INSIGHTS INTO

# SHIP VIBRATION ANALYSIS

JUNE 2022

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# 1. INTRODUCTION

As the size, power and speed of marine structures increase and their structures are becoming more optimized, vibration related problems are becoming more frequent. Recent design trends of locating machinery and accommodation spaces further aft and the predominant use of low-speed direct drive main diesel engines contribute to the increase of vibration related issues. At the same time, habitability related criteria for crew members and passengers are becoming more stringent.

Major classification societies have provided requirements and guidance to address vibration. For example, the ABS *Marine Vessel Rules* [1] 4-3-2/7 and the ABS *Guide for Enhanced Shaft Alignment* [2] provide requirements governing the design of propulsion shafting. Crew and passenger habitability, comfort and excessive noise levels are another area where both classification societies and international standards offer Rules and guidelines such as the ABS *Guidance Notes on Noise and Vibration Control for Inhabited Spaces* [3], the ABS *Guide for Crew Habitability on Ships* [4], IMO Code on Noise Levels On-board Ships (IMO Resolution MSC.337[91]) [5], International Labor Organization (ILO) Maritime Labor Convention (MLC) [6], ISO 6954 (1984) [7], ISO 6954 (2000) [8], ISO 20283-5 (2016) [9] and ISO 21984 (2018) [10].

Classification societies and other regulatory bodies have also published guidelines to identify and mitigate vibration issues on ships such as the ABS *Guidance Notes on Ship Vibration* [11], IACS Guidelines for the Identification of Vibration Issues and Recommended Remedial Measures on Ships [12] and SSC Ship Vibration Design Guide [13].

Furthermore, classification societies are engaged in continuous research related to structural vibration and continue to develop guidance for the maritime industry on structural vibration assessment and mitigation. Excessive structural vibration levels usually lead to three main types of problems:

- Fatigue cracks in local structural elements
- Decrease in habitability and comfort
- Malfunction of installed machinery and electrical instruments

It is important to consider vibration at a vessel's early design stage when significant improvements of the design are still possible at a relatively modest cost. As the design and construction of the vessel matures, the cost of addressing major vibration related issues grows exponentially.

Different design stages of the vessel may call for the application of different vibration analysis methods. At the early design stages, the details of the hull structure and machinery are usually not known. Therefore, it is appropriate to use robust analytical, semi-analytical, or empirical methods. The designer's experience will also have a significant effect on the outcome of the analysis. As the design matures and more details become available, more advanced methods based on numerical simulations may be used.

The objective of this paper is to highlight critical vibration issues on modern marine structures, to discuss possible mitigation measures and to inform the industry.

This paper presents the latest developments in vibration analysis, vibration criteria and mitigation measures. It also presents four different real-world cases where vibration issues were identified, analyzed and mitigated.

This paper focuses on the structural vibration issues of ship-shaped structures.

## 2. SHIP VIBRATION ANALYSIS

The purpose of vibration analysis is to estimate the dynamic response of a vessel to different excitation forces and to assist the design engineers in eliminating significant vibration problems. Depending on the ship design stage, vibration analysis approaches vary significantly in their complexity and accuracy. The effort to design against severe vibration in ship structures should start from an early concept design stage. Figure 1 shows typical ship vibration analysis approaches at various ship design stages.

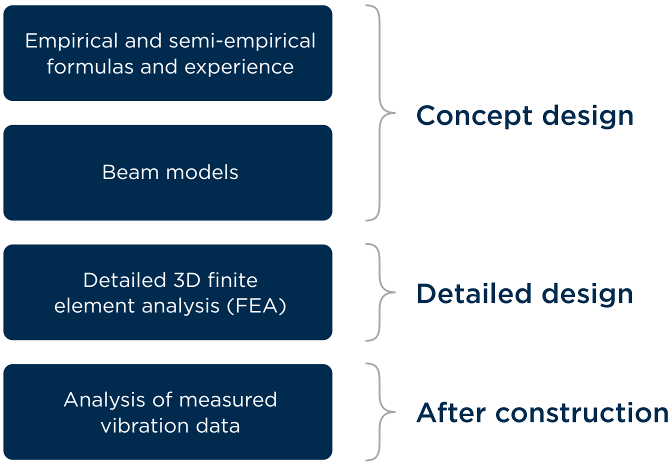



*Figure 1 – Different ship vibration analysis approaches*

Regardless of the ship design stage, vibration analysis methods can be divided into two main types:

- Free Vibration Analysis
- Forced Vibration Analysis

**Free Vibration Analysis** solves the equations of motion of an undamped system without any excitation forces. It seeks to identify the natural frequencies (also called eigenvalues or eigenfrequencies) of vibration of the structure at which it will tend to vibrate if disturbed. Associated with every natural frequency is its natural mode (also called eigenmode) which represents the relative deflection shape of the structure as it vibrates at a particular natural frequency. It is important to note that the free vibration analysis does not calculate the response of the structure to any particular excitation in absolute terms. It only identifies the natural frequencies of the structure, while its associated natural modes represent the structure's deflection shapes in relative terms. These mode shapes are usually scaled so that the largest amplitude of vibration in the structure is one unit. Damping in the structure has a minor influence on its natural frequencies and is usually neglected during the free vibration analysis. Figure 2 shows the first six global natural vibration modes of a typical cargo vessel.

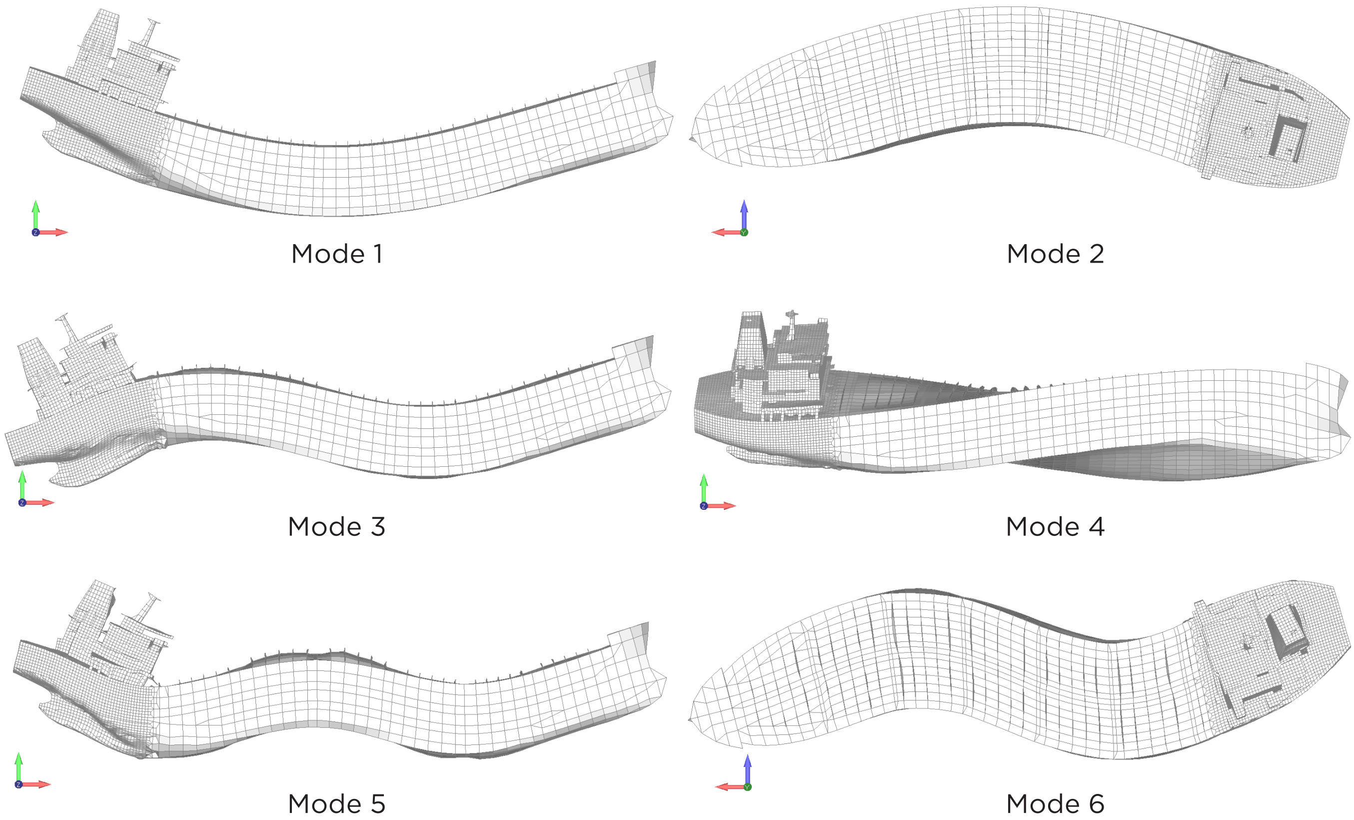



*Figure 2 – First six global natural vibration modes of a cargo vessel*

Even without estimating the response of the system to applied excitation forces or moment in absolute terms, free vibration analysis is crucial for identifying resonance conditions, especially at the early design stages. Resonance is an unwelcome phenomenon that occurs when the frequency of the excitation force or moment is equal, or very close to one of the natural frequencies of the system. In that case the excitation is only opposed by damping which could quickly lead to large dynamic responses of the structure, even for smaller excitation magnitudes. Many major vibration problems can be avoided by making sure the excitation frequency does not fall within 20 percent range around the natural frequencies of the vibration modes of the system. At the early design stages there are greater possibilities to influence the natural frequencies of the structure by changing its stiffness and mass distributions, or by changing the excitation force frequencies. This will be discussed in detail later in this paper. Sometimes, it is not possible to adequately separate the excitation and system natural frequencies. In that case, a more detailed forced vibration analysis should be performed to determine the dynamic response of the structure in absolute terms and to compare it with acceptance criteria.

