Guidance Notes

General Overview of Ship Structural Vibration Problems

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General Overview of Ship Structural Vibration Problems

1. General Description of Document

This document is intended to encapsulate the experience and best advice of Lloyd’s Register (LR) that has been assembled since circa 1980 regarding prediction and avoidance of potential problems in relation to structural vibration in ships.

This subject of structural dynamics can appear to be complex in nature and terminology. However, in this document, the intention is to make the subject's presentation as simple, concise and as useful as possible in both the analytical and practical sense.

Vibration is generally not subject to mandatory classification rules, except for the propeller shaft where excessive vibration can cause failure that would compromise the safety of a ship by loss of propulsion. However, classification societies do have optional class notations in relation to vibration and noise standards for habitability. The International Standards Organisation (ISO) also publishes standards. An owner usually specifies such criteria in the build contract, which are normally derived from ISO or a class notation that are with regard to measured values.

The primary objectives in order to meet vibration standards are avoidance of significant resonances (coincidence of structural natural frequencies and excitation frequencies in way of the operating range) and minimisation of excitation forces. Frequencies up to around 20 Hz are usually relevant in relation to global ship structure vibration, most notably 5 – 20 Hz. Somewhat higher frequencies can be relevant for local panels that are adjacent to propeller(s) and main engine(s).

This document does not cover prediction of noise levels. Although noise can be regarded as high frequency vibration, it is in the audible range above 30 Hz evaluated in decibels; and noise prediction methodology for large-scale structures is necessarily completely different to that for vibration.

This document covers structural vibration prediction and therefore does not include predictive methods for propulsion system vibration.

1.2 Importance of Vibration Prediction Analysis

Clients recognise the importance of performing cost-effective and meaningful structural vibration prediction analyses at the ship design stage, with the aim of avoiding potentially expensive and problematical situations occurring on ship trials and/or in service (see Chapter 8).

Contractual habitability standards are becoming ever more demanding, particularly for passenger ships. This is due to advancing expectations of crew and passengers in relation to comfort.

Invariably due to complicated structures, uncertainty of damping, fluid/structure interaction and excitation forces, it is not possible to guarantee a ship that is completely free of vibration problems by conducting dynamic analysis. The risk of unacceptable vibration occurring in practice can be greatly reduced, and any issues that may arise during ship trials or in service would normally be solvable, often by using an existing computer model.

1.3 Methodology

In the past, non-uniform beam methods were often used to represent global ship structure (such as LRHULVIB). This approach is essentially limited to fundamental vibration modes because three dimensional effects become significant for higher modes.

A development from beam element models was hybrid models such as a coarse three-dimensional model of an aft hull including a deckhouse and beam elements for the remainder of the hull.

However the ship is modelled, for dynamic response analysis it is necessary to represent the whole mass/stiffness system. Partial models can be used to evaluate natural frequencies of local structure, subject to suitable boundary conditions being arranged.

With increasing computer power and use of large scale finite element analysis (FEA), together with global stress analysis finite element models often being available, FEA is now the main vibration analysis method. LR mostly uses the NASTRAN finite element program. Therefore, FEA guidance and definitions in this document are in relation to that program, though comments for other FEA programs would be similar.

A typical ship finite element model created for global vibration analysis is shown in Figure 1:
1.4 Link to LR Guidance Document covering Vibration Measurement

LR also has a document entitled *Ship Vibration and Noise Guidance Notes* that primarily covers routine measurement surveys and investigation of problems occurring on ship trials or in service. This is a companion to this vibration prediction document. Content in the *Ship Vibration and Noise Guidance Notes* relating to vibration acceptance criteria is particularly relevant for this document.

Section 2

Theoretical Considerations

2.1 Basis Dynamic Equation

Theoretical descriptions in this chapter are of a high-level nature which are given for background information and as a lead to relevant textbooks for those who wish to go deeper into the theory of matrix algebra and differential equations. The general dynamic equation is:

\[ M \ddot{x} + B \dot{x} + K x = F \]

where

- \( M \) = mass matrix
- \( B \) = damping matrix
- \( K \) = stiffness matrix
- \( \dot{x} \) = displacement vector
- \( \ddot{x} \) = force vector

This equation can be visualised as:

(mass \( \times \) acceleration) + (damping \( \times \) velocity) + (stiffness \( \times \) displacement) = dynamic forces

Hence, in order to be able to predict dynamic responses accurately, it is necessary to know precisely mass, damping, stiffness and dynamic forces. As will be described in more detail later in this document, it is difficult to be precise about these parameters. Damping is the most problematical factor (see Chapter 5), 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 mostly known accurately, though not absolutely precisely in relation to added mass of sea water that vibrates with a ship, and outfit mass on local deck panels.

2.2 Normal Modes Analysis

Normal modes analysis yields natural frequencies and vibration mode shapes. It is associated with solutions of the undamped,
unforced system of equations, that is, \( B=0 \) and \( F=0 \) in the equation shown in 2.1. Thereby, the equation reduces to:

\[
M\ddot{z} + Kz = 0
\]

Using the form of solution for this differential equation \( x = ae^{\omega t} \) and substituting for \( \ddot{x} \) and \( x \) gives the formulation:

\[
\left(K - \omega^2 M\right)z = 0
\]

where \( \omega \) is frequency in radians per unit time and \( z \) is a vector of displacement amplitudes. It is often written:

\[
\left(K - \lambda M\right)z = 0
\]

that is called the characteristic equation, where \( \lambda = \omega^2 \).

Solution of this equation gives roots \( \lambda \) - eigenvalues, and \( a \) - eigenvectors which are mode shapes. Square roots of eigenvalues \( \lambda \) are natural frequencies in radians per unit time; and hence division by \( 2\pi \) then gives natural frequencies in cycles per unit time.

For each mode, \( \lambda = \frac{k}{m} \), where stiffness \( k \) and mass \( m \) are called ‘modal’ or ‘generalised’ values applying to that mode. This represents the value for an equivalent one degree of freedom system, such as a mass suspended on a spring that is constrained to displace in one direction only.

