

Guidance Notes On

Ship Vibration



September 2023



GUIDANCE NOTES ON

...
SHIP VIBRATION
SEPTEMBER 2023

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Foreword (1 September 2023)

The American Bureau of Shipping recognizes the study of overall ship vibration as an important measure in providing the habitability, safety and functionality of vessels. The ABS *Guidance Notes on Ship Vibration* have been developed to provide users with specific guidance on the design, analysis, measurement procedures and criteria to achieve the goal of reducing ship vibration to an acceptable level.

The design and construction of the hull, superstructure and deckhouse of a steel vessel are to be based on all applicable requirements of the *Marine Vessel Rules*. Specifically, for Container Carriers over 130 meters in length, the ABS *Marine Vessel Rules* require the consideration of vibratory responses of hull structures, as applicable (5C-5-3/13.1). For the LNG Carriers, the ABS *Marine Vessel Rules* require special attention to the possible collapse of membrane due to hull vibration (5C-8-4/13.5). For ship-type floating production units with spread mooring, the ABS *Rules for Building and Classing Facilities on Offshore Installations* require the flare tower/boom and hull natural frequencies to be separated to avoid resonance or near resonance (3-3/9.17). In conjunction with propulsion shaft alignment, the *Marine Vessel Rules* require the consideration of propulsion shaft vibrations (4-3-2/7). For cargo and passenger vessels, ABS provides optional classification notations for crew habitability and passenger comfort (ABS *Guide for Passenger Comfort on Ships* and *Guide for Crew Habitability on Ships*). Also, ABS offers a Condition Monitoring Program for machinery vibration (7-A-14/5.1.2 of the ABS *Rules for Survey After Construction (Part 7)*).

These Guidance Notes assist ship designers to avoid excessive shipboard vibration at an early design stage by providing guidelines on the concept design. These Guidance Notes also assist with the finite element analysis (FEA) based vibration analysis procedure to calculate the vibration response and evaluate the design at detail design stage. The analysis procedure **provides guidelines on finite element (FE) modeling, engine and propeller excitation, and free and forced vibration analysis** and represents the current analysis practice in ABS. These Guidance Notes also offer guidelines on the vibration measurement procedure at sea trials and the acceptance criteria on vibration limits based on the international standards and the practice in ABS.

The 2018 edition provided guidance on quick local vibration analysis methods for ships.

The 2021 edition introduced a new coefficient of the Kumai formula for calculating the hull girder 2-node vertical vibration mode natural frequency of ship-type floating production units.

The 2023 edition introduces new recommendations for modeling engine and propeller excitations, as well as combining excitations and modeling of cargo and fluid inside tanks. It aligns the measurement procedures with the ABS *Guide for Vibration of Machinery Equipment and Related Structures* and references that Guide for vibration criteria regarding machinery, equipment, and local structures.

These Guidance Notes become effective on the first day of the month of publication.

Users are advised to check periodically on the ABS website www.eagle.org to verify that this version of these Guidance Notes is the most current. Comments or suggestions can be sent electronically to rsd@eagle.org

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1 Introduction (1 September 2023)

With the increase of ship size and speed, shipboard vibration is an increasing concern in the design and construction of vessels. Excessive ship vibration should be avoided for passenger comfort and crew habitability. In addition to undesired effects on humans, excessive ship vibration may result in the fatigue failure of local structural members **and** malfunction of machinery and equipment.

These Guidance Notes provide users, specifically shipyards, naval architects and ship owners, with guidance on the concept design to avoid excessive ship vibration at an early design stage. If appropriate procedures are followed with insight and good judgment in the concept design stage, then the difficult countermeasures and corrections at the subsequent design stages may be avoided in most cases.

These Guidance Notes also assist with the finite element analysis (FEA) based vibration analysis procedure to predict the vibration response and evaluate the design in detail design stage. The vibration analysis procedure represents the most current analysis practice in ABS. These Guidance Notes also offer guidelines on the vibration measurement procedure during the sea trials and the acceptance criteria on vibration limits based on international standards and practice in ABS.

3 Application (1 May 2021)

These Guidance Notes are applicable to the vessels of all lengths, including ship-type floating production units.

5 Scope (1 September 2023)

These Guidance Notes provide overall guidelines on ship vibration excited by the main engine, propeller, or slamming. In these Guidance Notes, the following subjects are considered:

- i)* Concept Design
- ii)* Vibration Analysis
- iii)* Measurements
- iv)* Acceptance Criteria

The concept design in Sections 2, 3 and 4 provides users with guidance on dealing with ship vibration in the *concept design* stage. In the concept design, local vibration is not addressed because detail information is not usually available in the early design stage. Instead, the concept design is focused on those areas that have been known to be of critical importance in avoiding harmful ship vibration.

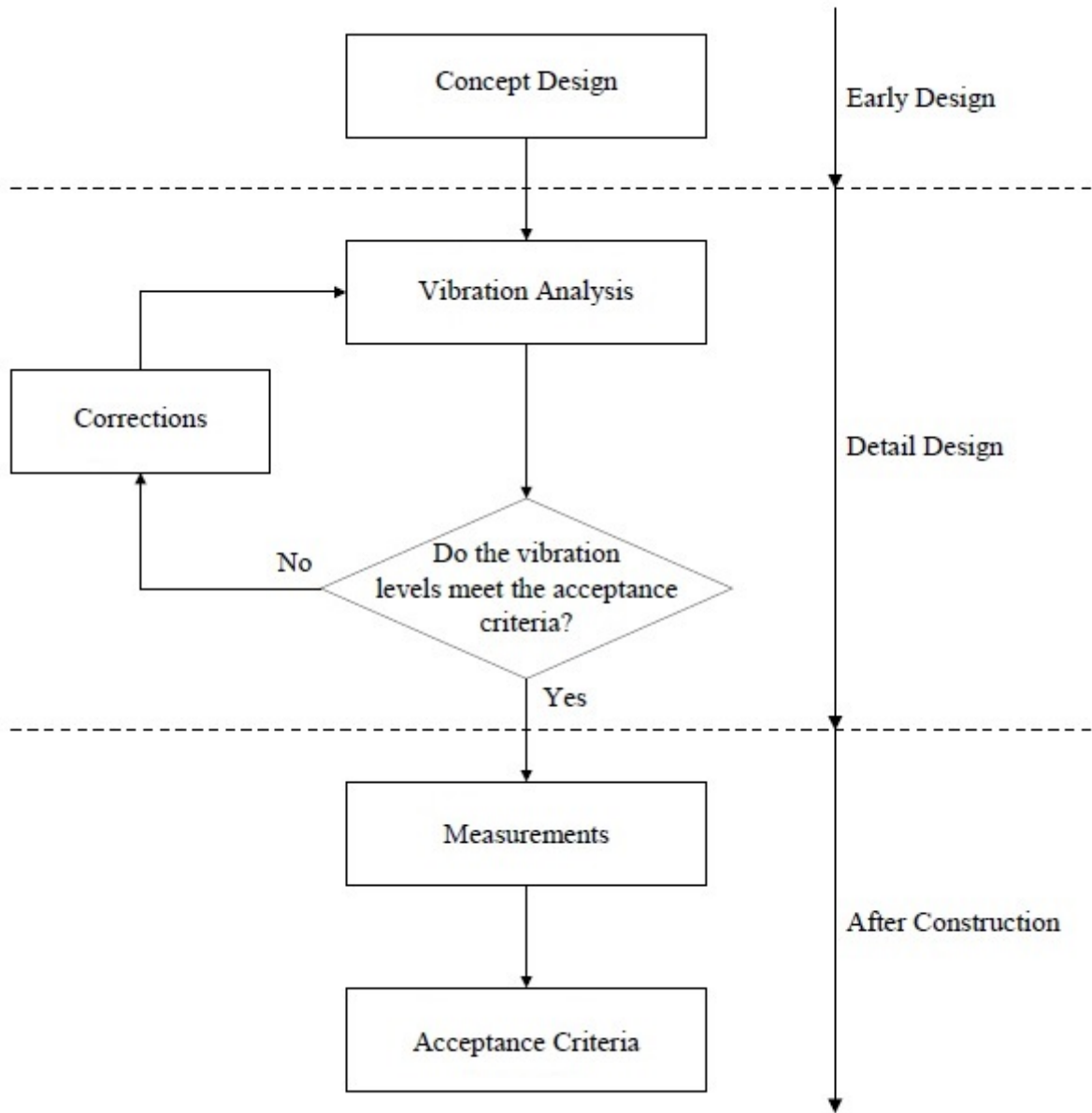
The vibration analysis in Section 5 provides the FE-based vibration analysis procedure based on first principles direct calculations. The FE-based vibration analysis is recommended to evaluate the design

during the detail design stage. If found necessary, the local vibration should be addressed in the detail vibration analysis. The analysis procedure provides guidelines on FE modeling, engine and propeller excitation and free and forced vibration analysis. The procedure for calculating slamming excitation can be found in the *ABS Guide for Slamming Loads and Strength Assessment for Vessels*. Site-specific wave scatter diagram should be used for slamming calculations **for ship-type floating production installations**.

For the assessment of ship vibration performance, the actual vibration levels at the most critical locations should be measured and evaluated during the sea trials. Section 6 provides guidelines on the vibration measurement procedure on the instrumentation, measurement conditions and locations, data processing and reporting. Section 7 provides acceptance criteria on the vibration limits for human comfort and habitability, local structures and machinery based on international standards and practice in ABS.

The shaft alignment and torsional vibration are not directly addressed in this document. For the requirements of the shaft alignment and torsional vibratory stress, refer to 4-3-2/7 of the *ABS Marine Vessel Rules*. The overall procedure for ship vibration assessment recommended in these Guidance Notes is shown in 1/5 FIGURE 1.

FIGURE 1
Overall Procedure for Ship Vibration Assessment



1 Introduction

The vibration avoidance process should begin at the concept design phase. It has been shown that if those vibration problems, identified by experience as the most important, are addressed at the earliest design stage, serious problems involving great cost in correction efforts may be avoided.

Considering vibration should begin early at the concept design stage, where there has been no development of details. Simple tools and rules of thumb available at that level can aid in avoiding major vibration problems. Major potential problems may often be present in the crude concept design definition. Simply identifying and addressing those potential problems in terms of the minimal technology available at the concept design stage is considered very important to the success of ship design.

Sections 2 through 4 provide guidelines on the concept design. Some of these guidelines are presented in *Principles of Naval Architecture, Chapter 7* (SNAME, 1988) and SNAME granted permission to be included in this document.

3 Design Considerations (1 September 2023)

The **five** elements of importance in ship vibration are:

- Excitation
- Stiffness
- **Mass**
- Frequency Ratio
- Damping

It is noted that any of the following contribute to vibration reduction:

- i)* **Reduction of excitation force amplitude, F .** In propeller-induced ship vibration, excitation may be reduced by changing the propeller unsteady hydrodynamics. This may involve lines or clearance changes to reduce the non-uniformity of the wake inflow or may involve geometric changes to the propeller itself. Specifics in this regard are addressed in Section 3.
- ii)* **Increase of stiffness, K .** Stiffness is defined as spring force per unit deflection. In general, stiffness should be increased rather than decreased when variations in natural frequency are to be accomplished by variations in stiffness. It is not recommended to reduce system stiffness in attempts to reduce vibration.
- iii)* **Mass of the structure and its distribution also directly influence the natural frequencies of the ship structure. However, the mass is usually determined by requirements of structural strength,**

installed equipment, cargo capacity, etc., and designers have less flexibility in changing it significantly to affect the natural frequencies.

- iv) *Avoidance of values of frequency ratio near unity; $\omega/\omega_n = 1$ is the resonant condition.* At resonance, the excitation is opposed only by damping. Note that ω/ω_n can be varied by varying either excitation frequency ω or natural frequency ω_n . The spectrum of ω can be changed by changing the RPM of a relevant rotating machinery source, or, in the case of propeller-induced vibration, by changing the propeller RPM or its number of blades. ω_n is changed by changes in system mass and/or stiffness; increasing stiffness is the usual and preferred approach. Specific measures for resonance avoidance in ships are addressed in Section 4.
- v) *Increase of damping, ζ .* Damping of structural systems in general, and of ships in particular, is small; $\zeta \ll 1$. Therefore, except very near resonance, the vibratory amplitude is approximately damping independent. Furthermore, damping is difficult to increase significantly in systems such as ships; ζ is, in general, the least effective of the **five** parameters available to the designer for implementing changes in ship vibratory characteristics.

5 Concept Design Approach (1 September 2023)

Five elements were identified in the preceding as being influential in determining ship vibratory response, and their relationship to vibration reduction was discussed. While quantification of all **five** elements is required in calculating the vibration response level, acceptable results may consistently be achieved with reasonable effort by focusing attention in concept design on two of the **five** elements. The two of the **five** elements of importance are *excitation* and *frequency ratio*. Application of these two objectives has improved ship design:

- Minimization of dominant vibratory excitations, within the normal constraints imposed by other design variables, and
- Avoidance of resonances involving active participation of major subsystems in frequency ranges where the dominant excitations are strongest.

Unlike vibration response, the excitation and frequency ratio elements involved in these objectives may be predictable with useful reliability for making design judgments. For propeller excitation, detailed hydrodynamic calculation procedures in conjunction with model testing have been established, at least to the level of reliable relative predictions in the post-concept, preliminary design stage. Natural frequencies involving the ship hull and its major subsystems are usually predictable using modeling and modern numerical structural analysis methods. Accuracy levels achievable in predictions of propeller and engine excitation and of sub-system natural frequencies are generally considered to be reliable enough to consistently achieve the two objectives cited above.

The detailed calculations and experiments required to avoid resonance and minimize excitation are usually performed by engineering analysis groups or model basins and are usually not the immediate responsibility of the ship designer. If addressed early enough while basic changes in a design can still be made, excessive vibration may be avoided effectively. The main functions of the ship designer in this regard are to:

- Establish a concept design to serve as the subject of the detailed investigations,
- Judge whether detailed analysis is warranted as a further step.

The quality of the concept design will be reflected in the number of detailed iterations that are required for achieving an acceptable final design. To establish high-quality concept designs which may require time consuming calculations and model testing, the designer requires both guidance to focus on areas of the design likely to be in most need of attention, and some simple methodology for identifying the critical aspects. Experience has shown that extra attention to vibration in concept design of large ships is warranted in the following areas:

- i) Main engine excitation (see Subsection 3/3)

- ii)* Stern lines and propeller clearance (see Subsection 3/5)
- iii)* Alternating thrust and cavitation (see Subsection 3/7)
- iv)* Hull girder vertical vibration excited by the main engine, propeller, slamming, or springing (see Subsection 4/3)
- v)* Main machinery/shafting system longitudinal vibration excited by the propeller (see Subsection 4/5)
- vi)* Superstructure fore-and-aft vibration excited by hull girder vertical vibration and/or main propulsion machinery/shafting system longitudinal vibration (see Subsection 4/7)

Various local vibrations, such as hand-rails, antennas, and plating panels, may be encountered on new vessel trials in addition to these **six areas**. However, local problems usually involve local structural resonances and are often considered to be minor problems, as the corrective approach of local stiffening may be easily achievable.

The concept design checklist is summarized in Appendix A4.

1 Introduction (1 September 2023)

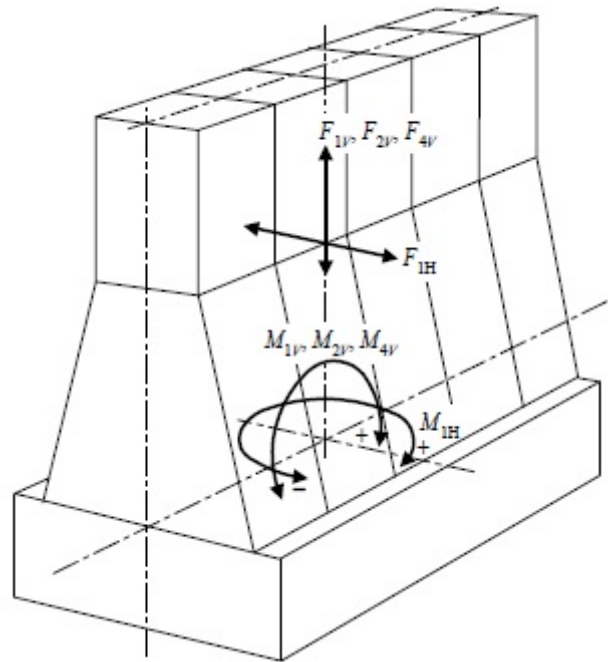
The principal vibration excitation sources should be addressed first, since with high excitation levels excessive vibration can occur almost independently of system structural characteristics. In general, the major sources are the low-speed diesel main engine and the propeller. Gas and steam turbines are generally considered to give less excitation than diesel engines. Thus, this section focuses on the excitation of the low-speed main diesel engine.

3 Low-speed Main Diesel Engine (1 September 2023)

Significant progress has been made in recent years by engine manufacturers in reducing vibratory excitation, largely by moment compensators installed with the engine. Steps to be taken by the engine manufacturer should be addressed in the main engine specifications for a new vessel. In this regard, it is important for the shipyard or owner to be technically knowledgeable on this issue. Diesel engine vibratory excitation can be considered as composed of three periodic force components and three periodic moment components acting on the engine foundation. The periodic force component along the axis of the engine is inherently zero. Certain other excitation components may also balance to zero, depending on particular engine characteristics.

Two distinctly different types of forces can be associated with the internal combustion reciprocating engine. These are: (a) gas pressure forces due to the combustion processes (guide force couples) and (b) inertia forces produced by the accelerations of the reciprocating and rotating engine parts (external forces). 3/3 FIGURE 1 shows the typical external force and moments acting on a diesel engine.

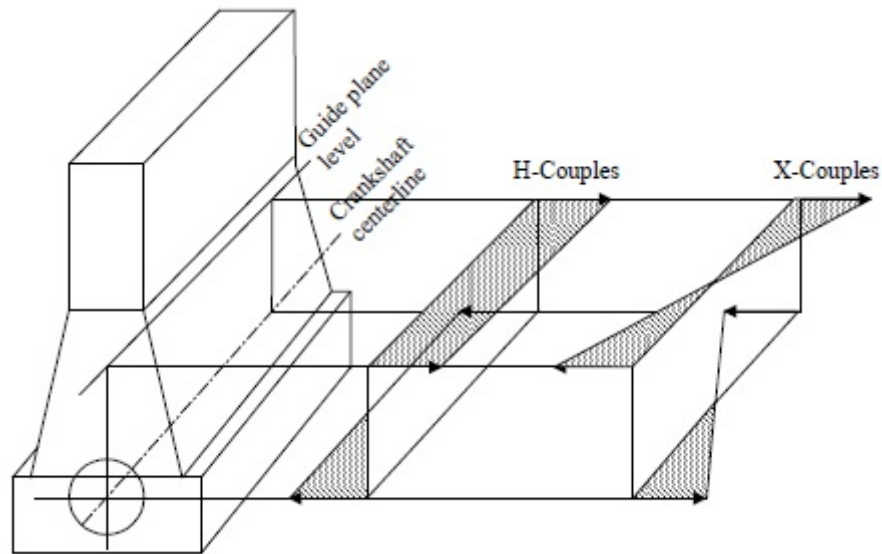
FIGURE 1
External Forces and Moments



Guide force couples acting on the crosshead result from transverse reaction forces depending on the number of cylinders and firing order. The guide force couples cause rocking (H-couples) and twisting (X-couples) of the engine, as shown in 3/3 FIGURE 2. The engine lateral vibration due to the guide force couples may cause resonance with the engine foundation structure. A possible solution to this resonance problem at the concept design stage may be a consideration of lateral stays (top bracings), which connect the engine's top structure to the ship hull.

The effectiveness of the engine stays (bracings) is significantly diminished if the engine is fitted inside a soft engine room, which is characterized by web frames with short webs, large openings, and pillars. On the other hand, the top bracings can be very effective if they are connected to a stiff engine room structure, which is characterized by tall web frames and strong connections between the double bottom and web frames with high radial stiffness. In stiff structures, the web frames are usually plated structures without any openings. The engine manufacturer's recommendations on minimum structural stiffness in the lateral or longitudinal directions at the point where lateral or longitudinal stays are attached to the hull structure, respectively, should be followed.

FIGURE 2
Guide Force Couples



The vertical force and moment, which are of primary concern with regard to hull vibratory excitation, and the transverse force and moment, are due to unbalanced inertial effects. For engines with more than two cylinders, which is the case with ships, the vertical and transverse inertia force components generally balance to zero at the engine foundation. This leaves the vertical and transverse moments as areas of concern. The values of the moment amplitudes are usually tabulated in the manufacturer's specification for a particular engine.

The majority of low speed marine diesels currently in service have six cylinders or more. Therefore, the 2nd order vertical moment M_{2v} is generally considered to contribute the most to the hull vibration. However, depending on the specific number of cylinders, the first order or higher order moment can be as large as the second order moment. In that case, further consideration should be given to the first or higher order moment.

A hull girder mode up to the third or fourth can have a natural frequency as high as the twice-per-revolution excitation of the 2nd order vertical moment. Hull girder modes higher than the first three or four have diminishing excitability and may be of less concern. The following steps are therefore recommended in concept design:

- i)* The 2nd order vertical moment M_{2v} is the diesel engine excitation of most concern. The potential danger is in resonating one of the lower hull girder vertical modes with a large 2nd order vertical moment. The value of M_{2v} should be requested from the potential main engine manufacturer, as early as possible **in the design process**.
- ii)* Power Related Unbalance (*PRU*) values may be used to determine the acceptable level of M_{2v}

$$PRU = \frac{M_{2v}[N - m]}{EnginePower[kW]}$$

TABLE 1

<i>PRU</i>	<i>Need for Compensator</i>
Below 120	Not likely
120-220	Likely
Over 220	Most likely

Further attention is recommended in cases where PRU exceeds 220 N-m/kW. The recommended action at the initial engine selection stage may be to either select a more suitable engine or to install moment compensators supplied by the engine manufacturer. Otherwise, the vertical hull girder response should be checked by calculation to be within an acceptable level without installation of compensators.

- iii) The engine lateral vibration due to X-type and H-type moments may produce excessive local vibration in the engine room bottom structure depending on the engine frame stiffness and engine mounting. The installation of lateral stays on the engine room structure should be addressed at the early design stage.

5 Hull Wake

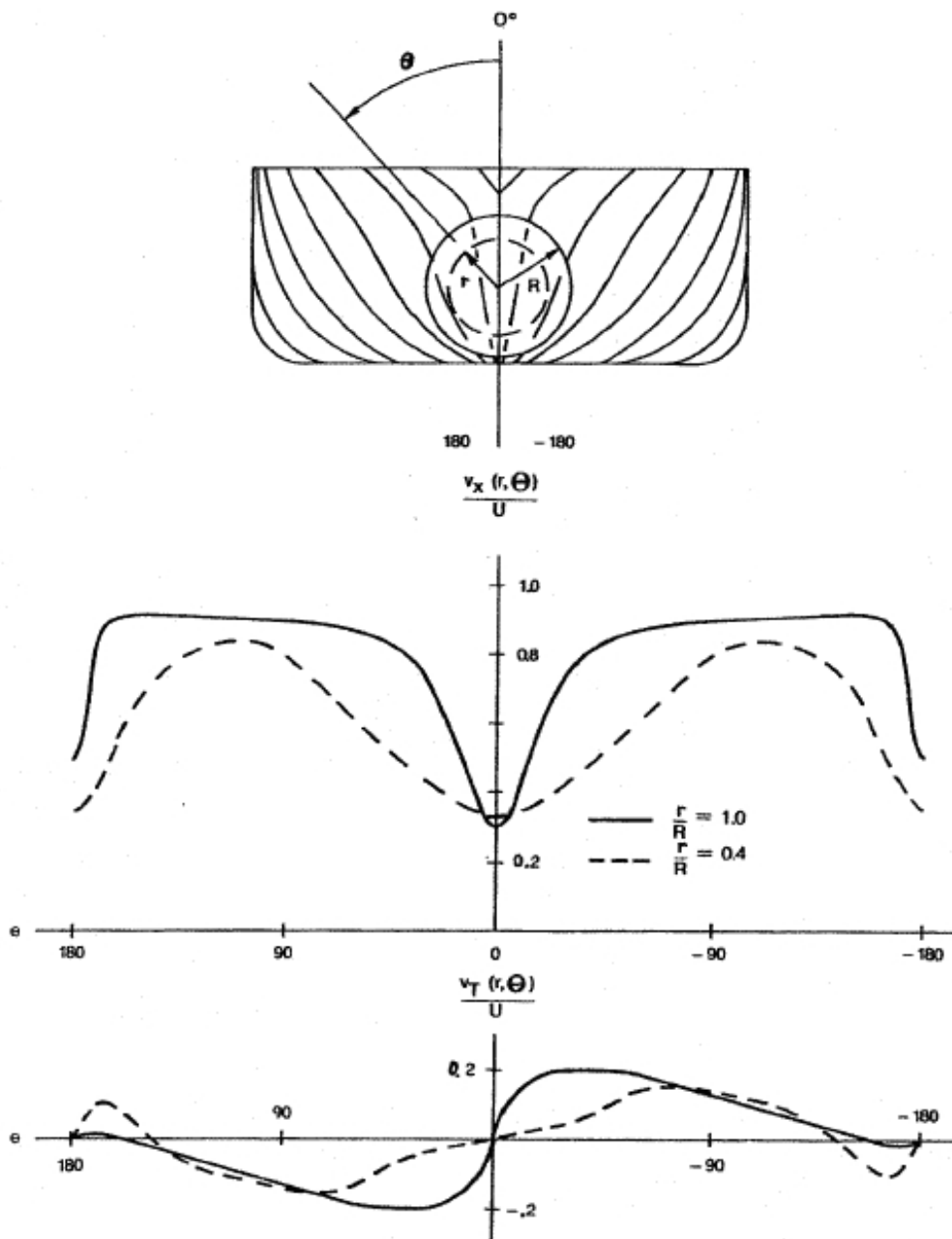
Hull wake is one of the most critical aspects in avoidance of severe ship vibration. Propeller-induced vibration problems in general start with hull lines in the stern aperture region leading to significant non-uniformity of wake in which the propeller must operate. Propeller excitation is far more difficult to quantify than the excitation from internal machinery sources. This is because of the complexity of the unsteady hydrodynamics of the propeller operating in the non-uniform hull wake. The non-uniform hull wake is the most complicated and critical aspect. Propeller-induced vibration would not be a consideration in ship design if the propeller disk inflow were circumferentially uniform. Any treatment of propeller excitation should begin with a consideration of the hull wake.

For engineering simplification, the basic concepts allow for the circumferential non-uniformity of hull wakes, but assume, for steady operation, that wake is time invariant in a ship-fixed coordinate system. Nominal wake data from model scale measurements in towing tanks are presented either as contour plots or as curves of velocity versus angular position at different radii in the propeller disc. 3/5 FIGURE 3 shows the axial and tangential velocity components for a typical conventional stern merchant ship.

The position angle, θ , on 3/5 FIGURE 3 is taken as positive counterclockwise, looking forward, and x is positive aft. The axial wake velocity v_x and tangential wake velocity v_T are dimensionless on ship forward speed, U . Note that the axial velocity is symmetric in θ about top-dead-center (even function) and the tangential velocity is asymmetric (odd function). This is a characteristic of single screw ships due to the transverse symmetry of the hull relative to the propeller disk; such symmetry in the wake does not exist with twin-screw ships.

