level then, when the engine moves athwartships towards the hull structure the piston compresses the oil in the primary chamber which flows via an overpressure relief valve to the secondary chamber. When the engine moves away again from the structure the pressure in the air chamber causes the oil in the secondary chamber to flow back into the primary chamber via a check valve. In this way the vibration motion is damped.

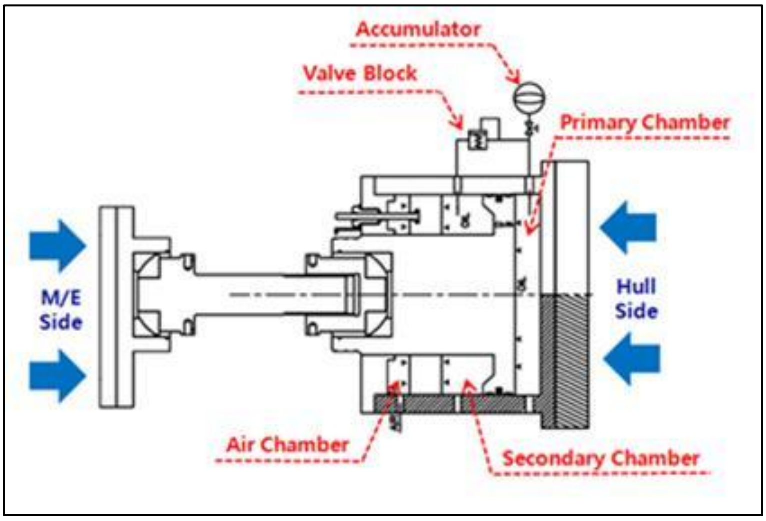


FIGURE 5-10 MAIN ENGINE HYDRAULIC TOP BRACING SYSTEM

5.4 Structural Modifications

In some cases, it may not be possible to solve the vibration at its source. In those cases where damage to structure is experienced or a risk due to vibration is identified, it may be possible to solve the problem by making structural modifications.

Significant structural damage can occur, typically to plate panels, such as bulkheads and decks, or mast structures, when the natural frequency of the panel/post is close to the excitation frequency of the vibration. This results in large oscillatory deformations of the panel/post which may ultimately lead to fracture of the plating, stiffeners or other structural elements.

Therefore, in some cases, it may be a simple task to prevent damage of the structure from the vibration by adding additional structural elements or increasing their size to increase the structure’s natural frequency away above from the vibration excitation frequency. This is typically seen in the form of carlings added between stiffeners, larger brackets at the base of posts, etc.

5.5 Vibration Compensators

In some cases, it is not possible to remove or improve the source of the vibrations and in these cases vibration compensators may be a solution. Vibration compensators work by rotating a weight at the same frequency as the vibration being experienced but in anti-phase, meaning the resultant oscillating force from the compensator and the vibration source is either zero or, more typically, greatly reduced.

Figure 5-11 below shows a typical system comprising of an RPM detection system to measure the frequency of the vibration (in this case the vibration source is the main engine, so the detector is measuring main engine rpm), a synchronizing unit and the compensator. Here the compensator is a horizontal type, to compensate for the horizontal vibration from the main engine, but vertical type compensators are also used where vertical vibrations are the problem.

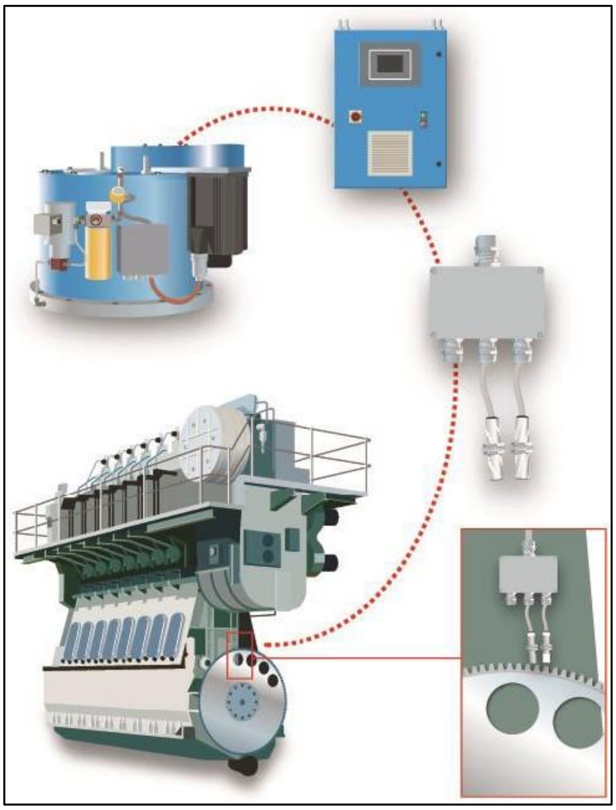


FIGURE 5-11 EXAMPLE OF VIBRATION COMPENSATOR SYSTEM

The location of the compensator can vary, in some cases being close to the source, in others being close to the area where high vibration levels have been measured.

In some cases, the compensators cannot be located at either the source or the high vibration area. In this case the compensator is instead located so that its distance to the hull structure node is maximized and therefore the force generated by the compensator can be large even when the compensator weights are small. In this way the vibration from the very large weight of the main engine can be compensated for by a lightweight compensator located in the steering gear room for example.

6 Recommendations for Shipowners

As opposed to noise requirements which are covered by IMO Resolution MSC.337(91) and where noise limits are explicitly stated, vibration is ruled by guidelines and a few, mostly optional, Class requirements.

Guidelines are subject to interpretation and a Shipowner must pay attention to common agreement how such guidelines are to be adhered to by the Shipyard. In general, a Shipowner can only expect the mandatory requirements in the Maritime Labour Convention 2006 to be included in a standard Newbuilding Specification as well as Class check of propeller shafting for axial and torsional vibrations.

The more requirements in a Newbuilding Specification the more expensive the vessel will be. Working with a limited budget, issues such as vibration prevention will often be down prioritized relative to extra costs related to the economical and safe operation of the vessel. It takes, however, only one vessel with vibration problems to discover how devastating this is to the welfare of the crew and the general condition of the vessel structure and outfitting. After vessel delivery the time, re-build, and off-hire costs to rectify a vibration problem will be several times the extra cost of adding a few additional requirements in the Newbuilding Specification.

In the following sections means to prevent, evaluate and measure vibrations in the various stages of the specifying, designing, construction, commissioning and operation of the vessel will be described.

As discussed above, additional requirements in a Newbuilding Specification will most often impose an extra cost to the Shipowner. The following sections will also be describing means for the Shipowner to make simple checks of the vessel vibration pattern and if this can impose unacceptable vibration.

The Shipowner shall at all times remember and remind the Shipyard that checking of local and global vibration issues during the design stage shall be a part of the Shipyard's internal standard quality procedure and shall not impose any extra cost to the Shipowner.

6.1 Newbuilding Specification

6.1.1 Standard Newbuilding Specification

A Standard Newbuilding Specification cannot be expected to include other rules or regulations regarding vibration than the Maritime Labour Convention 2006 (MLC2006).

MLC 2006 Sections A4.3.4, B4.3.1, B.4.3.3 acknowledge vibration as an exposure type which affects the occupational safety and health of the seafarer.

In Section B4.3.3 compliance with relevant international standards will be considered meeting requirements of MLC 2006. Today the most common used international standards are ISO 20283-5 and ISO 21984. Even if ISO 6954:1984 is superseded by ISO 6954:2000 and ISO 6954:2000 is superseded by ISO 20283-5 both are still regularly used by Shipyards even though not publicly accessible anymore.

The Shipowner shall insist that ISO 20283-5 or ISO 21984 is adhered to and explicitly entered in the "Rules & Regulations" section of the Newbuilding Specification.

Except for the above reference to international standards, MLC 2006 provides no specific requirements, but only recommendations which cannot be used without further agreement with the Shipyard as to how to interpret and follow. MLC 2006 refers to "excessive vibrations" which cannot be used as a basis for discussion with a Shipyard if a vessel experiences vibrations.

Guideline B3.1.12 "Prevention of noise and vibration" in MLC 2006 provides guidelines for design of the vessel for areas where the crew are expected to work and live.

Based on the above references from MLC 2006 to international standards, a standard Newbuilding Specification shall include a vibration standard covering recommendations to vibration levels and an onboard vibration measurement scheme to be performed during sea trials.

Apart from the above and Class requirements to propeller shaft axial and torsional vibrations, a standard Newbuilding Specification will only include few sections related to vibration guidelines and measurements if nothing else is agreed with the Shipowner. Many paragraphs will be loosely described providing the Shipyard the possibility to avoid any rectifying actions if vibrations occur. It is of importance that such diffuse requirements are discussed and if possible quantified by actual limits and required actions by the Shipyard. One shall however also remember that loosely described requirements can be used to the advantage of the Shipowner, but this will most often require long and time-consuming discussions with the Shipyard during both the design and the construction phases. A loosely described vibration paragraph could state:

"Routing and support of piping to reduce piping work on deck, pipe stress, vibration, etc. shall be considered in the design as per Shipyard's practice".