Eigenvectors are normalised usually by selecting each vector element to be divided by the maximum element in the vector, so that the maximum modal displacement is unity.

Eigenvectors are said to be ‘orthogonal’ or ‘normal’ with respect to the mass and stiffness matrices and their use to transform leads to mass and stiffness matrices that are diagonal, which is the so-called modal formulation.

### 2.3 Dynamic Reduction

Regarding finite element analysis, in general a significantly more coarse model is adequate for global vibration analysis than is required for stress analysis. Also, in the past, the size of dynamic models needed to be restricted in order to keep computational time and storage requirements within reasonable limits, because solution algorithms available then were such that the number of derived eigenvalues was equal to the number of degrees of freedom in the model. Hence, dynamic models had to be very coarse, or it was necessary to invoke dynamic reduction – in NASTRAN this is described as the Guyan dynamic reduction technique.

Guyan dynamic reduction is basically manual specification of degrees of freedom to be included in an ‘analysis set’ (ASET). The solution process then includes condensation to the analysis set, extraction of eigenvalues, and expansion back to the complete model. Selection of the ASET should be adequate to describe global structural configuration, mass distribution and the vibration modes to be derived. Only translational degrees of freedom need to be included. Provided that the ASET is chosen as previously described, a relatively small set sacrifices very little in accuracy for the global modes.

Later FEA solution algorithms are such that eigenvalue extraction can be restricted to a specified frequency range. However, with increasing use of ship FEA models that were made for stress analysis purposes, which are usually of much finer mesh than is required for global vibration analysis, dynamic reduction is still useful, in order to avoid a plethora of local modes being derived that are not desired or accurately represented.

There are other dynamic reduction methods than the manual ASET selection previously described, such as modal synthesis where substructures are analysed first and then subsequently represented by their vibration modes and combined. However, the degree of control of Guyan dynamic reduction that is afforded to an experienced user is useful. For example, if only main global structure configuration/intersection points of a ship model are specified in the ASET, including enough sections along the length to describe hull vibration modes, then essentially only global vibration modes will be derived. If it were desired to include modes of large deck panels, then points within these can be added to the ASET.

### 2.4 Direct Dynamic Response Analysis

In the direct method, the degrees of freedom are simply the displacements at grid points. The procedure involves the direct solution of the system of equations indicated in 2.1.

The procedure does not include derivation of vibration modes. It will usually be more efficient for problems in which a large proportion of the vibration modes are required to produce the desired accuracy, which is not generally the case for global ship vibration.

An example of a case where direct dynamic response analysis would be more efficient is the transient response of a LNG containment system to sloshing impacts. In relation to a modal solution, there would be global structure modes, local structure modes, and modes of the containment linings to take into account; thereby many modes over a wide range of frequency would be required in order to obtain sufficient accuracy for the dynamic responses. Hence, a direct dynamic response (transient) analysis would be the preferred choice for such a case.
2.5 Modal Dynamic Response Analysis

In the modal method of dynamic problem formulation, the vibration modes of the structure in a selected frequency range are used as the degrees of freedom, thereby reducing the number of degrees of freedom whilst maintaining accuracy within the selected frequency range.

The modal method will usually be more efficient in cases where a relatively small fraction of all of the modes is sufficient to produce the desired accuracy. This is invariably the case for global ship vibration.

The modal method tends to afford more flexible options for specification of damping, which will be described later in this document.

2.6 Transient Response Analysis

Transient response analysis is dynamic response analysis which is conducted in the time domain, that is, where force versus time is defined. It is most suitable for impact loads or loading of a shock nature, e.g. an explosion.

2.7 Frequency Response Analysis

Frequency response analysis is a dynamic response analysis which is conducted in the frequency domain, that is, where force versus frequency is defined. It is appropriate for loads that are of a continuous sinusoidal ‘steady state’ nature, such as cyclic loads from machinery. Hence, this is the primary method with respect to vibration analysis. It is, of course, possible to define force versus time for cyclic loads from machinery and conduct a transient response analysis, but this is not normally the most efficient method.

Section 3 Units

A section dedicated to the subject of units is included in this document because it is critically important to use a consistent set of units for dynamic analysis. This is one of the most common errors that will be found when reviewing the initial results of a dynamic analysis.

Whereas the distinction between mass and weight (force due to gravity acting upon mass) is not important for stress analysis, it is important for dynamic analysis. In fact, for stress analysis, mass is not required to be specified at all unless gravity is to be applied; otherwise all loads including weight can be described as static forces. For dynamic analysis, static forces are not directly recognised, and mass should be specified within a consistent system of units.

Five consistent sets of units are shown in Table 3.1. Using these will ensure the avoidance of errors in a dynamic analysis emanating from units. The first set is, of course, the SI system, which is the one that is most commonly employed.

Table 3.1 Five examples of consistent sets of units

<table>
<thead>
<tr>
<th></th>
<th>Force</th>
<th>Mass</th>
<th>Length</th>
<th>Time</th>
<th>Modulus</th>
<th>Density</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Newton</td>
<td>kilogram</td>
<td>metre</td>
<td>second</td>
<td>Newton/ metre²</td>
<td>kilogram/ metre³</td>
</tr>
<tr>
<td>2</td>
<td>Newton</td>
<td>tonne</td>
<td>millimetre</td>
<td>second</td>
<td>Newton/ millimetre²</td>
<td>tonne/ millimetre³</td>
</tr>
<tr>
<td>3</td>
<td>MegaNewton</td>
<td>kilotonne</td>
<td>metre</td>
<td>second</td>
<td>MegaNewton/ metre²</td>
<td>kilotonne/ metre³</td>
</tr>
<tr>
<td>4</td>
<td>kilogram/ g = 9.81</td>
<td>metre</td>
<td>second</td>
<td>kilogram/ metre²</td>
<td>kilogram/ g-metre³</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>kilogram/ g = 981</td>
<td>centimetre</td>
<td>second</td>
<td>kilogram/ centimetre²</td>
<td>kilogram/ g-centimetre³</td>
<td></td>
</tr>
</tbody>
</table>

Section 4 Mass

4.1 Structure

Structural mass is represented by including mass density in the definition of structural material properties. It is usually necessary to make a small increase to the material density to allow for minor structural details that are not specifically included in a model, in order to align with a defined mass of material.
4.2 Outfit

Outfit masses can be categorised into two types: those which can be considered to be of a distributed nature, and those which are concentrated. The former consists of items such as systems, insulation and minor equipment. The latter can include, for example, engines, generators, steering gear, rudders and propellers, which would then be specifically defined in a model.