Natural frequencies of the system can be estimated using empirical and semi-empirical formulas, beam methods, or advanced 3D finite element analysis (FEA), depending on the ship design stage.

**Forced Vibration Analysis** solves the equations of motion of the damped system with acting excitation forces or moments. There are two main subtypes of forced vibration analysis depending on the type of the excitation forces:

- Transient Response Analysis (TRA)
- Frequency Response Analysis (FRA)

TRA is the most general type of free vibration analysis wherein the time-dependent excitation forces can be arbitrary in nature, for example, not harmonic. An example of such an excitation force is an impulse loading due to *slamming* of the ship. TRA solves the equations of motions in time domain and can include nonlinear loads that are a function of displacement or velocity.

TRA can be subdivided into direct (explicit or implicit) and modal.

The direct method is based on the implementation of an explicit or implicit time integration rule for solving the equations of motion. The modal analysis method is used to solve the equations of motion using a subset of the structure's natural vibration modes, and is particularly suitable for analyzing a structure subjected to long-term excitation loads. For linear systems, the modal TRA is much more computationally efficient compared to the direct TRA.

FRA computes the system's structural response to a steady-state harmonic (sinusoidal) oscillatory excitation which is explicitly defined in the frequency domain. Each excitation force, moment, or pressure is defined as a harmonic represented by an amplitude, a frequency and a phase shift with respect to a reference point. FRA assumes a ship as a linear system in which the response to each harmonic excitation is also harmonic with the same frequency, but different amplitude and phase shift compared to the excitation (see Figure 3). The phase shift between the excitation and the response is caused by damping. Each excitation frequency is considered separately in FRA. If multiple excitations at different frequencies act on the ship at the same time, which is usually the case, then a separate FRA is performed for each frequency. Depending on what acceptance criteria is used, responses at different frequencies may need to be combined. This is discussed further in Section 4.

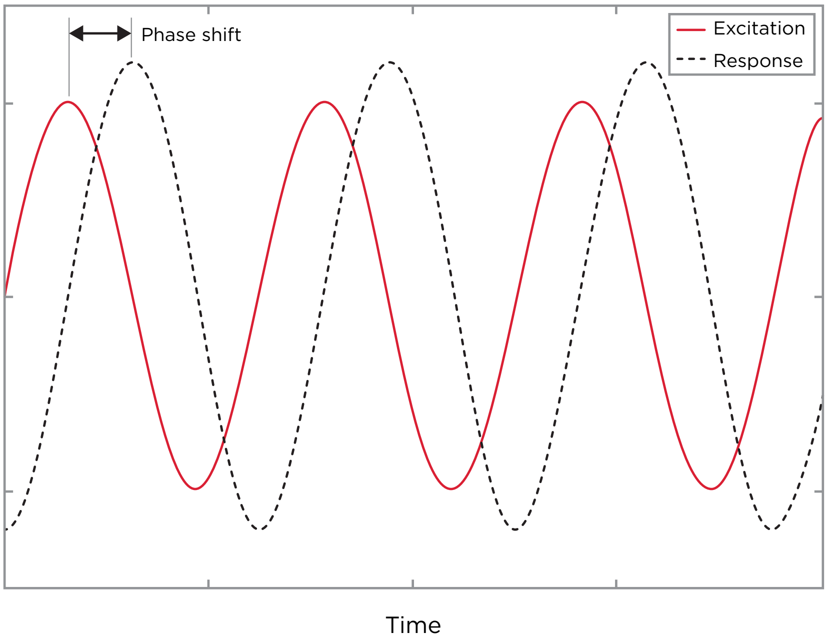



*Figure 3 – Phase shift between excitation and response*

FRA can also be subdivided into direct and modal.

Direct FRA solves the coupled equations of motion in the frequency domain for each forcing frequency. It is more accurate than the modal FRA method, but it is also more computationally intensive.

Modal FRA also solves the coupled equations of motion in the frequency domain for each forcing frequency by utilizing the mode shapes of the structure. Modal FRA relies on the fact that every response of the structure can be described as a linear superposition of its natural vibration modes. Usually, only a subset of natural modes of the structure are necessary to adequately describe the response of the system. Because not all the natural modes need to be calculated or retained, the problem size is significantly reduced. Usually, all natural modes with a frequency less than twice the highest excitation frequency are needed to achieve sufficient accuracy. Since modal FRA requires natural mode calculation, it will only be computationally more efficient compared to direct FRA when analyzing very large systems. This is usually the case when using large 3D full ship finite element models. Use of modal FRA is also advised when the free vibration analysis is calculated as an initial step in the vibration analysis. In that case, the natural modes will have already been calculated and modal FRA is then a natural continuation of the free vibration analysis. When there is no damping, or when damping can be defined for each natural vibration mode (modal damping), the modal FRA uncouples the equations of motion, making them much easier to solve.

Forced Vibration Analysis can be performed at any stage of the ship design process. However, it is normally performed at the detailed design level or after construction to determine the root cause of vibration problems. The uncertainty regarding the ship structure and its excitation forces at the concept design stage usually does not justify the use of advanced forced vibration analysis.

Figure 4 summarizes different types of ship vibration analysis.

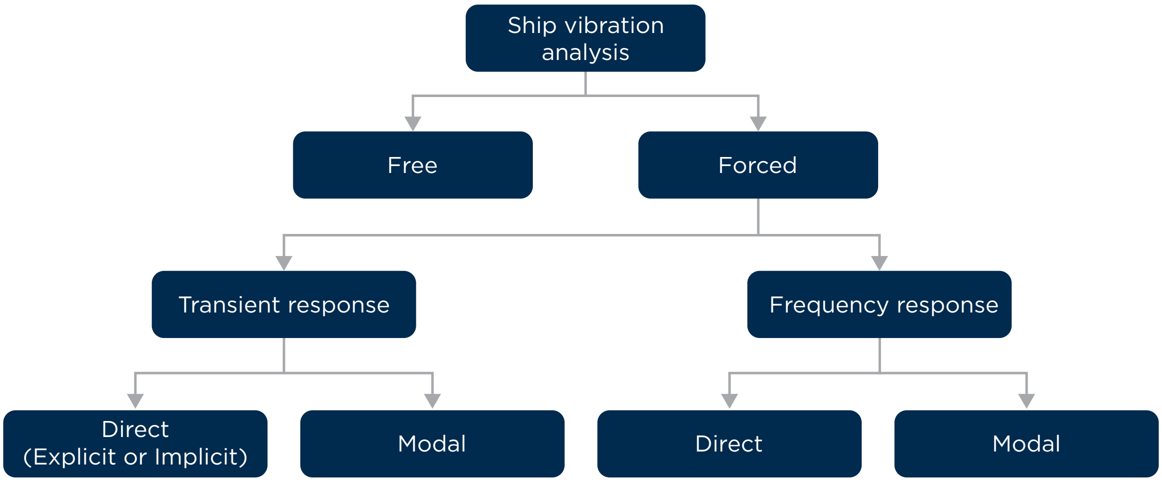



*Figure 4 – Different ship vibration analysis types*

Most major excitation forces on ships are caused by the main engine, propeller and significant rotating machinery. They are all steady-state and harmonic. Waves can also create steady-state harmonic excitation in the form of springing. In these cases, modal frequency response analysis is the best choice and offers a good balance between accuracy and computational efficiency. Full ship 3D finite element models are usually used so that both global and local vibrations of the structure can be analyzed. Such models should also have the main propulsion system modeled, which entails the finite element modeling of the main engine, shaft and the propeller. This enables the analysis of the interaction between the main engine and its foundation structure, as well as precise modeling of the engine and propeller excitation forces. Transient response analysis in time domain is used for non-harmonic excitations like slamming impacts when analyzing hull girder whipping.

## 3. EXCITATION FORCES

When performing the vibration analysis, it is important to consider all major vibration sources. This is not only important for the forced vibration analysis, but also for the free vibration analysis to avoid potential resonance conditions. Excitations can be described by their:

- Type (e.g., force, moment, pressure)
- Amplitude
- Frequency
- Phase lag
- Location and direction

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 it is often misrepresented in the forced vibration analyses. If more than one excitation source is acting on the ship structure, which is almost always the case, the phase lags between different excitation forces, moment, 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 must 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 the 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 knowing the phase lags between corresponding excitations at different orders. However, phase lags between excitations of the same order must 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 other location they may destructively interfere (cancel). In fact, for twin screw vessels, propellers may be phased in such a way to achieve destructive interference and reduce the total response.

It is widely accepted that excitation type, amplitude, frequency, phase lag and direction will affect the vibration response of a structure. Near the resonance, even small amplitude excitations can cause large vibration responses. However, what is not frequently well understood is that the location of the excitation force will also significantly affect the vibration responses. 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). Figure 5 shows two nodes and an antinode of a natural vibration mode of a structure. If the excitation force acts at, or is very close, to the node of the natural vibration mode, that vibration mode will have low excitability, even if the force amplitude is significant and even if its frequency is close to the natural frequency of that vibration mode. However, a node of one natural mode can be an antinode of another mode.