A Shipyard presenting such statement will always allow the Shipowner to ask for documentation how the current construction is designed to reduce vibrations. In most cases such discussion will be long and fruitless if the Shipowner is not able to question the Shipyard's argumentation in a qualified way.

Further to the above a standard Newbuilding Specification shall include an onboard vibration measurement plan to be performed during sea trials for measurement of propeller shaft torsional and axial vibrations.

Normally the Shipyard will include only sea trial measurements for the first vessel in a series. It is recommended to try and include vibration measurements for all vessels in a series, where normally different vessel loading conditions may be checked.

6.1.2 Additional Requirements to Newbuilding Specification

Recommendations to improve the standard Newbuilding Specification of a Shipyard are presented below. A selected variety of improvements are suggested, considered being the ones normally within the reach for a Shipowner to find mutual agreement with the Shipyard at no or reasonable extra cost. More sophisticated proposals will not be discussed.

6.1.2.1 Vibration Guidelines

ISO 6954:1984 and ISO 6954:2000 are only guidelines and contain no exact limits but only recommendations for acceptable vibration limits in passenger cabins (A), crew accommodation (B) and working areas (C). When one of the above standards is used in the Newbuilding Specification the acceptable levels must also be clearly stated. At all times the Shipowner shall insist that limits are below "Values above which adverse comments are not probable" in Table 1 of ISO 6954:2000. In the same table values are also given for "Values below which adverse comments are not probable". It should be possible to have the Shipyard accept maximum vibration levels in the mid- or lower range between the two levels.

Table 2 of ISO 20283-5 and Table 1 of ISO 21984 provide "guideline values" which are described to be regarded as acceptable values. In general ISO 20283-5 and ISO 21984 values are in the mid- or lower range between the two levels described in Table 1 of ISO 6954:2000. Additional advantages of ISO 20283-5 and ISO 21984 compared to ISO 6954:2000 is that additional areas are included with vibration limits such as "Offices", "Navigation bridge" and "Engine control room" which in ISO 6954:2000 were considered working areas (C) and new areas "Open-deck recreation spaces".

ISO 20283-5 and ISO 21984 includes "Work space" (areas allocated primarily for manual work, namely workshops, laundries, galleys, laboratories) and "Duty stations", work spaces where crew members typically stay over prolonged periods of time to monitor navigation or machinery.

The above guidelines focus on habitability (occupational health). None of these guidelines regulate allowed vibration levels in pure machinery spaces, pump rooms, stores, air compressors, boilers, oil fuel units, major electrical machinery, oil filling stations, thrusters, refrigerating, stabilizing, steering gear, ventilation and air conditioning machinery, etc. and trunks to such spaces. Spaces of this type can however be points of focus, which then must be agreed separately with the Shipyard.

Equally important as stating which standard to meet is to specify how Shipyard shall rectify it if the vibrations levels are measured to be non-compliant.

If the sea trial reveals measured vibration levels above the agreed limits, a paragraph specifying how to rectify the problem shall be included in the Newbuilding Specification and/or Contract. At all times the Shipowner shall at least have a statement included like:

"The vessel shall be built and tested to comply with either a Class notation and/or a specific standard, like ISO 20283-5 and shall be verified during the trials by a third party".

In order not to open for a discussion or interpretation it is advised to state the actual criteria in the Newbuilding Specification e.g. by copying the relevant tables. In this case it will be indisputable that this is a contractual requirement and the Shipyard shall rectify or compensate. This type of binding formulation may not be possible at all or will have a significant cost increase attached. Alternatively, it could be formulated like:

"Where measured vibration levels exceed the agreed values in the Newbuilding Specification, the Shipyard shall make the necessary improvements to a practical extent, which are to be mutually agreed between Shipowner and Shipyard".

The phrase "…to a practical extent…" is to be avoided if possible but based on experience it is difficult to remove without considerable extra cost associated.

For vibration requirements to machinery such as rotating machines, pumps, engines, bearings etc., supports, other equipment/electrical installations and structure, Classification Societies provide both Guidelines and Class Notations (see Section 3.1) which can be added to the Newbuilding Specification. For guidelines it shall be agreed with the Shipyard which limits to be below, and that such criteria shall apply to all permissible operating speeds and loads at stable running conditions.

An equipment maker may have different stated vibration limits, which can be used if without impact on other attached equipment or supporting structure. In some cases, they may be stricter with respect to function guarantees and maintenance intervals.

6.1.2.2 Calculations and Documentation for Shipowner's Approval

Most Shipyards will today make a 3D finite element model at design stage to prevent later harmful vibration patterns and Shipowner shall request such report is submitted for review. The report shall at least describe the vibration pattern for the accommodation block and the hull girder. Both vibration patterns shall be checked against propeller and main engine induced vibration for possible requirement for a moment compensator (see Sections 2.1 and 2.2).

For vessels built to CSR rules it is compulsory that the Shipyard makes a 3D model to verify scantlings. It is possible for the Shipyard to modify and extend the model to use as a basis for a vibration analysis.

If the Shipyard will not prepare a 3D finite element model of the vessel, the Shipowner shall request that simpler, semi-empiric, checks of global hull girder vibration and local bulkhead and deck vibrations are submitted for review. Such calculations shall form a natural part of the Shipyard's internal quality procedure and as such the Shipowner shall not accept when a Shipyard refuses to share these. If the vessel is the first in a series with a new design of the hull, propeller and main engine layout, the Shipyard shall be expected to prepare vibration calculations.

All results and reports are to be submitted to the Shipowner for possible third-party evaluation.

If the Shipyard does not carry out an analysis or is not willing to share information, the Shipowner is recommended to carry out their own analysis.

For piping systems that might be deemed critical or supported in an unconventional way, a separate detailed analysis shall be made, reflecting the actual supports and routing/bends, e.g. long pipes on deck or pump stack in tanks.

Resonance frequencies found should be at least +/-10% away from the excitation frequencies.

Simple checks that can be done by the Shipowner are presented in 6.2.

6.1.2.3 Hydrodynamics and Model Tests

In general, it is essential that the model and cavitation tests reflect the full operational range of the vessel and the propeller, i.e. different drafts, speeds and if a controllable pitch (CP) propeller, also different pitch. If only the contract speed conditions are tested, there may be an operational point where the wake field and/or propeller will cause unwanted pressure pulses or cavitation with severe vibration as a result.

Additional requirements for measurements of pressure pulses on aft ship from the propeller during model tests can be added subject to extra cost. Such model tests must specify the maximum allowed pressure pulses and loading conditions for measurements. Normally pressure pulses are not verified during sea trial due to the complexity of the onboard measurements.

Additional requirements for minimum clearance between propeller tip and hull can be added but might compromise vessel performance if the optimum propeller diameter cannot be reached.

Propeller tip clearance to the hull shall not be less than 20-25% of the propeller diameter. If the distance is smaller, investigations of propeller-induced excitation should be carried out.

If there are clearly identified operational conditions, e.g. slow steaming at constant RPM with CP propeller (for instance shaft generator arrangement), such conditions must be included in the test program.

6.1.2.4 Sea Trial and Quay Trial

Depending on which international vibration standard vessel will be built to comply with, additional requirements to sea trial measurements shall be added to the Newbuilding Specification and it is suggested that the following should be included as a minimum:

- How many vessels will be measured. Normally the Shipyard includes only measurements for the first vessel in a series.
- Who will do the measurements. A 3rd party consultant company experienced in this kind of measurements is preferred.
- Measurements shall be 3 axis and evaluation and reporting shall be according to the selected standard if specified in this.
- How many measurements will be done. Number of measurements shall be enough to characterize the vibration behavior with respect to habitability.
- The Shipyard normally performs vibration measurements at NCR. Subject to extra cost Shipowner can require additional main engine ratings to be tested e.g. intervals of 5 rpm, from 20% MCR to 100% MCR.
- The sea trial for the first vessel in a series is normally conducted at design draught whereas remaining series is often conducted at ballast draught as a request from the Shipyard. For 2nd vessel in a series which will be first vessel with sea trial in ballast condition it is beneficial if vibration test program from 1st vessel can be repeated.
- If a measurement is above the acceptable limit the Shipyard shall not only rectify to reduce vibration, but also accept that the same measurement is done on subsequent vessel to document vibration is mitigated by Shipyard's rectifying action.
- Maximum sea state of sea trial measurements. In ISO 20283-5 sea state 3 is the maximum allowed.
- Shipowner will have the right to share vibration measurements with 3rd party for analysis or alternatively the Shipowner will have the right for a 3rd party to carry out own vibration measurements.
- At the sea trial the Shipowner shall be able to request additional measurements based on visual observations.

Shaft torsional and axial vibrations shall be measured according to Class requirement in main engine RPM intervals.

6.1.2.5 Main Engine System

For the main engine the Shipowner must always specify that main engine top bracings shall be installed when recommended by the engine maker.

The vessel shall be designed and constructed for the installation of top bracings, which will be tested during sea trial.

Based on the result of the sea trial top bracings can either be deemed required or not. Most vessels have top bracings installed, but as described in Section 5.3 cases where the top bracing will worsen the vibration pattern can exist.

As also described in Section 5.3 hydraulic top bracings shall be specified compared to mechanical.