Distributed outfit masses can be included by factoring up the structural material mass density. In order to do this appropriately, it is usually necessary to divide the ship into material zones, because distributed outfit mass varies depending upon the type of compartment. Typically for a cargo ship, material zones would be arranged for aft peak, machinery spaces, the remainder of the hull, and the superstructure.

For passenger ships, where it is often desired to study the dynamic behaviour of large accommodation deck panels, it is necessary to be definitive about distributed outfit masses on decks. Hence, it is required to make separate material zones for these, and then apply non-structural mass individually. This can be done by:

- applying masses to a set of node points, or
- mass per unit area on plate elements, or
- by factoring the material density of the appropriate zones.

Outfit mass on accommodation decks can sometimes be available for an analysis, but often it is not. It has been found that an average value of 75 kg/m² is usually representative for internal accommodation deck panels, which includes deck coverings and fittings on the deck, together with systems and insulation attached below. However, variation between lightly and heavily loaded areas can often encompass a range such as 50 - 100 kg/m², so specific information should be obtained from the Shipbuilder or designer, if possible. A value of 30 kg/m² can be considered reasonable for an external deck panels above accommodation areas having only attachments beneath.

4.3 Internal Fluids and Cargo

For ships having large variations in loading conditions, such as tankers, LNG ships and container ships, it is usual to study a loaded and a ballast condition. For passenger ships, it is normally adequate to investigate a single loading condition.

Internal fluids can comprise fuel, water ballast and fresh water, together with fluid cargo, for example oil or LNG. Other notable types of cargo are: containers, vehicles on Ro-Ro ships, and bulk cargoes such as grain and ore.

The most rigorous approach for representation of internal fluids in a finite element model may initially appear to be fluid finite elements. However, for large scale global vibration models, this is unnecessarily complicated and impractical in computational terms. While use of boundary elements, such as those which can be employed to represent external fluids, is satisfactory for studying vibration behaviour of tank boundaries, they are not appropriate with respect to participation of the whole tank contents in global ship vibration.

Distribution of cargo 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 incorrect mass radius of gyration. However, normally this is not important, as hull torsion modes are not usually significant with regard to the acceptability of ship vibration behaviour. This method is satisfactory for small tanks in relation to all global vibration modes.

In relation to vehicle decks in Ro-Ro ships, the interface between vehicle masses and the decks comprises shock absorbers and tyres, or in other words, springs and dampers. Since the stiffness of these would be significantly weaker than that of the deck structure, they could be considered to be equivalent to a system of resilient mounts, thereby mostly isolating the vehicles from the deck.

4.4 Added Mass of Sea Water

Sea water surrounding a ship has an inertial effect on its dynamic behaviour. This can be treated as mass additional to the mass of a ship, and is known as ‘added mass’ or ‘virtual mass’. It is of significant magnitude: for a ship of full form in the vertical sense, it can be of the same order as the mass of the ship.

In early times, before the advent of sophisticated computational techniques, added mass was a difficult aspect to deal with in the dynamic analysis of ships.

The early method most commonly used was by Lewis: using a half-submerged cylinder of infinite length with a cross-section approximating to the midship section. Such a procedure exaggerates the effect of the water since it implies an assumption that the water can only move in a two-dimensional manner around the girth of a section. Since ship sections can vary rapidly along their length, the motion of water would not be so confined. Hence, a three-dimensional correction factor was incorporated into the method, which was based upon an equivalent ellipsoid of revolution.
Later methods that were developed included more specific definition of ship section shapes, such as the Frank Close Fit Method. The added mass of sea water varies with the direction of the dynamic motion and frequency. Early methods could reasonably account for direction in terms of the basic vertical and transverse modes, but essentially did not include the possibility of addressing the other variations.

With the availability of powerful computers, ‘boundary elements’, also known as ‘infinite elements’, defined on the wetted surface of ship finite element models, emerged as an efficient and accurate way of allowing for the virtual mass of sea water vibrating with a ship. The NASTRAN finite element program includes such a facility, where potential flow theory is used and the boundary elements constitute point sources.

Computational fluid dynamics (CFD) also affords increasing capabilities in relation to fluid-structure interaction and, in future, FEA and CFD may become more interrelated.

4.5 Shallow Water and Channels

Shallow water may be considered as a sea depth whose value is less than five times the vessel draught. It has the effect of increasing the virtual mass of entrained sea water, thereby reducing the vertical hull natural frequencies compared to those of a vessel travelling in deep water.

Horizontal modes are not significantly affected by the proximity of the sea bed; they are more affected if the vessel is travelling in a narrow channel.

The influence of the sea bed on the hull by way of virtual mass of entrained water is inversely proportional to the square of the distance between the sea bed and a hull.

Arbitrary plate elements together with boundary elements can be used to represent the sea bed or channel sides in order to produce the appropriate effect on added mass.

Section 5

Damping

5.1 General Description

Damping of ship vibration is complex and represents the greatest weakness in a dynamic analysis, particularly as most vibration problems occur in way of resonances, where the response is inversely proportional to damping.

This parameter has largely defied attempts to formulate a reliable theoretical predictive method, and measurements have indicated significant variations for similar and sister ships, without reasons being precisely quantified.

Damping occurs by way of dissipation of energy in various ways, which is finally converted into thermal energy.