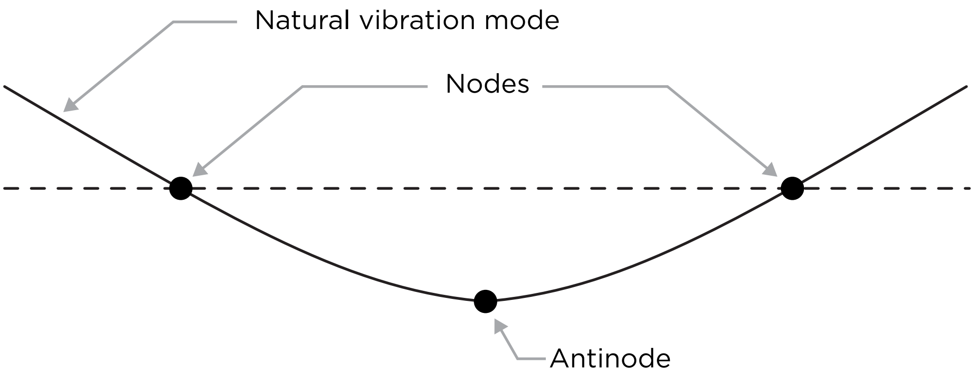



*Figure 5 – Nodes and antinode of a natural vibration mode*

In general, there are three major sources of vibration:

- Low-speed main diesel engine
- Propeller
- Waves

## 3.1 LOW-SPEED MAIN DIESEL EXCITATIONS

Excitations coming from the low-speed main diesel engines are most likely to cause global hull girder resonance, or resonance of the local structures in the engine room or the superstructure. This is because engine excitations can have amplitudes that are large enough and in the adequate frequency range to cause structural resonances. Low-speed main diesel engine excitation moments and forces can be divided into two main categories:

1. External (inertial) unbalanced moments and forces
2. Guide force moments
    a. H-Type moments
    b. X-Type moments

### 3.1.1 EXTERNAL MOMENTS AND FORCES

External (inertial) moments and forces arise from unbalanced rotating and reciprocating engine masses. Modern main engines with more than two cylinders, which is the case relevant to ships, are designed to cancel the external forces at the engine foundation. The external moments around the axis of the engine are inherently zero, which leaves only the vertical and transverse external moments. Most low-speed marine diesel engines have more than six cylinders. For these engines, the main contribution to hull vibration comes from the second order vertical external moment. This is generally the excitation of most concern for low-speed diesel main engines. However, depending on the number of cylinders and firing order, the first or higher order vertical external moments can be as large as the second order moment and may require consideration. The first order horizontal external moments may also need to be considered, again depending on the number of cylinders and firing order.

The second order vertical external moments tend to have frequencies that close to the natural frequencies of the hull girder vertical vibration modes having four or five nodes. The vibration response to the second order vertical external moment may be exacerbated if the engine is located at, or close to, a node of one of these natural modes. Higher natural vibration modes usually have lower excitability and may be of less concern.

The forced vibration analysis is usually performed for several different engine RPMs within its operating range. However, the amplitudes and phases of external moments are given by the engine manufacturer usually at the single RPM corresponding to the maximum continuous rating (MCR) of the engine. The amplitudes of external moments at different RPMs generally obey the following relationship:

$$M_A = M_1 \left(\frac{n_A}{n_1}\right)^2 \tag{1}$$

where $M_A$ is the external moment at arbitrary rotational speed $n_A$, and $M_1$ is the external moment at the rotational speed $n_1$, which corresponds to MCR. The phases of external moments to not generally change with the RPM of the engine.

### 3.1.2 GUIDE FORCE MOMENTS

Guide force (pressure) moments are caused by gas pressure forces developed during the combustion process and the subsequently developed transverse reaction forces in the crossheads from the connecting rod/crankshaft mechanism. These forces cause engine transverse vibrations, moving the engine athwartship and causing a rocking (H–type) vibration or twisting the engine (X–type). The largest component of the guide force H-type moments is of the order corresponding to the number of cylinders. Other components of the H-type guide force moments occur at orders that are multiples of the number of cylinders (firing order). X-type guide force components can be of any order. Nonzero X-type guide force moments generally exist up to the 16$^{th}$ order.

Guide force moments are usually harmless, except when they cause resonance within the local structures in the engine/double bottom foundation system.

Like the amplitudes of external moments, the amplitudes of the guide force moments also vary with the engine RPM. However, they do not obey the relationship in Equation 1. The phases of guide force moments depend on the engine RPM, unlike the phases of external moments. Therefore, both the amplitudes and the phases of guide force moments have to be provided by the engine manufacturer at each engine RPM.

Figure 6 schematically shows the engine-generated external and guide force moments. The most common orders are given in brackets.

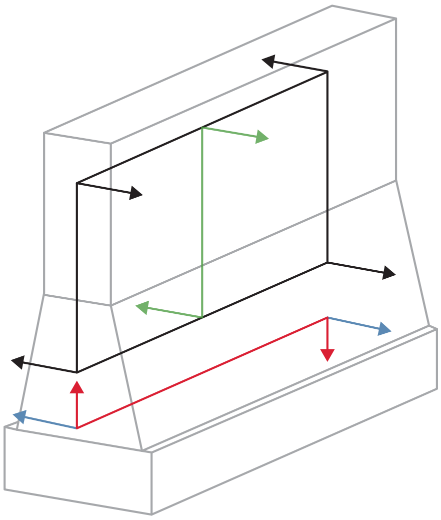



*Figure 6 – Engine generated external and guide force moments*

## 3.2 PROPELLER EXCITATIONS

Excitation from the propeller is transmitted to the rest of the ship structure through the shell plating above the propeller (surface forces) and through the shaft line (bearing forces). Both propeller surface and bearing forces are significantly exacerbated by hull wake nonuniformity. If the propeller disc fluid inflow is circumferentially uniform, propeller-induced vibrations would not have to be considered during the ship design. Therefore, attention should be paid to the design of the after-body and appendages to achieve a uniform wake field to the greatest extent possible.

### 3.2.1 PROPELLER SURFACE FORCES

Propeller surface forces are the predominant propeller excitations, especially for cavitating propellers. They account for approximately 90 percent of propeller caused ship vibrations. They act mainly in the vertical direction on the horizontal part of the ship's bottom above the propeller. Therefore, they directly affect the vertical vibrations of the stern ("fan-tail mode"), and consequently, the fore and aft superstructure vibrations by way of rotation. However, propeller surface forces are generally incapable of exciting the global hull girder modes to significant levels. The largest component of the propeller surface forces is a harmonic oscillation at the blade rate frequency. Harmonics at twice the blade rate frequency have, in general, half of the amplitude of the blade rate harmonic. Higher blade rate harmonics will have even smaller amplitudes. A full power blade rate harmonic has a frequency typically in the range of hull girder natural vibration modes with seven or more nodes. Surface pressure amplitudes of the full power blade rate harmonic more than eight kPa can be considered as high and those below two kPa considered as low.

Propeller surface forces arise from fluid pressure fluctuations on the stern bottom plating above the propeller which are generally proportional to the acceleration of the propeller cavity volume. The pressure fluctuations are very sensitive to the amount of intermittent propeller cavitation triggered by the nonuniform wake field. The calculation of propeller surface forces is a complex fluid structure interaction (FSI) problem. The methods for predicting them fall in one of these four categories:

- Empirical methods
- Advanced theoretical methods
- Numerical simulations
- Model testing

### 3.2.2 PROPELLER BEARING FORCES

Propeller bearing forces are the predominant factor for the vibration of shaft lines but can also cause engine room vibrations, vibrations of the main engine and its foundation and fore and aft vibrations of the superstructure, depending on the moment arm of the thrust forces with respect to the neutral line of the hull girder. The alternating propeller bearing forces consist of axial thrust, vertical and transverse forces, as well as torque and moments around the vertical and transverse axes. The blade rate and twice blade rate harmonics are usually considered. For the analysis of ship structural vibrations, only the thrust, vertical and horizontal bearing forces are typically considered. Alternating thrust can be applied at the thrust bearing block position, while vertical and transverse bearing forces can be applied at the stern tube bearing.

## 3.3 WAVE EXCITATIONS

Waves can excite hull girder vibrations either through slamming impacts, also known as whipping, or through resonance between the encounter frequency of the waves and one of the hull girder natural vibration modes (springing). Whipping is more transient in nature compared to springing, but both phenomena can be avoided by changing the speed and or heading of the vessel and they usually do not cause major structural or habitability issues. The exception may be when slender installations exist on the deck of the vessel, such as the flare tower on ship-shaped hydrocarbon production and process systems that are not able to change their speed or heading. Vertical whipping vibrations of the hull girder can, under certain circumstances, cause resonance of slender installations and subsequent fatigue issues. One such case is described in Section 6.1.

# 4. ACCEPTANCE CRITERIA

After performing a forced vibration analysis, the structural dynamic responses are evaluated by comparing them to the acceptance criteria which are typically agreed upon between the owner and the shipyard at the contractual phase. The main responses from the forced vibration analysis performed using the frequency response analysis and the 3D finite element model are displacements, velocities and accelerations at each node of the finite element model. All responses are harmonic and are given in terms of their amplitude and phase. Element stresses and strains can also be post-processed from the nodal displacements.

Most vibration criteria are based on displacements, velocities and acceleration measures. There are four types of measures for vibration responses:

- Narrowband maximum repetitive amplitudes (MRA)
- Broadband MRA
- Broadband root mean square (RMS)
- Frequency weighted broadband RMS
-

These different measures are often a source of confusion. They are described below in more detail.

## 4.1 NARROWBAND MRA

Narrowband MRA has been used in vibration acceptance criteria since the 1960s. Nowadays, narrowband MRA is used as a vibration criterion by some older habitability standards such as ISO 6954 (1984) [7] for assessing local structural vibrations [11], or assessing machinery vibrations [14].

Because the frequency response analysis inherently separates the response components at each forcing frequency, the narrowband (single frequency) MRA can be easily obtained as the amplitude of the calculated response at a particular frequency. For calculating the narrowband MRA using the forced vibration analysis, it is not necessary to consider the phase lags between excitations at different frequencies. However, the phase lags of different excitations having the same frequency must be considered to obtain a correct response at that frequency.

It is more difficult to estimate the narrowband MRA from the measured vibration time series. Measured time series contain all frequency components of the response. Therefore, precise filters need to be applied to separate components at different frequencies. Filtering will inherently introduce errors and there will be some leakage of energies from different frequencies called spectral leakage. Separating different frequency components of the response becomes even more challenging when components with the most energy have similar frequencies. Even precise filters cannot isolate a single frequency component. Rather, all the frequencies within a narrowband range will be included in the filtered signal. Therefore, the main frequency component, referred to as the carrier, will be modulated by other less significant frequency components within the narrowband range of interest. The narrowband MRA can be estimated as the peak value of the modulated narrowband response signal where the largest excursions of the response, originating from measurement errors, have been removed prior to calculating the narrowband MRA. The amount of modulation will depend on the range of frequencies passed by the filter and will directly impact the narrowband MRA. For these reasons, it is difficult to accurately determine the narrowband MRA from the measured data, which makes its use as a measure of vibration severity inappropriate in such cases.