As important as the top bracing is the structural connection of the top bracing to the vessel structure. If the connection is not rigid the main engine will vibrate as with no top bracing. The Shipowner is advised to specify this requirement and an example is given below.

"Supports of main engine top bracings shall be terminated efficiently towards transverse girder and bulkhead".

6.1.2.6 Local Supports

Local vibration analysis of fore mast and radar mast and other long slender structures should be stated in the Newbuilding Specification. These are also to be prepared with eye plates if vibrations are observed during sea trial. Shipyard shall install wires if vibrations are visually observed. Normally measurements are not done for these areas.

As described in Section 6.1.1 the Shipyard will often insert standard sentences that local structures will be designed using Shipyard's practice. Given the complexity of vibrations and the difficulty to predict where and how severe these will occur in local structures it is very difficult to agree on a sentence that both the Shipyard and Shipowner can accept. One way to mitigate the risk of unwanted vibrations can be that the Shipowner at sea trials can request several additional vibration measurements to be performed at the Shipowner's selection and where Shipyard must rectify by local stiffening.

6.2 Design and Engineering

If it has not been possible to add additional requirements to the Newbuilding Specification some simple calculations can be carried out by the Shipowner. If these calculations indicate that vibration might occur, these cases shall be presented to the Shipyard with requirement to document that there are no critical vibrations. In general, it will be cheaper for the Shipyard to prevent than rectify after sea trial.

6.2.1 Global Vibration Check

The objective of the check is to ensure that resonance of the hull girder does not occur due to propeller or main engine excitations (see Sections 1.2.1 and 2.1 for further details). If the Shipyard has prepared a 3D finite element report of the vessel the vertical hull girder natural frequencies of such report shall be used as input. If the Shipyard has not been able to provide such a report, the Shipowner can use semi-empiric methods as described in "ABS Guidance Notes on Ship Vibrations, 2023" to calculate 2-noded and higher vertical natural frequency. It is recommended to check at least 2-noded to 5-noded vertical natural frequency.

Plotting main engine RPM in graph as x-axis and vibration frequency as y-axis a relatively quick graphical check can be made as shown in Figure 2-2. The procedure to make the check is as follows:

1) Mark a vertical band of the area of operation of the vessel – in Figure 2-2 NCR to SMCR. When slow steaming is considered the lower band shall be less than NCR.

2) Mark all hull girder noded frequency bands with horizontal bands. Lower limit will be calculation for ballast draught and upper limit will be calculation for scantling draught. It is advisable that additional 5% is added as safety margin to the band.

3) Mark propeller passing frequency and relevant main engine excitation moments and forces as lines originating from 0,0.

4) Where one of the lines cross double hatched area further investigation is required.

Figure 2-2 is prepared for a medium range tanker with a 4-bladed propeller and MAN 6G50-C.5 main engine. The figure shows 3 areas where further investigation is required.

First area in Figure 2-2 is where propeller blade passing frequency is crossing the 5-noded vibration pattern. It is judged unlikely that a propeller can initiate a 5-noded vibration pattern as this requires too much energy. In general, it is difficult to predict if propeller passing frequency will induce vibrations without knowing the pressure pulses.

The second and third areas are where 3rd order main engine passes the 4-noded vibration pattern and where 2nd order passes 3-noded vibration pattern. To investigate these areas further it is required to consult the main engine maker Project Guide for the selected main engine. In the Project Guide for the MAN 6G50-C.5 main engine it is seen that main engine produces no 2nd and 3rd order vertical forces or 3rd order vertical moment. It does however produce a 2nd order vertical moment of 692 kNm. In the Project Guide the main engine maker introduces the term Power Related Unbalance (PRU) as a measure to check if vibration is likely to occur and a compensator will have to be installed.

The PRU is calculated as External moment [Nm] / Engine power [kW] which for the example results in PRU of 90 to 110, which shall be compared to the table in the Project Guide (see Table 6-1) for need for compensator:

Range PRU (Nm/kW)	**Need for Compensator**
0-60	Not Relevant
60-120	Unlikely
120-220	Likely
> 220	Most Likely

TABLE 6-1 NEED FOR COMPENSATOR

For the given example a compensator is not required but should the value have been higher than 120 it would be relevant to consult both main engine maker and Shipyard.

The PRU value increases with lower RPM and hence lower RPM values than NCR can be considered to ensure no vibrations when slow steaming.

6.2.2 Local Bulkhead and Deck Vibration Check

In the aft end area of the vessel, generally considered as aft ship, engine room, pump room if applicable and accommodation and engine casing blocks, the Shipowner has relatively simple ways to check if deck and bulkhead structures natural frequency will coincide with main engine and propeller. See also Sections 1.2.2 and 2.2 for further details.

For a tanker with a 2-stroke main engine, vibrations induced by the main engine and the propeller shall be lower than the natural frequencies of bulkheads and decks. The natural frequency of plane bulkheads can be checked relatively simply by finite element analysis or semi-empiric formulas as further described in Section 2.2.1. The guideline is that to increase natural frequency one shall consider to:

- Increase plate thickness
- Increase stiffener moment of inertia
- Shorten span of stiffeners, intermediate web/girders
- Having stiffener ends supported as “fixed”

It should be noted that in general it is weight efficient to introduce stiffeners and dividing the plate compared to increasing the plate thickness.

All the above will make the vessel heavier and more expensive which will be against the Shipyard’s interest. In the calculations one shall also remember that weight on a bulkhead or deck will lower the natural frequency. Examples are leveling material on accommodation decks and fluids inside tanks. Tank bulkheads shall always be checked with full tanks to get lowest natural frequency. As always boundary conditions are of high importance and it is recommended to keep these “simply supported” for conservative results, unless they are clearly fixed or free.

Based on the structural drawings it is recommended that the Shipowner selects a limited number of plate fields to check. As a safety margin natural frequency shall be at least 10% above propeller and main engine. The plate fields relevant to check are:

- Bulkhead plate fields with fluid on both sides with the longest span of stiffeners or smallest stiffener moment of inertia.
- Bulkhead plate fields with no fluid adjacent with the longest span of stiffeners or smallest stiffener moment of inertia.
- Deck plate field in accommodation including weight of levelling material.

In most cases it is enough to check only the plate field, but in cases where the plate thickness is less than 8.0 - 10.0 mm and the plate field also appears large, the stiffened panel should also be checked. Examples can be unloaded access platform in engine room or a platform inside a tank.

If the Shipowner finds any plate fields with natural frequencies coinciding with main engine and propeller these shall be presented to Shipyard for their further action. Experience is that the Shipyard will take action to prevent expensive rectifying works after sea trial.

6.2.2.1 Main Engine System

The support of the top bracing needs to be checked to ensure a stiff connection to the hull structure.

Shaft torsional and axial vibration reports shall be reviewed. These reports will be approved by Class and do not require a check by the Shipowner. The barred speed range shall also be checked by Class and the Shipowner.

6.2.2.2 Propeller and Hull

Reviewing the wake field for both ballast and loaded condition can give an indication if the vessel will encounter hydrodynamic induced vibrations. A full analysis shall however be left to people with in-depth hydrodynamic experience. For further description of the phenomenon of inhomogeneous wake field and the possible vibration consequences, please refer to Section 1.1.1.1.

To reduce the impact of propeller induced pressure pulses on the aft ship the structure above the propeller shall be examined. All stiffeners must have proper end connection and if possible, plates and stiffeners shall be above Class requirements which will come with extra cost.

6.3 Construction and Commissioning

6.3.1 Construction

During construction the Shipowner's site team shall check that all external components are installed according to maker's requirements or recommendations for reducing vibrations with special emphasis on rotating machinery. If there is flexible mounting in a support, all other connections must also be made flexible.

Location of vibration sensitive equipment shall be situated in areas where vibrations are judged unlikely to occur or additional fastening of the equipment shall be considered.

All piping must be proper fastened by clamping or other means to minimize vibration. In case of large pipes with fluctuating pressure, the pipe bends shall be supported to avoid vibrations in the direction of the pipe.

All details agreed during drawing approval process are to be verified.

The site team shall also ensure any rectifying action based on sea trial results are incorporated in future vessels in the series.

6.3.2 Commissioning

During the commissioning of the vessel (sea trial) and machinery systems (cargo trial) Shipowner's site team shall always be observant to vibration levels in machinery, outfitting details, piping and hull structure and report these when observed. Based on agreed wording in Newbuilding Specification proper actions shall be taken when the vibration level is agreed to be not acceptable.

Normally vibration measurements are carried out by handheld equipment, i.e. only easily accessible areas are checked at one time or possible two during a trial.

If there are areas of concern which are without access during the sea trial a temporary fixed installation should be made. This is also applicable for checking foundations and equipment that operate at many different conditions. The vibration measurements can be recorded continuously and analyzed separately before the sea trial is completed.

The Shipowner's site team is advised to be present during the local vibration measurements and the calibration of the equipment. If unwanted vibrations are observed in other places than agreed measurement points, these additional points shall be measured. Special attention to be considered for outfitting details where Class does not have requirements for construction supports such as catwalk supports, manifold drip tray supports etc.

Propeller shaft torsional and axial vibration will be measured according to Class requirement.