5.2 Components

Hydrodynamic damping is important for ship motions. However, this external damping is generally considered to be insignificant in relation to ship vibration.

Internal damping is generally considered to be predominant for ship vibration, for which the primary constituent is the behaviour of material under stress: essentially energy absorption by material hysteresis. This is increased by stress concentrations and residual stresses from welding. Hence, some variation in workmanship and welding quality may account for apparent differences in damping between similar ships. This assertion may be strengthened by measurements that have been taken on naval ships that tend to indicate lower levels of damping than those for merchant ships. This may be in connection with the higher standards of detail design, workmanship and welding quality required for naval ships.

Damping indicates a rising trend with increasing frequency. This may be explained by the effect of shear stresses becoming more significant for higher vibration modes.

Cargo damping is another constituent of internal damping, which, of course, varies according to the type of cargo. Reliable quantification of this effect is not presently available.

5.3 Critical Damping

Critical damping is defined as the value of damping such that the response of a given mode to an impulse returns to equilibrium just without oscillating.

Damping is normally expressed as a percentage of critical damping. Ship structures are lightly damped; thereby a small percentage of critical damping applies.
5.4 Variation and LR Values

Structural damping generally increases with increasing frequency.

Damping levels can vary significantly, even between sister ships, without precise reasons being quantifiable. Hence, LR's policy is to use damping levels that are ‘realistic to pessimistic’ in dynamic analyses, generally such that actual vibration responses are similar to, or less than, predictions.

As mentioned in 1.1, the most notable frequency range in relation to global ship vibration is usually 5 – 20 Hz. LR uses damping, as shown in Table 5.1. These assumed levels have essentially withstood the test of time over many years of LR conducting vibration prediction analyses.

Table 5.1 Damping levels and their associated frequency ranges

<table>
<thead>
<tr>
<th>FREQUENCY (Hz)</th>
<th>DAMPING (percentage of CRITICAL)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0 - 5</td>
<td>1</td>
</tr>
<tr>
<td>5 - 20</td>
<td>1 – 3 by linear interpolation</td>
</tr>
<tr>
<td>20 and above</td>
<td>3</td>
</tr>
</tbody>
</table>

### Section 6

**Marine Vibration Excitation Sources**

6.1 Definition of Excitation Order

A term that is often used in relation to vibration excitations is ‘order’. First order is invariably defined as propeller shaft frequency. For example, if operating shaft Revolutions per minute (RPM) is 90, then first order is 90 cycles per minute (CPM), or 1.5 cycles per second (Hz). Other orders may then be defined in relation to this reference order. For the example of a four-bladed propeller, blade frequency would then be 4 x 1.5 = 6 Hz, which could be termed propeller blade order, or more specifically, fourth order.

6.2 Machinery

6.2.1 Slow Speed Diesel Main Engines

Slow speed Diesel engines are usually selected as the main engine in large cargo ships for reasons of power and economy: however, they usually have significant imbalances at several frequencies, and as they have to be bolted directly on to ship structure due to their invariably large size and mass, vibration excitations can be fully transmitted into the ship.

This type of engine is direct drive to the propeller shaft, that is, there is no gearbox. Hence, engine crankshaft frequency and propeller shaft frequency are the same. ‘Slow speed’ usually implies an operating RPM that is somewhere in the region of 70 to 130 RPM.

Excitations typically include first order moments in the horizontal sense (engine rotation about a vertical axis) and vertical sense (engine rotation about a transverse axis), and also second order moment in the vertical sense. Sometimes, a fourth order moment in the vertical sense may also be present.

At engine firing frequency (the order equals the number of engine cylinders for this type of engine), invariably there is a transverse rocking moment (engine rotation about a longitudinal axis) that is termed the ‘H-moment’. Sometimes, there can also be a longitudinal force of significance that is generated in the crankshaft at engine firing frequency and transmitted into the ship structure via the thrust block.

Other slow speed Diesel engine excitations generally comprise engine twisting moments, which are termed ‘X-moments’. These normally feature at several orders, but because they are internal moments whereby they sum to zero in the external sense, usually they only need to be considered if they are of large magnitude.

Because of the significant size, mass and vibration excitation characteristics of a slow speed Diesel engine, it is recommended to include a coarse representation of the engine in a ship finite element model; this also facilitates application of engine excitation moments, and transverse top bracing, if fitted. Engine dimensions, mass and vibration excitation details may be obtained from the engine builders or the engine designers’ websites.

Figure 2 shows the typical vibration excitation moments described above.
6.2.2 Medium and High Speed Diesel Main Engines

Medium speed Diesel engines operate in the region of 500 RPM and high speed Diesel engines in the region of 1000 RPM, through reduction gearboxes to the propeller shaft.

They are significantly smaller and lighter than slow speed Diesel engines. Engine manufacturers normally state for these types of engines that there is no significant external imbalance that could affect global ship vibration, and they normally specify resilient mounts together with suitable system connections. This should always be arranged for passenger ships and ferries where low vibration is required.

6.2.3 Diesel Electric and Auxiliary Machinery

Electric motors do not constitute a significant source of vibration within the frequency range applicable to global ship vibration. Power generation for supplying electric motors or auxiliary services is usually provided by medium speed Diesel engines. Thereby, comments in 6.2.2 apply.

6.2.4 Turbines

Steam turbines and gas turbines are in balance and do not emit vibration. If they do, it is an indication of failure, such as a broken turbine blade.

6.2.5 Propeller Shafts

As mentioned in 1.1, the shafting system is subject to mandatory classification rules, in relation to torsion, lateral and, in some cases also axial, vibration.

Dynamic characteristics of shafting systems can affect vibration behaviour of ship structure. Axial vibration of the shafting system can be transmitted into ship structure via the thrust block. Torsion vibration of the shafting system would be translated into axial vibration at the propeller, and could then also be transmitted into the ship structure via the thrust block.