## 4.2 BROADBAND MRA

Broadband MRA refers to the peak response of the full broadband unfiltered response signal where the largest excursions of the response, originating from measurement errors, have been removed prior to calculating the broadband MRA. This measure of vibration severity has been developed because it has become widely accepted that hull vibration should be assessed by combining the effect from all the frequency components of the response.

Broadband MRA can be directly determined from the measured broadband signal of the response, as no filtering is necessary. The broadband MRA can also be determined from the results of the frequency response analysis, but in this case a broadband response signal in time domain must be obtained by combining individual harmonic response components at different frequencies. The peak value of the combined broadband response will depend on the phases of each response component. Therefore, the broadband MRA requires the consideration of phase lags between each excitation source, regardless of whether they have different or equal frequencies. ANSI S2.27-2002 [15] uses this measure in criteria for torsional stresses of the propulsion shaft that cause fatigue issues.

## 4.3 BROADBAND RMS

Broadband RMS value has been increasingly used as a measure of vibration severity in the industry. This is especially true for assessing structural and machinery vibrations. The RMS of a time-dependent broadband response signal is calculated as the square root of the mean of the squared signal and is related to the energy content of the response. For a sinusoidal single frequency response component, such as the one obtained by the frequency response analysis, the RMS value is equal to $1/\sqrt{2}$ times the amplitude. The RMS value is less affected by waves and ship motion than MRA, so it is considered as a more reliable measure of vibration severity. The broadband RMS can be calculated from the broadband measured response signal, or from the response signal obtained using the frequency response analysis. Unlike the MRA, the RMS value of the combined response does not depend on the phases of each response component due to the orthogonality property of response components at different frequencies. Therefore, unlike the broadband MRA, the broadband RMS only requires the consideration of phase lags between excitation sources at the same frequency. The broadband RMS of the combined response can be calculated as the square root of the sum of squares of the narrowband RMS values of individual response frequency components.

Broadband RMS value is used as a measure of vibration severity in ANSI S2.27-2002 [15] criteria for main propulsion machinery components and in ANSI S2.25-2001 criteria for the hull, superstructure and mast.

## 4.4 FREQUENCY WEIGHTED BROADBAND RMS

The frequency weighted broadband RMS value is used by newer habitability standards such as ISO 6954 (2000) [8], ISO 20283-5 (2016) [9] and ISO 21984 (2018) [10]. The narrowband MRA criteria considers each frequency component individually. However, it has been realized over the years that the effect of each frequency component on the human body should be combined. In that case, the broadband MRA or the broadband RMS values may be used. The RMS value proved to be a better choice since the human body is more sensitive to the average vibration energy (RMS) than to the maximum vibration amplitude (MRA). Instead of using the broadband RMS, modern vibration criteria regarding habitability use frequency weighted broadband RMS value because it is now known that the human body does not respond equally to different frequencies. A simple broadband RMS value could be obtained as a root square of the sum of squares of individual frequency component RMS values, assuming each frequency component has a unit weight. However, the frequency weighted broadband RMS values weights each frequency component RMS value before taking the square root of the sum of squares.

Frequency weighted RMS value requires filtering of the measured vibration signal to isolate each frequency component and entails all the drawbacks of the filtering process. No filtering is needed when using the results of the forced vibration analysis since different frequency components of the response are already separated and weighting can be performed in a relatively simple manner. Similar to the calculation of the broadband RMS value, the frequency weighted broadband RMS only requires the consideration of phase lags between excitation sources at the same frequency.

# 5. MITIGATION MEASURES

Vibration mitigation measures should start at the concept design stage where their cost effectiveness is the largest. The expertise and experience of the engineer/designer plays a significant role here. Identifying the potential for major vibration issues using very simple tools available at the concept design stage is very important for the ultimate success of the ship design.

Figure 7 shows the main elements influencing ship vibrations. Therefore, vibration mitigation measures can be taken by modifying any subset of these elements in such a way as to minimize the negative impact of vibrations on humans, ship structure, or machinery. The following subsections describe the mitigation measures usually applied to mitigate the most critical vibration problems identified by experience.

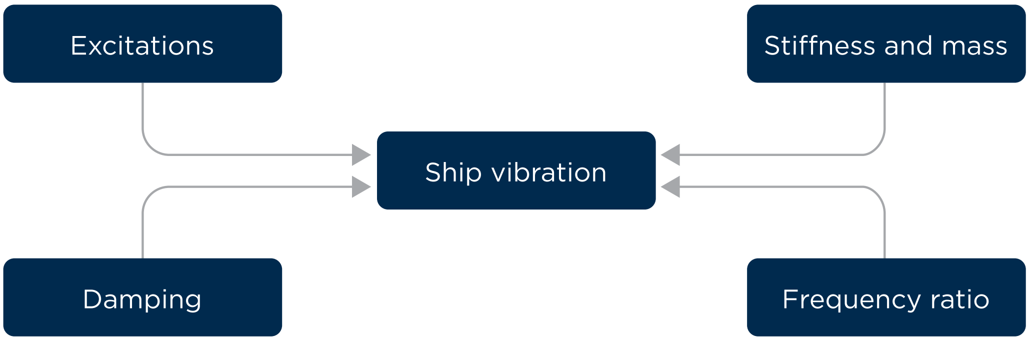



*Figure 7 – Main elements influencing ship vibration*

## 5.1 MITIGATING EXCITATIONS

As described in Section 3, excitations are characterized by their type, amplitude, frequency, phase, location and direction. Changing any of these characteristics will influence ship vibration.

### 5.1.1 MITIGATING MAIN ENGINE EXCITATIONS

Ship vibrations can be reduced by reducing the excitation force amplitude. The main concern for the main engine-induced vibrations, especially for six-cylinder engines, is the value of the second order external moment amplitude $M_{2v}$. It can be reduced by choosing a different engine or by installing the moment compensator. A value that is usually used to assess the acceptability of $M_{2v}$ is called the Power Related Unbalance (PRU), which is calculated as follows:

$$PRU = \frac{M_{2v}\,[Nm]}{Engine\ Power\ [kW]} \tag{2}$$

For values of $PRU$ higher than 220 Nm/kW, the need for a moment compensator is judged as highly likely, while for values below 120 Nm/kW it is considered unlikely.

The moment compensator consists of counterrotating masses at twice the shaft RPM (second order) and is usually geared directly to the engine. The rotation of the masses is properly phased to achieve partial or full cancelation of the undesirable second order external moment. Moment compensators can be installed at the aft end of the engine, forward end of the engine, or far away from the engine as electrically driven moment compensators. The location of the engine within the vessel greatly influences the choice of moment compensator. If the engine is located far away from the nodes of any natural hull girder modes that fall within the second order RPM of the engine at full power, the moment compensator will probably not be needed, as engine excitation moments are not effective at exciting hull girder natural vibration modes when located far away from the nodes.

If the engine is not fitted with a moment compensator, or a proper engine preparation has not been performed for future installation of the moment compensator, and the harmful vibrations of the hull and superstructure do occur, then an electrically driven moment compensator can be installed at a location far away from the natural vibration node.

The electrically driven moment compensators can be vertical and/or horizontal with the former being more common. The vertical moment compensator will generate a vertical force at twice the RPM, and if properly phased with respect to the engine second order external moment, it can partially or fully cancel its effect on the hull. The vertical moment compensator is usually installed in the gear room, which is distant from the nodes of the critical natural modes. That way the vertical force generated by the moment compensator has a significant effect. The vertical moment compensator can also be installed above the propeller to attenuate the propeller-induced vibrations.

The horizontal electrically driven 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 advantages of electrically driven moment compensators are that no preparation is needed for their installation, and they can be retroactively installed on existing ships. Figure 8 schematically shows a vertical electric compensator with two counterrotating masses.

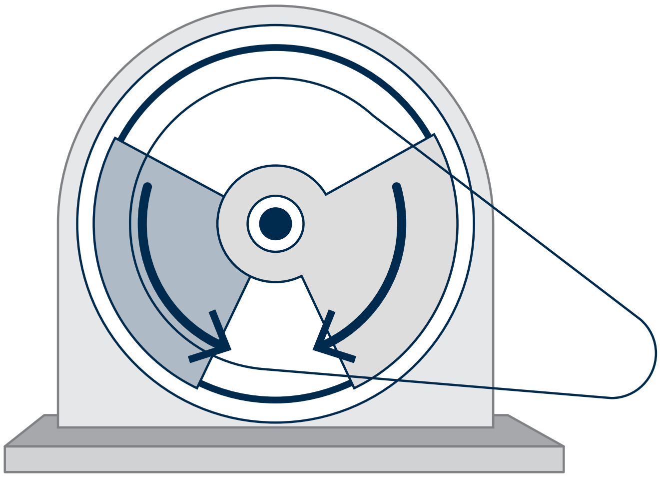


*Figure 8 – Electrically driven compensator*

#### 5.1.2 MITIGATING PROPELLER EXCITATIONS

For propeller-induced vibration, the propeller excitation amplitudes can be reduced by minimizing the wake nonuniformity. This can be achieved by changing the stern lines and locations and shape of the appendages, increasing the clearance between the propeller and the stern bottom plating and changing the propeller geometry. The following are the recommended stern arrangement guidelines to reduce propeller bearing and surface force amplitudes, depending on the stern configuration.