If the vessel is fitted with mechanical top bracing site team must check position of bracing when main engine is cold and hot to ensure that the structure can absorb the deflection due to the thermal expansion of the main engine. When the vessel is fitted with hydraulic top bracing the final position of this shall be adjusted based on hot main engine.

Preparations for vibration mitigation of masts should be in place, eye plates in deck and on mast should be welded. If the masts are observed to vibrate, the site team shall ensure that wires are fitted and check if vibrations are reduced to an acceptable level.

The Shipowner shall make a reservation on the vessel if the agreed vibration level is not met and the Shipowner shall also require measurements on the next vessel in the series. The captioned vessel should submit a guarantee claim if the vibration issue is not rectified.

Any unacceptable vibration must be agreed upon with the Shipyard in writing and rectifying actions to be agreed for current and subsequent vessel in the series.

6.4 In Service

During the guarantee period, a claim can be made towards the Shipyard if vibrations are observed. For vessels still under construction a claim shall make the Shipyard aware of the problem and have a chance to resolve the problem before the next sea trial.

If excessive vibrations are encountered in service, it is recommended to have these measured and analyzed by a 3rd party specialized company. Local vibration measurements around the vessel will be relatively inexpensive whereas investigation of propeller induced vibrations with cameras and pressure sensors will be costly.

If the vessel experiences vibrations in a condition which was also tested at sea trial it is worth investigating if the top bracings are working under the same parameters as during the sea trial.

If a vessel experiences heavy vibrations at 0 deg. rudder angle and the vibration disappears when the rudder is angled, the reason is most likely poor water inflow to the propeller as described in Sections 1.1.1.1 and 6.2.2.2. The vibration can be reduced by welding flow devices to the aft ship re-directing flow to uppermost part of the propeller disc. Given the flow devices can move the vortex generated by the hull lines causing the vibrations, the flow device can be installed with no impact on fuel consumption. However, when a flow device is installed which creates a new vortex this will result in increased resistance. Application of flow devices shall be done based of model tests and recommendations from hydrodynamic experts. For a reference project the vibrations in the aft ship of a medium sized tanker were reduced by more than 90% and the vessel performance slightly increased. For another reference project with the same vessel type vibrations were also reduced by 90% but vessel performance was decreased by 10%.

Rudder vibrations can also be caused by vortex shedding at the trailing edge of the rudder blade. If this occurs it can often be remedied by modifying the trailing edge, for example by welding a round bar on the trailing edge.

If a vessel shall be operated in the barred speed range this area can be moved or reduced by introducing a main engine torsional vibration damper.

7 Standards for Vibration

7.1.1 Habitability

ISO 6954:1984 has also been widely used as acceptance criteria for crew habitability and passenger comfort relating to mechanical vibration. The criteria are designed to ensure that vibration levels do not cause discomfort for crew and passengers.

ISO 6954:1984 has been revised to reflect recent knowledge on human sensitivity to whole-body vibration. The frequency weighting curves are introduced to represent human sensitivity to multi-frequency vibration for a broad range of frequencies, which are consistent with the combined frequency weighting in ISO 2631-2:2003.

ISO 6954:2000 provides criteria for crew habitability and passenger comfort in terms of overall frequency-weighted rms values from 1 to 80 Hz for three different areas i.e. Passenger Accommodations, Crew Accommodations and Work spaces

ISO 20283-5:2016 replaces ISO 6954:2000 with the following changes:

- Crew and Passenger spaces are clearly defined.
- Measurement conditions also include Dynamic Position Mode.
- Guideline values are changed from pairs of lower and upper values representing the range of commonly accepted vibration magnitude to just one maximum value.

ISO 21984:2018 is a supplementary document to ISO 20283-5:2016 and this contains slightly different guidance values of acceptable vibration than ISO 20283-5:2016. The document is intended for vessels with:

- Low speed 2-stroke main engine with fixed-pitch propeller.
- Deck houses with limited length.

ISO 2631-1:1997 defines methods for the measurement of periodic, random and transient whole-body vibration. It indicates the principal factors that combine to determine the degree to which a vibration exposure will be acceptable. It includes informative annexes that indicate current opinion and provide guidance on the possible side effects of vibration on health, comfort and perception and motion sickness.

BS 6841:1987 is regarded as a clearer guide for assessing vibration and repeated shocks in relation to human health. It specifically addresses whole-body vibration. While it does not set specific limits, it provides insights into potential human reactions to these vibrations and outlines a methodology for estimating them.

7.1.2 Structure

ISO 20283-2:2008 and ISO 21984:2018 provide guidelines, and specifies requirements and procedures for the measurement, diagnostic evaluation and reporting of structural vibration of ships, excited by the propulsion plant. Structural vibration can be of global or of local nature. Here, primarily global vibration is dealt with.

Occurrence of local vibration leading to fatigue damage is rare and strongly related to the individual configuration. Therefore, no general guideline for the measurement of such type of vibration is provided within the scope of this standard.

ISO 20283-2:2008 and ISO 21984:2018 do not consider transient ship vibration phenomena, e.g., as excited by slamming.

Even though torsional shaft or crankshaft vibration may in some cases cause relevant structural vibration, they are not considered here.

7.1.3 Equipment

ISO 20283-2:2008 and ISO 21984:2018 provide guidelines, requirements and procedures for the measurement of vibration generated by types of shipboard equipment, and which can be transmitted into a ship structure as structure-borne sound, as part of the factory acceptance test of the equipment unit

Operating machinery and equipment aboard ships can create vibration and excessive structure-borne sound. As a result, the limits for sound pressure levels specified by contracting partners for spaces occupied by crew and passengers may be exceeded. Where it is anticipated that structure-borne sound from machinery can adversely affect occupied spaces, this part of ISO 20283-3:2016 can be applied with the aim of selecting low-vibration and low noise machinery.

Measurement of the vibration of individual equipment units, conducted according to standardized procedures and compared with contractually agreed-on acceptance criteria, will provide the requisite information to the Shipyard for the proper selection and installation of the equipment.

It provides a framework for providing representative test results. It is applicable to shipboard equipment intended for passenger ships, merchant ships, yachts and high-speed craft.

7.1.4 Machinery

ISO 20283-4:2012 provides guidelines on measurement and evaluation of vibration of the ship propulsion machinery. This standard provides detailed guidance on test conditions, test procedures, measurements and data processing.

ISO 10816:2014 provides guidelines on the vibration criteria in terms of overall rms values (from 2 to 1000 Hz) for the non-rotating and, where applicable, non-reciprocating parts of general machines, measured at the bearings or bearing housings. It is noted that the criteria relate only to the vibration produced by the machine itself and not to vibration transmitted to it from outside.

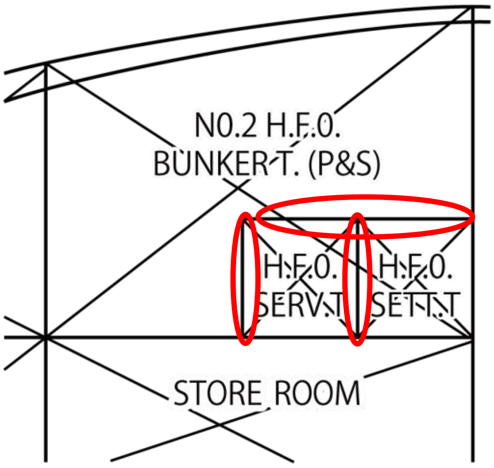
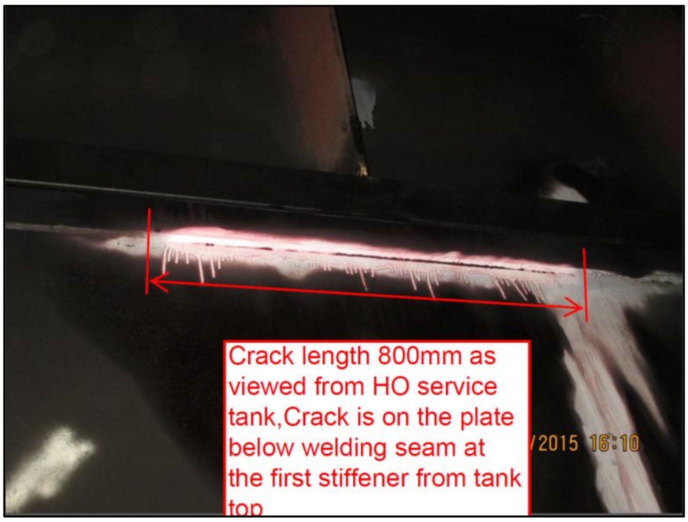
ISO 10816:2014 is complemented by ISO 20816:2022, which provides guidelines on the vibration criteria for the rotating parts of the machines.

ISO 20186:2018 replaces ISO 10816:2014.