The wake illustrated above represents one of the two characteristically different types of ship wakes. The flow character of the conventional skeg-stern is basically waterline flow; the streamlines are more or less horizontal along the skeg and into the propeller disk. The flow components along the steep buttock lines forward of the propeller disk are small. The dominant axial velocity field of the resultant wake has a substantial defect running vertically through the disk along its vertical centerline, at all radii.

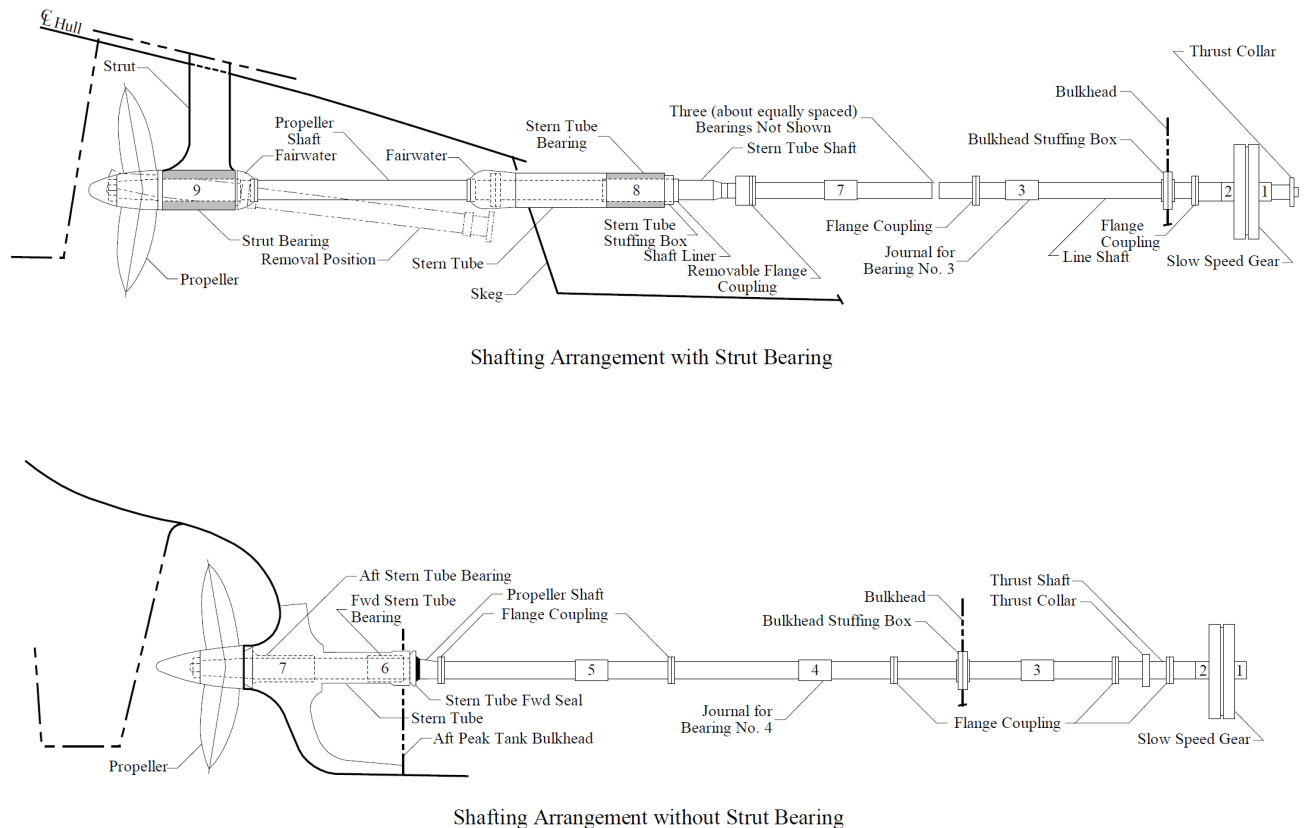
FIGURE 3
Nominal Wake Distribution for a Typical Merchant Ship
(DTMB Model 4370, $C_B = 0.6$)



This defect is the shadow of the skeg immediately forward. The tangential flow in the propeller disk, being the combination of the component of the upward flow toward the free surface and any disk inclination relative to the baseline, is much smaller.

The idealization of this wake, for conceptual purposes, is the two-dimensional flow behind a deep vertical strut placed ahead of the propeller. In this idealization, the axial velocity distribution is invariant vertically, and any tangential velocity distribution (due to disk inclination) is asymmetric about the vertical disk axis. This basic characteristic is exhibited in the 3/5 FIGURE 3.

FIGURE 4
Alternative Shafting Arrangements:
Open Strut Stern (upper); Conventional Skeg Stern (lower)



A characteristically different wake flow is associated with the *strut or barge-type* stern, the upper image of 3/5 FIGURE 4, which has a broad counter above the propeller disk and minimal irregularity immediately forward. The engine is further forward with this stern-type to accommodate the finer stern lines needed to minimize wave resistance in high speed ships, although an open strut stern would be beneficial for vibration minimization at any speed (provided the buttocks lines are not too steep). The flow character over this type of stern is basically along the buttock lines, versus the waterlines. Some wake non-uniformity may be produced by appendages forward, such as struts and bearings or by shaft inclination, but the main wake defect, depending on the relative disk position, will be that of the counter boundary layer overhead. In this case, assuming minimal shaft inclination, substantial axial wake again exists, but only in the top of the disk. Generally, only the blade tips penetrate the overhead boundary layer, and the axial wake defect occurs only at the extreme radii near top-dead-center, rather than at all radii along the vertical centerline, as in the case of the conventional single screw stern. Just as in the case of the conventional stern, the tangential disk velocity with the strut stern will generally be small; the vertically upward velocity ratio through the propeller disk will have average values on the order of the tangent of the sum of the buttock and shaft inclination angles. The idealization in the case of the barge stern, as a sequel to the vertical strut idealization of the wake of the conventional stern, is a horizontal flat plate above the propeller. Here, the degree of axial wake non-uniformity depends on the overlap between the propeller disk and the plate boundary layer. The tangential (and radial) wake components are due entirely to the shaft relative inclination angle in this idealization, as the flat plate boundary layer produces only an axial defect.

5.1 Hull-Propeller Clearance (1 September 2023)

The distinction between the two basically different wake types is useful in understanding the importance of clearances between the propeller blades and local hull surfaces. Firstly, it is helpful to consider the hull surface excitation by the propeller as composed of two effects:

- i) *Wake effect.* The effect of changing the wake inflow to the propeller according to a specified propeller relocation, but with the propeller actually fixed in position relative to the hull.
- ii) *Diffraction effect.* The effect of changing the propeller location relative to the hull, but with the wake inflow to the propeller held fixed.

In general, wake gradients become more extreme as propeller hull clearances decrease. The high sensitivity of propeller blade pressure and cavitation inception to the variations in wake non-uniformity creates the need for clearance minima and not the diffraction effect because analysis has shown that for wake inflow held invariant, propeller-induced excitation level is relatively insensitive to near-field variations in propeller location.

In light of the above, the conventional and strut-stern wake types should be considered. For conventional sterns, there is no significant boundary layer on the usually narrow counter above the propeller. Furthermore, taking into account vertical strut idealization of the skeg, the axial distribution would be invariant with vertical disk position. For conventional sterns, the waterline slope in the upper skeg region appears to be the critical design factor and propeller fore-and-aft clearances are generally less critical than vertical. Blunt waterline endings can result in local separation and substantially more severe wake gradients in the upper disk than predicted by simple strut idealization, in which case a “blunt strut” idealization would be more appropriate. Wakes attenuate very slowly with distance downstream so increasing the fore-and-aft clearances between the blade tips and the skeg edge forward certainly acts to reduce the wake severity, but the reduction is marginal for the usual range of such clearance variation. An exception to this are cases of separation in the upper disk due to local waterline bluntness whereby the closure region of the separation cavity exhibits large gradients in axial velocity.

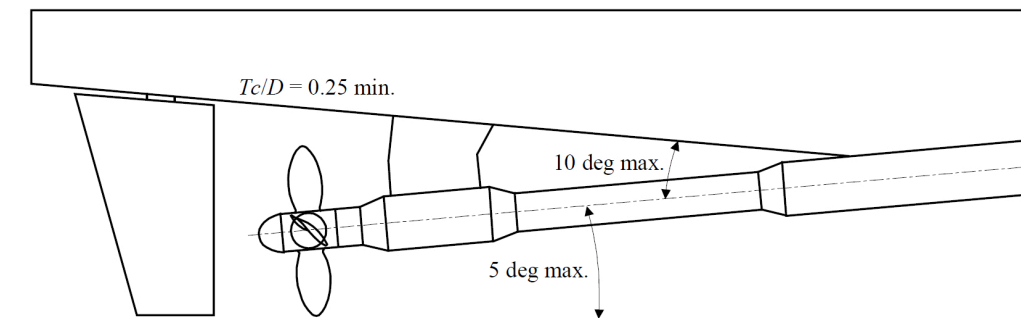
For the broad and flat countered strut stern vessel, the vertical tip clearance is a much more critical consideration. A relatively uniform wake will result if the propeller disk does not overlap the overhead boundary layer and the shaft inclination is moderate. This is the condition, in general, achieved on naval combatant vessels; the usual practice in naval design is a minimum vertical tip clearance of one-quarter propeller diameter. Vibration problems are almost unheard of on naval combatant vessels.

Some wake non-uniformity on strut stern ships results from shaft struts and from the relatively high shaft inclinations often required to maintain the 25 percent overhead tip clearances. With proper alignment to the flow, shaft struts produce highly localized irregularities in the wake which are generally not effective in producing vibratory excitation. The main effect of shaft inclination is a relative up-flow through the propeller disk, although separated flow from forward of a highly inclined propeller shaft can produce a serious axial defect. The possible cavitation resulting at the three and nine o'clock blade positions is of concern regarding noise and minor blade erosion, but the hull vibratory excitation produced has not been found to be of much significance for strut stern ships.

The minimum vertical tip clearance of 25 percent of a propeller diameter is accepted as the standard in both commercial and naval design. Consistent with the preceding analysis, the following lists the recommended configuration for stern arrangements in the order of preference for avoidance of excessive propeller induced vibration:

- i) *Single/Twin Screw Strut Stern*
 - The minimum vertical tip clearance should not be less than 25% of the propeller diameter.
 - The shaft inclination angle relative to the baseline should not be more than five degrees.
 - The shaft inclination angle relative to the buttocks angle of the counter should not be more than ten degrees. Refer to 3/5.1.i FIGURE 5.

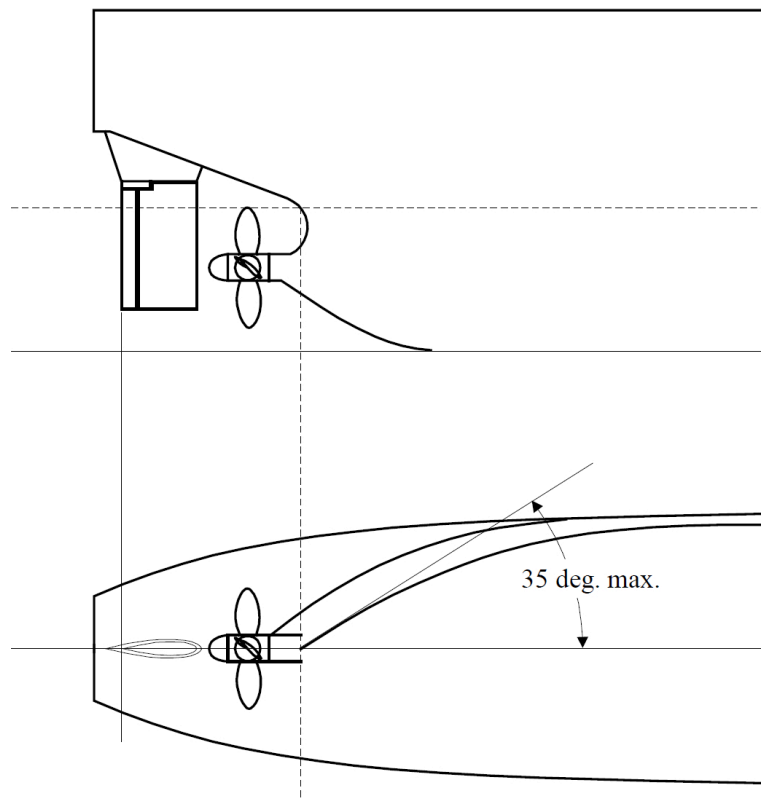
FIGURE 5
Open Strut Stern Arrangement



ii) Conventional Stern with Skeg and Bossing.

- The waterline angle from the vertical centerplane at the entrance to the aperture just forward of the top of the propeller disk should not exceed 35 degrees. Refer to 3/5.1.ii FIGURE 6

FIGURE 6
Conventional Skeg-Stern Arrangement



- With regard to the propeller tip clearance, the conventional skeg-stern ships are less critical than the strut-stern ships, as discussed earlier. In commercial practice, a minimum vertical tip clearance on order of 25% of propeller diameter and forward clearance of 40% of propeller diameter are often employed as a usual practice.

iii) If these limits cannot be achieved in concept design, model testing and/or direct calculation is recommended to confirm or establish the stern lines.

7 Propeller (1 September 2023)

With an unfavorable wake, propeller compromises are usually required to achieve compensation. Two types of excitation are of primary concern:

- i) Alternating thrust exciting longitudinal vibration of the shafting and machinery.
- ii) Vertical pressure forces on the stern counter exciting hull and superstructure vibration.

7.1 Alternating Thrust

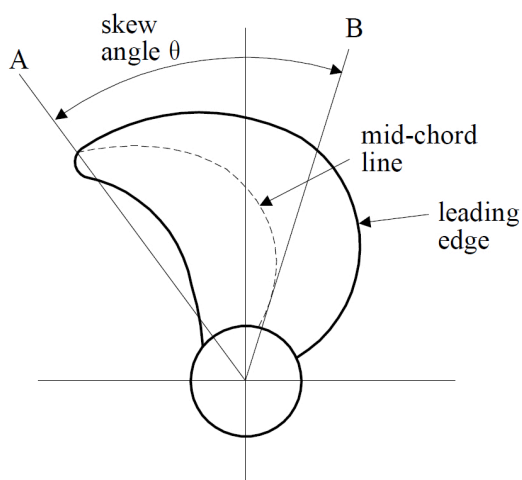
Alternating thrust is the excitation for longitudinal vibration of the shafting/main machinery system. It occurs at blade rate frequency (Propeller RPM \times Blade number N) and its multiples. However, the fundamental is usually much larger than any of its harmonics. Alternating thrust is produced by the blade number circumferential harmonic of the hull wake. This suggests that the more blades a propeller has the better, since the wake harmonic series converges. However, around the typical number of blades between four and six, the wake harmonic series convergence is not well organized, and is not a primary consideration. For the wake from a conventional single-screw stern, lower alternating thrust favors propellers with an odd number of blades. This is because the opposite blades of the propeller with an even number of blades line-up with the characteristic wake spike along the vertical center-plane above and below the propeller axis. This wake characteristic is discussed in Subsection 3/5 on hull wake. With strut-stern ships there is typically little blade number bias on the basis of wake.

Skew is the tangential “wrapping” of the blades with radius. Positive skew is in the angular direction opposite to the direction of rotation. For conventional single screw ship wakes, the shadow of the vessel skeg produces a heavy axial wake defect concentrated along the disk vertical centerline. The blades of conventional propellers ray-out from the hub (i.e., the blade mid-chord lines are more or less straight rays emanating from the hub centerline). Such “unskewed” blades abruptly encounter the axial velocity defect of the conventional stern wake at the top and bottom-dead-center blade positions. The radially *in-phase* character of the abrupt encounter results in high net blade loads and radiated pressure.

A more gradual progression of the blades through the vertical wake defect is accomplished by curving the blades. Different radii enter and leave the wake spike at different times; cancellation results in the radial integrations to blade loads and radiated pressure, resulting in reduced vibratory excitation.

Percentage skew is the blade tip skew angle, relative to the blade ray through the mid-chord of the propeller hub section, divided by the blade spacing angle (e.g., for 5 blades the blade spacing angle is 72 degrees; therefore, a tip skew angle of 72 degrees would be 100% skew). See 3/7.1 FIGURE 7, where ray ‘A’ is passing through the tip of blade at mid-chord line and ray ‘B’ is tangent to the mid-chord line on the projected blade outline.

FIGURE 7
Maximum Skew Angle



Highly skewed propellers may be designed with efficiency equivalent to propellers with minimal skew, but there may be issues associated with blade strength and flexibility. The modern designs have trended back toward more moderate positive tip skew, but with negative skew in the blade root regions. Here the percent skew may be defined in terms the total angle enclosed by the angular maxima of the blade leading and trailing edges. This provides blades with more balanced skew distributions, and highly *swept* leading edges. The radial load cancellations are achieved as effectively and blade strength is less problematic. The balanced skew is beneficial to limiting blade spindle torque with controllable pitch propellers, and is used widely on naval combatant vessels.

Skew works less effectively with strut-stern wakes since the axial velocity defect tends to be concentrated more in the outer extreme radii. The more radially uniform distributions of the conventional stern case are not available with which to achieve as high a degree of *dephasing* and radial cancellation. Of course, the strut stern vessel is in less need of propeller design extremes, as vibration problems are already largely eliminated by the stern form selection, provided proper clearances are incorporated.

Care should be taken in incorporating skew, particularly in replacement propellers, so that adequate clearance between the blade tips and the rudder is maintained. As the blades are skewed in the pitch helix the tips move downstream, closing-up blade tip to rudder clearances. The consequences can be increased hull vibratory forces transmitted through the rudder, as well as rudder erosion caused by the collapsing sheet cavitation shed downstream off the blade tips as they sweep through the top of the propeller disk. The solution is to incorporate warp into the blades along with the skew. Warping is a forward raking of the skewed blades back to (and sometimes beyond) the propeller disk. It is equivalent to skewing the blades in the plane of the disk rather than in the pitch helix.

Skew (and/or sweep) is beneficial in reducing vibration-producing unsteady sheet cavitation, even when such cavitation may be concentrated in the blade tips. The blade curvature is thought to result in a radially outward flow component in the vicinity of the blade tips which tends to sweep the cavity sheets into the tip vortex, where the critical collapse phase occurs more gradually. Unsteady sheet cavitation is found to have minimal effect on alternating thrust, but can dramatically amplify the hull pressure force components. This is addressed specifically in 3/7.3.

Guidelines offered on stern lines regarding hull wake serve to minimize the risk of unacceptable alternating thrust levels for properly sized propellers. This is without the need, particularly at the concept design stage, for detailed analysis to quantify alternating thrust level further along. Such quantification for detailed analysis is considered in Section 5.

Examples of ranges of the blade-rate alternating thrust amplitude are given below to serve as a reference for engineering decision making at the concept design level. The values are given as percentages of steady ahead thrust, and assume that due care has been taken with stern lines and that the propeller is properly sized for proper propulsive performance:

	<i>Alt Thrust/Steady Thrust</i>
Conventional Stern, moderately skewed/swept	0.02-0.06
Strut Stern	0.005-0.03

More details on propeller bearing forces and moments including alternating thrust for 20 real ship cases are given in Section 5. There is an outstanding issue here regarding proper sizing of the propeller, which is needed in the trade-offs at the concept design level. The old Burrill (1943) cavitation criterion is still useful in this regard. Alternating thrust is not very sensitive to blade cavitation, but limiting the steady cavitation to accepted levels reduces the propeller load which is necessary to achieve the alternating thrust ranges listed above.

The Burrill chart is provided in 3/7.1 FIGURE 8, which is a plot of a thrust loading coefficient at the 0.7 propeller blade radius, τ_c , versus the blade cavitation number, σ , also at the 0.7 radius.

$$\tau_c = \frac{T}{A_p q_T}$$

and

$$\sigma = \frac{p_a + \rho g h - p_c}{q_T}$$

where

- T = propeller thrust in kN
- A_p = propeller projected area in m²
- q_T = $\frac{1}{2}\rho V_R^2$ in kPa
- V_R = relative velocity of water at 0.7 radius in m/s
- p_a = absolute atmospheric pressure in kPa
- h = depth of the propeller axis in m
- p_c = cavitation cavity absolute pressure in kPa
- ρ = water density in ton/m³

The relative velocity V_R at 0.7 radius is:

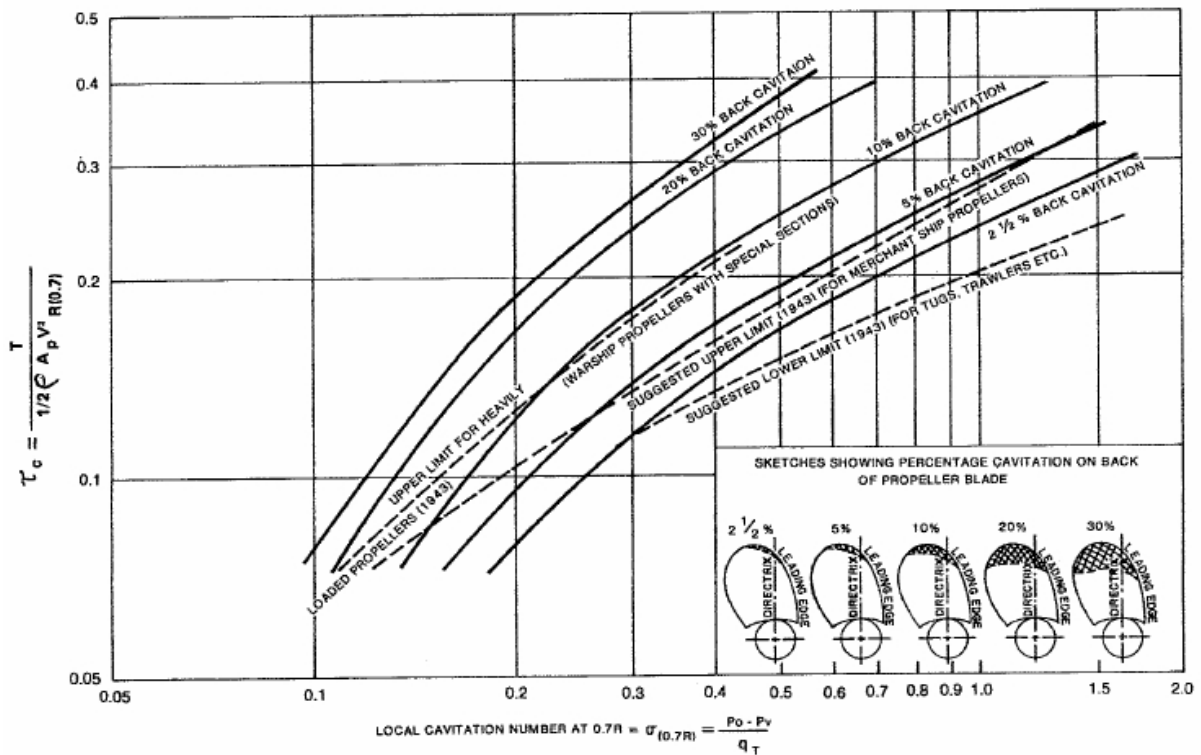
$$V_R(0.7R) = U(1-w)\sqrt{1 + \left(\frac{0.7\pi}{J}\right)^2}$$

where

- U = ship speed
- w = wake fraction
- J = design advance coefficient

The cavity pressure is often taken to be pure vapor pressure at the ambient water temperature, but for developed cavitation as is the case here, versus inception, p_c is higher due to dissolved air in the water; p_c around 6-7 kPa is often assumed. The curves of 3/7.1 FIGURE 8 correspond to the degree of steady cavitation development. The 5% back cavitation line was suggested as the limit for merchant ship propellers back in 1943. However, for modern commercial ship propellers, which are universally designed with aerofoil-type blade sections, the 10% back cavitation line is probably more appropriate. However, considering the cavitation developments shown, either the 5% or 10% lines may serve to avoid overloading the propeller in steady propulsion.

FIGURE 8
Burrill Cavitation Inception Chart



An example of the use of the Burrill chart for this purpose is as follows:

- Take a single screw ship with a delivered power $P_d = 22,380$ kW and a max rated speed $U = 22$ knots, corresponding to a thrust, $T = 1,400$ kN. The propeller diameter has been set at 6 meters consistent with tip clearance maxima. The propeller $RPM = 120$ and the wake fraction is estimated as $w = 0.2$. The pitch ratio at the 0.7 radius, $P/D]_{0.7R} = 0.9$. This gives an advance coefficient, $J = 0.755$ and resulting q_T of 399.3 kPa. For the hub centerline depth $h = 4.5$ m, $p_a = 101.3$ kPa, and $p_c = 6.9$ kPa, $\sigma = 0.35$. If the 5% back cavitation line is selected, $\tau_c \cong 0.16$. With the disk area $A_0 = \pi D^2/4$, the required projected area ratio would be:

$$\frac{A_p}{A_0} = \frac{4T}{\pi \tau_c q_T D^2}$$

- Substituting the values, $A_p/A_0 = 0.775$. It is preferable to work with the developed area ratio, A_d/A_0 , where A_d denotes propeller developed area with zero pitch. Taylor's approximate formula gives the projected area in terms of the developed area as:

$$\frac{A_p}{A_d} \cong 1.067 - 0.229 \frac{p}{D} \Big]_{0.7R}$$

Then the required developed area ratio is:

$$\frac{A_d}{A_0} = 0.90$$

- The projection here is that with a good wake, as that achieved by following the provided recommendations in these Guidance Notes, the subject propeller blades may not be heavily loaded in the mean, and the alternating thrust should be in approximately direct proportion. As an alternative, if it was decided to use a less conservative approach and accept steady cavitation to the 10% back cavitation at $\sigma = 0.35$, $\tau_c \cong 0.2$. Then the alternative developed area ratio would be:

$$\frac{A_d}{A_0} = 0.72$$

The more conservative propeller, with the larger blades, will have lower propulsive efficiency. The following is recommended in concept design, in addition to the preceding stern lines recommendations relative to hull wake, for achieving appropriate alternating thrust from the propeller design consideration:

- Perform the Burrill cavitation calculation demonstrated above with a blade area ratio as needed to limit the steady cavitation up to 10% back cavitation.
- For a conventional stern, a 5-bladed propeller is favored in reducing the alternating thrust, unless any harmful resonant vibration is anticipated on the superstructure, shafting system or local structures.
- In general, incorporate blade skew of no more than 50%, or if a blade with a highly swept leading edge is adopted (negative skew in the root), use no more than 25 degrees of tip skew (for either stern type). For a highly skewed propeller design other than the foregoing, refer to 4-3-3/5.5 of the *Marine Vessel Rules*.

7.3 Hull Pressure Forces (1 September 2023)

The dominant excitation for ship hull vibration, related to the third of the three critical items identified in the preceding section, is propeller cavitation-induced hull surface pressure force. If intermittent blade cavitation does not occur to a significant degree, then the main excitation of the hull is via the shafting system and the main engine, as already discussed. These sources will cause minor problems relative to, at times, the intense vibration from an intermittently cavitating propeller. It was in the mid-70's that propeller blade unsteady cavitation, triggered by the wake non-uniformity, was found to be the cause of most ship vibration issues.