1. Single or twin-screw strut stern
    - The vertical clearance between the tip of the propeller and the bottom of the stern shell plating (counter) should be larger than 25 percent of the propeller diameter.
    - The shaft inclination angle relative to the baseline should be less than 5 degrees (see Figure 9).
    - The shaft inclination angle relative to the buttocks angle of the counter should be less than 10 degrees (see Figure 9).

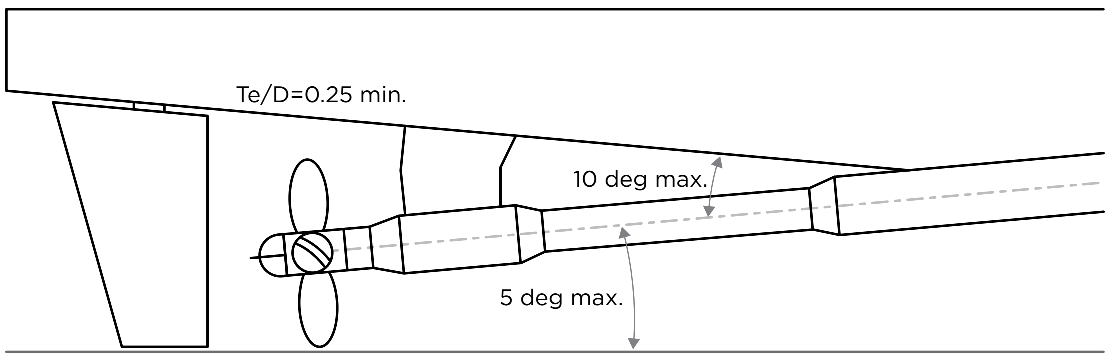



*Figure 9 – Open strut stern arrangement (Figure taken from [11])*

2. Conventional stern with skeg and bossing
   - The angle of the waterline at the tip of the propeller disk with respect to centerline at the entrance to the propeller aperture should be less than 35 degrees (see Figure 10).
   - The vertical tip clearance should be larger than 25 percent of the propeller diameter. Forward clearance of 40 percent of the propeller diameter is considered to be usual practice, even though the conventional skeg and bossing sterns are less sensitive to propeller clearance compared to the strut sterns.

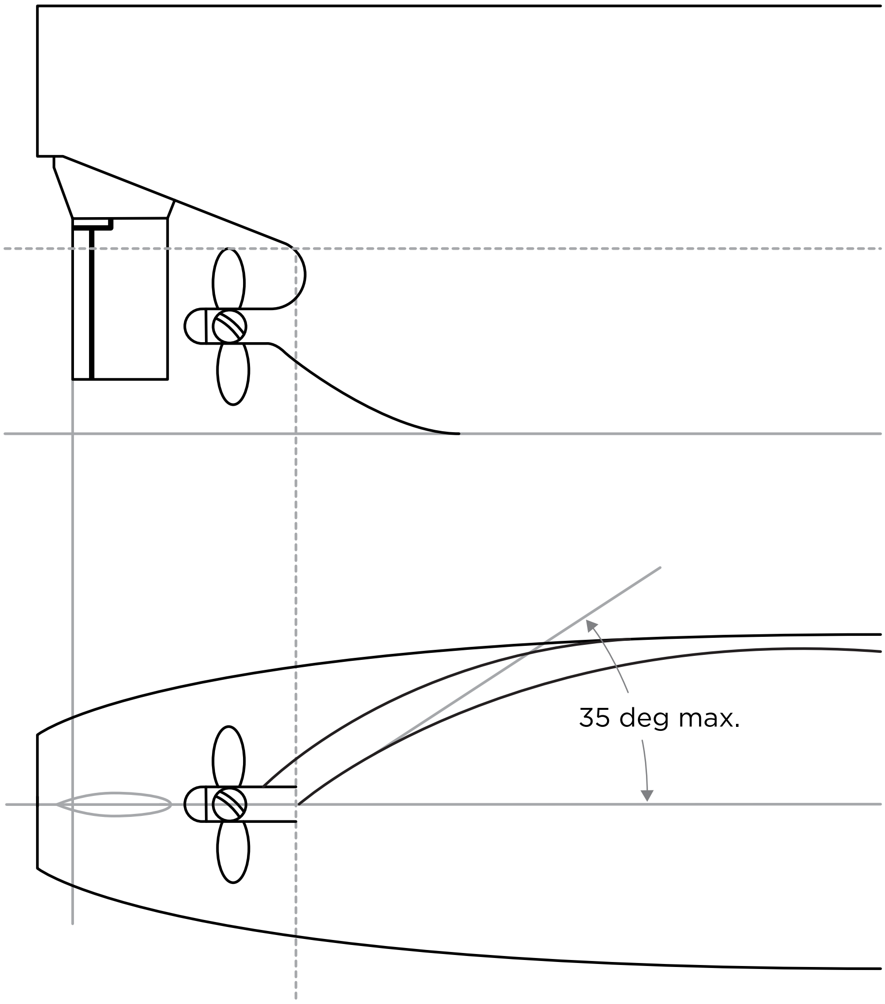



*Figure 10 – Conventional skeg-stern arrangement (Figure taken from [11])*

The amplitude of vertical propeller surface forces for a cavitating propeller can be an order of magnitude larger than that of the bearing forces. At the concept design stage, pressure and bearing forces should be assumed to be in-phase. Also, for twin screw ships, the forces from the two propellers should also be assumed to be in-phase. At the detailed ship design stages, proper phasing between different propeller forces from a single or multiple propellers can be considered during the forced vibration analysis.

## 5.2 ADJUSTING STIFFNESS AND MASS

Stiffness is defined as a force needed to cause a unit deflection of a structure, and it directly influences the natural frequency of each vibration mode. When changes to the natural frequency of the structure are needed to avoid resonance, this may be accomplished by increasing the stiffness of the structure. Decreasing the stiffness of the structure to avoid resonance is generally not a good practice.

Increasing the stiffness of the structure to avoid potential resonance is most frequently done at the local structural level (stiffened panel level) or at the sub-structural level (engine-foundation system, superstructure foundation, deck structure between bulkheads and longitudinal and transverse bulkheads). To significantly change the stiffness of large structural blocks, such as the superstructure, major structural changes are often required. Therefore, this may only be possible at the design stage before construction begins. Increasing the stiffness of existing structure may only be possible for local structural members, such as stiffened panels, masts, handrails and pipes.

Mass of the structure and its distribution will also directly influence the natural frequencies of the structure. However, the mass is usually determined by requirements of structural strength, installed equipment and cargo capacity, and designers have less flexibility in changing it significantly to affect the natural frequencies.

## 5.3 ADJUSTING FREQUENCY RATIO

Frequency ratio refers to the ratio of excitation frequency to the natural frequency of the structure (local or global). Resonance occurs for the frequency ratio equal to one, which should be avoided. The frequency ratio can be changed by either changing the excitation frequency, or by changing the natural frequency of the structure. As noted in Section 5.2, the natural frequency of the structure can be changed by either modifying its stiffness or its mass. Increasing the stiffness is the most common and preferred approach to changing the natural frequency. The excitation frequency can be changed by changing the RPM of the rotating and reciprocating machinery source, or by changing the RMP of the propeller or its number of blades.

### 5.3.1 HULL GIRDER FREQUENCY RATIO

The hull girder vertical vibration modes, usually with four or five nodes, may be excited to significant levels by the large vertical second order external moment of the low-speed main diesel engine. To a much lesser degree, the lowest two-node vertical vibration mode, may be excited by springing or whipping. These vibrations are usually not a problem from the habitability point of view, but may cause problems with structural fatigue. The vertical vibrations of the hull caused by springing or whipping can give rise to the resonance of various slender installations on the deck, such as flare tower.

During the concept design stage, it is important to estimate the hull girder natural frequencies of at least the first four vertical modes (up to the mode with five nodes). The natural frequency of the first (two-node) vertical vibration mode can be estimated using, for example, the Kumai empirical formula [16] as noted in [11]. Higher mode natural frequencies can also be estimated using, for example, the Johannessen and Skaar formula [17], but with less accuracy compared to the natural frequency of the first mode. For main engine-induced vertical hull girder vibrations, especially for *PRU* above 220 Nm/kW, twice engine RPM at full power should not fall within 20 percent range around the natural frequencies of any of the first four modes. For springing and slamming induced vertical hull girder vibrations, the natural frequency of slender deck installations on spread-moored ship-shaped hydrocarbon production and process systems should not fall within 20 percent of the natural frequency of the first vertical hull girder vibration mode.

If a 20 percent separation between the excitation frequency and the natural frequency of the structure cannot be achieved during the concept design stage, a more advanced estimation of the natural frequencies using 3D finite element analysis, followed by forced vibration analysis, is recommended during the detailed design stage.

### 5.3.2 SUPERSTRUCTURE FREQUENCY RATIO

Propeller excitations cannot generally excite the hull girder natural vibration modes to excessive levels but may excite vertical natural modes with more than seven nodes to relatively low vibration levels. These modes can be excited by the vertical pressure on the bottom shell above the propeller, or axial alternating thrust forces that create a moment relative to the hull girder neutral axis. These low-level vertical vibrations of the hull may serve as the base excitations for the superstructure and other attached subsystems that may fall into resonance with the blade rate propeller excitations, leading to excessive vibration levels. The superstructure fore and aft vibration are of the most concern. Since the hull and the superstructure are inherently coupled, all hull girder vibration modes will also excite the superstructure. However, it is the propeller-induced hull girder modes with more than seven nodes that are most likely to cause superstructure resonance and excessive vibrations.

Beginning in the 1960s, ship designers relocated the superstructure blocks of most cargo ships from amidships to the stern for reasons of longitudinal strength as well as cargo handling and maximization. However, placing the superstructure above the propeller and engine room machinery had a negative impact on vibrations. At the same time, the propeller surface and bearing forces became larger due to higher wake nonuniformity caused by fuller sterns, and the superstructure blocks became taller to allow unobstructed views over the bow from the aft-located navigation bridge. The resonance of the superstructure and other subsystems installed on the deck of the ship with full power blade rate excitations should be avoided by making sure that the blade rate frequency does not fall within 20 percent range around the superstructure fore and aft vibration natural frequency.