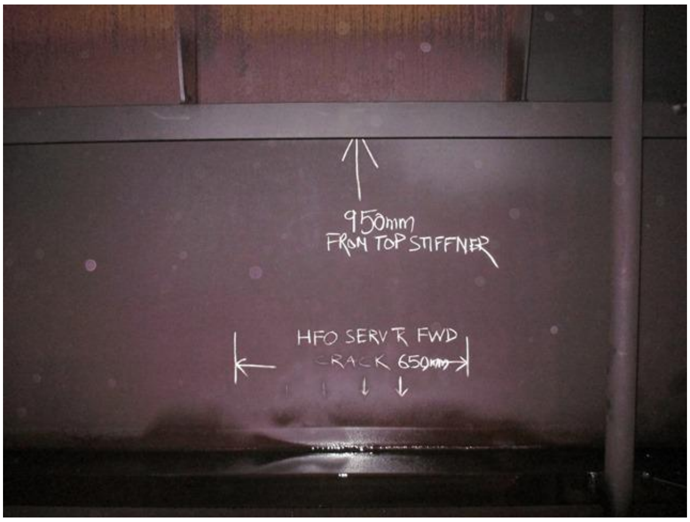
ISO 8528-9:2017 describes a procedure for measuring and evaluating the external mechanical vibration behaviour of alternating current generating sets driven by reciprocating internal combustion engines.

It applies to reciprocating internal combustion engine driven air condition. generating sets for fixed and mobile installations with rigid and/or resilient mountings.

Appendix – Case studies

Case 1			
Ship-Type	Crude oil tanker	**Capacity [tdw]**	110,000 [Aframax]
Year of build	2004	**Vessels age when damage found**	12 years
Location of damage	Damages occurred in the bulkhead plating between service/settling tanks and HFO tanks: 		
Description of defect/damage	Plating cracked at toe of weld of horizontal stiffeners: 		

Case 1



How discovered

The damage was discovered by visual examination. Oil seepage was observed through the cracks in the bulkhead plating.

Probable source/cause of damage

A calculation of the natural frequencies of representative plate fields was carried out and compared with the main potential sources of excitation. In general, there should be a 10% difference between the excitation and natural frequencies to minimize the risk of vibrations. As seen below both the propeller and main engine can excite possibly harmful vibrations. The effect of added mass is clear as it is primarily panels with liquid on both sides which have frequencies low enough to be excited.

b(mm)	l(mm)	t(mm)	Fluid sides	f(Hz)	Effect of carlings f(Hz)
945	3450	10	0	29.6	61.6
			1	14.2	33.9
			2	10.7	26
945	3450	12	0	36.5	
			1	18.3	
			2	13.9	

Excitation Frequencies:

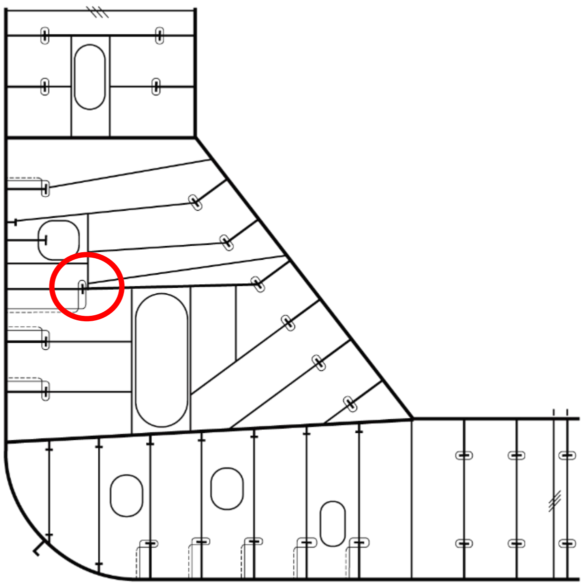
7 cylinder engine (probably running at around 97 rpm at NCR, when 105 is MCR)

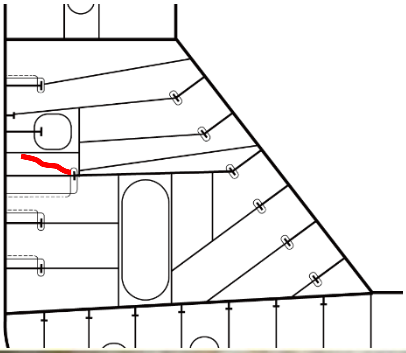
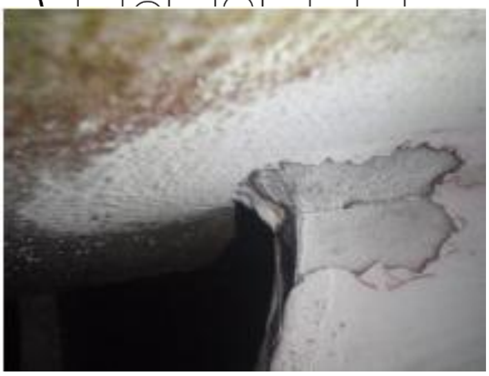
7^{th} order guide force moment will be at (97/60)*7 = 11.3 Hz

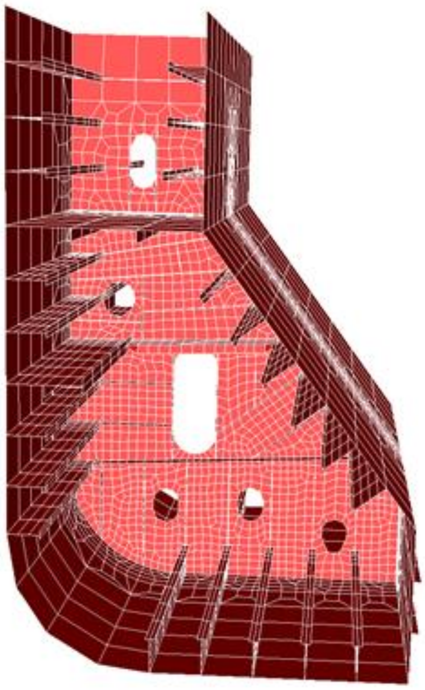
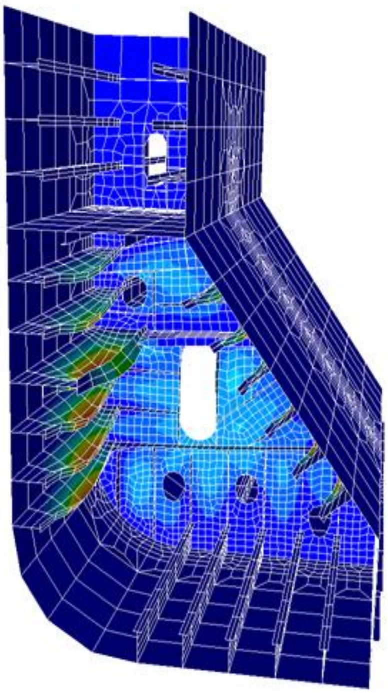
1^{st} order propeller blade passing frequency (97/60)*4 = 6.5 Hz

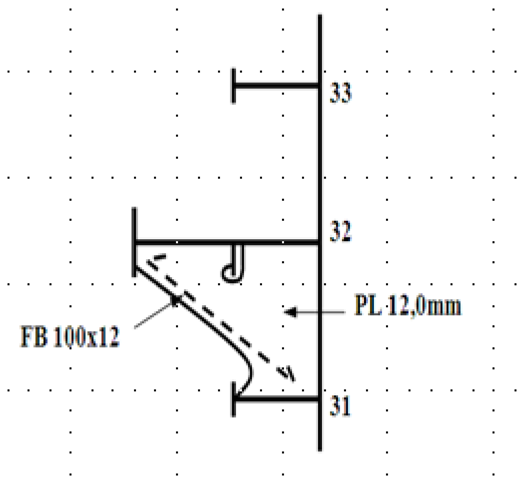
2^{nd} order propeller blade passing frequency (97/60)*8 = 13 Hz.

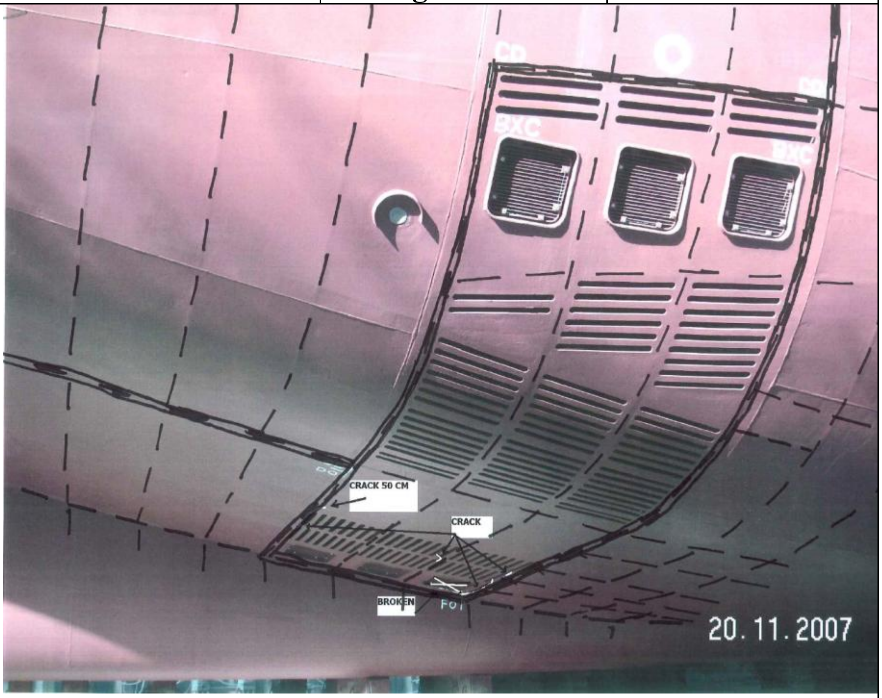
Case 1	
Description of repair	The natural frequency of a plate is a function of the thickness, the stiffener spacing and the aspect ratio of the plate field. As seen in the table above, adding vertical carlings between the horizontal stiffeners raises the frequency of the plate sufficiently to avoid harmful vibrations. The cracked plating was repaired by inserts and carlings were added.  Photo showing an inserted plate and vertical carlings.
Preventative action/lessons learned	i) Due to the complexity of the structure in the engine room it is not practical to check the natural frequency of all panels. However, it may be possible to check selected panels, particularly where they are often subjected to two sided filling. ii) In this case, only the damaged panels were initially repaired/reinforced. Additional cracks were observed shortly after the repair. These cracks were most likely already started but had not propagated through the plate. This highlights 1: It is important to do a thorough inspection, including NDT (dye pen in this case) to ensure that all cracks are addressed, and 2: If some panels are experiencing vibration problems it is important to consider all similar panels, i.e. panels with the same thickness, stiffener spacing and filling. In this case it was only the panels that had liquid on both sides that experienced problems (due to added mass).