The sheet cavitation expands and collapses on the back of each blade in a repeating fashion, revolution after revolution. The sheet expansion typically commences as the blade enters the region of high wake in the top part of the propeller disk. Collapse occurs upon leaving the high-wake region in a violent and unstable fashion, with the final remnants of the sheet typically trailing out behind in the blade tip vortex. The sheet may envelop almost the entire back of the outboard blade sections at its maximum extent. For large ship propellers, sheet average thicknesses are on the order of 10 cm, with maxima on the order of 25 cm occurring near the blade tip just before collapse.

The cavitation, while of catastrophic appearance, is usually not deleterious to ship propulsive performance. The blade continues to lift effectively; the blade suction-side surface pressure is maintained at the cavity pressure where cavitation occurs. The propeller bearing forces (i.e., alternating thrust) may be largely unaffected relative to non-cavitating values for the same reason. The cavitation may or may not be erosive, depending largely on the degree of *cloud cavitation* (a mist of small bubbles) accompanying the sheet dynamics. The devastating appearance of fluctuating sheet cavitation consistently manifests mainly in the field pressure that it radiates, and the noise and vibration that it thereby produces. The level of hull surface

excitation induced by a cavitating propeller can be easily an order of magnitude larger than typical non-cavitating levels. Vertical hull surface forces due to intermittent cavitation typically exceed propeller bearing forces by large amounts.

The cavitation-induced hull surface force is, like the bearing forces, composed of the fundamental blade-rate frequency and its harmonics. However, unlike the character of the alternating thrust discussed in the preceding, the harmonics of the cavitating surface force typically converge very slowly. The fundamental, at propeller RPM times blade number, is usually largest, but twice and three times blade-rate of the same order typically exists and can resonate structure and generate noise well above the normal excitation frequency range.

Cavitating vibratory force reduction is achievable with propeller design refinements, as sheet cavitation dynamics is sensitive to blade geometric, as well as wake, detail. As previously noted, blade skew can be beneficial, and tip loading can be reduced to reduce cavitation extent and intermittency by local blade tip pitch reduction, as well as changes in blade tip chord distributions. But measures of this type do reduce propeller efficiency and thereby compromise propulsive performance.

Propeller blade intermittent cavitation should be taken into account when attempting to control propeller induced hull vibration. The pertinent factor is again related to hull wake. With an unfavorable wake, characterized as exhibiting high gradients in the upper disk, or even a separation pocket, the resulting large cavity volume variations can produce hull surface force amplitudes of up to 30-40% of the steady thrust, with significant levels out to several harmonics of blade-rate frequency. But even with a good wake, as described in the preceding section, the conventional stern ship will still exhibit a wake defect in the disk over the depth of the skeg, with some intermittent cavitation being unavoidable. A vertical surface force with a blade-rate amplitude of 15% of steady thrust is probably a conservative reference for a conventional stern ship with a “good wake” as achievable by following the stern lines guidance in the preceding section.

With strut stern vessels with ample tip clearances and low relative shaft inclination, intermittent cavitation, or any cavitation at all, might be avoided entirely. This is largely achieved in the case of radiated noise from naval combatant vessels.

Prediction of the cavitating hull forces is much more complicated, with many more factors contributing, than is alternating thrust, which, as stated, is mostly insensitive to cavitation. Cavitating hull force analysis is certainly not an activity for concept design. However, with any conventional stern ship of modern power level, a propeller cavitation/hull pressure assessment program should be initiated in the preliminary design stage. For the direct calculation of the propeller-induced hull surface force, refer to the vibration analysis procedure in Section 5.

At concept design, which is the scope of interest here, the same analysis as in the preceding subsection, using the Burrill criteria for 5-10% steady back cavitation, should be applied to provide reasonable protection against under-sizing the propeller blades from the standpoint of unsteady cavitation. The example of the preceding subsection on alternating thrust is therefore proposed as applicable here as well.

SECTION 4 Structural Resonances

1 Introduction (1 September 2023)

The **six important** items in concept design are identified in Subsection 2/5. The relevant excitations are treated in Section 3, while this section focuses mainly on resonance avoidance.

3 Hull Girder Vertical Vibration (1 September 2023)

The vertical beam-like modes of vibration of the hull girders of modern ships or **ship-type floating production units** may become severe in three respects:

- i)* They can be excited to excessive levels by resonances with the dominant low frequency excitations of slow-running diesel main engines.
- ii)* Vertical vibration of the hull girder in response to propeller excitation is a direct exciter of objectionable fore-and-aft superstructure vibration.
- iii)* Vertical vibration of the hull girder excited by slamming can give rise to excessive vibration of various installations on the vessel's deck, especially slender structures such as a flare tower in hydrocarbon production and process systems.

The propeller is generally incapable of exciting the hull girder modes themselves to dangerous levels. This is primarily because the higher hull girder modes whose natural frequencies fall in the range where propeller excitation is significant have low excitability.

The vibration response at resonance depends on the magnitude of excitation, structural damping at the resonant frequency, and the location and direction of the excitation in relation to the nodes of the natural vibration modes. Forces will excite resonance type behavior most effectively if they act at the antinodes of the structural vibration mode (locations where the amplitude of the natural vibration mode is maximum), and moments will excite resonance most effectively if they act at the nodes (locations where the amplitude of the natural vibration mode is zero). 4/3 FIGURE 1 shows two nodes and an antinode of a hull girder natural vibration mode. The nodes of the higher hull girder vibration modes are further away from the main engine moment excitations, which explains their low excitability.

The low-level vertical hull girder vibration that does occur, either directly from the propeller or indirectly via the main shafting thrust bearing, serves as the base excitation for excessive vibration of superstructures and other attached subsystems which are in resonance with the propeller exciting frequencies. This will be addressed in a later section. The natural frequencies corresponding to the two-noded vertical bending modes of conventional ship hulls can be estimated using Kumai's formula:

$$N_{2v} = \varphi \sqrt{\frac{I_v}{\Delta_i L^3}} \text{ cpm}$$

where

- φ = 3.584×10^6 - for ship-type floating production units
 = 3.07×10^6 - for other vessel types
 I_v = moment of inertia, in m^4
 Δ_i = virtual displacement, including added mass of water, in tons
 = $\left(1.2 + \frac{1}{3} \cdot \frac{B}{T_m}\right) \Delta$
 Δ = ship displacement, in tons
 L = length between perpendiculars, in m
 B = breadth amidships, in m
 T_m = mean draft, in m

FIGURE 1
Nodes and Antinode of a Natural Hull Girder Vibration Mode (1 September 2023)

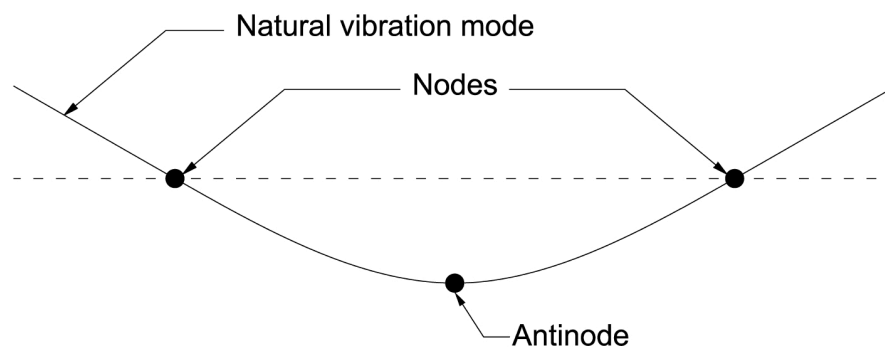


TABLE 1
Comparison of 2-node Vertical Vibration Natural Frequencies

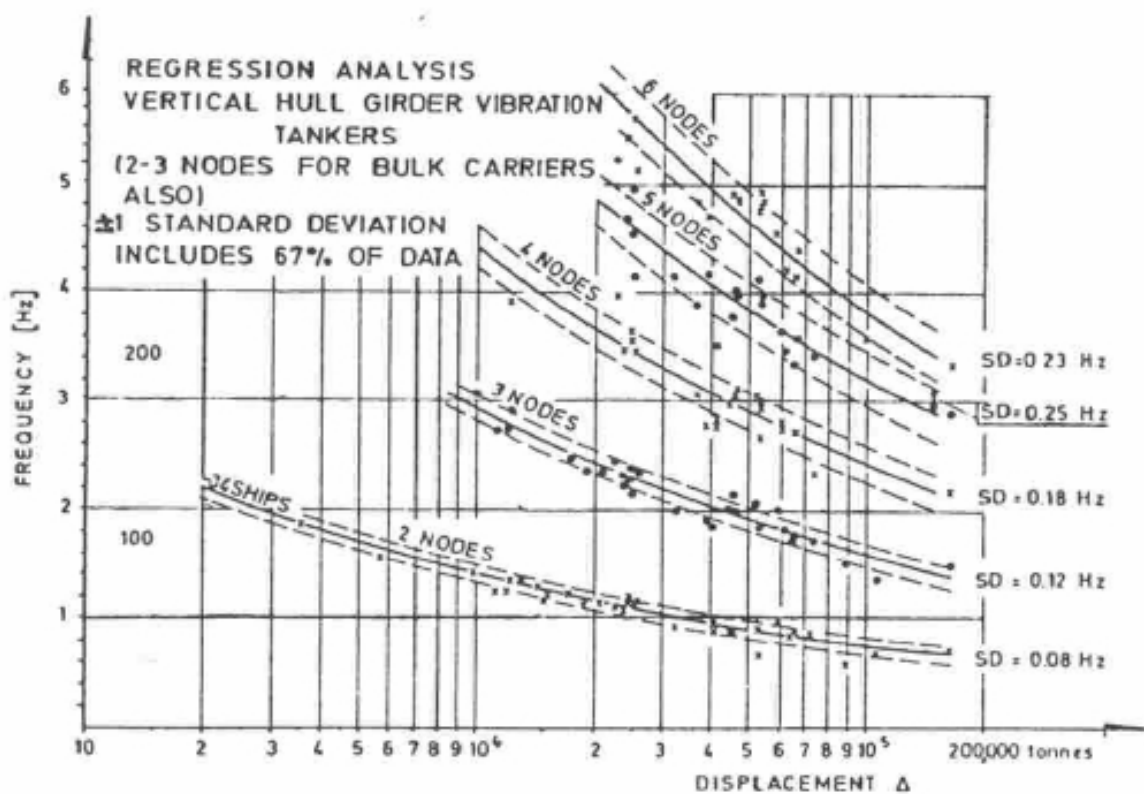
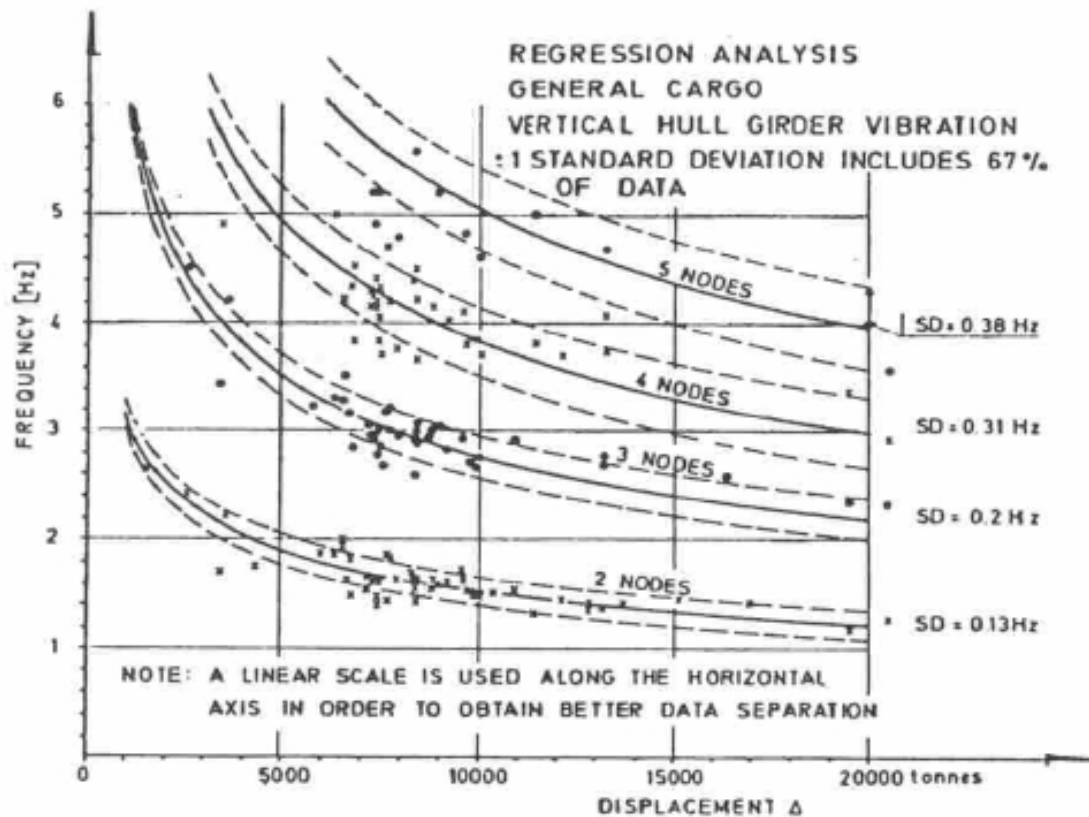
<i>Ship No.</i>	<i>Type</i>	<i>Size (tone)</i>	<i>Kumai (Hz)</i>	<i>FE Method (Hz)</i>	<i>Dev. (%)</i>
1	Reefer	15000	1.54	1.51	+2
2	Ro-Ro	49000	1.49	1.60	-7
3	Ro-Ro	42000	1.04	0.94	+10
4	Chemical	33000	1.00	0.93	+8
5	Bulk Carrier	73000	0.63	0.64	-2
6	Container Carrier	120900	0.41	0.49	-17.0
7	Large Container Carrier	200000	0.38	0.45	-15
8	VLCC	363000	0.40	0.46	-12.8

3/3.ii TABLE 1 indicates the accuracy that can be expected from Kumai's formula. The table compares the prediction of the 2-noded vertical hull bending natural frequency by Kumai's formula with the predictions of detailed finite element calculations performed on different ships. The deviation of the natural frequency calculated using the Kumai formula for ship-type floating production units with respect to collected measured or calculated natural frequencies ranges between -8.6% and 14.7%.

The 2-noded hull vertical bending natural frequencies lie well below the dangerous exciting frequencies of either typical diesel main engines or propellers and are, therefore, of little consequence in these considerations. As will be demonstrated further on, it is hull girder modes with typically a minimum of 4 or 5 nodes that can be excited excessively by the diesel main engine. In the case of the propeller, the hull girder vertical bending modes that fall near full-power propeller blade-rate excitation are typically more than 7-noded. Full-power blade rate excitation of large ships is usually in the range of 8 to 12 Hz. As indicated in 4/3 TABLE 1, the 2-noded vertical hull bending mode, on the order of 1 to 2 Hz, is well below the blade-rate excitation frequency level during normal operation.

It is observed that hull girder natural frequencies increase approximately linearly with node number from the 2-noded value for the first few modes. The data shown on 4/3 FIGURE 2, from Johannessen and Skaar (1980), provide estimates of the natural frequencies of the first four vertical bending modes of general cargo ships and of the first five vertical bending modes of tankers. Note the good agreement for 2-noded cases. Also note that the 6 Hz maximums represented by 4/3 FIGURE 2 still lie well below typical full-power propeller excitation frequencies, and the accuracy of the data fits indicated on the figure is deteriorating rapidly as modal order increases. The primary reason for the increasing data scatter with node number is the increasing influence of local effects (i.e., approaching resonances of deckhouses, decks, etc.) on the basic beam modes still identifiable.

FIGURE 2
Natural Frequencies of Vertical Hull Vibration



The Kumai's formula, in conjunction with 4/3 FIGURE 2, is, however, useful in preliminary steps to avoid resonances with a **slow-speed** diesel engine. The following formula, from Johannessen and Skaar (1980), representing the 4/3 FIGURE 2 data, expresses the first few vertical bending natural frequencies in terms of the 2-noded value:

$$N_{nv} \approx N_{2v}(n - 1)^\alpha$$

where

α	=	0.845	General Cargo Ships
		1.0	Bulk Carriers
		1.02	Tankers

Here N_{2v} is the 2-noded vertical natural frequency and n is the number of nodes; n is not to exceed 5 or 6 in order to remain within the range of reasonable validity. Note the approximate proportionality of N_{nv} to node number; this is also evident in 4/3 FIGURE 2. The use of this data is demonstrated by example:

- Assuming the “reefer,” ship number 1 of 4/3 TABLE 1, for example, $N_{2v} = 92.4$ cpm, and $\alpha = 0.845$. The first four vertical hull-girder modes of this vessel would then be predicted to have corresponding natural frequencies of:

n	N (cpm)
2	92.4
3	166
4	234
5	298

With the main engine RPM of 122 cpm, for example, the frequency of the 2nd order engine excitation is $2 \times \text{RPM} = 244$ cpm. It is relatively close to the predicted natural frequency of the 4-noded third hull girder mode. While the natural frequency estimate of 234 cpm is indeed rough, it provides at least useful guideline to dictate further analysis to refine the hull girder natural frequency estimates in this particular example.

In the case of projected high excitability in resonant vibration with the diesel engine moments, which sometimes develops in the course of design, the excitation moment components can usually be reduced effectively by incorporating compensators or electric balancers. These devices consist of rotating counterweights usually geared directly to the engine or are electrically driven. **The electrically driven moment compensators can be vertical or horizontal, with the former being more common.** They rotate at the proper rate and phase to cancel the undesirable 1st or 2nd order engine generated moments.

The electrically driven vertical moment compensator is usually installed in the steering gear room, away from the nodes of the critical natural modes. This way, the vertical force generated by the moment compensator is more effective. The vertical moment compensator can also be installed above the propeller to attenuate the propeller-induced vibrations.

The electrically driven horizontal moment compensators are usually installed on the engine to attenuate the effects of engine H-type and X-type guide force moments or in the superstructure to reduce excessive fore and aft vibrations.

The advantage of electric balancers (compensators) is that they can be positioned virtually anywhere within the ship structure.

An alternative that has seen some popularity is the installation of main diesel engines on resilient mounts (Schlottmann *et al*, 1999). Isolating the main engine on resilient mounts can be effective in minimizing hull vibration and structure-borne noise. The following steps are therefore recommended in concept design:

- i)* Assuming no moment compensation on the engine, compensators geared for $2 \times$ Engine RPM, should be installed on both ends of the crank if:
 - a)* The full power second-order vertical moment amplitude exceeds the PRU (Power Related Unbalance) value of 220 N-m/kW, as discussed in Subsection 3/3.
 - b)* Twice RPM of the engine at full power is within 20% of any of the vertical hull modes through at least 5-noded as predicted by the procedure of the immediately preceding example.
- ii)* Moment compensation is effective but if both a) and b) occur together a more precise analysis of the hull girder natural frequencies is recommended in setting structure in preliminary design to more accurately assess the proximity of a resonance with the engine within a 20% band around twice full power engine RPM.

5 Main Machinery/Shafting System Longitudinal Vibration Excited by the Propeller (1 September 2023)

Shafting/machinery longitudinal vibration can also be serious in several respects:

- i)* In a resonant or near resonant condition with the system mass and stiffness, thrust reversals at the main thrust bearing can result which, over a relatively short time period, are capable of destroying the thrust bearing.
- ii)* Engine room vibration, including vibration of the engine itself, can be excessive with regard to foundation and inner-bottom structural distress.
- iii)* The amplified thrust transmitted through the main thrust bearing and its moment arm relative the hull girder neutral axis can produce vertical response of the hull girder which excites resonant vibration of hull-mounted substructures.
- iv)* Longitudinal vibrations of the main machinery/shafting system can also be excited by the engine gas pressure acting on the crank throw through the connecting rod mechanism. The arms of the crank throw will deflect in the longitudinal (axial) direction of the crankshaft causing axial vibrations. Most engine designs come with axial dampers which serve to mitigate the effects of axial vibrations.

Interest in longitudinal machinery vibration has a long history, starting seriously with the steam turbine-powered battle ships in WWII. It is considered necessary that longitudinal vibration be a subject of concept design. The main machinery items have long lead times and any problems should be detected early.

Main propulsion system fore-and-aft natural frequencies tend to fall in the range of propeller blade rate excitation frequency. For short shafting systems, the one-node (first) mode can be easily coincident with blade rate excitation, but with the second mode well above. For long shaft systems, the system just cannot be designed with the first mode above the blade rate excitation frequency, and should therefore be configured so that it lies far enough below. But then the second mode becomes of concern with long shaft systems.

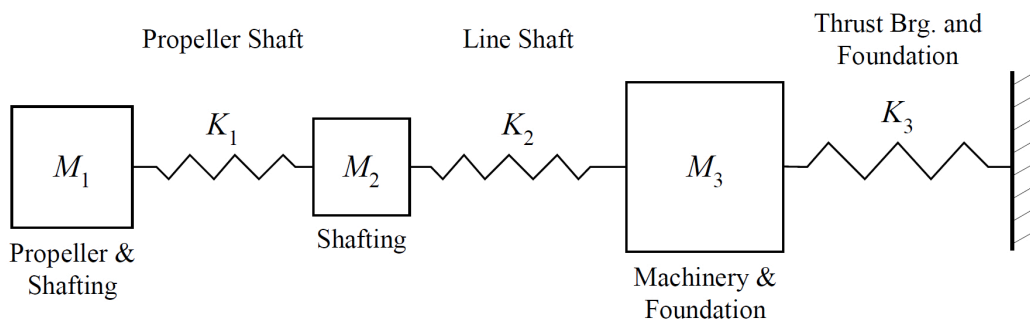
The dominating uncertainty with regard to longitudinal vibration is the stiffness of the main thrust bearing and its “foundation.” The thrust bearing “foundation” is the serial ship structure that deflects, as a spring, in response to the thrust transmitted through the thrust bearing. The thrust bearing on a diesel is normally located in the engine casing aft. The long engine casing provides some extra stiffness over that with the steam plant. It is the serial stiffness in the engine room and ship bottom structure that can be the critically

weak link; recall that in serial stiffness the overall stiffness is less than the stiffness of the most flexible element. This supporting structure should be carefully designed early to properly place the first two system natural frequencies relative to blade rate excitation to avoid serious problems. The control is through shipyard responsibility for the design of the engine room bottom structure and machinery foundations.

The three mass model of 4/5 FIGURE 3 can be used for first estimates of the first two shafting/ machinery system modes. Three masses are considered the minimum number needed for estimating the first two system modes with reasonable accuracy. Definition of the mass and stiffness data shown on the figure is as follows:

- M_1 = lumped mass at the propeller in kg, composed of propeller mass, increased 60% for hydrodynamic added mass, one half of the propeller shaft mass
- K_1 = stiffness of propeller shaft in N/m, from propeller to coupling with line-shaft
 $= A_{ps}E / \ell_{ps}$
- A_{ps} = propeller shaft cross-sectional area in m^2
- E = elastic modulus in N/m^2
- ℓ_{ps} = propeller shaft length in m
- M_2 = lumped mass at propeller shaft/line shaft coupling in kg, composed of one-half of propeller shaft mass plus on half of line-shaft mass. (For very short propeller shafts, M_2 could be located in the line-shaft with mass and stiffness contributions appropriately adjusted.)
- K_2 = stiffness of line-shaft in N/m, from coupling to thrust bearing (thrust bearing assumed to be at aft end of engine casing).
 $= A_{ls}E / \ell_{ls}$
- A_{ls} = line shaft cross-sectional area in m^2
- ℓ_{ls} = line shaft length in m
- M_3 = lumped mass at thrust bearing in kg, composed of one-half of the line-shaft mass, the engine, including the thrust bearing, plus a thrust bearing/engine foundation structural mass allowance of 25%.
- K_3 = stiffness of thrust bearing elements and engine foundation in N/m

FIGURE 3
3-mass Longitudinal Model of Main Propulsion System



It is convenient to view the foundation stiffness K_3 as an unknown, to be determined so that the two natural frequencies lie at appropriate levels with respect to the blade-rate excitation frequency. The coupled equations of motion lead to the eigenvalue problem:

$$\begin{bmatrix} -\omega_n^2 M_1 + K_1 & -K_1 & 0 \\ -K_1 & -\omega_n^2 M_2 + K_1 + K_2 & -K_2 \\ 0 & -K_2 & -\omega_n^2 M_3 + K_2 + K_3 \end{bmatrix} \begin{bmatrix} \psi_{n1} \\ \psi_{n2} \\ \psi_{n3} \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \\ 0 \end{bmatrix},$$

where ψ denotes the mode shape vector. It is necessary to expand the determinant of the coefficient matrix to form the characteristic equation whose roots are the three natural frequencies. First define the following for convenience of notation:

$$\Omega_n = \omega_n^2, \Omega_{11} = \frac{K_1}{M_1}, \Omega_{12} = \frac{K_1}{M_2}, \Omega_{22} = \frac{K_2}{M_2}, \Omega_{23} = \frac{K_2}{M_3}, \Omega_{33} = \frac{K_3}{M_3}$$

Then the characteristic equation from the determinant expansion is the following cubic in Ω_n :

$$\Omega_{11} \cdot \Omega_{22} \cdot \Omega_{33} - \Omega_n \cdot (\Omega_{12} \cdot \Omega_{23} + \Omega_{11} \cdot \Omega_{23} + \Omega_{12} \cdot \Omega_{33} + \Omega_{11} \cdot \Omega_{22} + \Omega_{11} \cdot \Omega_{33} + \Omega_{22} \cdot \Omega_{33}) + \Omega_n^2 \cdot (\Omega_{12} + \Omega_{23} + \Omega_{11} + \Omega_{22} + \Omega_{33}) - \Omega_n^3 = 0$$

The unknown K_3 , in Ω_{33} , can be calculated for specified distributions of Ω_n in the ranges of interest.

$$\Omega_{33} = \frac{\Omega_n^3 - (\Omega_{11} + \Omega_{12} + \Omega_{22} + \Omega_{23}) \cdot \Omega_n^2 + (\Omega_{11} \cdot \Omega_{22} + \Omega_{11} \cdot \Omega_{23} + \Omega_{12} \cdot \Omega_{23}) \cdot \Omega_n}{\Omega_n^2 - (\Omega_{11} + \Omega_{12} + \Omega_{22}) \cdot \Omega_n + \Omega_{11} \cdot \Omega_{22}}$$

The corresponding value to the required foundation spring constant is:

$$K_3 = M_3 \Omega_{33}$$

K_3 can then be plotted as a function of the arbitrary Ω_n to decide the stiffness of the structure that the shipyard is to design and build to provide the proper support stiffness for the system. Upon increasing the Ω_n from low values, K_3 will increase, and then will become negative as the second mode is reached. The stiffness subsequently turns back to positive again with further increasing frequency in the 2nd mode. The relevant ranges are the ranges of positive K_3 only.