The fundamental fore and aft natural frequency of the superstructure is mainly determined by its geometry, the arrangement of longitudinal bulkheads, the support structure at the main deck and its total mass. Usually, the natural frequency of the superstructure is within six to ten Hz and may be estimated using semi-empirical methods, such as the method of Hirowatari and Matsumoto [18] as indicated in [11]. Resonance with the full power blade rate propeller excitations can be avoided by changing the RPM of the main engine, usually by selecting a different engine or by changing the number of propeller blades. If other design considerations prevent changing the engine RPM or the number of propeller blades, the natural frequency of the fore and aft vibration mode of the superstructure can also be changed by increasing the stiffness of the superstructure foundation. The semi-empirical methods also enable estimating the necessary superstructure foundation stiffness to avoid resonance. Changes to the superstructure stiffness are realistically only possible at the concept and detailed design stages of the vessel. Once the vessel is in operation, required stiffness changes would usually require major structural modifications and may not be feasible.

Sometimes the fore and aft vibrations of the superstructure can be caused by the transverse H-type vibrations of the main engine at the firing frequency (number of cylinders x RPM). Transverse bracings (stays) can be installed between the engine upper platform and the casing side. Transverse bracings primarily increase the H-mode and X-mode natural frequencies of the engine-foundation system to levels above the normal operating range of the main engine. This can help to avoid large H-type and X-type engine resonant vibrations and minimize the vibration responses throughout the engine room and the superstructure.

As already mentioned, alternating thrust at full power blade rate can excite vertical hull girder modes with more than seven nodes which typically serve as the basis for the superstructure fore and aft vibrations. Moreover, the alternating thrust can also cause main engine L-mode resonance. To avoid it, main engine top bracing in the longitudinal direction can also be installed. Figure 11 shows the three basic main engine vibration modes.

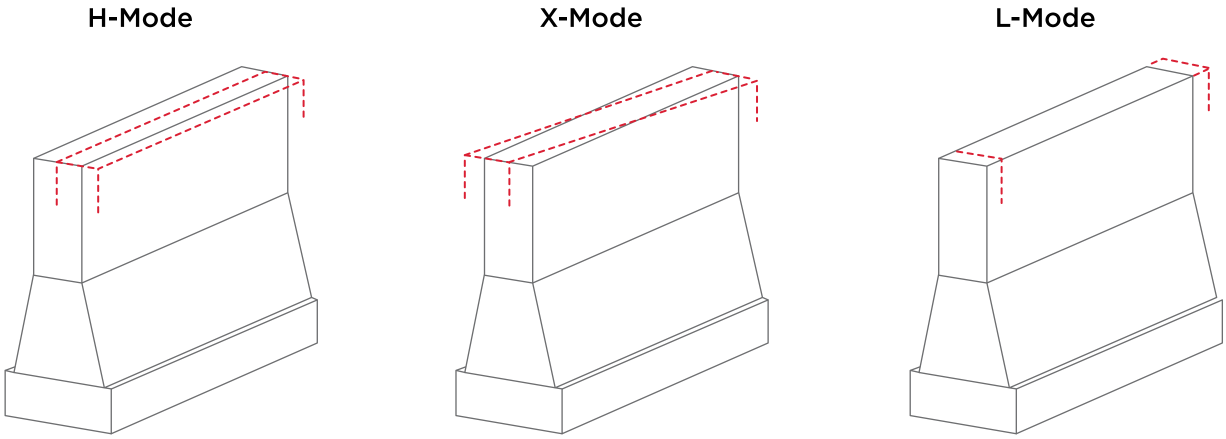



*Figure 11 – Basic main engine vibration modes*

### 5.3.3 FREQUENCY RATIO OF MAJOR SUBSTRUCTURES

Other larger substructures may also have natural frequencies similar to the superstructure and may start to resonate at the blade rate or engine firing frequencies. Examples of such substructures are large span decks aft of amidship and bridge masts. The resonance can be avoided by increasing the stiffness and, thereby, the natural frequency of these substructures. Navigation bridges may be fitted with additional structs or brackets, while large span decks can be modified by increasing the scantlings of primary structural members and stiffeners, by increasing their number, or by installing additional vertical supports like pillars or longitudinal and transverse bulkheads. Instead of increasing the natural frequency of bridge wings by increasing their stiffness, the natural frequency can also be lowered by adding mass at the ends of the bridge wings. Natural frequencies of substructures can be estimated using simple analytical or semi-analytical methods or by using local finite element models with appropriate boundary conditions. The natural frequencies can also be estimated using full ship global finite element free vibration analysis.

### 5.3.4 FREQUENCY RATIO OF LOCAL STRUCTURES

Local superstructure stiffened panels or panels in the aft end/engine room close to the propeller, the engine, or other machinery items may also start to resonate at blade rate, twice the blade rate, or even higher harmonics. A common problem is cracking of the freshwater tank panels in the aft peak. Stiffened panel natural frequencies can be evaluated using analytical or local finite element analysis and should include the outfitting mass of the panel or the added mass of fluid. The resonance can be avoided by increasing the stiffening of the panel in terms of scantlings and/or number of stiffening elements.

## 5.4 DAMPING

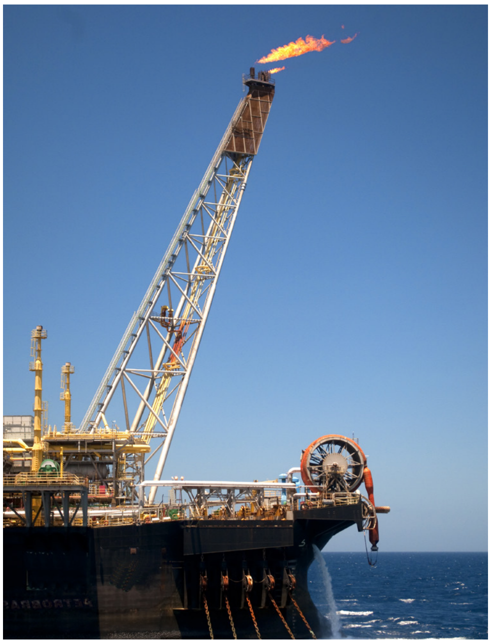
Damping in ship structures comes from external hydrodynamic damping and internal structural and cargo damping. Hydrodynamic damping is usually negligible compared to structural and cargo damping. Structural damping comes from material energy absorption in each hysteresis loop during cyclic loading. Residual stresses and stress gradients increase structural damping which means that the quality of workmanship will influence damping. This introduces a lot of uncertainty in precisely determining damping which remains the weakest point in the vibration analysis.

Damping in ships is generally small and the vibration response is approximately damping independent, except when very close to resonance where the vibration responses are inversely proportional to damping. Furthermore, damping is difficult to increase and precisely control for ships and ship-like structures. Therefore, increasing damping is the least effective way to reduce vibration responses.

Medium- and high-speed marine engines, as well as other machinery items, can be connected to their foundation by resilient mounts which isolate them from the rest of the structure. Resilient mounts have elastic elements but may also have damping characteristics to reduce the transfer of vibrations from machinery to the structure. Hydraulic top main engine bracings have damping characteristics and allow adjustments to different loading conditions of the ship. Axial and torsional dampers are usually installed at the end of the main engine crankshaft to prevent the engine generated longitudinal and torsional crankshaft vibrations to transfer to the rest of the propulsion shaft.

## 6. CASE STUDIES

The following subsections present four different case studies conducted by ABS in recent years to address various vibration issues. Their summary serves to raise awareness of vibration issues identified on modern vessels and to highlight various methods for vibration assessment and mitigation of identified issues.



### 6.1 FLARE TOWER RESONANCE

Fatigue cracks appeared at the foundation of a flare tower installed on a floating production storage and offloading (FPSO) vessel after a short period in operation. Large vibrations were previously reported by the crew. Therefore, the flare tower was instrumented with accelerometers to measure the dynamic responses. A significant vibrational displacement was measured at the top of the flare tower. At the same time, wind, waves and draft of the FPSO were also recorded.

A Fast Fourier Transform (FFT) analysis of the measured data has been performed identifying three distinct peaks in the response spectrum (see Figure 12). The first peak is related to the rigid body motion of the ship caused by waves. The second peak occurred at the natural frequency of the two-node vertical vibration mode of the hull girder, while the third peaks occurred at the first longitudinal natural vibration mode of the flare tower. Because the hull girder and the flare tower are coupled, the hull girder two-node mode vertical vibrations will be transferred to the flare tower. However, even larger flare tower responses were observed at the flare tower natural frequency that was excited by the hull girder. It soon became apparent that the excessive flare tower vibrations were mainly caused by the resonance phenomena as the flare tower natural frequency is very close to the natural frequency of the hull girder two-node vertical mode.