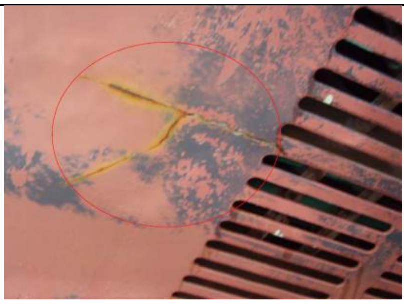
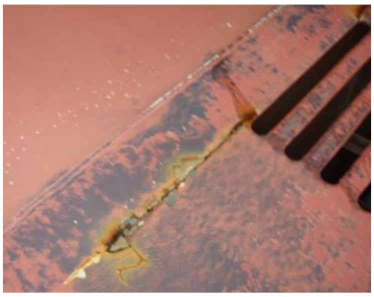
Case 2			
Ship-Type	Crude oil tanker	**Capacity [tdw]**	320,000 [VLCC]
Year of build	2007	**Vessels age when damage found**	3 years
Location of damage	Cracks were found in web frames in way of the cut-out for a PMA (Permanent Means of Access) platform. 		
Description of defect/damage	Cracks started at the radius of the cutout for the PMA and in the weld between the PMA and the web frames.		

Case 2	
	   
How discovered	Found by visual examination.
Probable source/cause of damage	A stress analysis based on normal operating loads did not reveal stress levels that could explain the cracks. Based on the age of the vessel it was suspected that vibrations could be a contributing

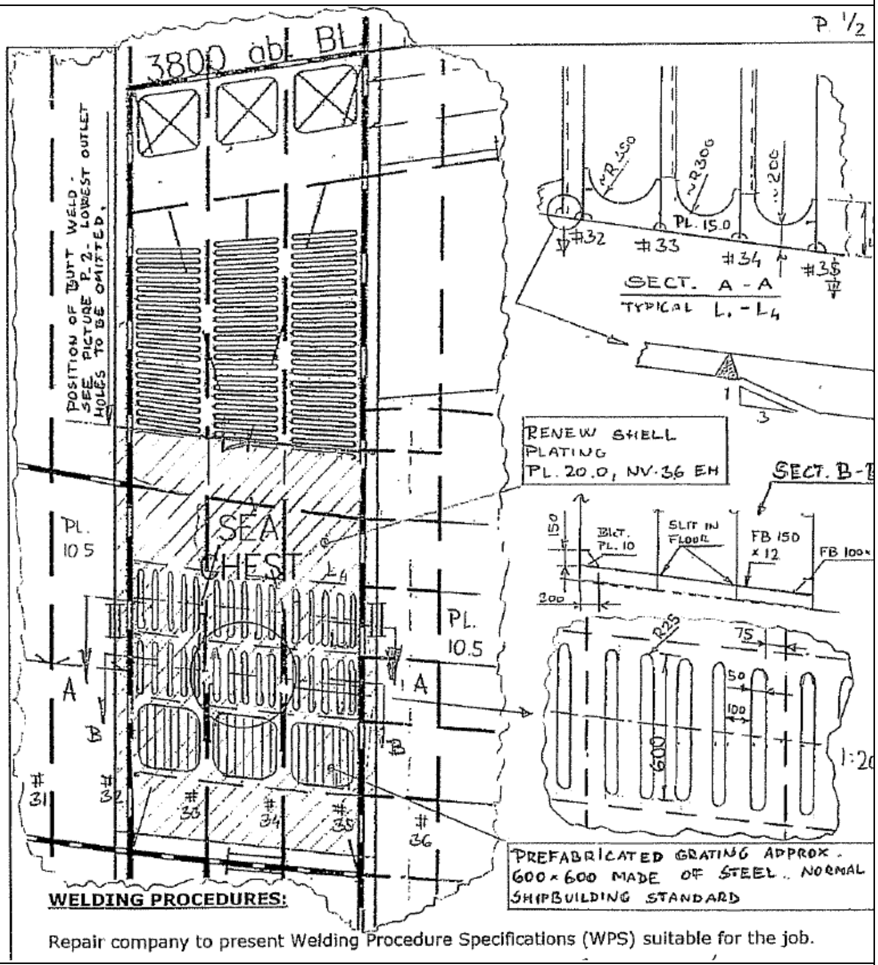
Case 2	
	factor (the vessel would not have experienced a sufficient number of wave cycles to initiate fatigue cracks).   The natural frequencies of the PMA were calculated and compared with the most likely excitation sources. Reviewing the actual operating profile of the vessel showed that she was slow steaming in the ballast condition. In this condition the natural frequency of

Case 2	
	the PMA was close to the propeller blade passing frequency and therefore a likely contributing factor.
Description of repair	An initial temporary repair was carried out by gouging and re-welding the cracks, and closing the openings with collar plates. This will reduce the stress in way of the openings, but when vibrations are suspected minor changes in details will often only move the problem to a new area or postpone cracking since the excitation is still there. A permanent solution should address the source of vibration to reduce the excitation forces, or change the natural frequency of the affected element to reduce the response. In this case intermediate brackets (one bracket between web frames) were installed. This raised the natural frequency of the PMA sufficiently to avoid harmful vibrations. 
Preventative action/lessons learned	It is not possible to check all structural elements for the risk of vibration. What is important is to identify the cause of a damage when it occurs so that the correct repairs/reinforcements are applied. For this example, the initial repair of closing the openings would have been appropriate if it was a typical wave induced fatigue damage. For vibrations the new brackets are much more effective.

<table>
<tr><td colspan="4" align="center">Case 3</td></tr>
<tr><td>Ship-Type</td><td>Chemical tanker</td><td>Capacity [tdw]</td><td>16,000</td></tr>
<tr><td>Year of build</td><td>2007</td><td>Vessels age when damage found</td><td>2 years</td></tr>
<tr><td>Location of damage</td><td colspan="3">

Cracks occurred in way of sea chest shell plating (port and starboard) on a number of 2-3 year old sister vessels. In order to ensure sufficient flow for box cooler installed in sea chests, numerous grid holes had been arranged from newbuild according to box cooler makers instructions. Above picture show grid pattern as well as the structural arrangement (webs, girders and watertight bulkheads. The cracks occurred in the area where the grid holes were positioned in transverse direction, i.e. perpendicular to the waterflow around the hull.</td></tr>
<tr><td>Description of defect/damage</td><td colspan="3"> </td></tr>
</table>

Case 3	
	  The cracks developed in the shell plating starting at the ends of the transverse oriented grid holes in the shell plating. The longest cracks were up to 1 m in length. In one case the cracks led to a hole in the shell plating. All cracks were within the area of the sea chests. The plating was 14.0 mm grade A.
How discovered	Since delivery, significant structure born noise had been observed in engine room in way of the sea chests. A vibration measurement report made 1 year after delivery concluded that the cause could be flow induced vibration in way of the sea chest grid. Two years after delivery diver inspections started reporting cracks in way of the sea chest shell plating.
Probable source/cause of damage	The cause of the problem was concluded to be flow induced vibration in combination with extensively perforated shell plating with limited stiffening inside. Structural lay-out in way of the sea chest is indicated above.
Description of repair	• Cracked hull plating was cropped and renewed. • Plate thickness in subject area was increased from 14.0 to 20.0 mm. • Intercoastal brackets were introduced in subject area in order to increase stiffness of shell plating in way of the sea chest grillage. See also repair sketch below.

Case 3	
	
Preventative action/lessons learned	Increased awareness of sea chest structural arrangement for future new-buildings.

<table>
<tr><td colspan="4">Case 4</td></tr>
<tr><td>Ship-Type</td><td>Product tanker</td><td>Capacity [tdw]</td><td>29,000</td></tr>
<tr><td>Year of build</td><td>2005</td><td>Vessels age when damage found</td><td>2 years</td></tr>
<tr><td>Location of damage</td><td colspan="3">• Cracks in main engine top bracings.
• Discomfort to crew in accommodation area.</td></tr>
<tr><td>Description of defect/damage</td><td colspan="3">Since the delivery of the first vessels in the series, there were complaints about the vibration levels on board.