Example:

Take the case used with the propeller example of the preceding sub-section, with the propeller RPM = 120 with 5-blades, implying a blade rate frequency of 10 Hz. The data used in this example are:

Propeller mass, $M_p = 24,098$ kg

Propeller shaft diameter, $d_{ps} = 76.31$ cm

Propeller shaft length, $\ell_{ps} = 15.24$ m

Line shaft diameter, $d_{ls} = 51.09$ cm

Line shaft length, $\ell_{ls} = 21.34$ m

Engine mass, $M_e = 136,080$ kg

By the lumping scheme described above, the masses and springs are:

$M_1 = 51,455$ kg

$K_1 = 6,182 \times 10^6$ N/m

$M_2 = 44,528$ kg

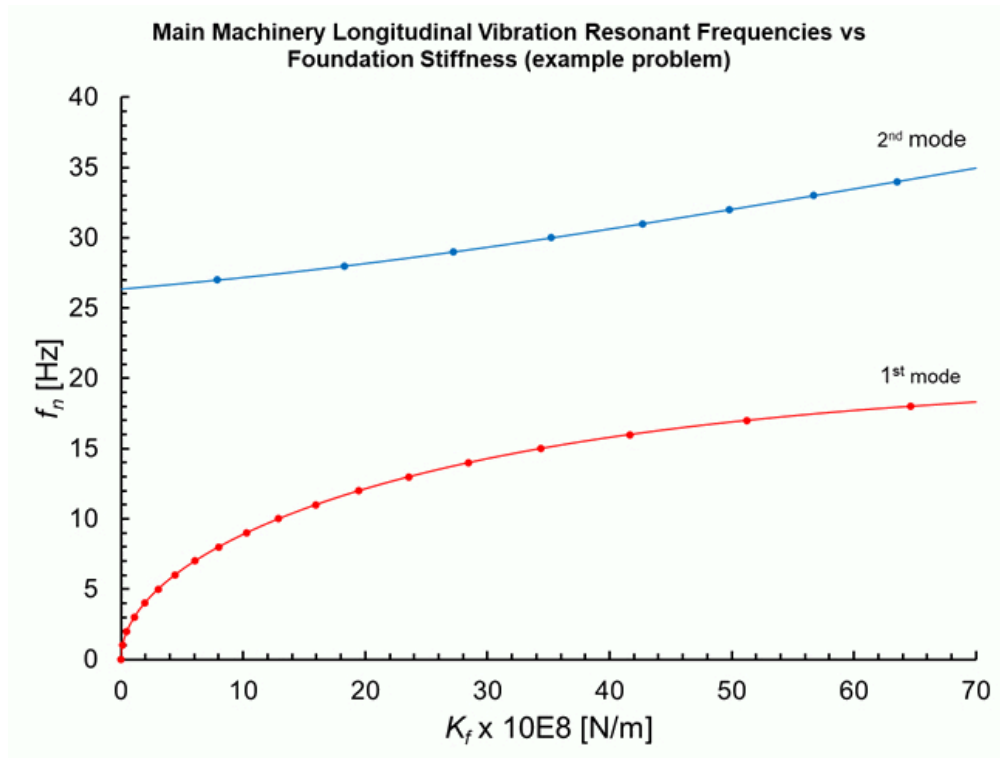
$K_2 = 1,979 \times 10^6$ N/m

$M_3 = 204,940$ kg

$K_3 = K_f$ to be determined

5/3.3.3 FIGURE 3 shows the plot of natural frequency versus thrust bearing and foundation stiffness for the first two modes.

FIGURE 4
Example of Natural Frequencies vs. Foundation Stiffness (1 September 2023)



The blade-rate excitation frequency is 10 Hz, so that the foundation stiffness should be either sufficiently above or below approximately 12.9×10^8 N/m. The restricted frequency band is conventionally taken to be from 20% below the resonance to 20% above. Therefore, by this criterion, the stiffness should be either below 7.98×10^8 N/m or above 19.46×10^8 N/m. Recall that K_f is the serial stiffness of the thrust bearing and the foundation; the thrust bearing stiffness is typically around 36×10^8 N/m. By the serial addition, this gives a higher required stiffness range for the foundation proper: below 10.25×10^8 or above 42.36×10^8 N/m. It is desirable to set the excitable natural frequencies above the excitation frequency range. Otherwise, a resonance falls within the operating range; in that case it should be confirmed that the excitable natural frequencies are far enough below the full power blade-rate frequency.

In this particular example, which is not atypical, the 2nd mode is not of relevance as it lies well above the exciting frequency range for all foundation stiffnesses. As for the first mode, experience has shown that it would be impractical for the shipbuilder to build a foundation with a stiffness as high as 42.36×10^8 N/m. The available choice would be a foundation stiffness no higher than 10.25×10^8 N/m.

Designing a thrust bearing foundation for a specified stiffness is not simple. It typically involves considering the shear deflection of the girders and bending deflection of the inner bottom up to some distance away from the engine. The best recourse is probably to design the double bottom to be as deep and stiff as reasonably possible. This limits the deflecting structure to the foundation proper above the double bottom where it can be dealt with more reliably in detailed structural design.

The following steps are therefore recommended in concept design:

- i) Approximate the constants in the 3-mass model and perform the calculation of the combined thrust bearing and foundation stiffness.
- ii) Estimate the thrust bearing stiffness and perform the serial subtraction to establish a first estimate of the foundation stiffness. Graph the result in the form of 4/5 FIGURE 4.
- iii) If propeller RPM and blade number N have been tentatively set, select a foundation stiffness from the graph for which neither of the natural frequencies are within 20% of the full power blade-rate excitation frequency.
- iv) If propeller RPM and/or blade number have not been set, or *iii*) above cannot be achieved, select propeller RPM $\times N$ such that step *iii*) can be achieved. Preferably, select propeller RPM and N such that the first mode resonant frequency falls at least 20% above the full power blade-rate excitation to avoid resonance in the power range. $N = 5$ blades is the best choice for minimum blade-rate alternating thrust for conventional stern ships with conventionally configured propellers, as discussed in Section 3.
- v) Provide the estimated required foundation stiffness to the hull structural designers for consideration in the preliminary/detailed structural design stages.

7 Superstructure Fore-and-Aft Vibration (1 September 2023)

The relocation of commercial ship engine rooms in the 1960s from amidship to the stern was a technological advancement in all respects except one: **the increase of** propeller-induced vibration. The relocation was prompted by longitudinal strength as ships became longer. Removing the structural discontinuity from the midship region and shortening the shafting system also reduced cost. However, wakes inherently became more irregular with the fuller sterns, and with the increasing ship power at that time, the role of propeller blade cavitation as a dominant vibration exciter quickly became apparent. To have an unobstructed view over the ship bow from the aft-located bridge, it was necessary to elevate the bridge atop a towering deckhouse. Because the deckhouse was now located over the engine room **and propeller** cavity, adequate structural stiffness was difficult to incorporate, and propeller-excited deckhouse vibration became the industry's troublesome problem.

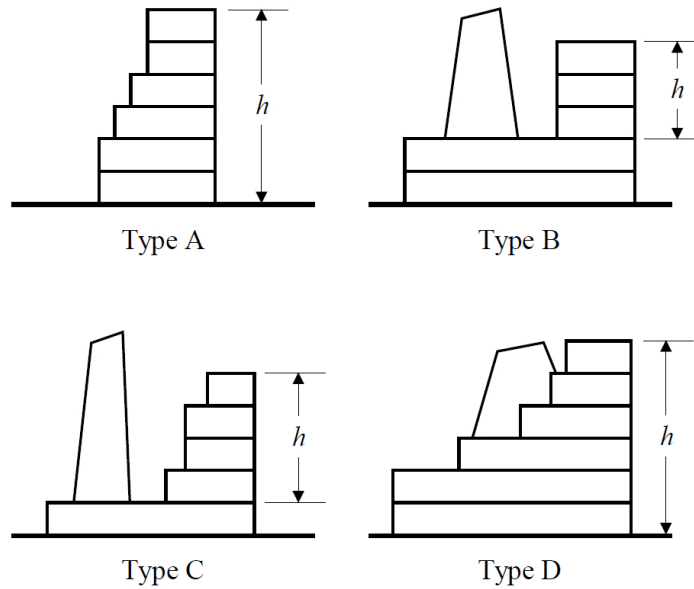
As suggested in Subsection 4/3, a hull girder beam analysis which ignores the dynamics of superstructure produces useful estimates of the hull girder lower natural frequencies for purposes of resonance avoidance with a main diesel engine. It is indeed fortunate that the lower rocking/bending natural frequencies associated with stern superstructures, which usually fall in the range of propeller blade-rate exciting frequencies, can conversely be estimated by ignoring the dynamics of the hull girder. This is the case when the mass of the superstructure is small, relative to the effective mass of the hull girder near the coupled natural frequencies of interest. However, any consideration of vibratory response versus natural frequencies alone should allow for the dynamic coupling. This is due to, in most cases, superstructure vibration excited by the hull girder vibration at its base. The superstructure vibration mode of primary concern is a fore-and-aft rocking/bending mode excited most often through vertical blade-rate vibration of the hull girder. This mode can also be excited indirectly at blade-rate frequency by longitudinal excitation from longitudinal resonance of the shafting/machinery system via the main thrust bearing (Subsection 4/5).

For obtaining preliminary estimates of superstructure fore-and-aft rocking/bending natural frequencies the semi-empirical method of Hirowatari and Matsumoto (1969) has proved to have great utility (Sandstrom and Smith, 1979). This method was developed from correlation of simple analysis and measured fore-and-aft superstructure natural frequencies on approximately thirty ships. In this method, the fore-and-aft “fixed base” natural frequency of the superstructure (i.e., superstructure cantilevered from the main deck) is determined according to deckhouse type and height. The fixed base natural frequency is then reduced by a correction factor to account for the rotational flexibility of the underdeck supporting structure. Specifically, the procedure of Hirowatari **and Matsumoto** is as follows:

- i) Select superstructure type from 4/7.iv FIGURE 5.
- ii) Determine superstructure height, h .
- iii) Read f_{∞} (fixed-base natural frequency) as a function of h from 4/7.iv FIGURE 6.

iv) Read f_e/f_∞ (the correction factor) from 4/7.iv TABLE 2.

**FIGURE 5
Deckhouse Types**



**FIGURE 6
Fixed-base Superstructure Natural Frequencies**

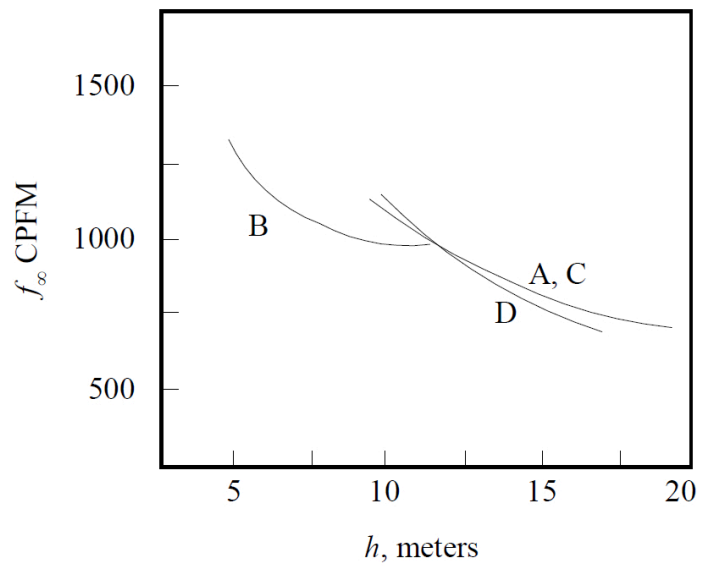


TABLE 2
Flexible Base Correction Factors

<i>Type</i>	f_e/f_∞
A, C	0.625
B	0.602
D	0.751

- v) Compute f_e (the expected deckhouse natural frequency in the first fore-and-aft mode) from the following formula:

$$f_e = f_\infty(f_e/f_\infty)$$

This procedure generally produces results that are within 15 percent of measurements from shaker tests. However, the method becomes inapplicable when the superstructure type varies significantly from those given in 4/7.iv FIGURE 5. Furthermore, there is some uncertainty regarding the use of the correction factors for superstructure support flexibility given in 4/7.iv TABLE 2, since the supporting structure may vary from deep beams to column supports to structural bulkheads. Despite these complexities, the method seems to work quite well in most cases, considering the limited input that is required. This feature makes the Hirowatari and Matsumoto method particularly attractive in the concept design stage when the design data are sparse or non-existent.

The two basic effects influencing the fundamental fore-and-aft superstructure natural frequency are exemplified in this approach:

- i) Cantilever (fixed base) bending and shear of the superstructure as a beam over its height h , 4/7.iv FIGURE 6.
- ii) Rocking of the superstructure as a rigid box on the effective torsional stiffness of its supporting structure.

Ordinarily, one of the superstructure main transverse bulkheads will be a continuation of one of the two engine room transverse bulkheads. The intersection of the continuous bulkhead and the deck identified with the superstructure base can usually be taken as the axis about which the rocking of the house occurs.

The fore-and-aft natural frequency of the superstructure due to the combined effects of rocking and bending/shear can be estimated using Dunkerley's equation (Thomson, 1973) as,

$$f_e = \sqrt{\frac{1}{1/f_\infty^2 + 1/f_R^2}}$$

Here, f_∞ has been identified as the fixed base cantilever natural frequency, from 4/7.iv FIGURE 6, or by detail analysis. f_R is the rocking natural frequency of the rigid superstructure, of height h , on its supporting stiffness,

$$f_R = 60/2\pi\sqrt{K_f/J} \text{ cpm}$$

J is the mass moment of inertia of the superstructure about the rocking axis and K_f is the effective torsional stiffness of the superstructure foundation, also about the axis of rotation. The Hirowatari and Matsumoto procedure, in conjunction with the above formula, has utility in design or post design corrective studies where estimates should be made of the relative effects of structural changes. This is demonstrated by the following example:

Example:

Assume that a conventional Type A superstructure, 4/7.iv FIGURE 5, has been concept designed and that for the selected engine and propeller at full power, propeller RPM = 100 and N = 5, giving a blade-rate excitation frequency of 500 cpm. The house height, h, is 15 meters. Referring to 4/7.iv FIGURE 5 and 4/7.iv FIGURE 6 and 4/7.iv TABLE 2:

$$f_{\infty} \cong 800 \text{ cpm}$$

and

$$f_e/f_{\infty} = 0.625$$

This gives the estimated fore-and-aft house natural frequency, f_e , as:

$$f_e = 0.625(800) = 500 \text{ cpm}$$

which is precisely the full power blade rate excitation frequency. Proceed with the idea of stiffening the system in order to raise the natural frequency by the 20% minimum to 600 cpm. The rocking frequency is first estimated as,

$$f_R = \sqrt{\frac{1}{1/f_e^2 - 1/f_{\infty}^2}} = 640 \text{ cpm}$$

Then,

$$K_f/J = (2\pi/60)^2 f_R^2 = 4490 \text{ 1/rad-s}^2$$

Now assume that the mass of the house, m , has been estimated as 300 tonnes. Also assume that the house front is a continuation of the engine room forward transverse bulkhead, so that the house effectively rotates about its front lower edge, 4/7 FIGURE 7. Assume a radius of gyration, \bar{r} , of the house about this axis of 10 meters. The house mass moment of inertia, J , is then:

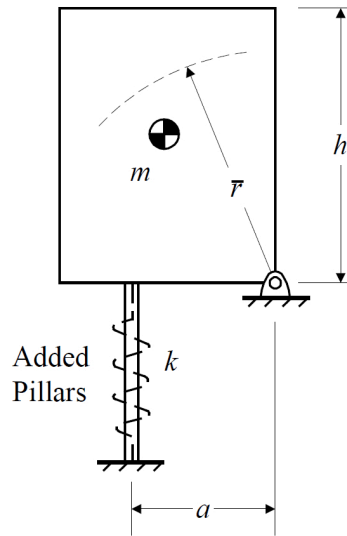
$$J = m \bar{r}^2 = 3 \times 10^7 \text{ kg-m}^2$$

The effective rotational stiffness of the foundation is then estimated as:

$$K_f = 1.35 \times 10^{11} \text{ N-m/rad}$$

Furthermore, assume that stiffening is proposed in the form of two parallel pillars made up of 20 cm extra-heavy steel pipe, each 6 meters long, and located under the house side bulkheads, as indicated on 4/7 FIGURE 7.

FIGURE 7
Deckhouse Stiffening



The effective axial stiffness of the parallel pillars is calculated to be $k = 5 \times 10^8$ N/m. The pillars are located at a distance $a = 5$ m aft of the forward bulkhead, so that the incremental rotational stiffness added by the pillars is:

$$\delta K_f = ka^2 = 1.25 \times 10^{10} \text{ N-m/rad}$$

The stiffness of the stiffened foundation is

$$K'_f = K_f + \delta K_f = 1.475 \times 10^{11} \text{ N-m/rad}$$

which represents a 9.3 percent increase. The new rocking frequency becomes

$$f'_R = f_R \sqrt{1.093} = 669 \text{ cpm}$$

Then, the house fore-and-aft natural frequency is raised to:

$$f'_e = \sqrt{\frac{1}{1/f_\infty^2 + 1/f_R'^2}} = 513 \text{ cpm}$$

This represents a 2.6 percent increase over the value of 500 cpm without the pillars, and far less than the 600 cpm judged to be needed. In order to achieve $f'_e = 600$ cpm, reversing this last calculation gives $f_R = 907$ cpm, and $K_f = 2.72 \times 10^{11}$ N-m/rad. This is about double the rotational stiffness of the original design. It would be essentially impossible to double the support structural stiffness if standard design features were employed.

This example demonstrates the common misconception that vibration problems are always eliminated by stiffening the structure. This may be true for light structures such as mast, hand-rails, and the like, but with major substructures, such as a deckhouse, once the scantlings and principal dimensions are set incorrectly, the problem tends not to be correctable by simple stiffening, as the preceding example suggests. It may be possible to reduce stiffness to avoid a resonance, which was the spirit of the previous example on thrust bearing foundation stiffness in Subsection 4/5. However, reductions in stiffness are generally undesirable. Stiffness reduction can result in excessive static deflection. As in the thrust bearing foundation stiffness example, the simplest recourse might be to change the engine RPM and/or the propeller blade number. In

this current example, a switch from a five to a six-bladed propeller with no **other** structural changes would place the resonance within the operating range, but at the desired 20% below the new full power blade-rate exciting frequency of 600cpm.

The following specific steps are suggested for reducing excessive propeller induced deckhouse vibration at the concept design level:

- i)* Perform the preceding analysis on the candidate deckhouse and support structure design to predict the fore-and-aft rocking bending/shearing natural frequency.
- ii)* Compare the natural frequency with the blade-rate exciting frequency of any proposed engine and propeller characteristics. If the engine and/or propeller have **not yet been selected**, note the blade rate frequency to be avoided in that ultimate selection.
- iii)* If the engine and propeller characteristics have been **decided upon**, establish the stiffness range of the deckhouse supporting structure calculated as necessary to avoid resonance within 20% of the full power blade rate frequency. This is just as recommended with the longitudinal vibration issue at the concept design stage. This defers the ultimate implementation to the preliminary/detailed designers who will have been made aware of the potential difficulty in an initial phase.

1 Introduction

1.1 Scope and Objective (1 September 2023)

The design and construction of a ship free of excessive vibration continues to be a major concern and, as such, it is prudent to investigate, through analysis, the likelihood of vibration problems early in the design stage. Vibration analysis is aimed at the confirmation of the many design considerations associated with:

- Stern configuration
- Main propulsion machinery
- Propeller and shafting system
- Location and configuration of major structural assemblies

The ship hull structure includes the outer shell plating and all internal members, which collectively provide the necessary strength to satisfactorily perform the design functions in the expected sea environment. The hull structure behaves as a free-free beam (both ends free) when subjected to dynamic loads. The vibration induced by the propulsion system is a common source of ship vibration. The vibration from this source manifests itself in several ways. Dynamic forces from the shafting system are transmitted to the hull through shaft bearings. The propeller induces fluctuating pressures on the surface of the hull, which induces vibration in the hull structure. The main and auxiliary engines can directly cause vibrations through dynamic forces transmitted through their supports and foundations. The response to this forcing can cause the vibration of the hull girder, deckhouse, equipment, deck and other structures. When attempting to determine the source of vibration, it is necessary to establish the frequency of excitation and to relate the frequency of excitation to the shaft rotational frequency by determining the number of oscillations per shaft revolution. **Usually only the first (one cycle per revolution) and the second (two cycles per revolution) order external moments need to be considered for engines with six or fewer cylinders.** The engine manufacturer is to provide the magnitude of **engine excitation** forces and moments.

The response of the hull structure may be resonant or non-resonant. The hull structure will normally vibrate in the following modes:

- Vertical bending
- Horizontal bending
- Torsional (twist)
- Longitudinal
- Coupling between horizontal and torsional modes, especially in containerships.

Typical major substructures include deckhouses, main deck structures and large propulsion machinery system, which because of the direct coupling with hull structure vibration can significantly influence the total or global pattern of ship vibration. In analyzing vibration patterns of such large complex structures, it is necessary to identify the principal cause of observed excessive vibration. Excessive vibration of a major substructure may be due to structural resonance in the substructure or in the attachment detail for the substructure and hull structure.

Local structural components are the minor structural assemblies, relative to major substructures previously referred to. Local structural components may be decks, bulkheads, platforms, handrails, minor equipment foundations, or the main engine coupled with its seating and inner bottom structure, and are components of larger structures (major substructures) or of the entire hull structure. Most ship vibration problems occur in local structural components and are the result of either strong inputs received from the parent structure amplified by resonance effects in the local structure or are the response to vibratory forces generated by mechanical equipment attached to the local structure.

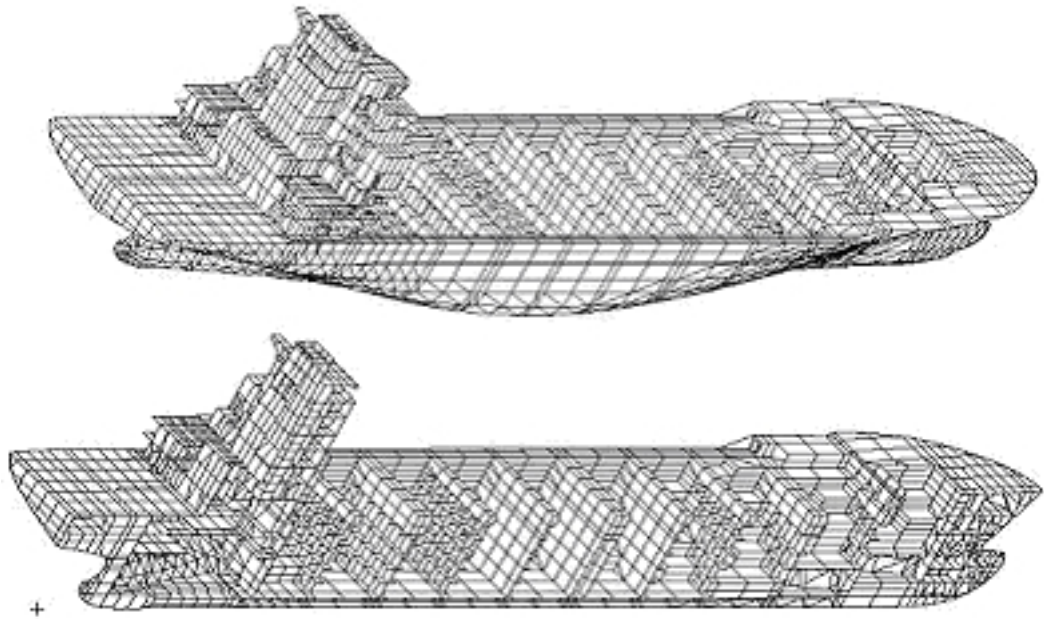
Local structural panels are basic elements forming local structural components. Local structural panels are identified as plate panels between stiffeners, stiffener panels enclosed by strong supports, or girder panels enclosed by bulkheads and/or decks. Typically, the natural frequencies of local structural components or local structural panels should be out of the range between 85% and 115% of major excitation frequencies to avoid resonance.

This section describes the analysis procedure for propeller and main engine induced ship hull vibration using a three-dimensional finite element method. The objective of vibration analysis is to determine the overall vibration characteristics of the hull structure and major superstructures so that areas sensitive to vibratory forces may be identified and assessed relative to standard acceptance criteria on vibration level.

1.3 Procedure Outline of Ship Vibration Analysis

5/1.3 FIGURE 1 provides a flowchart of the procedure. It is recommended to perform the vibration analysis using a three-dimensional finite element model representing the entire ship, including the deckhouse and main propulsion machinery system. Both free and forced vibration analyses are included.

FIGURE 2
Global FE Model Example



3.3 Engine, Propeller Shaft and Stern/Skeg

3.3.1 Main Engine (1 September 2023)

Care should be taken when modeling the engine and surrounding structure. The engine foundation should be modeled in detail. The engine can be modeled using solid or shell elements such that the mass, center of gravity (CG), and the engine stiffness accurately represents the specifications of the engine. By assigning different densities of the material, mass and center of gravity of the engine can be controlled. In order to match the engine stiffness, different Young's moduli can be used for longitudinal and transverse structural members, respectively, based on the engine stiffness data provided by the engine manufacturer. An example of simplified diesel engine FE model is shown in 5/3.3.3 FIGURE 3. An example of a turbine engine FE model is shown in 5/3.3.3 FIGURE 4. Main engine top stays can be modeled as spring elements with spring coefficients obtained from manufacturer. **Hydraulic stays provide some damping, which can be modeled using dashpot elements with a damping coefficient obtained from the manufacturer.**

3.3.2 Propeller Shaft and Propeller (1 September 2023)

The propeller shaft should be explicitly modeled such that it accurately represents the stiffness of the shaft. It may be modeled with beam or bar elements. Care should be taken when modeling the support bearings. The transverse supporting structure should be modeled with the help of membrane shell elements. The tail shaft should begin from the center of the propeller hub as shown in 5/3.3.3 FIGURE 5. The propeller may be modeled as a mass element **considering propeller immersion.**

3.3.3 Stern/Skeg Structure

Proper care should be taken when modeling the skeg structure. Typically, the skeg structure is a thick casting. If it is modeled with shell elements, the thickness should be increased appropriately to account for the stiffness of that structure.