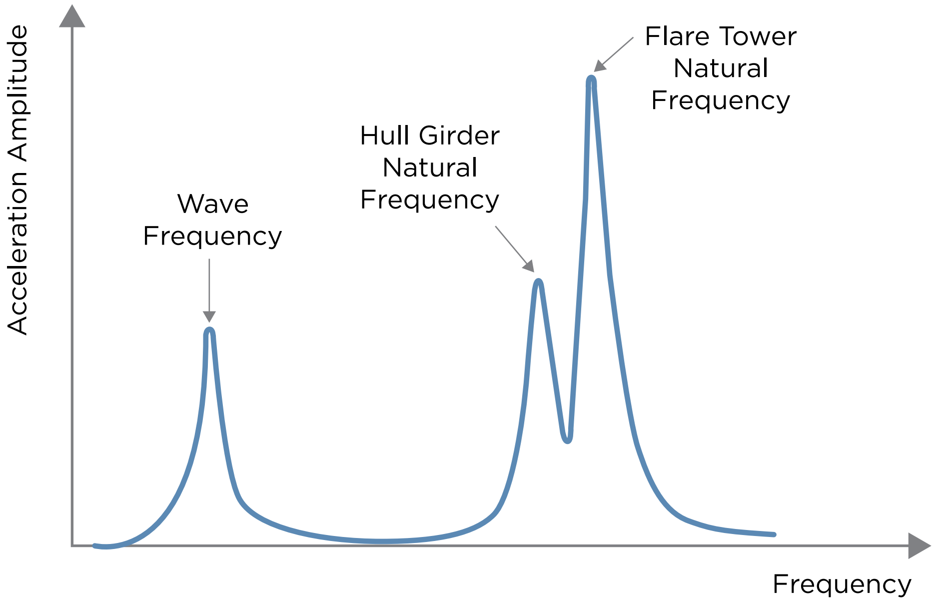



*Figure 12 – Flare tower amplitude response spectrum*

The FPSO was spread-moored at the site. Without any excitations from the main engine and the propeller, the vertical two-node vibration mode could only be explained as the whipping response to slamming impacts. The observed structural damage in shell plates and stiffeners in way of the aft end also indicated the presence of strong stern slamming.

A fatigue analysis has been conducted using both low-frequency (rigid body motions in waves) and high-frequency (vibratory response to slamming) responses. A Rule-based approach has been used for the low-frequency responses. For the high-frequency responses, actual irregular time series of stress have been used to count the number of stress cycles at different magnitudes of stress range using the rainflow counting method. The stress time series have been obtained from measured displacement time series at various locations on the flare tower. This procedure assumes that the total displacement of the flare tower at any point in time can be represented as a linear combination of its different natural modes. The same is true for the total stress. The relative displacements and stresses of natural modes have been obtained by performing the free vibration analysis using a local finite element of the flare tower together with its foundation and appropriate boundary conditions. An accurate prediction of the stress time series has been obtained by considering only several natural modes. More information about this measurement-based approach can be found in Section 4/3.2 of the ABS *Guidance Notes on Fracture Analysis for Marine and Offshore Structures* [19].

The fatigue damage caused by low- and high-frequency responses were combined to get the total fatigue damage. The analysis showed that the fatigue life of the critical locations, where the damage was observed, was less than one year. It has also been observed that the major contribution to the fatigue damage came from the high-frequency vibratory responses of the flare tower induced by the whipping of the hull girder. As the draft of the FPSO decreases, its total mass decreases so that the natural frequency of the vertical two-node mode increases and moves closer to the natural frequency of the flare tower. Therefore, the vibratory response of the flare tower was found to be more severe for lighter drafts.

Following the fatigue analysis, a permanent repair strategy was formulated which consists of renewing the cracked plates, reinforcing the foundation area and flare tower cords by adding brackets and other reinforcements and softening the bracket toes. The fatigue analysis of the repaired structure has been conducted to show that all critical locations, including the cracked locations in the original structure, meet the fatigue criteria for the intended operations profile of the FPSO.

It is realized that the resonance of the slender deck installations induced by the hull girder vibrations should be addressed as early as possible in the design of the vessel. Flare tower natural frequencies can be calculated using a simple analytical method or the more advanced finite element method. Estimating the hull girder natural frequencies is not a straightforward process unless a reliable empirical formula exists. As part of the effort to avoid flare tower resonance problems in the future, ABS formed a Joint Working Group (JWG) involving FPSO owners and operators with the objective of calibrating a formula for estimating FPSO hull girder two-node vertical vibration mode natural frequency. Various nonlinear regression models have been fitted to measured or calculated two-node vibration natural frequency data obtained for various FPSO vessels at various drafts. The best fit was obtained using the well-known Kumai formula (see Section 5.3.1), but its parameters were calibrated specifically for FPSO vessels and introduced into the ABS *Guidance Notes on Ship Vibration*. Additionally, a new requirement that the natural frequencies of the first two modes of the flare tower are to be separated from the two-node vertical hull natural frequency by at least 20 percent has been added to the ABS *Rules for Building and Classing Facilities on Offshore Installations*. The requirement applies to ship-type units with spread mooring and which are susceptible to slamming or springing.

## 6.2 RUDDER RESONANCE

Unexpected vibration-induced cracks appeared on the rudders of a series of vessels. Four subsequent rudder designs have been tested on actual ships to mitigate the vibration response of the rudders by reducing the excitation amplitude, reducing the lever arm of the excitation and local strengthening of the rudder. Vibration measurements have been completed on all five types of rudder design (the original and four different modifications).

As part of the root cause analysis (RCA), ABS performed measurement data analysis, structural vibration analysis and computational fluid dynamics (CFD) analysis of rudder trailing edge vortex shedding to identify the root cause of the cracks and evaluate different rudder designs with respect to their structural vibration performance.

A modal decomposition method, as described in Section 6.1, was used to convert the measured rudder displacements into rudder stresses. Only several natural modes of the rudder were used. It has been found that the annual fatigue damage at the crack locations was high and it was concluded that the rudder trailing edge vortex shedding was inducing rudder resonant vibration.

In addition to calculating the fatigue damage on the rudder where the cracks were first identified, fatigue analysis has also been completed on other rudder types. A fatigue damage index has been developed for relative comparison between the rudders. The analysis showed that all subsequent modifications progressively lowered the vortex shedding and the overall vibration levels. The vibration levels of the last design iteration (which included all the previous modifications) were found to be insignificant and not likely to develop fatigue cracks.

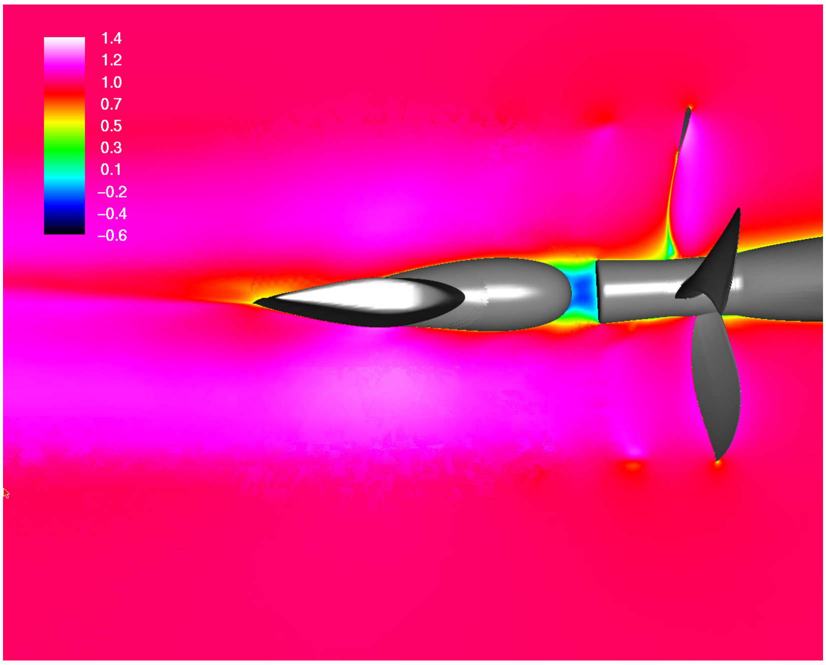



*Figure 13 – CFD analysis of the rudder design*

## 6.3 PANAMAX CONTAINERSHIP STRUCTURAL VIBRATION ISSUES

Relatively high structural vibrations have been observed by the crew of a newly built containership. Vibration measurements were conducted to identify the vibration transmission paths, the cause of the vibration issues and to propose possible solutions. Even though the structural vibrations did not exceed the vibration limits on most measurement locations, several locations have been identified where the vibrations exceeded the allowable values, namely the lashing bridge, frames and plating in the engine room and main engine starting air pipe. In addition to this, relatively high relative displacements have been measured between the active and passive side of the hydraulic top engine bracings installed on the starboard side of the engine.

The main diesel engine was identified as the main source of vibrations with excitations of the $4^{th}$, $7^{th}$, $14^{th}$ and $18^{th}$ order. Propeller excitations of the $6^{th}$, $12^{th}$ and $18^{t}$h order (first, second and third blade rate frequency) were found to be less dominant.

The analysis of vibration measurements showed that the most likely contributing factor to the high structural vibrations is the softness of the stern structure, including the engine room. This was supported by an external study conducted by the engine manufacturer [20]. The study examines the effectiveness of the top engine bracings when fitting the engine inside a "soft" engine room (see Figure 14 left) and inside a "stiff" engine room (see Figure 14 right). It should be noted that the engine room structure of the containership closely resembles the structure in Figure 14 left. The study showed that the effectiveness of the engine bracings is significantly diminished if the engine is fitted inside the soft engine room which is characterized by web frames with short webs, large openings and pillars. The radial stiffness of the connection between the double bottom and web frames is very low, as well as the stiffness of the engine room platforms in the vertical and transverse directions. On the other hand, the top bracings can be very effective if they are connected to a stiff structure like the one shown in Figure 14 right. The stiff engine room structure is characterized by strong web frames and strong connections between the double bottom and web frames. The web frames are usually plated structures without any openings.

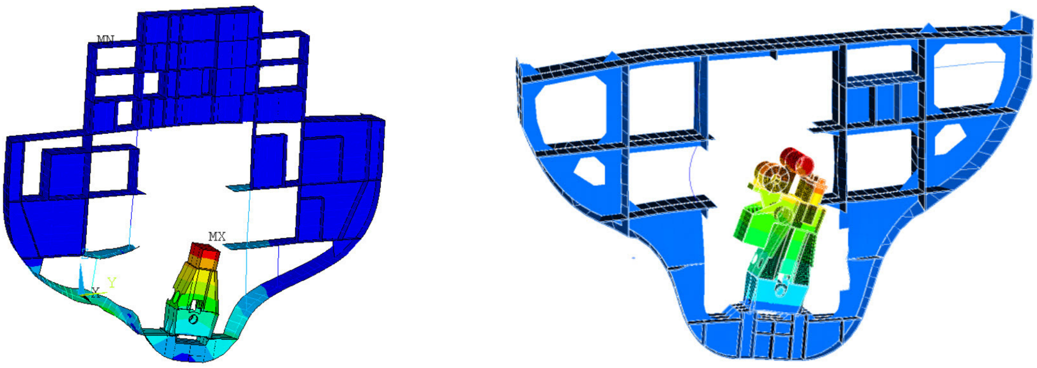


*Figure 14 – Soft and stiff engine room structure (Figure taken from [20] )*

The top engine bracings in the containership have not been connected to the hull at the engine room platform level, but at the plated structure above it, which reduced the stiffness of the connection even further.