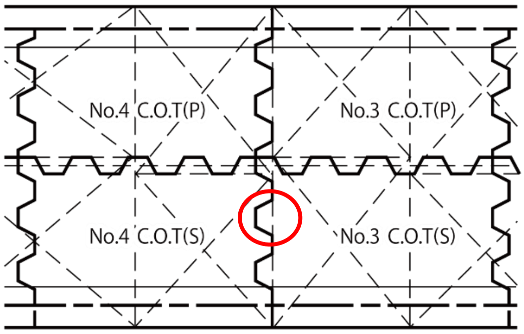
The vibration problems caused mechanical damage to structural members (top bracings) and discomfort for the crew. On the later vessels in the series, changes were made in the design at the following spots in order to improve the vibrational behaviour:

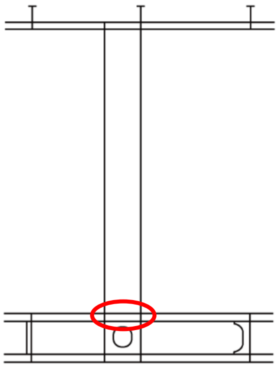
• Top bracings were increased in dimensions.
• Reinforcements were made in way of the structural support of the top bracings at the interface to the hull.
• Reinforcements were applied to the bridge deck.

Compared to the earlier delivered vessels, the vibration levels improved on the latest delivered vessels. The vibration level were however still not considered to be satisfactory on any of the vessels.</td></tr>
<tr><td>How discovered</td><td colspan="3">• Complaints from crew
• Cracks in main engine top bracings
• Vibration measurements</td></tr>
<tr><td>Probable source/cause of damage</td><td colspan="3">A vibration survey was carried out by a consultant on one of the vessels. The report concluded that the main contributor to the high vibration levels experienced, was the main engine firing frequency causing transverse vibrations of the main engine top at a frequency of approx. 10,5 Hz.</td></tr>
<tr><td>Description of repair</td><td colspan="3">The survey revealed that the vibration levels were decreased when the hydraulic top bracings were made inactive. The report therefore recommended that it was considered to operate with the top bracings inactive. This was done for a period as a temporary solution. As a permanent solution, the engine maker recommended to install a magnetic switch electronically controlled by the main engine speed. This was able to be programmed so that the bracings could be activated/de-activated depending on the main engine revolution speed.

Despite deactivating the top bracings, the crew did not notice a significant reduction in vibration levels. Consequently, a vibration compensator was installed at the top of the main engine. This</td></tr>
</table>

Case 4	
	compensator effectively reduced the vibration levels, thereby resolving the issue.
Preventative action/lessons learned	• Assess potential vibration risks during the design process with special attention to type of main engine and number of cylinders. • Evaluate if top bracings are relevant and (if yes) ensure efficient support of top bracings at the interface to the hull structure.

<table>
<tr><th colspan="4">Case 5</th></tr>
<tr><td>Ship-Type</td><td>Oil/Chemical Tanker</td><td>Capacity [tdw]</td><td>4,000</td></tr>
<tr><td>Year of build</td><td>2015</td><td></td><td></td></tr>
<tr><td>Location of damage</td><td colspan="3">Crack occurred at the welding connections between corrugated bulkhead and inner bottom plating


<Plan>


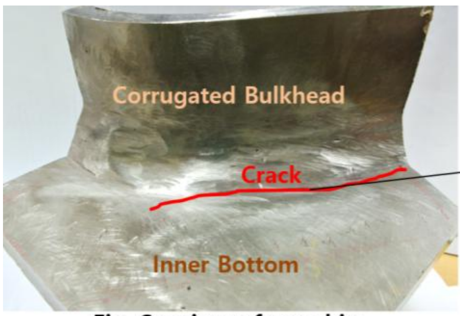
<Elevation></td></tr>
<tr><td>Description of defect/damage</td><td colspan="3">Crack penetrated both bulkhead and inner bottom plates. Beach marks were found on the crack surface which indicates that the crack may have been caused by vibration. Duplex stainless steel was used for the plates.


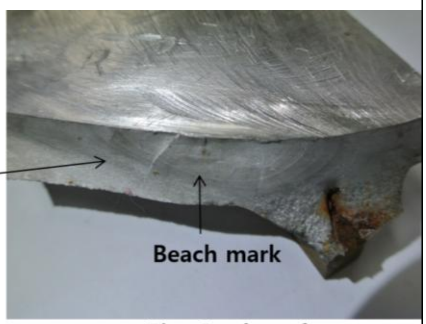
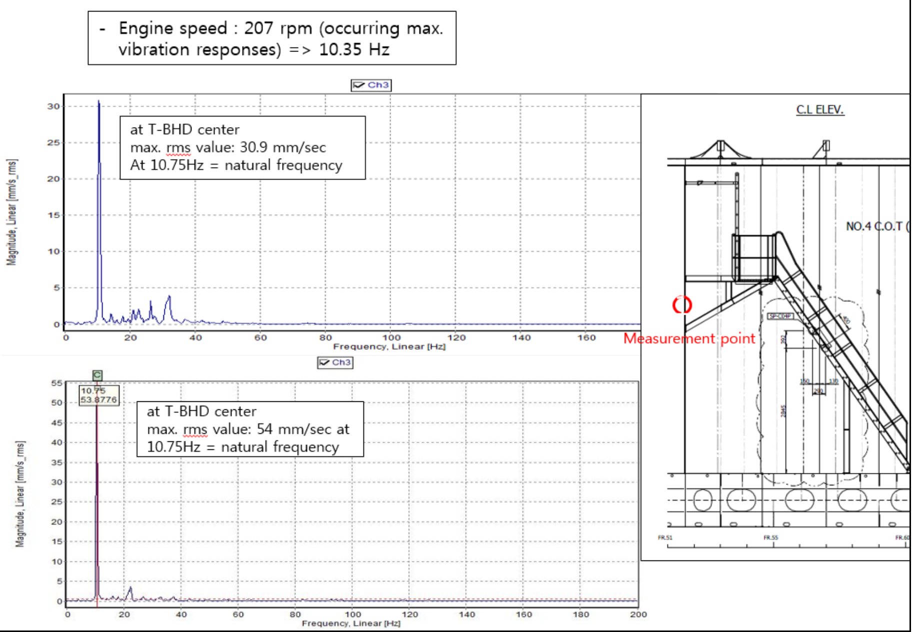
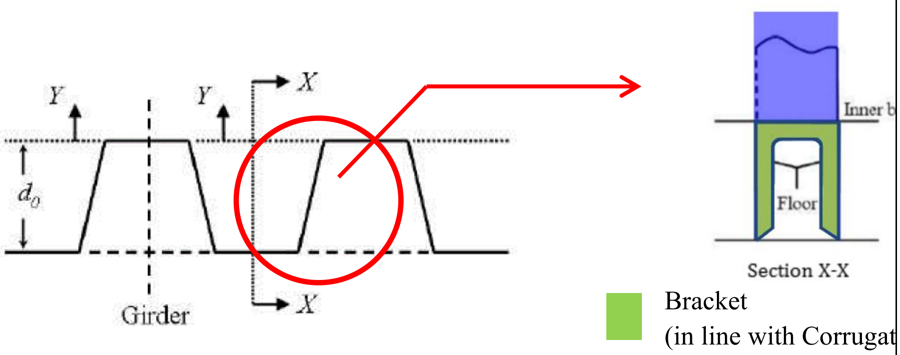
Fig. Specimen from ship


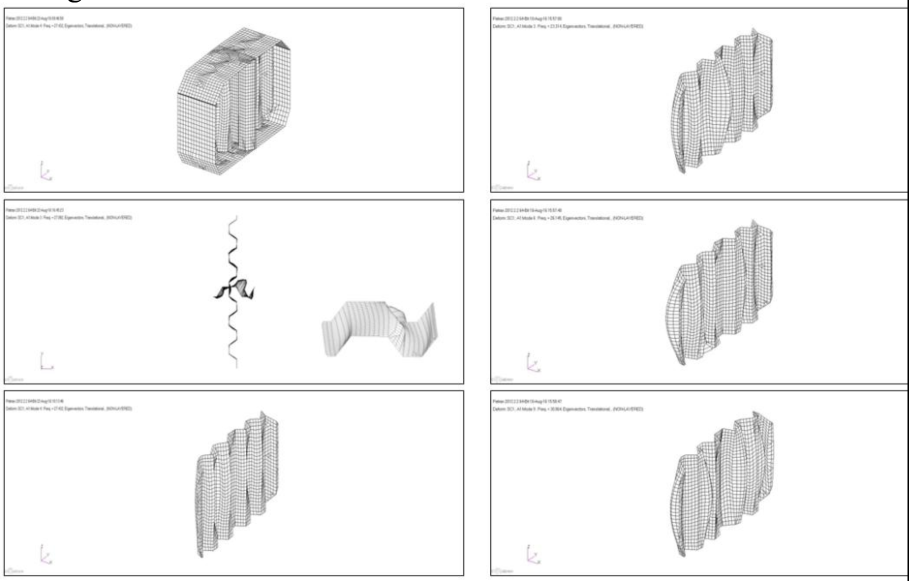
Fig. Crack surface</td></tr>
</table>

<table>
<tr><th colspan="2">Case 5</th></tr>
<tr><td>How discovered</td><td>The crack was discovered when cargo leaking was observed by the crew in the empty cargo tank while the ship was sailing with cargoes alternatively loaded.</td></tr>
<tr><td>Probable source/cause of damage</td><td>Due to several cracks in welds between the ladder pads and transverse bulkhead in cargo tank, resonance of the transverse bulkhead was suspected as the cause of crack. To verify this, vibration measurements were made at the location marked red adjacent to cracked parts of ladder support pad in No.4 Cargo Hold. It was found that the natural frequency of the transverse bulkhead web plate was resonant with the main engine third order excitation frequency at 207 rpm. The maximum vibration level at the bulkhead was measured to be 54 mm/s in the spectrum as below.