A non-uniform wake will cause fluctuating forces on a propeller which will be transmitted to the shaft. Lateral modes of shafts may be excited by transverse and vertical forces and moments which are developed in this way. Vibration may be transmitted to the ship structure via the bearing housings. Avoidance of the coincidence of a lateral mode of the shaft within 20 per cent of an excitation...
source is typically required, though lateral modes may be sufficiently damped that excessive vibration does not occur despite coincidence. The propeller blade passing order is typically one of the more important sources of excitation for this phenomenon.

Propulsion shafting system vibration analysis, including the engine crankshaft, is often carried out by engine manufacturers and submitted via the Shipbuilder to classification societies for approval. Alternatively, the calculations can be carried out by consultants and submitted for approval, or performed by the classification society.

In some cases therefore, it may be relevant to consider longitudinal dynamic forces emanating from the propulsion shafting system in relation to ship vibration analysis, which would be primarily with respect to engine firing frequency of slow speed Diesel engines.

For certain passenger ships which have relatively stringent vibration criteria, in a few cases, excitation from the propulsion system at first order (propeller shaft frequency) has been noticed in measurements. In theory, this should not exist unless there is an imbalance in the system due to a fault. Where no fault is evident, the behaviour essentially remains unexplained. Another feature that has sometimes been noticed during measurements for multiple screw passenger ships is gradual alteration in phase between the propulsion systems, causing variation in measurements over time. This ‘propeller beating’ effect can be subjectively annoying. It can often be controlled by phase-locking the propeller shafts, preferably at the most favourable phase relationship for minimising vibration response.

6.3 Propulsors

6.3.1 Open Propellers

Propellers gain in terms of propulsive efficiency by being located behind the ship, in water that is moving at a velocity relative to the ship, which is lower than the ship speed; the so-called ‘wake’. However, there is a penalty in terms of vibration, because the water flow speed varies around the propeller disk. Hence, when the propeller rotates, dynamic forces are generated at the propeller shaft bearings, and through the water on to the hull surface above the propeller. Where propeller cavitation exists, the latter effect dominates propeller excited vibration.

The degree of non-uniformity of the wake, that is, irregularity of flow into the propeller disc, depends upon the position of the propeller and the after-body form of the ship. For a full form ship with a single propeller, such as a typical tanker, the wake may be quite irregular, with low velocity flow at the top of the propeller disk, and high at the bottom: in such conditions, significant cavitation may be generated around the top blade position. For a ship of finer form with a single propeller, the wake would tend to be more uniform, leading to lower dynamic forces. For twin screw ships and pod arrangements, propellers would usually be positioned in relatively uniform flow, which is favourable from the vibration point of view.

Propeller excitation of the hull can be thought of as consisting of a non-cavitating part, resulting from blade loading and thickness, and a cavitating part. For ships fitted with fixed pitch propellers, the non-cavitating part will dominate at low ship speeds where cavitation is not very pronounced. The non-cavitating pressure excitation resembles a sine wave with the blade passing frequency. At the ship’s normal continuous running, hull pressure is, typically, dominated by pressure radiated by cavitation dynamics. Cavitation is relatively repeatable per blade passage; however, temporal variations in the inflow can produce differences in practice.

Open propellers normally have four, five or six blades. During the design phase, propeller dynamic forces for input to a ship vibration analysis can be obtained from specialist calculations or model measurements. Reliable predictions from calculations are essentially limited to blade frequency and, somewhat less accurately, at twice the blade frequency; however, these are usually adequate to cover the frequency range that is relevant for global ship vibration. Calculations to predict excitation levels at higher orders are generally not reliable: these orders can be relevant in relation to noise and vibration of local panels close to the propeller. Model measurements can provide propeller excitation forces at blade and at twice the blade frequencies: also at higher orders, however, these have not always proven to be reliable.

Propeller dynamic forces at shaft bearings input to a ship vibration analysis would typically include longitudinal force (thrust variation) at the thrust block position, and transverse and vertical forces at the stern tube bearing position (shaft bracket locations for a twin screw arrangement).

Hull surface forces are applied to the hull above the propeller. LR uses total integrated hull surface force and applies it to a single substantial structural point above the propeller. In the past, forces distributed over an area of points above the propeller, together with phase differences between the points, were investigated, but the increased complication was generally not justified by the improved correlation with full-scale measurements for global ship vibration responses.

Recommended minimum propeller clearances are stated in LR Rules and Regulations for the Classification of Ships, Part 3, Chapter 3, 3.15.

6.3.2 Ducted Propellers

Ducted propellers can include:
- propellers in conventional positions with nozzles fitted (a common arrangement for tugs that require high thrust at low speed);
- water jets for high speed small craft;
- azimuthing thrusters with propellers that incorporate nozzles; and
- tunnel thrusters for manoeuvring purposes (usually only in port or restricted waters).

Generally, it is not possible to obtain reliable predictions of dynamic forces from ducted propellers by calculation or model tests. Hence,
predictive analysis is usually confined to resonance avoidance for large deck panels local to these devices. Thrusters can be of the ‘fixed RPM, variable pitch’ or ‘variable RPM, fixed pitch’ types. The former type is easier to deal with from the resonance avoidance point of view.

6.4 Sea Waves

6.4.1 Springing

Sea waves can induce ship vibration of a continuous nature, mostly at the lowest vertical natural frequency of the hull (2-node vertical) in way of the resonance with wave encounter frequency, which is termed ‘springing’. Generally, this is not significant with regard to acceptable levels of ship vibration from the habitability point of view, but may be of relevance in relation to structural fatigue for certain types of vessel.

6.4.2 Slamming and Whipping

Slamming impacts at the ends of a ship, that depend upon hull form, draught, sea conditions, ship speed and heading, can cause transient global vibration in the form of whipping, as well as local response. In most cases, structural strength issues are more relevant to consider than aspects of vibration, since comfort issues can usually be addressed by a change of ship speed and/or heading.