FIGURE 3
Engine Model Example

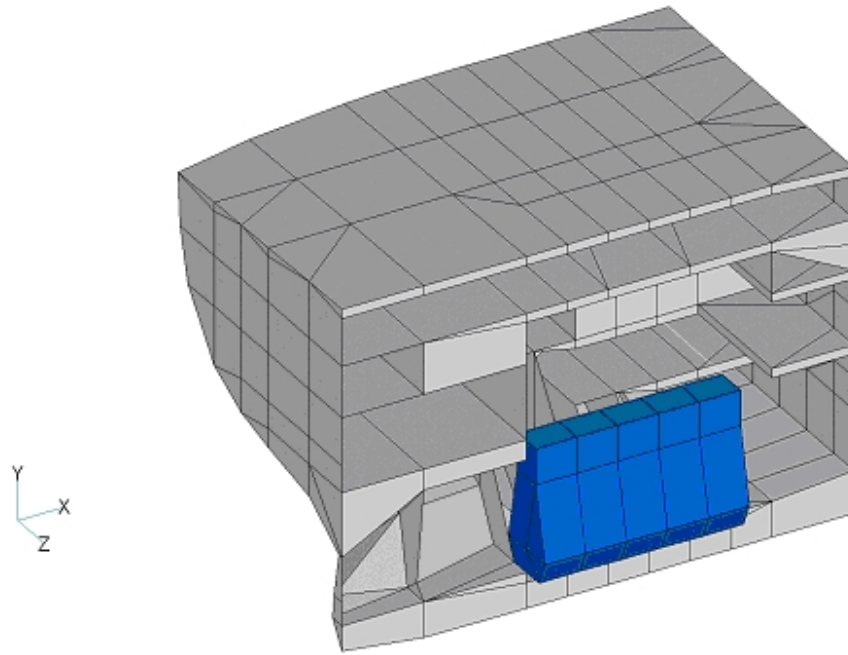


FIGURE 4
Turbine Engine and Propeller Shaft Modeling Example

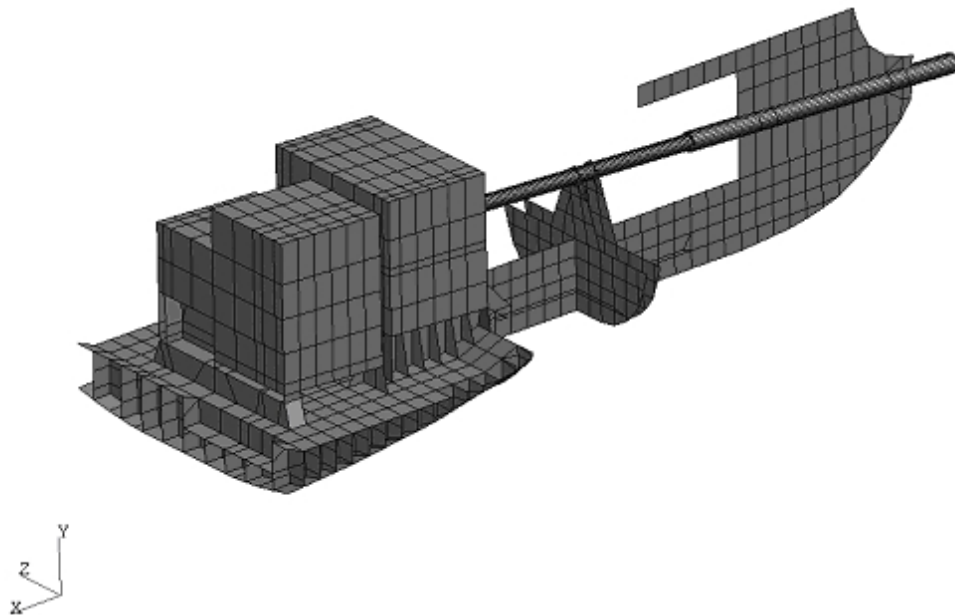
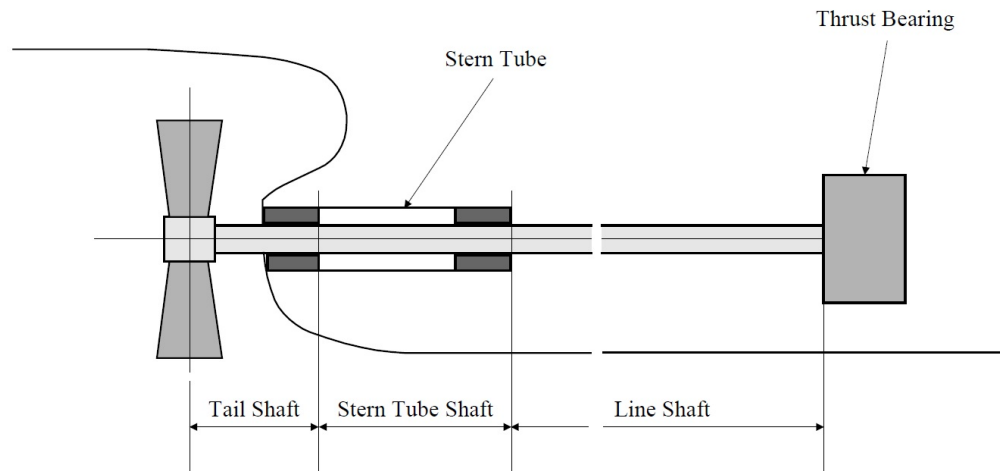


FIGURE 5
Propeller Shaft



3.5 Lightship Weight Distribution (1 September 2023)

Lightship weight distribution is an important factor in any vibration analysis. Results can change dramatically if it is not properly represented. Typically, masses of all the heavy equipment should be modeled as mass elements and **should be connected to well-supported structural nodes using rigid elements** such that the total mass and center of gravity (CG) of that equipment are maintained. This distribution should be as accurate as possible in the areas of interest. In general, the CG from the FE model and the CG from the T & S booklet should be within 0.5% of the total ship length.

The deckhouse superstructure weight and mass distribution, longitudinally and vertically, should also be accurately represented. Typically, the non-structural masses that account for furniture, outfitting and deck coverings in the deckhouse superstructure should be modeled by changing local structural density or modelled as mass elements uniformly placed at well-supported nodes. This distribution should be provided by the ship designer or shipbuilder. If an accurate distribution is not available, sensitivity studies should be conducted to determine the most critical values between 40 kg/m² to 100 kg/m².

3.7 Cargo, Water Ballast in Tanks and Fuel Oil in Tanks (1 September 2023)

Mass elements can be applied to the tank boundaries to represent cargo, water ballast in tanks, and fuel oil in tanks. **The distribution of the mass elements to tank boundaries should satisfy that the total mass and the center of gravity of the cargo or liquid tank content are maintained.**

Cargo and liquid tank content can also be modeled as concentrated mass located at the mass CG node and connected to the tank boundary nodes using rigid elements that allow the motion of the concentrated mass node (dependent node) to be calculated as an average motion of all the tank boundary nodes. This can be achieved, for example, by using RBE3 elements in NASTRAN (Siemens, 2020a).

When the tank boundaries vibrate, the tank liquids are only partially disturbed, depending on the direction of tank boundary vibration. Therefore, tank liquids can also be treated as an added mass problem and modeled using boundary element method (BEM) such as 'MFLUID' card of NASTRAN (Siemens, 2020b).

3.8 Local Structural Component Models (1 February 2018)

For local structural components, the FE models should also include surrounding structures which should be extended to the adjacent strong supports, such as primary transverses or girders, bulkheads, or decks, etc. For example, for local vibration analysis of the navigation deck, the FE model should include not only the navigation deck but also the surrounding bulkheads between the navigation deck and adjacent decks. In the

calculations, fixed boundary conditions are applied to the FE model in order to reduce the modes of the extended structures. The mesh size can range from one or two elements between stiffeners to about six elements between stiffeners.

3.9 Local Structural Panel Models (1 February 2018)

A number of local structural panels are selected from the various decks, platforms, flats and bulkheads within the superstructure and regions of the hull for the natural frequency calculations. The panels selected are either typical locations or those locations expected to have the lowest natural frequencies in the areas of concern. In the calculations, pinned boundary conditions are applied to stiffener panels and plate panels. For girder panels, suitable boundary conditions should be assessed from the local design arrangements if pinned boundary conditions result in lower natural frequencies than major excitation frequencies. The mesh size can range from three elements between stiffeners to about six elements between stiffeners.

5 Loading Condition

5.1 Selection of Loading Conditions and Ship Speed

The objective of the vibration analysis is to investigate the ship vibration performance at intended service conditions. Therefore, the loading conditions, such as full load condition and ballast condition, in which the ship operates at ship design speed, 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 to be taken, it is recommended that vibration analysis be performed for two selective conditions; either

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

Depending on the loading condition, the mass of the cargo and ballast is then distributed in the structural model using mass elements. The corresponding added mass and the buoyancy springs are then calculated and added to the model.

5.3 Added Mass (1 February 2018) (1 September 2023)

For local vibration analysis, when local structures are in contact with fluid, the added mass effect of the fluid should be considered in the calculation. The added mass can be considered either through the boundary element method (e.g., 'MFLUID' in NASTRAN) or using empirical formulas. For global vibration analysis, **the added mass of water surrounding the hull may be modeled using** a three-dimensional seakeeping analysis program. The analysis includes three main tasks:

- *Development of a hydrodynamic panel model:* The hydrodynamic panel model is to represent the geometry of the ship's hull below the still-water line when using a linear seakeeping program. It is recommended that a total of about 2000 panels be used for the ship's hull surface, including port and starboard sides. A computer program may generate the panel model using the ship's offsets. The main particulars required for the hydrodynamic model include ship displacement, drafts, location of center of gravity and radii of gyration.
- *Hydrodynamic analysis:* The purpose of the analysis is to obtain the distributed added mass. In general, the natural frequency of the ship is much higher than the wave frequencies considered in the seakeeping analysis. To obtain the added mass at such high frequencies, infinite frequency added mass option should be used.
- *Mapping hydrodynamic results onto FE model:* The distributed added mass from the seakeeping analysis is represented as an added mass on each hydrodynamic panel model. An interface program can be used to map the heave added mass onto the FE model. It should be confirmed that the total added mass on the FE model is equal to the total added mass on the hydrodynamic panel model.

Alternative approaches may be considered on a case-by-case basis by ABS.

5.5 Buoyancy Springs (1 September 2023)

The effect of buoyancy on hull vibration is generally regarded as small. Where it is necessary to consider the buoyancy effect, it may be modeled by adding rod elements to the wetted surface of the model. The rod elements work as springs and the total stiffness of the rod elements should be equivalent to the ship's vertical buoyancy stiffness calculated as:

$$\rho g A_{WP}$$

where

ρ = water density

g = gravitational acceleration

A_{WP} = waterplane area

The distribution of the rod element stiffness properties should be based on the local breadth of the waterline at each longitudinal location where the rod elements are placed.

5.7 Special Conditions

Shipyards and/or ship owners may consider additional loading conditions to be investigated during the vibration analysis process.

7 Free Vibration

7.1 Analysis Procedure (1 September 2023)

Computation of the natural frequencies and mode shapes should be performed by solving an eigenvalue problem. The natural frequencies (eigenvalues) and corresponding mode shapes (eigenvectors) of the three-dimensional finite element model can be obtained by solving the following equation of motion:

$$[M]\{\ddot{u}(t)\} + [C]\{\dot{u}(t)\} + [K]\{u(t)\} = \{F(t)\}$$

where

$[M]$ = mass matrix

$[C]$ = damping matrix

$[K]$ = stiffness matrix

$\{\ddot{u}\}$ = vector of accelerations

$\{\dot{u}\}$ = vector of velocities

$\{u\}$ = vector of displacements

$\{F\}$ = vector of harmonic forces

For free vibration, damping $[C]$ and forces $\{F\}$ are zero. The solution would then be found from:

$$([K] - \omega^2[M])\{\Phi\} = 0$$

where

$\{\Phi\}$ = eigenvector or mode shape

ω = natural frequency

This is an eigenvalue problem that can be solved by normal mode analysis. An important characteristic of normal modes is that the scaling or magnitude of the eigenvectors is arbitrary. Mode shapes are fundamental characteristic shapes of the structure and are therefore relative quantities. Examples of mode shapes of a typical LNG carrier are shown in 5/7.1 FIGURE 6 and 5/7.1 FIGURE 7. The natural frequencies obtained from the analysis can then be compared to the excitation frequencies to check for resonance.

FIGURE 6
First Two Vertical Mode Shapes

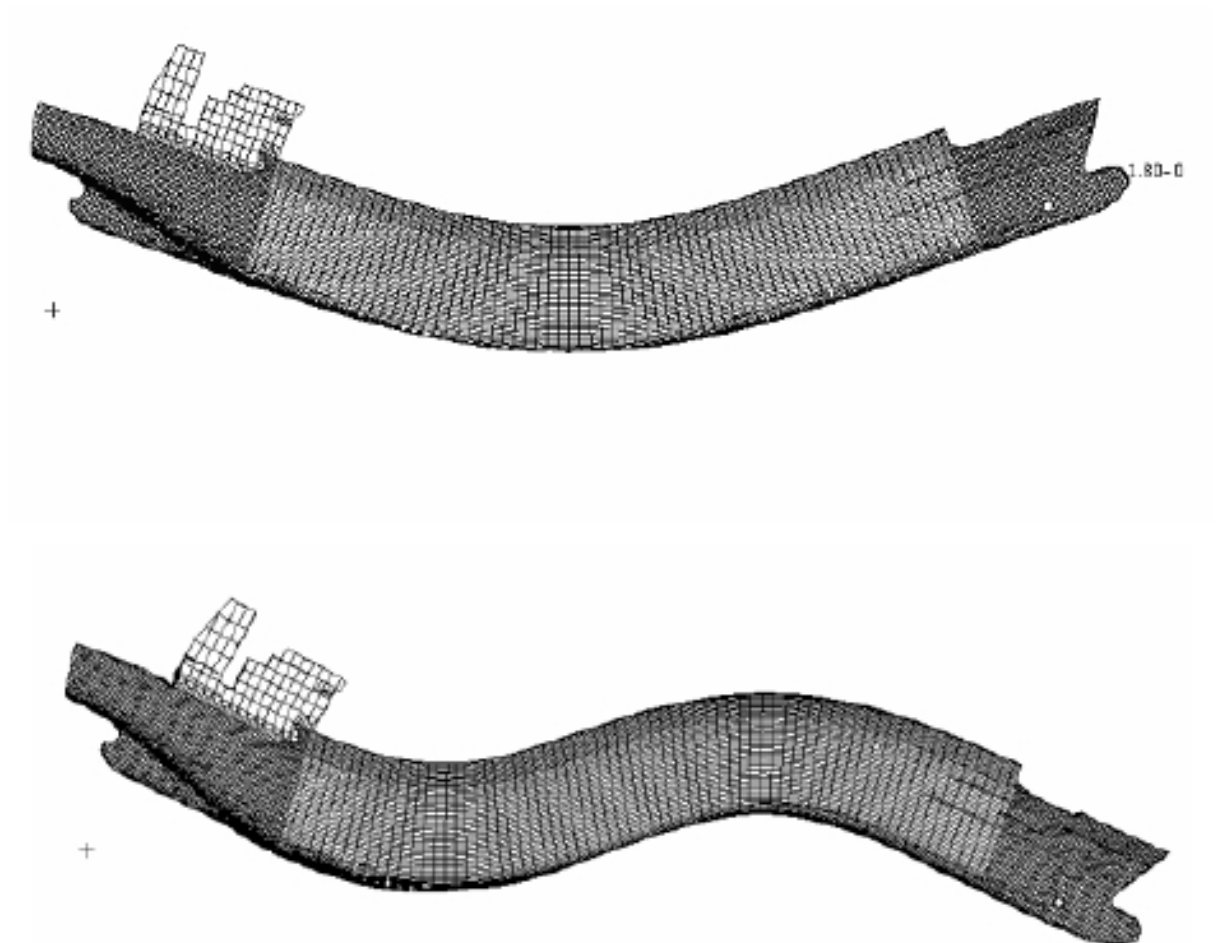
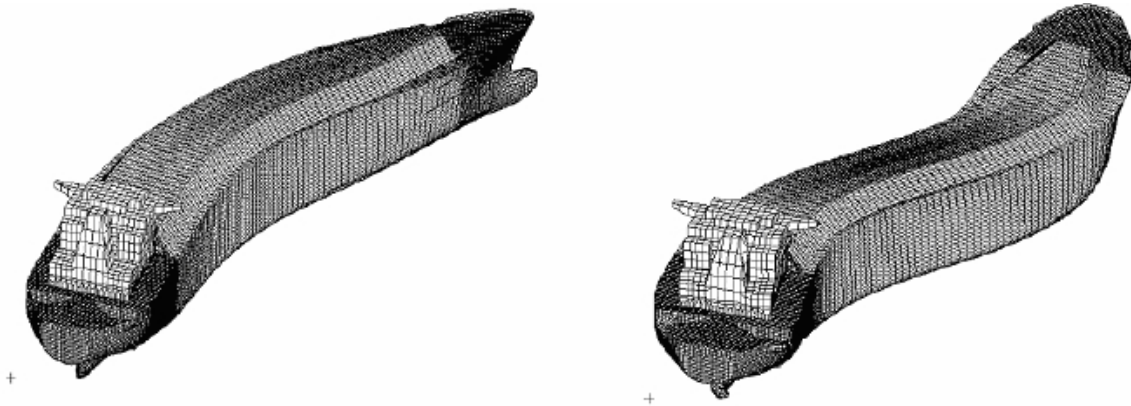


FIGURE 7
First Two Horizontal Mode Shapes



7.3 Checking Points

7.3.1 Mode Shapes

When performing free vibration analysis, it is important to validate the calculated results by investigating their mode shapes. There are several checking points:

- The first six mode shapes should be rigid body modes and these six rigid body mode shapes, namely Pitch, Roll, Yaw, Surge, Heave and Sway, are not to display elastic distortion. Frequency is generally very low, well below the first elastic natural mode. Any mixing of rigid body modes and/or missing rigid body mode(s) would be a good indication of an erroneous FE modeling, especially when incorrect multi-point constraints are applied to the FE model.
- The first several elastic modes should be for the hull structure. For typical commercial ships, there is usually no local modes below 4 Hz. Any “local modes” with low natural frequencies would be an indication of incorrect FE modeling. Too low frequencies are usually an indication of too high a mass or too low stiffness, or both.

It is a usual practice to investigate mode shapes up to at least twice the blade-rate frequency.

7.3.2 Natural Frequencies

Another checking point is to check the calculated natural frequencies from the free vibration analysis. Kumai’s formula in Subsection 4/3 for two-node vertical vibration may be used to preliminarily evaluate the calculated two-node natural frequency in order to verify that the FEM frequencies are in a proper range.

9 Propeller Excitation

9.1 Introduction

Excitation forces from the propeller are transmitted into the ship via the shaft line and also in the form of pressure pulses acting on the ship hull surface above the propeller. Whereas propeller shaft forces (bearing forces) are the most significant factor for vibrations of shaft lines, the predominant factor for vibrations of ship structures are the pressure fluctuations on the hull surface (hull surface forces).

9.3 Propeller Shaft Forces

9.3.1 General (1 September 2023)

Propeller shaft forces (bearing forces) are fluctuating load effects caused by the non-uniform wake inflow to the propeller. Usually, bearing force fluctuations transmitted from propeller to propulsion shafting are not significantly affected by cavitation processes if blade cavitation is not severe. In the absence of high shaft inclination (see 3/5.1), the magnitude of the bearing forces depends on the characteristics of the wake field, the geometric form of the propeller (in particular the skew and blade number), the ship speed and the rotational speed of the propeller. For a given application, the forces and moments generated by a fixed pitch propeller without severe cavitation are, in general, proportional to the square of the revolutions, since for a considerable part of the upper operational speed range the ship will work at a nominally constant advance coefficient.

Although the thrust fluctuation can be as high as 10% of the mean thrust it usually falls between 2 and 8%. The force fluctuations in transverse and vertical directions are between about 1 to 2% of the mean thrust. In most cases, the moment fluctuation about the transverse axis is predominant (5 ~ 20% of the mean torque) compared to fluctuations of the moment about the vertical axis and of the torque (1~10%).

9.3.2 Previous Studies (1 September 2023)

In the 1980's, an analytical investigation of 1st and 2nd blade rate bearing forces/moments was performed for 20 ships (Veritec, 1985). The results of these calculations are useful for making preliminary estimates of the dynamic forces at the early stages of design. 5/9.3.2 TABLE 1 shows the results of these calculations in terms of the mean values and their ranges. In the Table, each of the six loading components is expressed in terms of the mean thrust T_0 or the mean torque Q_0 .

Propeller bearing forces and moments can excite various modes of vibration. For computations of axial and torsional vibrations of the shaft line, fluctuations in thrust and torque should be taken into account. Bending vibrations of the shaft are influenced by transverse forces in horizontal and vertical directions as well as by bending moments about the corresponding axes.

Longitudinal thrust variations should be applied at the thrust bearing location, while the other bearing forces and moments may be applied at the stern tube bearing.

TABLE 1
Propeller Bearing Forces and Moments for 20 Real Ship Case Study

				<i>Blade Number</i>		
				<i>4</i>	<i>5</i>	<i>6</i>
Blade rate frequency component	Thrust	$F_x(1)$	Mean	$0.084 T_0$	$0.020 T_0$	$0.036 T_0$
			Range	$\pm 0.031 T_0$	$\pm 0.006 T_0$	$\pm 0.0024 T_0$
	Vertical force	$F_z(1)$	Mean	$0.008 T_0$	$0.011 T_0$	$0.003 T_0$
			Range	$\pm 0.004 T_0$	$+ 0.009 T_0$	$\pm 0.002 T_0$
	Horizontal force	$F_y(1)$	Mean	$0.012 T_0$	$0.021 T_0$	$0.009 T_0$
			Range	$\pm 0.011 T_0$	$\pm 0.016 T_0$	$\pm 0.004 T_0$
	Torque	$M_x(1)$	Mean	$0.062 Q_0$	$0.0011 Q_0$	$0.030 Q_0$
			Range	$\pm 0.025 Q_0$	$\pm 0.0008 Q_0$	$\pm 0.020 Q_0$
	Vertical moment	$M_z(1)$	Mean	$0.075 Q_0$	$0.039 Q_0$	$0.040 Q_0$
			Range	$\pm 0.050 Q_0$	$\pm 0.026 Q_0$	$\pm 0.015 Q_0$
	Horizontal moment	$M_y(1)$	Mean	$0.138 Q_0$	$0.125 Q_0$	$0.073 Q_0$
			Range	$\pm 0.090 Q_0$	$\pm 0.085 Q_0$	$\pm 0.062 Q_0$
Twice blade rate frequency component	Thrust	$F_x(2)$	Mean	$0.022 T_0$	$0.017 T_0$	$0.015 T_0$
			Range	$\pm 0.004 T_0$	$\pm 0.003 T_0$	$\pm 0.002 T_0$
	Vertical force	$F_z(2)$	Mean	$0.008 T_0$	$0.002 T_0$	$0.001 T_0$
			Range	$\pm 0.004 T_0$	$+ 0.002 T_0$	$\pm 0.001 T_0$
	Horizontal force	$F_y(2)$	Mean	$0.001 T_0$	$0.006 T_0$	$0.003 T_0$
			Range	$\pm 0.001 T_0$	$\pm 0.003 T_0$	$\pm 0.001 T_0$
	Torque	$M_x(2)$	Mean	$0.016 Q_0$	$0.0014 Q_0$	$0.010 Q_0$
			Range	$\pm 0.010 Q_0$	$\pm 0.008 Q_0$	$\pm 0.002 Q_0$
	Vertical moment	$M_z(2)$	Mean	$0.019 Q_0$	$0.012 Q_0$	$0.007 Q_0$
			Range	$\pm 0.013 Q_0$	$\pm 0.011 Q_0$	$\pm 0.002 Q_0$
	Horizontal moment	$M_y(2)$	Mean	$0.040 Q_0$	$0.080 Q_0$	$0.015 Q_0$
			Range	$\pm 0.036 Q_0$	$\pm 0.040 Q_0$	$\pm 0.002 Q_0$

9.5 Hull Surface Forces Induced by Propeller Cavitation

9.5.1 General (1 September 2023)

Propeller-induced hull surface pressure fluctuations are more significant than shaft bearing forces. In merchant ships, for which a certain degree of propeller cavitation is generally tolerated for the

sake of optimizing the propeller efficiency, about 10% of propeller-induced vibration velocities are caused by bearing forces, whereas approximately 90% are due to pressure fluctuations, or hull surface forces. In the design of ships having propellers with weak cavitation, the ratio may be reversed, while at the same time the absolute excitation level is much lower.

Pressure fluctuation is generally proportional to the acceleration of the propeller cavity volume. Calculation of the cavity volume may require knowledge of the pressure distribution on the propeller blade, both in the radial and circumferential direction. The flow condition in the tip region of the propeller blades has a particularly strong influence on cavitation processes. The formation and subsequent detaching of tip vortices additionally complicate the flow condition at the blade tips. Computer programs for the prediction of cavitation volumes are correspondingly complex.

To avoid strong cavitation-induced pressure amplitudes, the volume curve is to exhibit the smallest possible curvatures (note that pressure fluctuation is proportional to the acceleration of cavity volume). This can be achieved by influencing the wake (minimizing wake peaks) and by utilizing appropriate propeller geometry. However, improved cavitation characteristics are usually offset by reductions in efficiency. Selection of a larger area-ratio and reduction in propeller tip loading by utilizing smaller pitch and camber at the outer radii are the most effective measures. In addition, by means of skew, a situation can be achieved where the individual profile sections of a propeller blade are not all subjected to their maximum loading at the same time, but instead the volume curve is rendered uniform by the offset in the circumferential direction. Some concepts tolerate comparatively severe cavitation and are aimed at making the growth and collapse of the cavitation layer as slow as possible.

From ABS experience, the pressure amplitude above the propeller alone is not adequate to characterize the excitation behavior of a propeller. Therefore, no generally valid limits can be stated for pressure fluctuation amplitudes. These amplitudes depend not only on technical constraints such as achievable propeller tip clearance and power to be transmitted, but also on the geometry-dependent compromise between efficiency and pressure fluctuation. Nevertheless, pressure amplitudes at blade frequency of 1 to 2, 2 to 8 and over 8 kPa at a point directly above the propeller can be categorized as low, medium and high, respectively. Total vertical force fluctuations at blade frequency, integrated from pressure fluctuations, would range from 10 kN for a small ship to 1000 kN for a high performance container ship. Whether these considerable excitation forces result in high vibrations depends on the dynamic characteristics of the ship's structure and can only be judged on the basis of a forced vibration analysis.

The propeller-induced hull surface force may be applied as a concentrated force at a well-supported point directly above the propeller, or as a pressure acting over a certain area of the stern bottom plating above the propeller. Special attention should be paid to phasing between the propeller-induced hull surface forces, the shaft bearing forces, and all other excitation forces.

9.5.2 Empirical Formulae

There are three methods for predicting hull surface pressure: by empirical methods, by calculations using advanced theoretical methods and by experimental measurements.

With regard to empirical methods, the most well-known and adaptable method is that of Holden, et al (1980). This method is based on the analysis of full-scale measurements for some 72 ships. The method is intended as a first estimate of the likely hull surface pressures using a conventional propeller form. Regression based formula for estimation of the non-cavitating and cavitating pressure are proposed as follows by Holden, et al.

$$p_o = \frac{(ND)^2}{70} \frac{1}{Z^{1.5}} \left(\frac{K_o}{d/R} \right) \text{ N/m}^2 \quad (\text{non-cavitating pressure})$$

and

$$p_c = \frac{(ND)^2}{160} \frac{V_s(w_{Tmax} - w_e)}{\sqrt{(h_a + 10.4)}} \left(\frac{K_c}{d/R} \right) \text{ N/m}^2 \quad (\text{cavitating pressure})$$

where

N	=	propeller RPM	
D	=	propeller diameter, in m	
V_s	=	ship speed, in m/s	
Z	=	blade number	
R	=	propeller radius, in m	
d	=	distance from $0.9R$ to a position on the submerged hull when the blade is at the T.D.C. (Top Dead Center) position (m)	
w_{Tmax}	=	maximum value of Taylor wake fraction in the propeller disc	
w_e	=	mean effective full scale Taylor wake fraction	
h_a	=	depth of shaft centerline	
K_o	=	$1.8 + 0.4(d/R)$	for $d/R \leq 2$
K_c	=	$1.7 + 0.7(d/R)$	for $d/R < 1$
K_c	=	1.0	for $d/R > 1$

The total pressure impulse, combining both the cavitating ' p_c ' and non-cavitating ' p_o ' components, is then calculated from

$$p_z = \sqrt{(p_o^2 + p_c^2)}$$

Empirical methods of this type are particularly useful as a guide to the expected pressures. They should not, however, be regarded as a definitive solution, because differences, sometimes quite substantial, will occur when correlated with full-scale measurement. For example, p_c and p_o regression formulae give results having a standard deviation of the order of 30% compared to the base measurements from which it was derived.