</td></tr>
<tr><td>Description of repair</td><td>Brackets were provided below inner bottom in line with the web of corrugated transverse bulkhead. Full penetration welding was applied for the connection between corrugated bulkhead and inner bottom plating.

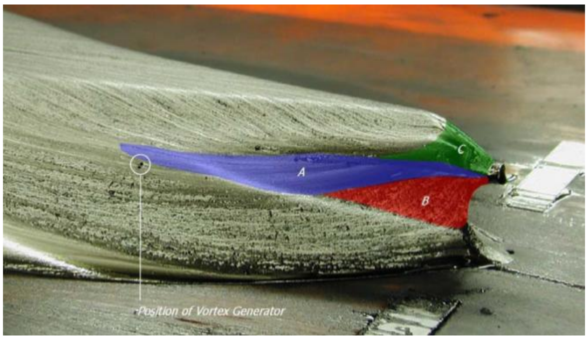
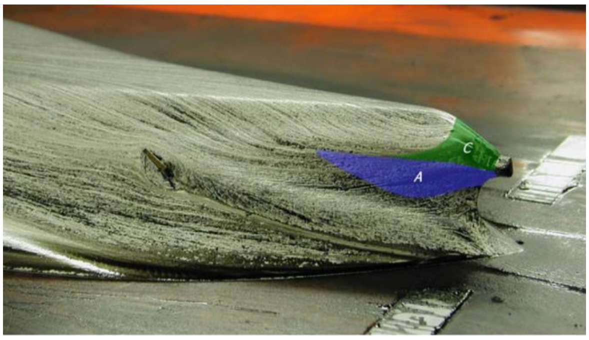
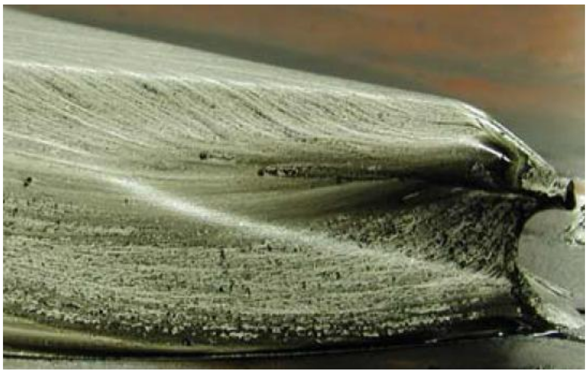


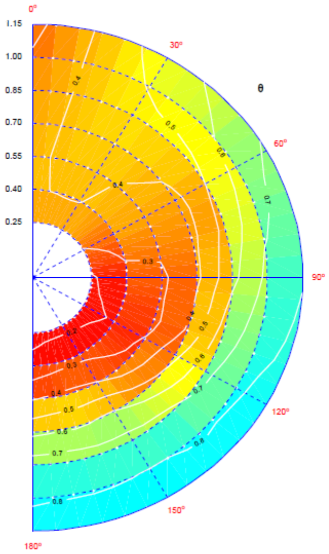
In order to reduce the vibration in the transverse bulkhead, a vibration compensator was installed to reduce the M/E third order H-</td></tr>
</table>

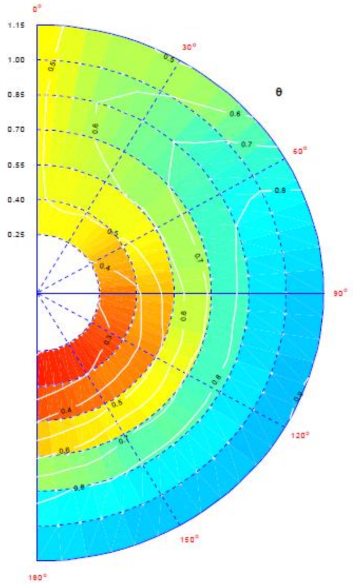
Case 5	
	moment excitation force. The compensator reduced the vibration level at transverse and longitudinal bulkhead by around 70 % at NCR
Preventative action/lessons learned	In case of a new design, vibration analysis is recommended to avoid resonance at corrugated bulkhead. Close attention needs to be paid to corrugated bulkhead without lower stool. 

Case 6			
Ship-Type	Product tanker	**Capacity [tdw]**	50,000
Year of build	2002	**Vessels age when damage found**	0 years
Location of damage	No actual damages were observed, but the crew experienced significant discomfort due to the vibration level.		
Description of defect/damage	Severe vibration level in aft region and accommodation block of vessel.		
How discovered	When vessel was sailing in light ballast condition severe vibration level was observed. Vibrations were only observed with the rudder in 0° position. When rudder was angled, the vibration level was considerably reduced.		
Probable source/cause of damage	As vibrations were only present with the rudder at 0° angle the vibration was established to be hydrodynamically induced with large chance that poor wake field of propeller was the likely cause. When wake field parameters change too much over the 360° rotation of the propeller blade, the varying pressure distribution can lead to vibration of the propeller blades which can be transferred to shaft bearings. The varying wake field can also lead to increased pressure pulses at the propeller blade top position which, if the propeller is cavitating, can be of an order of magnitude which can cause vibrations.		
Description of repair	Wind-tunnel tests were performed to document wake field of propeller and to establish if by means of flow guides it would be possible to improve wake field by making it more uniform over the 360° rotation of a propeller blade. Wind tunnel tests showed that the aft hull lines especially at ballast condition had a large region where water separated from the hull and hence created a large shadow zone where the top of the propeller disc, which experienced very low inflow axially and considerable vertical velocity. By introducing flow guides (one at each side) the flow separation was eliminated and the inflow to the propeller disc was re-distributed to be of a more uniform velocity distribution over the 360° rotation. Vibration measurements conducted before and after installation of flow guides confirmed a reduced vibration level of about 90%. As flow guides introduce a tip vortex these can generate additional drag. As the flow guides however also eliminated original hull separation vortex no additional drag or fuel consumption could be recorded.		
Preventative action/lessons learned	Sea trial shall also be conducted at light ballast condition.		

Case 6	
	Acceptable vibration levels shall be explicitly stated in Newbuilding Specification, and also include areas not covered by ISO 6954. A Newbuilding Specification shall include rectifying actions by shipyard if sea trial reveals vibration levels above the agreed limits.

Before	After
 Flow areas marked by streamline tests and categorized by A: Flow separation area. B: Area dominated by low axial and high vertical downwards flow. C: Flow separation from propeller boss termination.	 Flow separation area (A) reduced significantly and flow area with vertical velocity (B) eliminated.
 Close-up of flow pattern. Notice “smooth” appearance without visible flow lines indicating flow separation. Notice also large downwards streamline just forward of propeller.	 Close-up of flow pattern. Notice reduced separation and vertical flow areas.

<table>
<tr>
<td>

Axial velocity distribution measured by means of hotwire technology. Notice steep velocity gradients at upper region for all r/R fractions.</td>
<td>

More constant axial velocity distribution measured for all r/R fractions.</td>
</tr>
</table>

All results, photos and graphs by courtesy of Force Technology, Denmark (www.forcetechnology.com)

Case7			
Ship-Type	VLCC	**Capacity [twd]**	310,000
Year of build	1999	**Vessels age when damage found**	9 Years
Location of damage	Cross tie end detail support bracket crack, multiple locations.		
Description of defect/damage	Aftmost center cargo tank cross tie end detail crack Local cracking, non-integrity related Mild steel structure		
How discovered	Visual and also NDT		
Probable source/cause of damage	Stress and vibration effects both suspected. Cross tie twisting mode natural frequency at 5 Hz found to be close to propeller blade frequency.		
Description of repair	FE analysed. Long term local repairs carried out by cropping and renewing affected material. Operational measures implemented: (a) Prohibit ocean going passages with the center tanks partial loaded to a level below 75% (avoid sloshing); and (b) Limit propeller RPM to <72 in ballast passages. Bracket support detail improvements studied. If web were changed in shape, increased in thickness, and flange altered in a certain fashion, local stresses could be reduced by a factor of up to 3, as predicted by FE analysis.		
Preventative action/lessons learned	Stress and vibration effects in some combination suspected to be involved. Design stage considerations should where possible include greater consideration of vibration effects than at present. Good structural detailing as part of design is always important to start with.		