Section 7

Vibration Acceptance Criteria

As mentioned in 1.4, LR also has a document entitled Ship Vibration and Noise Guidance Notes that covers measurements and is a companion to this document. It includes a comprehensive description of vibration criteria, which will not be repeated here. However, criteria strictly apply to measurements, whereas comments in this document will be with reference to predictions.

7.1 Habitability Standards

Section 6 of LR’s Ship Vibration and Noise Guidance Notes cover this subject. Human tolerance of vibration is essentially subjective, and perceptions vary between individuals. However, the ISO produced criteria ISO 6954 in 1984. These are based upon single amplitude peak responses at a discrete frequency, that is, for narrow frequency band or single order. Vibration levels in the form of displacement, velocity and acceleration are presented in the acceptance chart, which are of course interrelated by simple harmonic motion. Vibration velocity in millimetres per second (mm/s) is usually chosen as the preferred parameter for assessment.

With reference to Figure 3, the acceptance chart shown on the next page, frequency is measured on the horizontal axis and vibration velocity is measured on the vertical access, and it essentially features three zones. These reflect the subjective nature of vibration acceptability: ‘adverse comments probable’, ‘adverse comments not probable’, and a transition zone between these. Above a frequency of 5 Hz, the vibration response level at the top of the transition zone is 9 mm/s and at the bottom is 4 mm/s. Below 5 Hz, a linear increase in acceptable vibration velocities is indicated, which is consistent with the generally increased human tolerance relating to lower frequencies.

In the past for frequencies above 5 Hz, 9 mm/s was often viewed as an acceptable vibration level for habitable areas of cargo ships. More recently, in connection with improved standards of crew accommodation, 4 mm/s is usually stated as the contractual requirement for cargo ships in relation to ISO 6954 (1984). Contractual requirements for passenger ships are, of course, invariably much more demanding: in accommodation areas, a criterion of 2 mm/s is typical; and this is even as low as 1 mm/s for some mega yachts.

As previously mentioned, vibration criteria are designed for application to measurements, which are made on ship trials or in service. The ISO 6954 (1984) standard recognises that peak responses in relation to measurements would not be expected to be in accordance with a perfect sinusoidal curve, as in a vibration prediction analysis, and it uses the description ‘maximum repetitive value’ instead. In practice, this is somewhat open to interpretation, and it has often led to a difference of opinion about the maximum repetitive value in a time series of measurements.

Mainly to resolve the issue of quantifying maximum repetitive value in relation to measurements, ISO 6954 (2000) was developed. This is based upon a weighted root mean square average vibration response for all frequencies in the range 1 - 80 Hz, that is, for broad frequency band or sum of all orders. Using appropriate measurement equipment then gives a single value for comparison with criteria, which makes the assessment of measurements easier and not open to interpretation.

Although an algorithm to adapt and combine predicted vibration responses may be applied for comparison with ISO 6954 (2000) criteria, this standard tends to be less well suited and informative in relation to vibration prediction analysis. Consequently, LR usually assesses predicted vibration responses on the basis of ISO 6954 (1984) as appropriate, which enables review of the response to each vibration excitation order separately.
7.2 LR Recommendation for ‘Occasionally Occupied Spaces’

Such areas could include bridge wings, areas in the hull aft and vehicle decks in Ro-Ro ships. Single amplitude peak responses of 10 mm/s at discrete frequency would be judged to be appropriate by LR.

7.3 LR Recommendation for Structure

Sections 4 and 5 of LR’s *Ship Vibration and Noise Guidance Notes* cover this subject.

In general, the usual recommendation for the avoidance of structural damage by vibration is not to exceed single amplitude peak responses of 30 mm/s at discrete frequency. However, it is appropriate to apply a lower level for certain critical structures, such as 15 mm/s for pump towers in membrane LNG ships, and in way of stern tube bearings.

7.4 LR Recommendation for Machinery

Section 7 of LR’s *Ship Vibration and Noise Guidance Notes* covers this subject.
Section 8
Vibration Problems and Solutions

8.1 Main Engine

As mentioned in 6.2.1, slow speed Diesel engines often feature significant dynamic forces at several orders. Hence, comments in this section relate to this type of engine. Engine manufacturers normally make recommendations regarding vibration countermeasures for fitting to their engines, if appropriate.

Engine excitation moments at first and second order can be resonant with the lower hull natural frequencies of large cargo ships, which can be a problem if the excitation magnitude is large and the engine is located around a node point of a corresponding vibration mode shape. First order moments feature in the horizontal and vertical sense, and static balance weights can be used to reduce moments in one direction at the expense of the other, depending on the potential problem. Second order moments are in the vertical sense only, and these can be neutralised by balancers fitted at the aft and forward end of the engine that consist of a system of rotating weights driven by the engine.

Some engines have a significant H-moment (sideways rocking) at firing order, for which the engine manufacturers may recommend transverse top bracing. These stays may be of the friction pad or hydraulic type, and they should be attached to substantial ship structure in order to be effective. They essentially alter the dynamic behaviour of the engine together with the ship structure in way of the engine room. Transverse top bracing may also be recommended for engines with large X-moments (twisting).

Troublesome H- and X-moments may also be reduced by changed the firing order of the engine, though this may not always be possible due to other restrictions, such as torsional stress limits on the shaft and certification of the engine for emissions etc.

In ships with more than one engine, phasing of the engines’ firing has been used effectively to reduce certain orders that were problematic. The fundamental concept is the same as that of using a compensator. The systems typically require relatively inexpensive additional instrumentation to be fitted to allow the ship’s engineers to read the engines’ relative phasing.

8.2 Propeller Shaft

As mentioned in 6.2.5, dynamic forces in the propulsion shafting system can be transmitted into the ship structure via the thrust block, primarily at engine firing frequency.