9.5.3 Theoretical Method (1 September 2023)

In the case of a more rigorous calculation, more details can be considered, which results in a higher level of accuracy. Theoretical models, which would be used in association with this form of analysis, can be broadly grouped into the lifting surface or vortex lattice categories. In particular, unsteady lifting surface theory is a basis for many advanced theoretical approaches in this field. Notwithstanding the ability of analytical methods to provide a solution, care should be exercised in the interpretation of the results since these are particularly influenced by factors such as wake scaling procedures, the description of the propeller model and the hull surface, the distribution of solid boundary factors and the harmonic order of the pressure considered in the analysis. Furthermore, propeller calculation procedures assume a rigid body condition for the hull, and thus do not account for the self-induced pressure resulting from hull vibration. These should be taken into account by other means; typically, finite element models of the hull structure. As a consequence of all of these factors considerable care should be exercised in interpreting the

results. Analysis procedures for direct calculations of propeller-induced loads are briefly described in 5/9.7.

9.5.4 Model Tests

Model tests to predict hull surface pressures can be conducted in either cavitation tunnels or specialized facilities such as depressurized towing tanks. In the past, the arrangement in a cavitation tunnel comprised a simple modeling of the hull surface by a flat or angled plate above a scale model of the propeller. Although this technique is still used in some establishments, a more enlightened practice is to use a partial model with shortened body. Wake screens are typically used to fine-tune the wake field when a partial model is used for wake measurements. In some large facilities, however, a full-length ship model may be used. The advantage of using a model of the actual hull form is twofold: first, it assists in modeling the flow of water around the hull surface and does not require wake screens, and second, it makes the interpretation of the measured hull surface pressure easier since the actual hull form is simulated.

To interpret model test results, reference can be made to dimensional analysis, from which it can be shown that the pressure at a point on the hull surfaces above a propeller is dependent on the following set of dimensional parameters:

$$p = \rho n^2 D^2 \Phi(J, K_T, \sigma, R_n, F_n, z/D)$$

where

ρ	=	water density in ton/m ³
n	=	propeller RPM
D	=	propeller diameter in m
J	=	advance coefficient
K_T	=	propeller thrust coefficient
σ	=	cavitation number
R_n	=	Reynolds number
F_n	=	Froude number
z	=	distance from propeller to point on the hull surface in m

As a consequence of the above equation, a pressure coefficient K_p can be defined as follows:

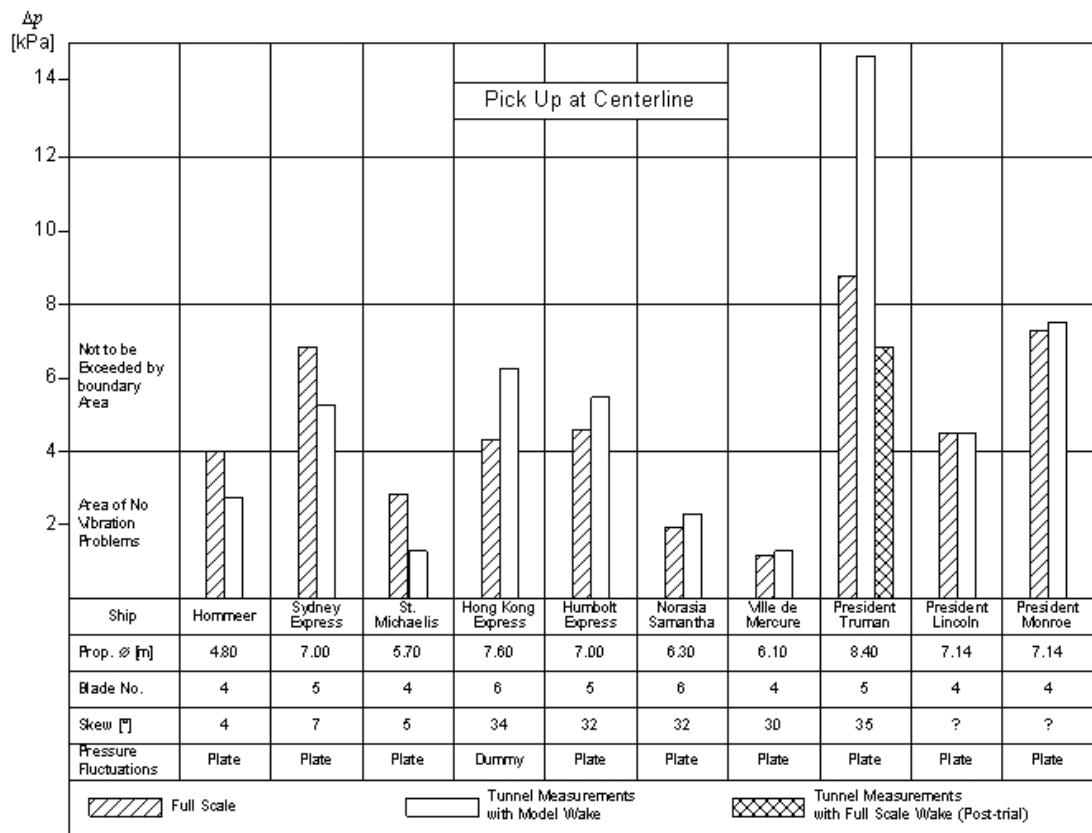
$$K_p = \frac{p}{\rho n^2 D^2}$$

As defined, dimensionless hull surface pressure is a function of propeller loading, cavitation number, geometric scaling, Froude number and Reynolds number.

In propeller cavitation tests, the test conditions are usually chosen such that the average propeller thrust loading (expressed by K_T and J -identity) is equal for model and full scale. In addition, the pressure is lowered to such a level that model and full-size cavitation number are equal at corresponding points in the propeller disc. If the model test needs to fulfill Froude number identity, the test can only be realized in a depressurized towing tank or a tunnel with free surface. For a cavitation tunnel without a free surface, a rate of rotation for model scale is chosen within practical limits related to the tunnel capacity, the particular test set-up and the ranges of static pressure to be adjusted.

Although cavitation number and Froude number can be satisfied in most model tests, Reynolds number identity can seldom be fulfilled. This causes the well-known scale effect on the model test measurements. To take the scale effect into account in the model measurement, sometimes leading edge roughness for model propeller is used for tripping the flow over the propeller blades to turbulence. However, it also should be noted that scale effects exist not only on the flow around the propeller blade surface but also on the inflow toward propeller. 5/9.5.4 FIGURE 8 shows the comparisons of maximum propeller induced pressure between mode scale and full-scale data. As seen, the discrepancies are notable. To obtain reliable model test results, model test measurement should be appropriately calibrated based on full-scale data.

FIGURE 8
Scale Effect due to Propeller Inflow Condition



9.7 Direct Calculation of Bearing and Surface Forces

The overall calculation procedure for propeller loading calculation is as follows:

- Open water propeller flow analysis
- Nominal wake harmonic analysis
- Effective wake calculation – propeller/nominal wake interaction
- Propeller cavity pattern simulation
- Bearing forces/moments calculation
- Propeller induced hull pressure calculation

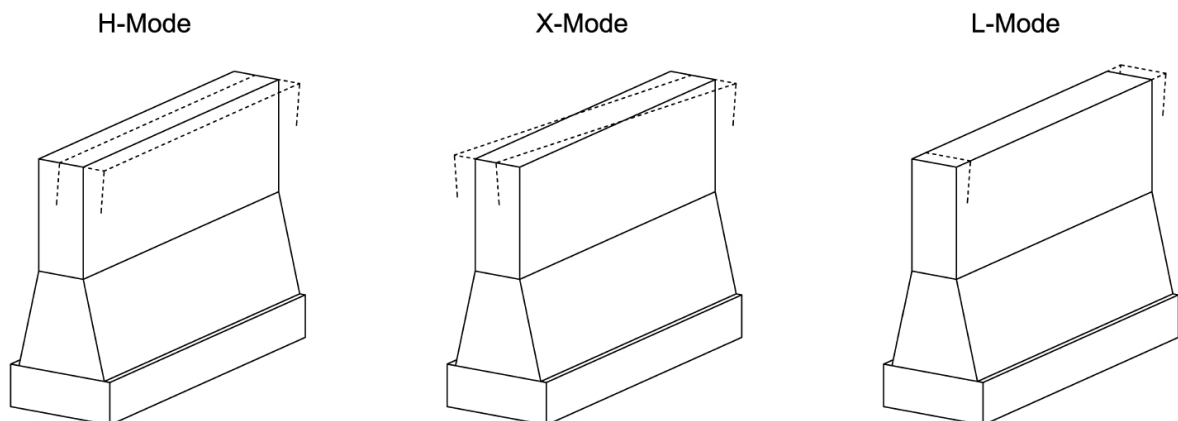
In order to perform the aforementioned analyses, the following information should be required:

- Measured nominal wake
- Stern wave height
- Water temperature
- Ship speed
- Propeller RPM
- Draft
- Propeller blade number
- Propeller geometry (diameter, pitch, rake, skew angle, chord length, section profiles)
- Propeller location (related to ship hull)
- Ship stern geometry

11 Engine Excitation (1 September 2023)

The excitation forces and moments of low-speed diesel engine can be a significant source of hull vibration. Dominant forces acting on the engine structure originate from the combustion process (guide force **moments**) and from the inertia of mass in motion (external forces **and moments**). Guide force **moments** result from transverse reaction forces acting on the crosshead, causing rocking (**H-mode**) and twisting (**X-mode**) of the engine (see 5/11 FIGURE 9). These forces may cause resonance with the engine foundation structure. A possible solution for the potential vibration problem may be the installation of transverse stays (top bracings), connecting the engine's top structure to the ship hull. **The alternating thrust or axial shaft vibrations caused by the gas pressure acting on the crank throw through the connecting rod mechanism can cause main engine longitudinal (L-mode) resonance** (see 5/11 FIGURE 9). Installation of main engine top bracing in the longitudinal direction can mitigate this longitudinal (**L-mode**) resonance. Additionally, most engine designs have axial dampers to minimize axial vibrations.

FIGURE 9
Basic Main Engine Vibration Modes (1 September 2023)



Detailed consideration of the engine excitation forces and their effect on the ship hull structures is an important part in prevention of damaging structural vibrations. The magnitude **and phases** of these forces and moments should be obtained from the engine manufacturer.

External and guide force moment amplitudes change with the change in rotation speed of the engine crankshaft. External moments at different rotation speeds typically obey the following quadratic rule:

$$M_A = M_1 \left(\frac{n_A}{n_1} \right)^2$$

where

M_A = external moment for arbitrary rotational speed n_A

M_1 = external moment at n_1 , which corresponds to maximum continuous rating (MCR) of the engine

The phase difference for each external moment order is constant and does not depend on the engine RPM. The phase differences are typically given with respect to cylinder one top dead center (TDC).

Guide force moment amplitudes typically do not obey the quadratic rule. Also, the phase differences of guide force moments depend on the engine RPM.

External vertical moments may be applied to the engine structure using a force couple at the first and last engine bearings as denoted by red arrows in 5/11 FIGURE 10. The amplitude of these forces can be obtained as

$$F_E(f) = \frac{M_E(f)}{L_x}$$

where

$F_E(f)$ = frequency dependent force amplitude arising from the external vertical moment $M_E(f)$

L_x = horizontal distance between application nodes of the forces (distance between first and last engine bearing).

H-type guide force moments may be applied to the engine structure using two force couples at the longitudinal locations of the first and last engine bearings as denoted by blue arrows in 5/11 FIGURE 10. Each couple should consist of a force at the bearing level and an equal but opposite force at a well-supported engine node at the cylinder level in the centerline.

The amplitude of forces arising from the H-type guide force moments can be obtained as:

$$F_{GH}(f) = \frac{M_{GH}(f)}{2L_z}$$

where

$F_{GH}(f)$ = frequency dependent force amplitude arising from the guide force H-type moment $M_{GH}(f)$

L_z = vertical distance between application nodes of the forces (distance between bearing level and the cylinder).

Factor of two in the denominator means that the total moment has been divided into two force couples.

X-type guide force moments may be applied to the engine structure using force couples at the longitudinal locations of the first and last engine bearings as denoted by green arrows in 5/11 FIGURE 10. Each couple should have the same application points as in the case of H-type guide force moments, but the X couples should have the opposite directions at the two longitudinal locations.

The amplitude of forces arising from the X-type guide force moments can be obtained as:

$$F_{GX}(f) = \frac{M_{GX}(f)L_{CR}}{L_zL_x}$$

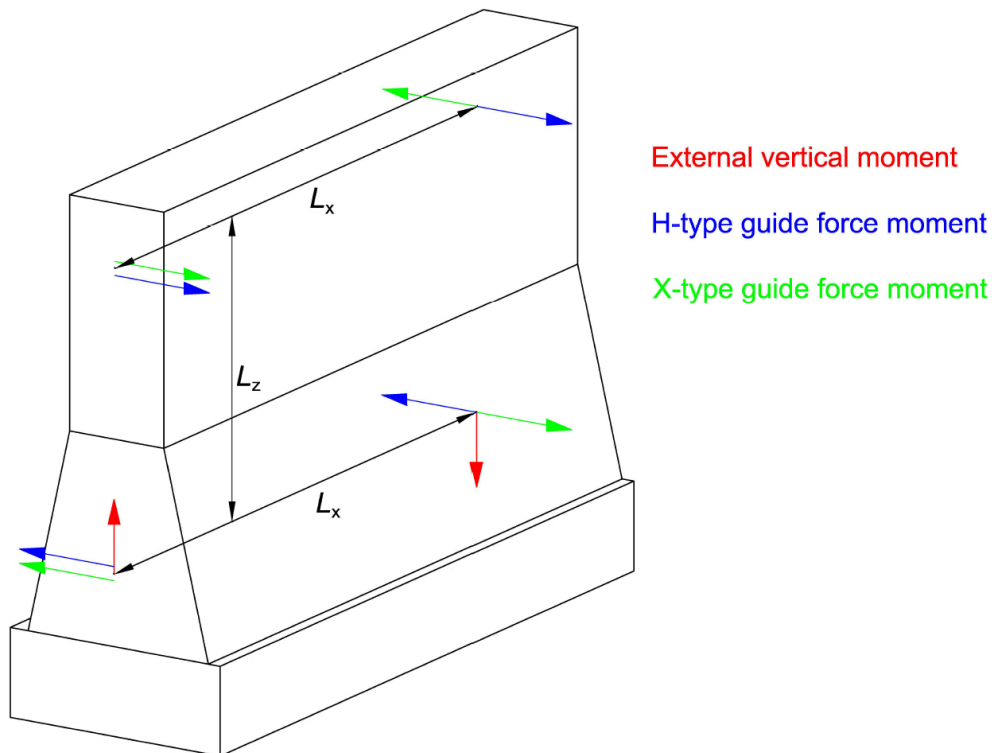
where

$F_{GX}(f)$ = frequency dependent force amplitude arising from the guide force X-type moment
 $M_{GX}(f)$

L_{CR} = length of the connecting rod as specified by the engine manufacturer

L_x and L_z are as previously defined. Connecting rod connects the crosshead pin and crankpin. Its length is specified by the engine manufacturer.

FIGURE 10
Application of Engine Moments (1 September 2023)



13 Forced Vibration

13.1 General (1 September 2023)

Frequency response analysis is used to compute structural response to steady-state oscillatory excitation. The three-dimensional FE model is subject to systems of harmonic loads which represent the oscillatory excitation forces induced by propeller and main engine. Two different numerical methods can be used in frequency response analysis. A direct method solves the coupled equations of motion in terms of forcing frequency. A modal method utilizes the mode shapes of the structure to reduce and uncouple the equations of motion. The solution for a particular forcing frequency is obtained through the summation of the individual modal responses. The choice of the method depends on the model size, number of the excitation frequencies and the frequency of excitation. If the modal method is used, all modes up to at least two to three times the highest forcing frequency should be retained.

A frequency response is obtained by applying the cyclic propeller and engine forces and moments with varying frequencies, related to the shaft RPM and the number of blades per propeller and the order of main

engine, to the model and solving the resulting dynamic problem given by the following equation of motion.

$$[M]\{\ddot{u}(t)\} + [C]\{\dot{u}(t)\} + [K]\{u(t)\} = \{F(t)\}$$

where

- $[M]$ = mass matrix
- $[C]$ = damping matrix
- $[K]$ = stiffness matrix
- $\{\ddot{u}\}$ = vector of accelerations
- $\{\dot{u}\}$ = vector of velocities
- $\{u\}$ = vector of displacements
- $\{F\}$ = vector of harmonic forces

The structural response to this harmonic load is a steady state response at the frequency of the load. (It is assumed that the harmonic load continues long enough so that the transient response damps out). Therefore:

$$F(t) = Fe^{i\omega t}$$

$$u(t) = ue^{i\omega t}$$

Upon substitution, the equations of motion reduce to

$$[K + i\omega C - \omega^2 M]\{u\} = \{F\}$$

It is noted that both $\{F\}$ and $\{u\}$ are complex quantities due to damping C and to the fact that the various components of propeller and engine induced vibratory forces are not in phase with one another.

13.3 Critical Areas (1 September 2023)

In order to demonstrate the frequency response in terms of peak velocities, **displacements, or accelerations** within a certain structure without losing generality, a fairly large quantity of grid points within the structural model may be selected and their responses may be obtained in the form of magnitudes and phase angles rather than complex components. Here is a list of the typical critical areas that should be taken into consideration in any vibration analysis.

- Transom at Centerline
- Transom at Port and Starboard Side
- Point right above the propeller
- Deck house:
 - Port and Starboard side at Bridge
 - Centerline at Bridge
 - Port and Starboard side at top of deck house (Wheel House)
 - Centerline at top of deck house (Wheel House)
 - Centerline at upper deck (Forward and Aft)
- Centerline at stern

- One point on each cargo Hold on the transverse Bulkhead.
- Engine room areas
- Machinery areas.
- Skeg area.

Depending on the ship type, there may be additional areas of investigation. Care should be taken to model the masses of the equipment. They should be located on well-supported nodes as much as possible. A computer program may be used to post process the results obtained from the forced vibration analysis.

13.5 Damping

The total damping associated with overall ship hull structure vibration is generally considered as a combination of the following components:

- Structural damping;
- Cargo damping;
- Water friction;
- Pressure wave generation;
- Surface wave generation.

For the forced vibration analysis, it is assumed that the effects due to structural damping, cargo damping, water friction and pressure wave generation can be lumped together. The effect of surface wave generation needs only to be considered for very low frequencies of vibration. This effect is generally neglected. For simplification, a constant damping coefficient of 1.5 percent of the critical damping may be used for the entire range of propeller RPM and main engine orders. Otherwise, more detail frequency-dependent damping coefficients may be used, if applicable.

13.7 Excitation Combination (1 September 2023)

Phase lag of any excitation is given with respect to a reference point, which is usually the top dead center (TDC) of the first main engine cylinder. This information is crucial but is often misrepresented in forced vibration analysis. If more than one excitation source acts on the ship structure, which is a realistic case, the phase lags between different excitation forces, moments, or pressures will impact the final response of the structure. Whenever a combined broadband time series of a response is needed (e.g., a stress time series needed for fatigue analysis consisting of a wide range of different response component frequencies), then the phase lag between different excitations should be considered.

At any given point in time, multiple excitation sources on the vessel give rise to multiple response components. Excitation sources can be at the same frequency or different frequencies. At a given main engine speed, measured in shaft rotations per minute (RPM), the main engine will generate complex excitations that can be decomposed into the base harmonic component oscillating at engine RPM (first order) and higher order harmonics oscillating at multiples of the RPM (e.g., 2 x RPM, 3 x RPM, 4 x RPM, etc.). The propeller will also generate complex excitations, but its base harmonic will oscillate at number of propeller blades x RPM. This is called the blade rate frequency. Higher order propeller excitation harmonics will be multiples of the blade rate frequency. For a propeller with four blades, the base harmonic will be of the fourth order, the second harmonic will be of the eighth order, and so on.

The vibration criteria sometimes allow the independent combination of responses at different orders (frequencies) and do not require knowledge of the phase lags between corresponding excitations at different orders. However, phase lags between excitations of the same order should always be considered as their phase angle will directly impact the response at that frequency. Depending on the phase angle, the responses from each same-order excitation may constructively interfere (add) at one location, while at another location they may destructively interfere (cancel). For example, propellers of twin screw vessels may be phased in such a way to achieve destructive interference and reduce the total response.

If the phase lags between different excitation sources of the same order are not known, then the vibration responses to each excitation may be calculated separately and assumed to be in phase at any point on the ship structure. When a combined broadband time series of a response is needed and the phase lags between excitation sources of the same or different orders are not known, then the vibration response time series to each excitation may be calculated independently and combined in the time domain assuming that the maximum values of each response occur simultaneously.

1 General

1.1 Scope

These Guidance Notes provide general guidelines on the measurement and evaluation procedures for the vibration of hull girder, superstructure, local structures, main propulsion machinery and shafting, and other machinery or equipment. This section covers overall procedures for acquiring, processing, and presenting data collected during sea trials based on ISO 4867 and SNAME T & R No. 2-29.

1.3 Application (1 September 2023)

The procedures provided are applicable to both turbine and diesel-driven vessels of all lengths. Actual items for measurements should be determined as appropriate based on ship specifications or specific measurement requirements.

Section 3 of the *ABS Guide for Vibration of Machinery Equipment and Related Structures* contains general requirements for the measurement and evaluation of vibration of machinery, equipment, and their associated local structures when an optional classification notation **VIB-M** is requested.

1.5 Terminology (1 September 2023)

- *Average Amplitude Value*: The value obtained from a Fast Fourier Transform (FFT) signal analyzer, set in the averaging mode, equal to $\sqrt{2}$ times the root mean square (RMS) value. This is equivalent to the average sinusoidal value obtained during steady speed trial conditions, and for design purposes, it is comparable to predicted response.
- *Free Route*: The condition achieved when the vessel is proceeding at a constant speed and course with minimum **helm adjustment (± 2 degrees or less) and no throttle adjustment**.
- *Hull Girder*: The primary hull structure such as the shell plating and continuous strength decks contributing to flexural rigidity of the hull and the static and dynamic behavior of which can be described by a free-free non-uniform beam approximation.
- *Hull Girder Vibration*: That component of vibration which exists at any particular transverse cross section of the hull so that there is little or no relative motion between elements intersected by the plane of the cross section.
- *Local Vibration*: The dynamic response of a structural element, deck, bulkhead or piece of equipment which is significantly greater than that of the hull girder at that location.
- *Maximum Repetitive Amplitude (MRA)*: This is the largest repeating value of the modulated signal obtained on sea trial. It can be observed in the envelope of peak values on a time compressed strip chart of the modulated signal. This is the acceptance value specified in the codes.

- *Conversion Factor (C_F)*: The ratio of the MRA to the average amplitude obtained during steady speed trial condition.
- *“Peak Hold”*: Most FFT analyzers include a feature called “Peak Hold” which generates a spectrum of values which are the maximum spectral values in each frequency component for the set of all individual spectra comprising the “Peak Hold” spectrum. This is not the same as the MRA. In practice, it represents an average amplitude of a number of cycles of the evaluated signal rather than the maximum value of the single largest amplitude observed. The number of cycles averaged is based on the “window” or viewing time in seconds and, of course, increases with frequency.
- *RMS Value*: The root mean square value is the square root of the **the arithmetic mean** of the squares of all points of a signal. For a sinusoid, it is $1/\sqrt{2}$ times the amplitude. **It is an indication of an equivalent steady state energy contained in an oscillating signal and can be calculated from a time series of a measured signal or from a frequency spectrum obtained using the Fast Fourier Transform (FFT). Refer to ISO 20283-4 for methodologies of RMS calculations in frequency domain or in time domain.**
- *Severity of Vibration*: The peak value of vibration (velocity, acceleration, or displacement) during periods of steady-state vibration.
- *Windowing*: A technique used to minimize the spectral leakage during FFT of random signals. This is accomplished by multiplying the time series of the measured signal with a window function. The window is a symmetric function in time and usually begins and ends with a zero value and has a value of one in the middle.

3 Instrumentation

3.1 General Requirements

Measurements are preferably to be made with an electronic system producing a permanent record. The transducers may generate signals proportional to acceleration, velocity or displacement. Recording can be made either on magnetic tape, paper oscillographs, or in digital format. Use of paper oscillographs during the tests means that the vibration traces can be inspected directly and is very helpful in evaluating existing vibration problems. When displacement rather than either velocity or acceleration is recorded, the desired low-frequency signals associated with significant vibratory motion are the major components of a recorded trace. Thus, they are readily evaluated since they overshadow possible higher frequency signals with low displacement amplitudes. Provisions should be made for suitable attenuation control to enable the system to accommodate a wide range of amplitudes.

An event marker should be provided on the propeller shaft. Its position with respect to top dead center of cylinder number 1 and a propeller blade should be noted.

3.3 Calibration (1 September 2023)

Calibration procedures are categorized as either system calibration or electrical calibration. In general, system calibration refers to a procedure performed before installing instrumentation on board of the ship or as the transducers are installed. It should be a complete reckoning of the sensitivity of the transducers, signal conditioning and recording equipment.

Electrical calibration refers to a procedure that can be accomplished usually at the recording center, is considered a “spot check,” takes only a few minutes, and can be performed periodically during the vibration trials. Calibration procedures are different for different types of gauges and are discussed in this section.

Instrument calibration should be performed every other year or following the calibration interval specified by the manufacturer, whichever interval is shorter.

3.3.1 Accelerometers

All accelerometers can be calibrated over the frequency range of interest by mounting on a shaker table or calibration device that oscillates at known amplitudes. Normally, this is the type of system calibration that is used. Strain gauge and piezoresistive accelerometers can be calibrated for zero and ± 1 g by laying them on their sides, their bases, and upside down, respectively. This provides a direct current (D.C.) calibration only and is useful only if the conditioning and recording equipment operates at a frequency of zero Hz.

Once the transducers are installed, the “electrical calibration” is usually accomplished by an internal (to the amplifier) signal of known value being applied to the conditioning and recording equipment. In the case of strain gauge and piezoresistive accelerometers, a shunt resistor can be applied across one arm of the bridge and the value of the resistor can be equated to a certain acceleration. The latter results in a D.C. step being recorded.

3.3.2 Velocity Gauges

System calibration for velocity gauges should be performed on a shaker table that oscillates at known amplitudes and frequencies. There are no D.C. types of calibration suitable for these gauges. Electrical calibration is done by means of an internal signal of known value being fed into the conditioning and recording equipment. Since the conditioning equipment usually does not operate for D.C. signals, the known signal is usually a sinusoid.

3.3.3 Proximity Probes

Proximity probes should not be sensitive to changes in frequency and therefore should be able to be calibrated for D.C. steps only. They should be calibrated at several distances from the target by means of “feeler” gauges. Plastic feeler gauges are available that do not affect the signal and can be left in place while recording. Preliminary calibration can be done before installing the probes on board ship, but the final calibration should be done with the probes in place because each target has a slightly different effect on the gauge. The only practical type of “electrical” calibration would be the substitution of a signal of known value.

3.3.4 Strain Gauges

The type of calibration needed for strain gauges depends on how they are used. Again, the signal output should not be sensitive to frequency and D.C. calibrations are adequate. If possible, the strain-gauged object should be subjected to known loads and the resulting strains calculated and related to the signal output. If the object has a complex shape or if known loads are difficult to apply, the only choice is to accept the manufacturers’ listed Gauge Factors and use a shunt resistor for calibration.

In most applications on board ship, the signal leads from the strain gauges are fairly long. This reduces the sensitivity of the gauges, requiring the shunt to be applied at the gauge rather than at the amplifier to obtain accurate results.

5 Measurement Conditions

Vibration tests should be conducted under the conditions that are agreed by the shipyard and owner. The following measurement conditions are provided as a reference.

5.1 Environment Condition

5.1.1 Sea State (1 September 2023)

Refer to 3/3.1.1 of the *ABS Guide for Vibration of Machinery Equipment and Related Structures*.

5.1.2 Water Depth (1 September 2023)

Refer to 3/3.1.2 of the *ABS Guide for Vibration of Machinery Equipment and Related Structures*.

5.3 Loading Condition (1 September 2023)

Refer to 3/3.2 of the *ABS Guide for Vibration of Machinery Equipment and Related Structures*.

5.5 Course

Tests should be conducted for the following courses:

5.5.1 Free-route Runs (1 September 2023)

Refer to 3/3.3 of the *ABS Guide for Vibration of Machinery Equipment and Related Structures*.

5.5.2 Maneuvers (1 September 2023)

In addition to free-route runs, for naval combatants or small craft, tests may also be conducted while completing the following maneuvers:

- Hard turn port
- Hard turn starboard
- Crash astern

5.7 Speed and Engine Power (1 September 2023)

For free-route runs, refer to 3/3.4 of the *ABS Guide for Vibration of Machinery Equipment and Related Structures*. If there are multiple shafts or propellers, all should be run as close as possible to the same speeds (generally within two percent). In certain instances, it may also be appropriate to run with a single shaft or propeller.

Tests for hard turns to port and starboard should be conducted at maximum speed. The **crash astern** test should be performed starting from the maximum speed and decreasing the propeller RPM to 10% of the maximum RPM in a minute or less.

7 Measurement Locations (1 September 2023)

Vibration measurement locations for local structures, machinery, and equipment for a vessel to obtain the optional classification notation **VIB-M** are given in the *ABS Guide for Vibration of Machinery Equipment and Related Structures*.

Other vibration measurement locations that may also be considered are given in this section.

7.1 Stern

Vertical, transverse, and longitudinal measurements of the hull girder should be taken at the centerline and as close as possible to the stern by means of a transducer. These measurements are for reference purposes. A pair of transducers for vertical vibration are also to be placed at the deck edge (one port, one starboard). They should be located on the same deck and longitudinal location as the aforementioned transducer. The vibration of the hull girder should be measured, excluding local effects.

7.3 Superstructure

Vertical, transverse, and longitudinal vibrations should be measured at the following locations:

- Wheel house (navigation bridge), centerline at front of bridge
- Bridge wing aft end
- Main deck, centerline at front of deck house
- A pair of transducers to measure torsional motions of an aft deck house, when torsional vibration should be determined

Measurements should be made in the range of 90 to 100 % normal shaft rotational frequency.

7.5 Main Engine and Thrust Bearing

7.5.1 Geared Drive Turbines

Vertical, transverse, and longitudinal measurements should be taken on top of the thrust bearing housing. Recordings are also to be taken on one additional point on the thrust block foundation in the longitudinal direction. Measurements at other locations may be executed as optional. Other measurements to achieve the same results are permissible.

7.5.2 Direct Drive Diesels

Recordings should be taken at the following locations:

- Top and foundation of the thrust bearing; if applicable;
- Top forward end of the main engine, in the longitudinal direction;
- Top forward and aft ends of the main engine, in the vertical and transverse directions;
- Forward end of the crankshaft, in the longitudinal direction;
- Forward and aft ends of the engine foundations, in the vertical and transverse directions.

For the other measuring points, optional recordings may be taken at constant shaft rotational frequency. Measurements should be made throughout the normal operational range of the ship.

7.7 Lateral Shaft Vibration

If measurement is conducted, vertical and transverse vibration measurements should be made on the shaft relative to the stern tube. Additional measurement points are optional and may be taken. These measurements should be made throughout the normal operational range of the ship.

In order to eliminate possible error, shaft run-out should be checked by rotating the shaft by the turning gear and recording the first-order signal. This signal should be phased and the shaft vibration measurement corrected accordingly.

7.9 Torsional Shaft Vibration

If measurement is conducted as per 4-3-2/7.5.8 of the *Marine Vessel Rules*, torsional vibration measurements should be taken either at the free end of the propulsion machinery, using a suitable torsional vibration transducer, and/or on the main shafting, using strain gauges. Alternatively, depending on the system characteristics, a mechanical torsigraph, driven from a suitable position along the shafting or free end, may be used for this purpose.

7.11 Local Structures

If severe local vibration exists during sea trials, vertical, transverse, and longitudinal measurements should be taken at the suspect location. This is necessary to determine the need for corrective measures.

7.13 Local Deck Transverse

Vertical and transverse bending vibration measurements should be taken at the deck edge using as many points as necessary to determine the mode shapes at low frequencies while avoiding local resonances. These types of measurements should be made by use of a reference transducer at the stern along with a portable transducer. Torsional modes may require phased deck-edge measurements.

7.15 Local Machinery and Equipment

Vertical, transverse and longitudinal measurements should be taken at the outside of machinery where there is evidence of large vibration amplitudes. Types of machinery and equipment include diesel generator, electric motors, air compressors, ballast pumps, etc. Vibration measurement should be taken at the bearing position.

7.17 Shell Near Propeller

If necessary, the measurements of hull surface pressure should be taken to confirm design estimates, to obtain design data or to investigate potential cavitation problems. The pressure transducers should be located as follows:

- Two transducers in the propeller plane;
- One transducer at 0.1D forward of the propeller plane

To minimize the effect of plate vibration, all transducers in the hull plating should be located as close as possible to adjacent frames or partial bulkheads.

9 Data Processing Analysis

9.1 Measured Data

9.1.1 Displacement, Velocity, Acceleration

The type of data that should be recorded (displacement, velocity or acceleration) may depend on the appropriate frequency range. For example:

<i>For Frequencies</i>	<i>Measure</i>
Below 300 RPM	displacement
Between 300 RPM and 6,000 RPM	velocity
Above 6,000 RPM	acceleration

If a particular data type is critical in determining damage to machinery or structure or for human response, the appropriate data type may be used in-lieu of that above. For example:

<i>For</i>	<i>Measure</i>
Mechanical or structural fatigue	displacement
Human response	velocity
High frequency machinery components	acceleration

9.1.2 Frequency Range

The frequency ranges of transducers, signal conditioners and recording equipment should be chosen to match the frequency components of interest in the data being recorded. Frequencies that are known to be present but not of interest should be excluded.

Unwanted frequencies should not be recorded, since this makes it difficult to separate the level of the frequency components of interest from the "noise". Also, a frequency analyzer with a broader range than necessary should not be used. This decreases the resolution and accuracy of the results.

TABLE 1
Typical Frequencies Ranges

<i>Item</i>	<i>Frequency Range</i>
Ship motion	Below 1 Hz
Major structure (deckhouses, masts, etc.)	Between 1 Hz to 10 Hz for the lower modes
Propeller shaft rotation, gear frequencies, and blade rate	To be determined from the machinery characteristics

9.1.3 Time History or Frequency Spectra

In cases where the vibration is self-excited, such as with rotating machinery, obtaining the frequency spectra directly is appropriate. In this case, amplitudes are fairly constant, and a large sample is not required. However, most vibration data gathered from ships are excited by the sea, propeller or some combination (e.g., hull girder vibration). For this type of data, the amplitudes are modulated and have a randomness associated with them and may require a sample large enough to account for the randomness, which for blade frequencies and its harmonics should be three minutes.

9.3 Performance of Measurements (1 September 2023)

- For the free-route run, data should be recorded for at least three minutes after steadying at each ship speed, or for at steadily increasing shaft speed less than 1 RPM per 5 minutes.
- For the hard turns to port and starboard, recording is to commence before the turn and continue until maximum vibration has passed (when settled into the turn).
- For the **crash astern** test, data should be recorded before the propeller RPM is reduced and continue until the ship comes to rest.

9.5 Analysis Methods

The methods of analyzing shipboard vibration data are presented in SNAME T & R No. 2-29. The manual method should be used by an engineer during sea trial to complete a quick assessment of test results and to identify potential problems. For the final test report, the use of the fast digital electronic (FFT) analyzers should be employed. For more details, refer to SNAME T & R No. 2-29.

9.5.1 Manual

The manual method of analysis involves measuring frequencies and amplitudes of vibration components from an oscillograph record. The analysis of periodic waveforms is based on the principle that any periodic record is a superposition of sinusoids having frequencies that are integer multiples of the lowest frequency present. The lowest frequency is determined by the smallest portion of the record that repeats itself, or one cycle.

Ship vibration often involves more than one component, so that decomposing the signal into its separate components involves discerning beating and modulating and applying the appropriate analysis techniques. Thus, having magnetic tape of the actual recorded signal available for detailed analysis, as applicable, is highly recommended.

9.5.2 Envelope

This method of analysis consists of filtering the signals to obtain the MRA of a narrow frequency band of interest and recording the result on a chart recorder at slow play out speed. This condenses a several minute record into an envelope just several inches long, from which the MRA can be visually noted. Care should be taken that the filter used does not pass a significant amount of any component other than that of the component of interest. For this reason it is important to use precision filters.

This method is the most direct approach to the determination of the MRA of individual components. If the MRA of the multi-frequency amplitude is required, the desired response is obtainable by simply recording the real-time record at a slow speed and measuring the overall amplitude and multiplying by the sensitivity of the measuring/recording system. However, for obtaining the amplitudes of individual components for use in evaluating design analyses and/or developing improved crest values to be used in conjunction with spectral analyses, it is necessary to include filtering systems.

9.5.3 Spectral (1 September 2023)

Two common types of spectra that can be found with most analyzers are the “average” and the “peak” amplitude. For both types, the record is broken down into segments for frequency analysis. The segments vary in length with the frequency range of the analysis, but for ship vibration studies they would be several seconds long. For each segment, the analyzer finds the RMS level of each frequency component (i.e., in each frequency interval, or for each “line”). The number of lines (resolution) varies with the analyzer, but most often falls between 100 and 1000. **For the frequency range between 1 Hz and 200 Hz, a resolution of at least 400 spectral lines and a Hanning window should be used. For the frequency range up to 1000 Hz, a minimum of 1600 spectral lines resolution with flat top window should be used.** Ship vibration records will normally be one or several minutes long and will contain many segments. If the “average” spectrum is chosen, the analyzer will average all the RMS levels found in like frequency intervals and produce the “average” spectra as “average RMS” values. This value will correspond to the RMS value shown in the ISO-6954 formula:

$$(MRA) = (C_F\sqrt{2}) \times \text{RMS value}$$

where C_F denotes “conversion factor” and $C_F\sqrt{2}$ denotes “crest factor”. The average value, or average amplitude is equal to $\sqrt{2}$ times the “average RMS” value read from the “average” spectrum. This is equal to the average sinusoidal value obtained during steady speed trial conditions and, for design purposes, is comparable to predicted response.

The term “peak” is frequently misinterpreted when used in conjunction with the application of the spectral analyzer in the evaluation of shipboard hull vibration. To clarify, the peak value of any vibration cycle is the difference between the mean and maximum values of that cycle. However, in the case of a modulating signal it is noted that the “peak” value increases to a “maximum” value, every few cycles. Thus, the “peak” of the modulating signal represents the maximum repetitive value or MRA and is actually equivalent to the peak value of each segment and corresponds to “peak hold” for that segment.

When choosing the “peak” spectrum, the largest RMS values are obtained in each frequency interval found among several segments. These values are called “peak RMS” values with some variation in results, depending on the frequency interval used. A larger frequency interval will yield higher results. The **peak** spectral value will always be less than the MRA. The relationship between the latter two will depend on the rate of modulation compared to the length of the segment. If the modulation is very slow, the amplitude will be near its maximum for the entire length of some segments and the two will be close. If the modulation is fast, the peak will be closer to the average amplitude.

The amplitude variation at blade-frequency between “average spectral”, “peak spectral” and MRA (derived by the envelope method) is shown in 6/9.5.5 TABLE 2 for example. The data represents the average of three full-power runs measured at 114 RPM, and shows alternating pressure measured over the propeller in psi, and alternating displacement in mills at both the stern of the hull in the vertical direction and at the hub of the bull gear in the longitudinal direction. This data indicates that the conversion factor (C_F) of 1.8 and the crest factor of 2.5 are reasonably satisfactory for these trials, particularly regarding the displacement amplitudes obtained on the hull and bull gear measurements. It also indicates that the hull and machinery vibration measurements, on which the acceptance criteria are based, supports the position that the peak spectral measurements will be significantly lower than the “true” MRA obtained by the envelope method of analysis.

9.5.4 Histograms of Instantaneous Values

Some analyzers sample the signal and obtain histograms of the instantaneous values. This capability might be useful with ship vibration records, but there are several considerations. First, the primary interest is the amplitudes of blade-frequency and its harmonics, so that filtering is

necessary as it is in the envelope method. The limitations and necessary steps in the use of the filtering of the higher harmonics also apply to the statistical method.

Second, we are concerned with the MRA of the major frequency components. The histograms are usually obtained by sampling all the points on the record, not just the peaks. The amplitude, which is exceeded by only one or two percent of the samples, would probably involve only the tips of the largest cycles and may be comparable to the MRA. The amplitude, which is exceeded by some percentage of the samples, could be determined from the cumulative probability plot.

9.5.5 Histograms of Peak Values

This procedure would be closer to the envelope method and can be obtained with an analog to digital (A/D) converter and a microcomputer in conjunction with filters. It entails sampling the filtered signal, as described for the histograms of instantaneous values, then finding the peak (and trough) of each cycle. The peak amplitudes are then put into a histogram or cumulative probability plot. The top three, five, or ten percent of the peak amplitudes may be comparable to the MRA. Although it is not likely that this procedure is currently being used for routine analysis of ship vibration data, it is the most accurate of those discussed. However, considering the uncertainties in the measurement and analysis of ship vibration, this degree of complexity is not warranted.

TABLE 2
Examples of Alternate Vibration Measurements (1 September 2023)

	<i>Avg Spect (RMS)</i>	<i>Peak Spect (RMS)</i>	<i>MRA (Envelop)</i>	<i>Crest Fctr $C_F\sqrt{2}$</i>	<i>Conv Fctr (C_F)</i>
Hull Pressure, psi	1.3	1.65	2.65	2.04	1.45
Stem Vertical Disp., mils	1.7	3.29	5.80	3.41	2.42
Bull Gear Hub, Long Disp, mils	1.87	3.22	4.79	2.56	1.81
Average					1.89

11 Measurement Report

11.1 Analysis and Reporting of Data

11.1.1 Analysis

The contents of analysis should be selected from the following depending on the necessity:

- Severity of vibration at the propeller shaft rotational frequency for hull girder transducers
- Severity of vibration at blade rate frequencies for hull girder and machinery transducers
- Severity of vibration of each detectable harmonic of shaft rotational frequency or blade rate for hull girder and machinery transducers, as applicable. Also the severity of each detectable multiple of crankshaft rotational frequency in the case of geared diesel installations
- Severity of local structural vibration at all measurement locations
- Mode shape of local vibrations. Use hull girder vibration as reference for the mode shape.
- Severity of vibrations of local machinery or equipment at all measurement locations
- Phase relation between various transducers at blade rate, as applicable, using a suitable reference datum (e.g., a hull girder or machinery transducer or event marker)
- For diesel engines, phase relation should be provided between all transducers measuring in the longitudinal direction and for the transducers on top of the engine measuring torsional motions; therefore, each group should be measured simultaneously

- Severity of vibration at hull girder and machinery resonances

Note:

If beating exists, it should be noted by recording the maximum and minimum values of the amplitude and the frequency of the beat.

11.1.2 Reporting of Data (1 September 2023)

The contents of reporting should be selected from the following depending on the objectives of the report:

- Particulars of test ship
- Particulars of propulsion-shaft system
- Particulars of main diesel engine or turbine driven plants
- A sketch of inboard profile of hull and superstructure;
- A lines plan
- A midship section
- A sketch of propeller aperture
- A sketch showing locations of hull girder and machinery transducers and their directions of measurement. Transducer locations for local vibration measurements should be shown on a separate sketch where the precise location of the transducer is noted.
- Trial conditions during vibration measurements including sea state, wave direction, water depth, draft at FP and AP, propeller immersion
- Plots of displacement, velocity or acceleration amplitudes versus speed for shaft rotation frequency, blade rate, or machinery excitation frequency or any harmonic thereof.
- Profiles of local deck vibration at each resonance from port to starboard and from the nearest aft to the nearest forward structural bulkhead
- Tables of all significant vibration severities and their location and frequency. Include the shaft rotational frequency for machinery-excited vibration
- Hull girder natural frequencies identified from stern measurements and any unusual vibration condition encountered
- Results of vibration measurements at local areas
- Results of vibration measurement during maneuvers
- Longitudinal vibration of the propulsion system during maneuvers
- Method of analysis of results
- Type of instrument used

The report is to note the hull natural frequencies and modes which have been identified. Also, the report is to address any undesirable or unusual vibration condition encountered.

The requirements for vibration test plans for local structures, machinery, and equipment that are to be submitted to ABS for review for obtaining the **VIB-M** optional notation are given in Subsection 3/6 of the *ABS Guide for Vibration of Machinery Equipment and Related Structures*.

1 General

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

- i)* Vibration limits for crew and passengers
- ii)* Vibration limits for local structures
- iii)* Vibration limits for machinery

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

3 Vibration Limits for Crew and Passengers

3.1 ABS Criteria for Crew Habitability and Passenger Comfort (1 September 2023)

ABS provides optional classification notations for crew habitability and passenger comfort. The following ABS Guides provide acceptance criteria based on the evaluation of both sea-motion induced and mechanically-induced vibrations for the purpose of granting ABS optional notations:

- *ABS Guide for Passenger Comfort on Ships*
- *ABS Guide for Comfort on Yachts*
- *ABS Guide for Crew Habitability on Ships*
- *ABS Guide for Crew Habitability on Offshore Installations*
- *ABS Guide for Crew Habitability on Workboats*
- *ABS Guide for Crew Habitability on Mobile Offshore Drilling Units (MODUs)*
- *ABS Guide for Habitability of Industrial Personnel on Accommodation Vessels*

3.3 Other Relevant Industry Standards for Crew Habitability and Passenger Comfort (1 September 2023)

The following standards are also widely used in the industry as acceptance criteria for crew habitability and passenger comfort:

- ISO 6954 (1984)
- ISO 6954 (2000)

- ISO 20283-5 (2016)
- ISO 21984 (2018)

5 Vibration Limits for Local Structures (1 September 2023)

Excessive ship vibration should 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:

- Machinery foundation structures,
- Structures in compartments of machinery and equipment that may be affected by the operation of the associated machinery and equipment, and
- Local structures excited by other vibration sources such as springing and whipping .

The vibration acceptance criteria for local structures are provided in Subsection 2/2 of the *ABS Guide for Vibration of Machinery Equipment and Related Structures*.

7 Vibration Limits for Machinery

7.1 Main Propulsion Machinery (1 September 2023)

The vibration of main propulsion machinery tends to be severe because of the excitation from the propellers. The area of primary concern should be the main propulsion machinery, principally longitudinal vibration at propeller blade rate frequency (see Section 4). The vibration **acceptance** criteria for the main propulsion machinery are provided in 2/3.1 of the *ABS Guide for Vibration of Machinery Equipment and Related Structures*. Alternatively, for main propulsion machinery items not specified in the *ABS Guide for Vibration of Machinery Equipment and Related Structures*, the acceptance criteria in the following paragraph may be used. All vibration limits are provided in terms of the broadband RMS velocity values with multi-frequency components.

The longitudinal vibration for free-route at thrust bearing and bull gear hub for geared turbine drives should be less than 5 mm/s RMS. For other propulsion machinery components exclusive of engines, propellers, and shafting aft of the thrust bearing, the longitudinal vibration should be less than 13 mm/s RMS. For stern tube, the lateral vibration should be less than 7 mm/s RMS. For direct diesel engines over 1000 HP (slow and medium speed diesels connected to the shafting), the vibration limits are 13 mm/s at the bearings, in all three directions. 7/7.1 TABLE 1 presents these vibration limits.

TABLE 1
Vibration Limits for Main Propulsion Machinery (1 September 2023)

<i>Propulsion Machinery</i>	<i>Direction</i>	<i>Limits (Broadband RMS)</i>
Thrust Bearing and Bull Gear Hub	Longitudinal	5 mm/s
Other Propulsion Machinery Components	Longitudinal	13 mm/s
Stern Tube	Lateral	7 mm/s
Diesel Engine at Bearing (direct, >1000 HP)	All directions	13 mm/s

A primary concern in the longitudinal propulsion machinery vibration is the avoidance of thrust reversal at the thrust bearing due to system longitudinal resonance, which can result in destruction of the thrust bearing. ANSI S2.27 provides the criteria on the alternating thrust at thrust bearing. During free-route runs, the broadband peak value of the alternating thrust on the main thrust bearing should be less than 75% of the mean thrust for that speed, or less than 25% of the full power mean thrust, whichever is less.

As shown in 4/3, it may not be possible to avoid a longitudinal resonance in the operating range with ships having long shafts. A critical speed of at least 20% below the full power RPM is the usual practice. Even if the critical speed occurs at an RPM less than 20% of full power RPM, it may be very difficult to achieve the criteria proposed by ANSI S2.27 at the resonant point, depending on the blade-rate alternating thrust amplitude and damping. In this case, an RPM band around the critical **speed** might have to be restricted for operation as the only reasonable recourse.

7.3 Machinery and Equipment (1 September 2023)

The machinery **and equipment** include but are not limited to generators, turbochargers, **gears, turbines, motors, centrifugal pumps, compressors, boilers, blowers, fans, pipes, and electrical equipment, as applicable**. The application of vibration limits may vary depending on specific type, size, configuration and mounting of the machinery.

The machinery and equipment vibration acceptance criteria are provided in 2/3.2 of the *ABS Guide for Vibration of Machinery Equipment and Related Structures*. Alternatively, the vibration criteria for machinery and equipment may be provided by the manufacturers.

ISO 10816 provides guidelines on the vibration criteria in terms of the broadband 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 is complemented by ISO 7919, which provides guidelines on the vibration criteria for the rotating parts of the machines.

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American Bureau of Shipping, *Guide for Crew Habitability on Workboats*

American Bureau of Shipping, *Guide for Habitability of Industrial Personnel on Accommodation Vessels*

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Siemens (2000a), *Simcenter Nastran 2020.1 Quick Reference Guide*

Siemens (2000b), *Simcenter Nastran Advanced Dynamic Analysis User's Guide*

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ANSI S2.27 (2002), *Guidelines for the measurement and evaluation of vibration of ship propulsion machinery*, American National Standard

ISO 20283-4 (2014), *Mechanical vibration - Measurement of vibration on ships - Part 4: Measurement and evaluation of vibration of the ship propulsion machinery*

ISO 20283-5 (2016), *Mechanical vibration - Measurement of vibration on ships - Part 5: Guidelines for measurement, evaluation and reporting of vibration with regard to habitability on passenger and merchant ships*

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ISO 2631-2 (1997), *Evaluation of human exposure to whole-body vibration -Part 2: Continuous and shock-induced vibration in buildings (1 to 80 Hz)*

ISO 4867 (1984), *Code for the measurement and reporting of shipboard vibration data*

ISO 4868 (1984), *Code for the measurement and reporting of local vibration data of ship structures and equipment*

ISO 6954 (1984), *Mechanical vibration and shock- Guidelines for the overall evaluation of vibration in merchant ships*

ISO 6954 (2000), *Mechanical vibration - Guidelines for the measurement, reporting and evaluation of vibration with regard to habitability on passenger and merchant ships.*

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Note: This Appendix is adapted from PNA Chapter 7 (SNAME, 1988) with permission from SNAME.

1 Corrective Investigations (1 September 2023)

The approach to resolving ship vibration problems involves two steps:

- i)* Clearly establish the cause of the problem.
- ii)* Implement the changes required to eliminate the problem in an efficient manner.

In about 80% of cases, the main cause of ship vibration is the propeller. Whether or not the vibration of a particular ship has its source in the propeller is easily established from underway vibration measurements. If at some shaft RPM the measured frequency of the vibration is predominantly RPM times propeller blade number, and varies directly with shaft RPM, then the propeller is definitely the exciting source. If blade-rate frequency, or its multiples, is not strongly detectable in the records, then it is likely that the propeller is not the primary excitation, unless the records exhibit a strong shaft-rate frequency, which could indicate propeller mechanical or hydrodynamic unbalance difficulties, but these are rather rare.

Once the excitation frequency has been established from the underway measurements, next in order is to establish whether resonance with structural natural frequencies plays a significant role in the magnitude of the vibration. For non-cavitating propellers, excessive hull vibration is expected to be resonant vibration. Resonant vibration is established by varying shaft RPM in steps and recording vibration amplitude successively at each RPM in the region where the problem has been identified as being most intense. If a plot of displacement amplitude versus RPM shows a definite peak with increasing RPM followed by decline, then resonant vibration is established and the position of the peak establishes the natural frequency of the resonant structural mode. If the amplitude/RPM characteristic does not peak but has an increasing trend as roughly RPM squared in the upper power range, then structural resonance is not playing a major role. If, alternatively, the amplitude/RPM characteristic increases very rapidly only in the immediate vicinity of full power without establishing a definite peak up to the maximum obtainable RPM, a full power resonance may or may not be indicated. This exhibition can be entirely the manifestation of the onset of propeller intermittent sheet cavitation, which tends to produce an almost discontinuous amplification of the hull surface excitation at the onset RPM. The sudden appearance of strong higher harmonics of blade-rate frequency in the vibration records, accompanied by violent pounding in spaces above the counter, are good indications of a full-power non-resonant vibration problem caused by excessive propeller cavitation.

If non-resonant vibration due to propeller cavitation is established, attention can then be focused on hydrodynamic design changes in the stern/propeller configuration. This course of action is considered in the next Subsection.