The magnitude of the dynamic forces depends upon the design of the shafting system; most notably the shaft diameter in relation to the torsional natural frequency, together with the engine characteristics. The ‘thin shaft’ design is usually employed, where the fundamental torsional natural frequency of the shaft is below engine firing frequency at operational RPM, and this leads to a ‘barred speed range’ of RPM in way of this resonance, whereby the ship is not to operate continuously within this range in order to avoid excessive vibration. The ‘thick shaft’ design has sometimes been used in the past, where the fundamental torsional natural frequency of the shaft is above engine firing frequency at operational RPM, in order to avoid a barred speed range. However, it can mean operating on the rising flank of this resonance, leading to unfavourable vibration responses, so it is not usually recommended.

Axial and torsional dampers are often a standard fitting for slow speed Diesel engines, at the forward end of the engine crankshaft.

8.3 Propeller

Vibration problems resulting from propeller excitation are, in some cases, the result of excessive excitation levels; and, in others, a result of resonances of structural natural frequencies with propeller excitation frequencies. A reduction in propeller excitation will be beneficial in both cases.

Propeller dynamic forces depend upon propeller design, loading and flow conditions into the propeller (wake).

The characteristics of the wake are primarily determined by the after-body design. Further, in some cases, the wake has been adversely affected by disturbances, such as cooling water discharges on the hull forward of the propeller. The wake is usually assessed by model measurements, though CFD is expected to have increasing application in this regard, and also leads to prediction of cavitation extent on propeller blades.

As mentioned in 6.3.1, propeller excitation may initially be evaluated by calculations or model measurements. As an approximate guideline for cargo ships, at propeller blade frequency, hull surface pressures above the propeller in excess of 8 kilopascals (kPa) would have a high probability of producing vibration problems, irrespective of notable structural resonances. Levels in the region of 4 to 8 kPa would probably lead to problems if there are significant structural resonances. Pressures of less than 4 kPa would be a favourable indication in most cases. In relation to twice the propeller blade frequency, the guideline would be half of the pressures stated for blade frequency. A further reduction of amplitudes is required for higher harmonics.

For passenger ships, which have more demanding vibration criteria, these guideline values typically could be halved, or even further reduced, for some mega yachts.

As may be discerned from the hull surface pressures mentioned above for blade and for twice the blade frequency (that is, first and second blade harmonics), it is expected that magnitudes should decrease in a similar manner for increasing harmonics. If that is not the case, problems may be indicated for the higher orders in relation to propeller noise and excitation of local panels of structure close
to the propeller. In relation to the latter, it is generally not feasible to deal with such problems by alteration of local structure to avoid
resonances, in view of the spread of panel natural frequencies and the broad frequency band of the excitation.

High propeller excitation levels usually imply high incidence of cavitation. Improvements can be effected by alterations to the propeller
design or modification of the wake. The latter is more easily effected and can be investigated using CFD, which for example may lead
to a recommendation to fit a flow alteration device such as a vortex generator, usually of elongated pyramid shape, to the hull forward
of the propeller for modifying the wake pattern. A typical example is shown in Figure 4, where the right hand side of the picture is
forward.

![An example of a vortex generator fitted to the hull, forward of the propeller](image)

In a few past cases of problematical propeller excitation in service, air injection into the top blade region has usefully been employed.
However, such a system is expensive to fit and maintain.
8.4 Hull

In general, it is not feasible to address overall hull vibration problems by structural alterations. Hence, remedial measures would comprise reduction in excitation levels, such as through balancing of the main engine. The lowest hull natural frequency is normally the 2-node vertical mode (as shown in Figure 5) or 1-node torsional mode. For the super long ships, this is often in the region of 0.5 Hz.

![Fig. 5](image1)  FE analysis result showing the 2-node vertical vibration mode

As mentioned in 8.1, slow speed Diesel main engines can cause significant hull vibration, normally occurring at a resonance with the fundamental natural frequencies of the hull. If such a problem were indicated, then the usual solution would be the prescribed balancing for the engine. This requirement is shown as an example in Figure 6 (4-node vertical mode). The engine is below the superstructure, and thereby unfavourably located for vertical moments, as it is at a node point.

![Fig. 6](image2)  FE analysis result showing the 4-node vertical vibration mode

Unfortunately, cases have occurred where both a vertical and a horizontal hull mode were in the operational speed range and were excessively excited by the first order moment of the engine. In these cases, balancing to reduce either the horizontal or vertical engine first order moment may be undertaken, and a compensator fitted to counter the remaining out-of-balance engine first order moment.

In relation to high propeller excitation, this can sometimes bring about significant vertical vibration at the aft end of the hull, particularly when there is resonance between propeller blade frequency and a vertical aft end vibration mode – often known as a ‘fan-tail’ mode (see Figure 7). Local structural alterations, particularly regarding longitudinal bulkheads in the aft end of the hull may be feasible at an early design stage; otherwise attention can be given to propeller excitation or the number of blades. It should be noted that the aft end of the hull in cargo ships does not contain permanently occupied spaces that are subject to habitability standards.
In 6.4.2, it was stated that sea waves can cause uncomfortable vibration by way of slamming impacts at the ends of a ship. Slamming forward can, of course, be addressed by a reduction in ship speed and/or a change of heading. For a few passenger vessels with flat counter sterns at a particular height above the sea surface, slamming impacts can cause unpleasant vibration in accommodation spaces aft. However, this tends to be at low speed whilst manoeuvring, or stationary in response to wave trains caused by passing vessels in port, since at higher speed a stern wave usually immerses the flat counter stern. If such a relatively unusual phenomenon is experienced in service, a known solution is to create an air cushion under the flat counter stern by incorporating a ‘skirt’ arrangement around the area, including internal diaphragms for strength purposes and air pressure relief apertures.

8.5 ‘Tower Block’ Superstructure

Most cargo ships have crew accommodation and operating spaces placed in tower block deckhouses at the aft end of the hull, for reasons of payload maximisation and cargo handling. However, this is the least favourable location from the vibration point of view, since it is above machinery compartments and close to the propeller. Furthermore, the tower block is relatively high for visibility from the navigation bridge.

Hull and superstructure vibration modes are invariably coupled, such that the superstructure will follow hull vibration mode shapes. Notwithstanding, it has been indicated many times by past practical experience that the fundamental longitudinal natural frequency of a superstructure accommodation block is particularly important to consider in relation to resonance avoidance. Relevant excitations in this respect are firing frequency of slow speed Diesel engines and propeller blade frequency.

Excitation at engine firing frequency can arise from an H-moment and, although this is a transverse moment, it can sometimes lead to some longitudinal vibration in the superstructure. Further, as previously discussed for some cases, there can be a longitudinal dynamic force imposed on the thrust block by the propulsion shafting system.

The primary excitation at propeller blade frequency is usually the hull surface force on the aft end of the hull above the propeller, as mentioned in 8.4, whereby vertical movement of the aft end in turn brings about longitudinal displacement, by way of rotation for a tower block superstructure that is located just forward of the aft end (see Figure 8).
The fundamental longitudinal natural frequency of a tower block superstructure is primarily determined by its height and length, followed by the configuration and continuity of the longitudinal bulkheads, the support below the superstructure, and the total mass including outfit. This natural frequency is often within the range 6 – 10 Hz.

Transverse vibration and torsional vibration of the superstructure, which are mostly induced by the main engine H-type moment and X-type moment through transverse top bracings and double bottom structure, may also occur during vessel sea trials and in normal operation. Quite often, the natural frequencies of transverse and torsional vibration modes are higher than those of longitudinal modes, but they may vary in different cases.

If a problem of resonance with the fundamental longitudinal natural frequency of a superstructure accommodation block is indicated, significant structural alterations are not usually feasible or very effective, except for a connection of the accommodation block to a separate funnel casing block for applicable cases (at the expense of increased noise transmission). Normally, structural alterations are also not feasible if resonance happens with transverse and torsional vibration modes of the superstructure block.

The excitation source may be addressed to effect a solution. If it is the propeller, attention can be given to excitation level and/or number of blades. If it is the engine, attention can be given to the H-moment / X-moment by way of top bracing, or damping devices for the propulsion shafting system, as appropriate.

Another possible solution that has been employed in some cases, usually post-trials, is the fitting of a vibration compensator. This is a compact, electrically controlled device incorporating rotating weights that can be installed on or near to the navigation bridge. It can be adjusted for force magnitude, phase and frequency to neutralise or reduce vibration. It tends to be more effective as a countermeasure for machinery excited vibration, rather than from the propeller, as the former is of a more consistent nature. Overall dimensions are within about one metre.

8.6 Bridge Wings

Cantilever bridge wings are often incorporated in cargo ships, for the purpose of visibility along the ship sides.

Undesirable vibration levels can occur if there is resonance between the fundamental longitudinal or vertical vibration modes of the bridge wings, with slow speed Diesel engine firing frequency or propeller blade frequency in way of operating RPM and, to a lesser extent, at twice the propeller blade frequency. Further, if there is resonance between engine firing frequency or propeller blade frequency, together with the fundamental longitudinal natural frequency of the tower block superstructure and, simultaneously, the fundamental longitudinal natural frequency of the bridge wings, the vibration responses can be very high.

Consequent to the above, it is recommended to calculate the fundamental longitudinal and vertical vibration modes of the bridge wings at the design stage. This can be part of a global ship vibration analysis, or a bridge wing local model with suitable boundary conditions at the bridge side, to evaluate natural frequencies. An example of such a model is shown in Figure 9, indicating the fundamental longitudinal mode.
If a problem is identified, it can be addressed by structural modifications, such as struts or large brackets. An addition of mass at the ends of the bridge wings, in order to lower their natural frequency, may also be feasible.

8.7 Large Span Deck Panels

These are of primary concern for accommodation areas in passenger ships, which are subject to habitability standards. Vehicle decks in Ro Ro ships also feature, but these would only be described as ‘occasionally occupied spaces’.

From the resonance avoidance point of view, it is possible to study large span deck panels by using local models, subject to suitable boundary conditions being arranged. However, they would normally be incorporated in a global vibration analysis that would evaluate natural frequencies and vibration responses. Typical large deck panel vibration modes are shown in Figure 10.

The relevant excitation frequency would be the propeller blade frequency and the main engine firing order frequency; thereby essentially being relevant for panels aft of amidships. In order to avert possible undesirable vibration while occasionally using manoeuvring devices, such as tunnel thrusters, resonance avoidance can be implemented for large deck panels close to their locations.

Problems that are identified can be attended to by increasing numbers and/or scantlings of primary girders/stiffeners or by introducing additional vertical supports such as pillars or bulkheads with structural continuity below, subject to arrangement considerations.
8.8 Panels Local to Propeller

A resonance avoidance procedure should be applied for these, using analytical or local finite element analysis to evaluate natural frequencies. For naval ships, this is a LR class requirement.

Resonance avoidance should be implemented for aft peak panels, including the effect of the added mass of fluid where appropriate, in order to avoid potential structural damage. The relevant excitation frequencies for comparison are at propeller blade and at twice the blade frequencies. The most common problem experienced in this regard is cracking of fresh water tank bulkheads that are located in the aft peak.

Resonance avoidance should be implemented for local panels in accommodation decks close to propellers such as in tower block superstructures aft, including outfit mass, where appropriate, in order to avoid potential infringement of habitability standards. The relevant excitation frequency for comparison is propeller blade frequency.

Problems are usually addressed by an increase to stiffening of the panels.

8.9 Masts on Naval Ships

Masts on naval ships often have sensitive surveillance equipment mounted on them, so it is important to avoid resonance of fundamental mast natural frequencies with any significant excitation frequencies that can be transmitted at the base of the mast.

Hence, it is recommended to calculate the fundamental longitudinal and transverse vibration modes of the masts at the design stage.

The masts should have substantial support arranged, so that local vertical vibration is not an issue. This can be calculated as part of a global ship vibration analysis or as a local analysis, using a model of the mast with suitable boundary conditions applied. The structural design can then be amended, if unfavourable resonances are indicated.