If the problem is established as highly localized resonant vibration of plating panels, piping, and the like, then the vibration survey likewise needs to go no further. It is usually very apparent in such cases how natural frequency changes, through local stiffening, can be effectively and expediently accomplished to eliminate the locally resonant conditions.

If the vibration problem is established as a resonant condition of a major substructure, such as a deckhouse, which has been all too often the case, then the vibration survey should proceed to obtain mode shape information in the interest of an expeditious correction program.

3 General Approach

Just as in developing a vibration-sufficient ship design, there exist three possibilities for correcting a vibration-deficient one in normal practice:

- i) Reduce vibratory excitation
- ii) Change natural frequencies to avoid resonance
- iii) Change exciting frequencies to avoid resonance.

Most excessive diesel engine excited hull vibration can usually be corrected by the following provisions:

- i) Mechanical or electrical moment compensators against engine external moments, or
- ii) Engine lateral stays of weld, hydraulic or friction type against engine lateral vibration, or
- iii) Axial damper against crank shaft vibration, or
- iv) Dynamic absorber against fore-and-aft vibration of superstructure.

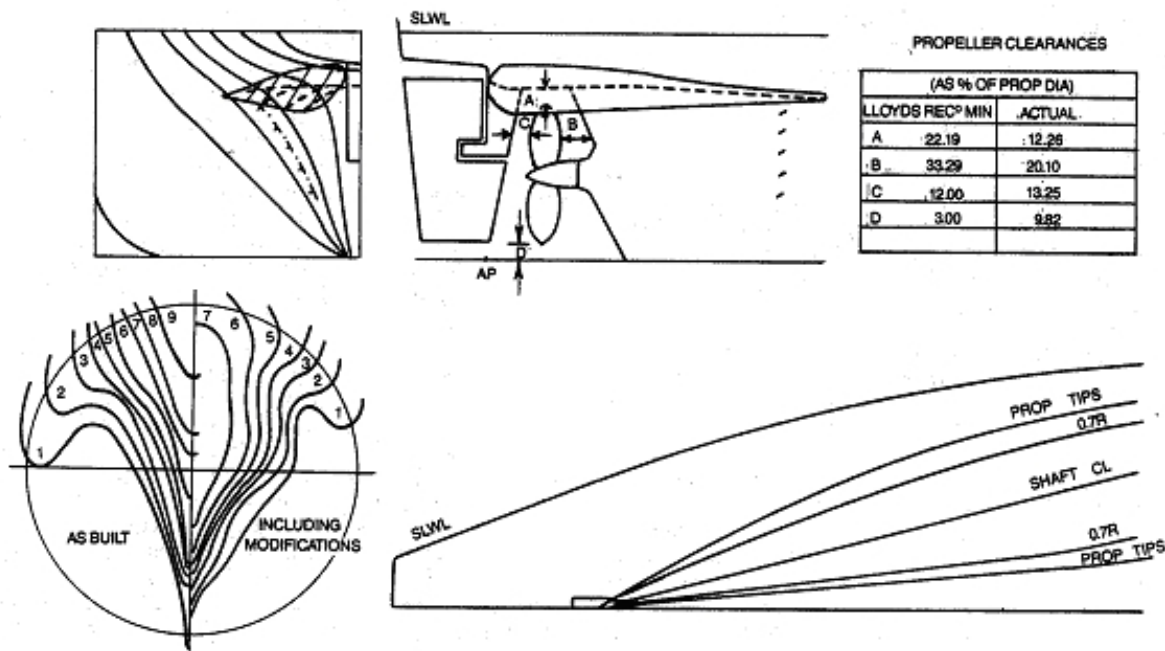
The detail information necessary for the installations of above provisions should be provided by engine or equipment manufacturers. Otherwise the achievement of any of the three correction possibilities identified above may involve modifications in either stern/propeller hydrodynamics or hull structure.

5 Hydrodynamic Modifications (1 September 2023)

The most effective way to reduce propeller vibratory excitation is to reduce the circumferential non-uniformity of the hull wake in which the propeller operates, as discussed in detail in Section 3. In the early design stage, acceptable wakes can be achieved by taking proper care with stern lines. (Refer to Subsection 3/5 for the specific guidelines). In a post-design corrective situation, basic lines changes are, of course, not possible. However, with good luck in the case of poor stern lines, considerable improvements in wake might be accomplished by back fitting one of the several types of wake adapting stern appendages. The partial tunnel A2/5 FIGURE 1, below, has been the most broadly applied of the wake adapting appendages, which also include vortex generators and wake adapting propeller ducts.

Diverting the upward-aft flow along the buttock lines forward longitudinally into the upper propeller disc may reduce the wake spike near top-dead-center. This device will work most effectively on the buttock-flow type of stern; the partial tunnel was applied successfully over the years on Great Lakes ore carriers, most of which have barge-type sterns with very steep buttock angles. For sterns which exhibit a waterline-flow character, the partial tunnel would be expected to be **less effective** due to the lack of upward flow to divert. However, the effectiveness of the partial tunnel cannot always be accurately judged by simply classifying a prospective application as one of the two limiting cases of buttock versus waterline flow. For example, the stern shown on A2/5 FIGURE 1, from Rutherford (1978-79) might be classified as more of a waterline flow, yet the modifications shown produced significant improvement in the nominal wake, as exhibited by the before and after axial velocity contours. The contour lines on A2/5 FIGURE 1 are the lines of constant wake fraction, so that 9 is 0.9, or 10% of the axial free stream velocity, which represents an unfavorable wake. The A2/5 FIGURE 1 modifications do however include vortex generators as well as the partial tunnel, and the contributions of each to the wake improvements shown are not known.

FIGURE 1
Wake Improvement with Special Lines-adapting Stern Devices Conventional Stern Cargo Ship



The decision to retrofit a wake adapting stern appendage should not be made lightly without quantification of the advantages and disadvantages. A price is paid for appendages in increased hull resistance. As a minimum, model tuft-tests with and without the appendage should be performed to observe the change in stern surface flow. The absence of any noticeable smoothing may be misleading, as a wake survey can show improvements in the propeller plane not discernable in the tuft behavior. Furthermore, aside from nominal wake considerations, it has been found that greatest wake improvements are sometimes achieved through propeller/appendage interaction. This implies that model tuft-tests should be conducted both with and without the operating propeller. In these cases, the best indicator of significant effective wake improvements from the standpoint of vibratory excitation may be an improvement by several percentage points in the propulsive efficiency from the model SHP test conducted with and without the wake adapting appendage. This is as explained in Hyllarides (1978).

Aside from wake improvements, the only recourse for reducing propeller excitation is modification or replacement of the propeller. Some instances of successful modifications of troublesome propellers have been reported. For example, trimming blade tips to reduce wake severity at the extreme propeller radii can produce improvements, but some degree of RPM increase should then be tolerated. Successful modifications of existing propellers are rare because of the usually unacceptable trade-offs of performance degradation against vibration improvement. The same disadvantages exist in propeller replacement considerations. Replacement propellers, with modified features, such as changed blade number, reduced diameter (for increased hull clearance), increased blade area, and reduced pitch in the blade tips may relieve the vibration problem, but often with a trade off in vessel performance. With the exception of moderate blade skew, essentially all of the measures available in propeller design for reducing vibratory excitation, once the stern lines are established, also result in reduced propeller efficiency. It cannot be emphasized strongly enough that the greatest insurance against propeller-induced vibration problems is to place high emphasis on wake uniformity in making trade-offs in the original establishment of vessel lines at the concept design level.

7 Structural Modifications

The most cost-effective approach for eliminating structural resonances is usually to shift natural frequencies through structural modifications. The alternative is to shift exciting frequency by changes in engine RPM or number of propeller blades.

Just as with hydrodynamics related problems, the most effective way to approach the correction of a vibration problem that promises to involve significant structural modifications is by using the tools of rational mechanics. A structural math model should be first calibrated to approximately simulate the existing vibration characteristics. Modification possibilities are then exercised with the model, and their probability of success is established on paper. In this way the probability of a success when shipboard modifications are subsequently implemented is maximized. The alternative “cut-and-try” approach to solving serious ship vibration problems brings the possibility of expending large amounts of time and resources without achieving complete success.

Of course, the paper-studies proposed as a tool for use in correcting a serious ship vibration problem should be concluded quickly. Several months, or even several weeks, is not available when delivery of a vessel is stalled, awaiting the resolution of vibration deficiencies. This places a premium on formulating the simplest possible structural models which still retain adequate fidelity to provide the basis for the required assessment as to the relative effects of vessel modifications. This is why the collection of thorough trial vibration data is important. Measurement of vibratory mode shape data is often a near necessity for securing guidance in formulating calibration models of the desired simplicity, but with sufficient accuracy. This is illustrated by the following case study.

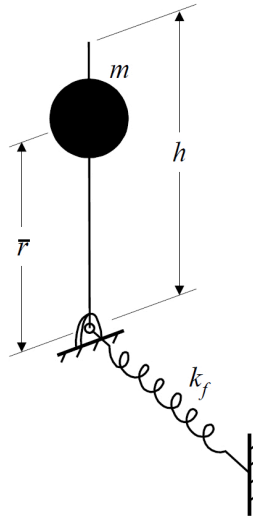
9 Case Study (1 September 2023)

Excessive vibration of a Type A deckhouse, 4/7.iv FIGURE 5, occurs on the builder’s trials of an 80,000 DWT product carrier. Vibratory displacement amplitude data are recorded with a multi-channel recorder via phase-calibrated accelerometers mounted at points on the house and on the main deck. The records establish the following information.

- i) The vibration occurs at predominantly blade-rate frequency, confirming the propeller as its exciting source.
- ii) The vibration amplitude peaks at 100 RPM, and the propeller has five blades. A resonance of the house at 500 cycles per min (cpm) is therefore established.
- iii) Vibration records recorded at 100 RPM show that the vibration of the house is predominantly fore-and-aft, with fore-and-aft amplitude increasing with a quasi-linear characteristic from low levels at main deck to a maximum of 39 mm/s at the housetop. The housetop is 15 meters above main deck.
- iv) The 100 RPM record also shows that the amplitude of the vertical vibration at main deck is approximately uniform at $V = \omega X = 5$ mm/s over the house length. The vertical vibration amplitude is also approximately constant at this same level up the house front, which is a continuation of the forward engine room bulkhead.

The above measured characteristics of the vibration under consideration are judged to allow the use of the simple rocking/bending house model in conjunction with the Hirowatari and Matsumoto method, Subsection 4/7, 4/7.iv FIGURE 5 and 4/7.iv FIGURE 6 and 4/7.iv TABLE 2.

FIGURE 2
Mass-elastic Model of Deckhouse and Support Structure



9.1 Determination of Model Constants

For a Type A house with $h = 15$ meters, the fixed-base fundamental house natural frequency is estimated from 4/7.iv FIGURE 6 as $f_\infty = 800$ cpm. Using the Dunkerley formula in Section 4 with the measured house natural frequency $f_e = 500$ cpm, the effective rocking frequency is predicted to be

$$f_R = \sqrt{\frac{1}{1/f_e^2 - 1/f_\infty^2}} = 640 \text{ cpm}$$

The two frequencies f_∞ and f_R can be used to determine the effective stiffness of the house and its under-deck supporting structure for use in the simple model of A2/9 FIGURE 2. For a house mass established as $m = 300$ t, with a radius of gyration, r , about the house forward lower edge of 10 m, the effective torsional stiffness of the under-deck supporting structure is, from Subsection 4/7:

$$K_f = (2\pi/60)^2 f_R^2 J = 1.35 \times 10^{11} \text{ N-m/rad}$$

where

$$J = m\bar{r}^2 = 3 \times 10^7 \text{ kg-m}^2$$

An approximate effective bending/shear stiffness of the house, K_h , is obtained by first lumping the house mass at the radius of gyration above the assumed pin support on the main deck at the forward bulkhead. This preserves the mass moment of inertia in the A2/9 FIGURE 2 model. Then, for the house base fixed,

$$K_h = (2\pi/60)^2 f_\infty^2 m = 2.1 \times 10^9 \text{ N/m}$$

The effective combined torsional stiffness for use in the equivalent reduced one-mass system is then:

$$K = \frac{1}{1/K_f + 1/(K_h\bar{r}^2)} = 0.822 \times 10^{11} \text{ N-m/rad (A2.1)}$$

This combined stiffness can also be deduced directly from the measured natural frequency and the house mass moment of inertia as:

$$K = (2\pi/60)^2 f_e^2 J = 0.822 \times 10^{11} \text{ N-m/rad}$$

The effective exciting moment due to the vertical hull girder vibration can be estimated using the following formula:

$$M_e = \omega m \bar{\xi} V \quad (\text{A2.2})$$

Here, ωV is the vertical acceleration of the house via the hull girder, so that $m\omega V$ is a vertically downward inertia force acting at the house CG. $\bar{\xi}$ is the longitudinal coordinate to the house CG, measured aft from the house front, which is 5 m. V is the velocity amplitude of the main deck vertical vibration, which is 5 mm/s. In terms of arbitrary hull girder vibration frequency ω , the exciting moment is estimated as:

$$M_e = 150\omega^2 \text{ N-m}$$

As a final remaining element of the A2/9 FIGURE 2 equivalent one-mass model, the damping factor ζ , is estimated as follows using the measured 39 mm/s house top vibration amplitude. With Φ being the amplitude of the equivalent vibratory rocking angular velocity of the house, the fore-and-aft velocity amplitude of the housetop is approximated as,

$$U = \Phi h$$

where h is the 15 m house height above main deck. Substituting the response formula for the A2/9 FIGURE 2 model,

$$U = \frac{\omega h M_e / K}{\sqrt{[1 - (\omega/\omega_n)^2]^2 + (2\zeta \omega/\omega_n)^2}} \quad (\text{A2.3})$$

But at resonance, $\omega = \omega_n$, so that:

$$U = \frac{\omega h M_e}{2\zeta} = 39 \text{ mm/s} \quad (\text{A2.4})$$

or

$$\zeta = \frac{\omega h M_e}{2 KU}$$

For $\omega = \omega_n = 2\pi f_e/60 = 52.4 \text{ rad/s}$,

$$M_e = 4.12 \times 10^5 \text{ N-m}$$

The damping factor is then,

$$\zeta = \frac{52.4 \times 15 (4.12 \times 10^5)}{2(0.822 \times 10^{11})(39 \times 10^{-3})} = 0.05$$

With the calibrated model so established as an equivalent one degree of freedom system, with constants, J , K , ζ and M , the above formula can be reused to evaluate changes in the house-top vibratory velocity amplitude, U , resulting from selected changes in the array of design variables included in the simple formulation.

9.3 Structural Rectification Analysis

To demonstrate the procedure, stiffening in the form of the added parallel pillars of the 4/7 FIGURE 7 example is contemplated. Following that example, the torsional stiffness of the under-deck supporting

structure is raised from the above value of 1.35×10^{11} N-m/rad to 1.475×10^{11} N-m/rad by the pillar addition. From Equation A2.1, the increased combined stiffness of $K = 0.866 \times 10^{11}$ N-m/rad results in a 2.6% increase in natural frequency from the measured value of 500 cpm to 513 cpm. The full power RPM of the vessel is 105, which corresponds to a full power blade-rate exciting frequency of 525 cpm. The critical has therefore been raised only to a higher level in the operating range from 100 to 102.6 RPM.

At 102.6 RPM, the 5 mm/s vertical hull girder vibration measured at 100 RPM would be increased by at least the frequency increase cubed, since the propeller exciting force would vary as RPM squared. This is assuming a flat frequency response characteristic of the hull girder (not close to a hull girder critical) as well as a non-cavitating propeller. Assuming a frequency cubed increase in V , the hull girder vertical vibration velocity amplitude becomes,

$$V = 5(102.6/100)^3 = 5.4 \text{ mm/s}$$

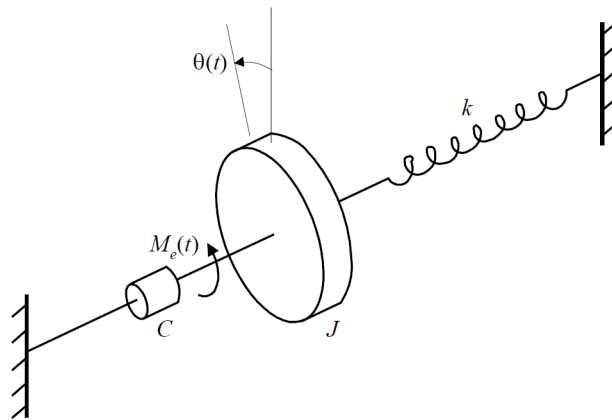
which would still be acceptable. However, the exciting moment from Equation A2.2 would be increased to:

$$M_e = 4.56 \times 10^5 \text{ N-m}$$

at the new resonant frequency $\omega = \omega_n(2\pi)(513/60) = 53.72 \text{ rad/s}$. The house top fore-and-aft vibratory velocity amplitude resulting from the foundation stiffening, assuming the damping factor unchanged, is predicted to be increased from 39 mm/s to:

$$U = \frac{53.72 \times 15 (4.56 \times 10^5) / 0.866 \times 10^{11}}{2(0.05)} = 40 \text{ mm/s}$$

FIGURE 3
Equivalent One-mass System



This is an increase in vibration of 2.5% over the original 39 mm/s level. The inadequate stiffening has simply raised the critical to a higher point in the operating range where the excitation is more intense. Some care is required here in order to achieve a satisfactory result.

In view of the above result, it would be intelligent at this point to evaluate the stiffness increase required in order to achieve a satisfactory vibration level. It is most desirable to move the critical above the full power RPM of 105, rather than decreasing stiffness to get below it. This establishes the exciting frequency at the full power RPM:

$$f = 525 \text{ cpm} = 8.75 \text{ Hz}$$

$$\omega = 8.75(2\pi) = 55 \text{ rad/s}$$

For crew accommodation areas, the limiting housetop fore-and-aft vibratory velocity amplitude of 5 mm/s is selected at this frequency. Continuing to assume a frequency-cubed variation in the hull vertical velocity amplitude, the exciting moment is now from Equation A2.2:

$$M_e = (55)(3 \times 10^5)(5)(5)(525/500)^3/1000 = 4.775 \times 10^5 \text{ N-m}$$

From Equation A2.3, for $2\zeta\omega/\omega_n < < [1 - (\omega/\omega_n)^2]$ for $\omega/\omega_n < 1$:

$$U = \frac{\omega h M_e / K}{1 - (\omega/\omega_n)^2} \quad (\text{A2.5})$$

Then with $\omega_n = \sqrt{K/J}$, the required combined stiffness is:

$$K = \omega h M_e / U + J \omega^2$$

Substituting the values, the required combined stiffness is:

$$K = 1.7 \times 10^{11} \text{ N-m/rad}$$

This requires more than doubling the as-built combined effective stiffness of 0.822×10^{11} N-m/rad. Little can normally be done to change the house stiffness; functional requirements of the house usually will not permit the modifications necessary to accomplish any significant increases in house casing section moment of inertia and shear area. Stiffening of the under-deck supporting structure is the only possibly effective structural modification that can be accommodated. The required K_f , is, from Equation A2.1:

$$K_f = \frac{1}{1/K - 1/K_h \bar{r}^2} = 1.02 \times 10^{12}$$

Therefore, meeting the vibration limit of 5 mm/s at the housetop will require increasing the effective torsional stiffness of the under-deck supporting structure by a factor of:

$$\frac{1.02 \times 10^{12}}{1.35 \times 10^{11}} = 7.5$$

and this would be impossible to accomplish. For example, if the two parallel pillars of the Subsection 4/7 example were doubled in number from 2 to 4 and moved 3 meters aft to line up under the house after bulkhead, rather than under the house sides (4/7 FIGURE 7), K_f would be increased to only:

$$\begin{aligned} K_f &= 1.35 \times 10^{11} + 2(5 \times 10^8)(8)^2 \\ &= 1.99 \times 10^{11} \text{ N-m/rad} \end{aligned}$$

which is still a factor of more than 5 below the required value. At this point, the virtual impossibility of rectifying the problem through structural modifications is clear, and attention would be turned to the propeller.

9.5 Propeller Change (1 September 2023)

Considering an alternative 4-bladed propeller, the critical would be shifted to:

$$100(5/4) = 125 \text{ RPM}$$

which is 25% above the critical, which meets the rule of thumb. With the foundation unchanged, the housetop vibration at the full power RPM of 105 would be, from Equation A2.5:

$$U = \frac{(55)15(4.775 \times 10^5)/0.822 \times 10^{11}}{1 - (105/125)^2} \times 1000 = 16 \text{ mm/s}$$

The vibration at the housetop is reduced by almost 50%, but is still too high, on the basis of the habitability criterion. Another possibility for the propeller would be to change it to six blades and lower the critical below full power to:

$$100(5/6) = 83 \text{ RPM}$$

At full power in this case:

$$U = \frac{(55)15(4.775 \times 10^5)/0.822 \times 10^{11}}{|1 - (105/83)^2|} \times 1000 = 8 \text{ mm/s}$$

This level might be marginally acceptable as a workspace on the bridge deck at the top of the deckhouse. The disadvantage to 6 blades is the resonance at 83 RPM. At 83 RPM (43.5 rad/s), the exciting moment, Equation A2.2, goes down by at least frequency cubed, as has been previously assumed.

$$M_e = 4.775 \times 10^5 (83/105)^3 = 2.36 \times 10^5 \text{ N-m}$$

With 6 blades, $f_e = 6(83)/60 = 8.3 \text{ Hz}$ (52.2 rad/s). The resonant amplitude is from Equation A2.4,

$$U = \frac{(52.2)15(2.36 \times 10^5)/0.822 \times 10^{11}}{2(0.05)} \times 1000 = 22.5 \text{ mm/s}$$

While predicted resonant amplitude level is excessive, it would not necessarily disqualify a 6-bladed propeller, as continuous operation at any particular lower RPM is usually not critical, and 83 RPM could be avoided except in passing.

In this case study, it was decided to change to a 6-bladed propeller. A reconditioned propeller of the correct diameter and pitch was refitted on the vessel. It was confirmed in advance that the blade area met the Burrill Criteria 10% back cavitation as suggested at 4/7.iv FIGURE 6. The deckhouse vibration was judged by the owner to be acceptable regarding the habitability criteria from [the ISO 6954 \(2000\) standard](#). A vibration increase in passing through the 80-85 RPM range was detectable by personnel in the deckhouse but it was moderate and found to be acceptable. Successful acceptance trials were completed within the delivery guarantee margin mandated by the contract.

APPENDIX 3 Seaway Excitation and Response

1 General

Ship flexural vibration excited by the seaway can be of three types:

- i)* Springing
- ii)* Bow Flare Slamming
- iii)* Bottom Slamming

3 Springing

Springing is a continuous-type vibration of the hull excited by the short wave (high frequency) wave components of the seaway. This is versus transient slamming, which is addressed in the next Subsection. The most well-known cases of springing were with the Great Lakes Ore Carriers built in the 1970's. There, the locks put limitations on vessel beam and depth, requiring longer and longer ship length to meet desired cargo capacity. Therefore, the ships had a very low 2-noded vertical flexural natural frequency. This coupled with the short-length (high frequency) spectrum of the Great Lakes seaway, resulted in measurable response magnification in the first flexural mode. The level of the flexural response could, however, not be explained by sub-resonant magnification alone. It was concluded by ABS that nonlinear hydrodynamics in producing low frequency “difference wave” excitation also played a role. Although monitored carefully for some years, no failures of the Great Lakes bulk carriers have ever been attributed to springing.

Springing could however be brought back to the forefront of attention by increasing vessel speed. The higher the speed the higher the “frequency of encounter” with the sea waves. The result could be the same as what appeared on the Great Lakes 30 to 40 years ago, but for a different reason. Care needs to be exercised in designing the new generation of high-speed ships with enough hull girder stiffness to avoid significant dynamic magnification of hull flexure by high frequency of encounter relative to sea waves.

5 Bow Flare Slamming and Whipping

Bow flare slamming occurs due to flattening of the bow (and stern) lines toward horizontal in the vessel body plan. Here the bow flare slamming tends to be more continuous and of longer duration than fore-foot bottom impact slamming. Therefore, for large container carriers as a typical example, the bow flare slamming can produce a significant transient hull girder vibratory response called ‘whipping’, which may cause the amplification of global hull girder bending loads. The bow flare slamming is also known to be capable of producing local damage to bulwarks and side plating in the flare regions.

Adding flare in the bow to increase hydrodynamic stiffness reduces bow submergence in waves at the expense of high flare loads and potential structural damage, both globally and locally.

Usual practice is that with high-speed ships, flare angles of the bow side shell and bulwarks are not to exceed angles of 40 to 45 degrees relative to the vertical.

7 Bottom Impact Slamming

Emergence of the fore-foot bottom above the water surface followed by reentry tends to produce very intense slamming that occurs for a short duration (fractions of a second). Here, the essentially instantaneous application of a point force will simultaneously excite all of the ship's normal modes, but to different degrees, depending on the spectral content of the slam impulse. Substructures with very low natural frequency may be excessively excited by the hull girder. It can also produce local plating damage in the fore-foot due to the high hydrodynamic impact pressure loadings.

Aside from designing for a deep forefoot, the best protection against bottom slamming damage is the adjustment of seaway operations to suit the local sea characteristics.

APPENDIX 4 Concept Design Checklist

The following provides a summary of the check list that should be used in the typical concept design process:

- i)* During the initial engine selection, check the 2nd order vertical moment M_{2v} from potential vendor, by the procedure of Subsection 3/3. If the PRU (Power Related Unbalance) exceeds 220 N-m/kW, consider either change of engine selection or installation of moment compensators. Also, the installation of engine lateral stays on the engine room structure should be addressed at the early design stage
- ii)* During the initial stern lines fairing and propulsion system arrangement studies, check the maximum angles and minimum propeller clearances in 3/5.1.i FIGURE 5 or 3/5.1.ii FIGURE 6.
- iii)* Confirm that aperture clearance provided in *ii)* is adequate to accommodate a propeller diameter satisfying the Burrill cavitation criteria for a maximum of 10% back cavitation, in 3/7.1 FIGURE 8.

If either *ii)* or *iii)* are not satisfied, plans should be made to perform detailed wake and propeller cavitation studies based on model testing and/or numerical simulation.

- i)* Once preliminary hull principal dimensions and weight are established, check the vertical hull girder natural frequencies up to at least 5 nodes to avoid the resonance or near resonance (20% separation minimum) with the engine second-order vertical moment at full power. This is by the procedure of Subsection 4/3.

If twice the RPM of the engine at full power falls within 20% of any of the estimated natural frequencies, the more precise FE-based free vibration analysis of the hull girder natural frequencies is recommended.

- ii)* At the time of propulsion machinery and foundation arrangement design, follow the procedure of Subsection 4/5 to check the thrust bearing structural foundation stiffness needed to avoid blade-rate resonance with either of the first two system longitudinal natural modes in the upper power range.

If this cannot be achieved, the change of propeller blade number and/or RPM should be considered so that a minimum 20% margin on the full power RPM is provided.

- iii)* In developing the scantlings and supporting structure of the deckhouse, perform the analysis in Subsection 4/7 to avoid fore-and-aft blade-rate resonance, within the 20% margin at full power.