To reduce the excessive vibration levels at the global level, the engine room structure needs to be stiffened, specifically the frames in way of the main engine. Also, the top engine bracings should be connected directly to the engine room platform or the existing connection requires further stiffening. It has also been recommended to install additional transverse engine bracings on the port side of the engine and one longitudinal bracing at the aft end of the engine. As the main engine had already been fitted with a moment compensator, further reduction of engine excitations has not been deemed feasible as it would involve the installation of a different engine or limiting the existing engine output.

Local reinforcements have also been proposed to increase the stiffness of local structures exhibiting excessive vibration levels.

## 6.4 CONTAINER STACK RESONANCE

Container stacks on decks of large containerships are inherently nonlinear systems due to the nonlinearities present in the twistlocks, lashing rods and the coupling between large stack transverse displacements and gravity forces. As containerships become larger and the container stacks and lashing bridges taller, it becomes increasingly more important to consider nonlinearities in the analysis of lashing systems. In addition to this, dynamic effects, especially potential resonance between the container stacks and hull girder, become important as well. Figure 15 shows a typical container stack with twistlocks between container corners and lashing rods that tie it down to the lashing bridge. The twistlocks and lashing rods are nonlinear structural elements: twistlocks because they contain clearances (or gaps) and lashing rods because they can only bear tension forces, while easily buckling under compression.

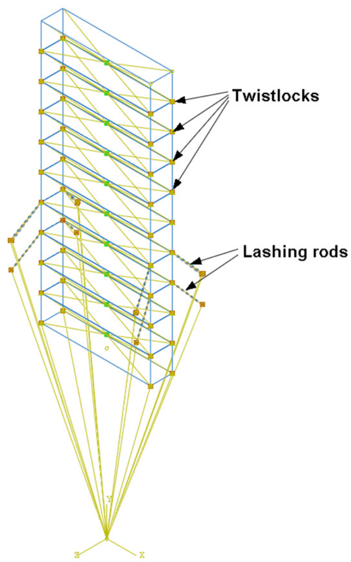



*Figure 15 – Container stack*

The standard industry approach for stack analysis is to perform a linear quasi static analysis using dynamic accelerations obtained from extreme roll and pitch motions. ABS was the first classification society to develop the fully nonlinear stack analysis procedure and the accompanying software C-Lash [21], which became the industry standard. However, dynamic effects, such as stack snapping and stack resonance, are still not considered with the quasi-static analysis procedure. In recent years ABS has developed an advanced dynamic simulation procedure to properly consider the dynamic effects during stack snapping and resonance.

Stack snapping occurs during extreme roll motions when the twistlocks between the containers suddenly open and the stack snaps and starts to oscillate at its natural frequency. The snapping usually occurs before the vessel reaches its maximum roll angle. The higher frequency oscillations due to the snap are then superimposed on the lower frequency oscillations due to roll motion. The added forces in the lashing elements due to stack snapping can be significant, as shown in Figure 16.

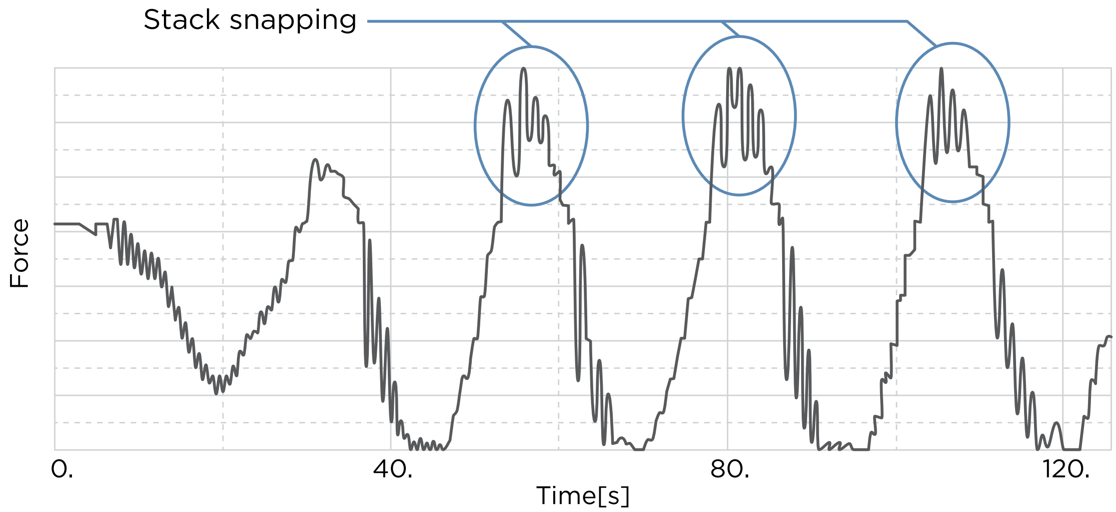
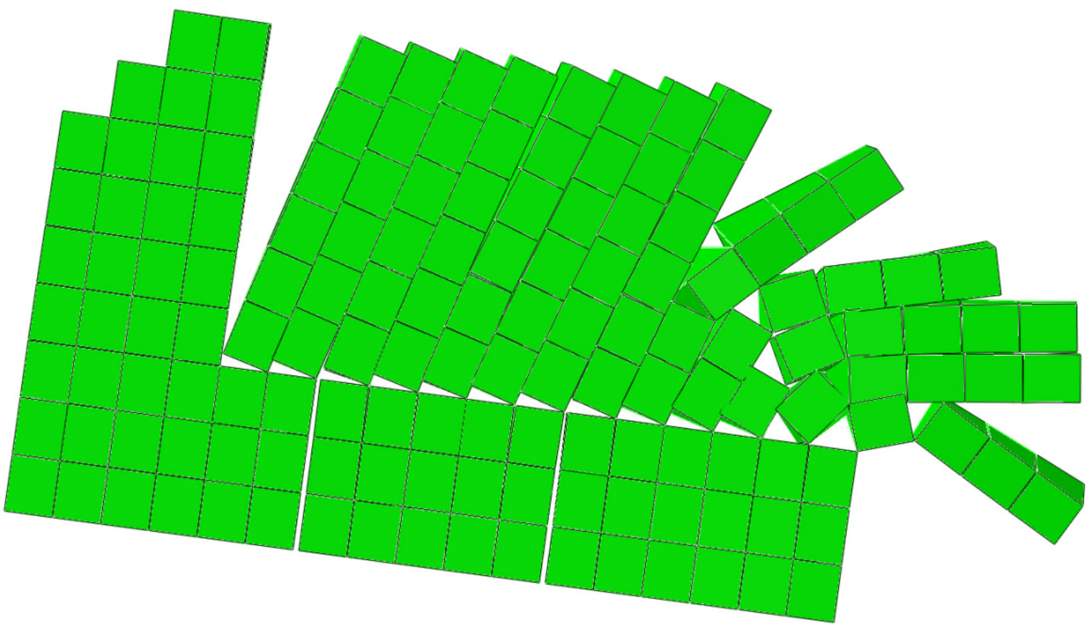



*Figure 16 – Stack snapping*

Stack resonance occurs when the natural frequency of one of the hull girder natural vibration modes coincides with the natural frequency of the stack. This can, and usually does, occur in sea states with moderate to no roll motion of the vessel. Springing and whipping of the hull girder may give rise to stack resonance as has been observed many times, especially on large vessels with tall stacks (10 or more tiers). Stack resonance can quickly give rise to the large transverse motion of the stack with displacements at the top of the stack in excess of one meter. The period of stack vibrations is usually two to three seconds.



In the case of a flare tower, the resonance can be avoided by making sure the natural frequency of the flare tower does not coincide with the natural frequency of the hull girder. The natural frequency of the flare tower, being a linear structure, can easily be calculated using modern analysis tools. However, avoiding stack resonance can be difficult for a few reasons. First, there are hundreds of on-deck container stacks on large containerships. There can be an infinite number of combinations of stack height, mass distribution, lashing pattern, lashing bridge height and container size and stiffness, all of which affect the natural frequency of the stack. The second reason is that the container stack is a highly nonlinear system whose natural frequency will also depend on the magnitude of the excitation force. Therefore, it is difficult to calculate the natural frequency of every stack on board the vessel, and chances are some of them will fall into resonance with the hull. Hence, it is necessary to design the lashing systems and prepare the stowage plans considering the possibility of stack resonance.

ABS has extensive experience in simulating stack resonance using state-of-the-are nonlinear finite element programs. Figure 17 shows the time series of a maximum twistlock force in the stack subjected to moderate harmonic roll superimposed with a high frequency transverse vibration of the hatch cover on which the stack is located. Four cases are shown on the same plot. First case (back line) shows roll only, while other cases have the same underlying roll motion superimposed with the high frequency transverse hatch cover motion of equal amplitude, but varying frequency from 0.2 Hz to 1.12 Hz. One can immediately see that the stack enters resonance when the frequency of the hatch cover transverse motion is equal to about 0.4 Hz (red line) and that its response is significantly higher compared to the response to the roll only motion. The response of the stack to roll motion superimposed with a very high frequency transverse hatch cover motion of 1.12 Hz (blue line) is also quite moderate as the excitation cleared the stack resonant frequency range.

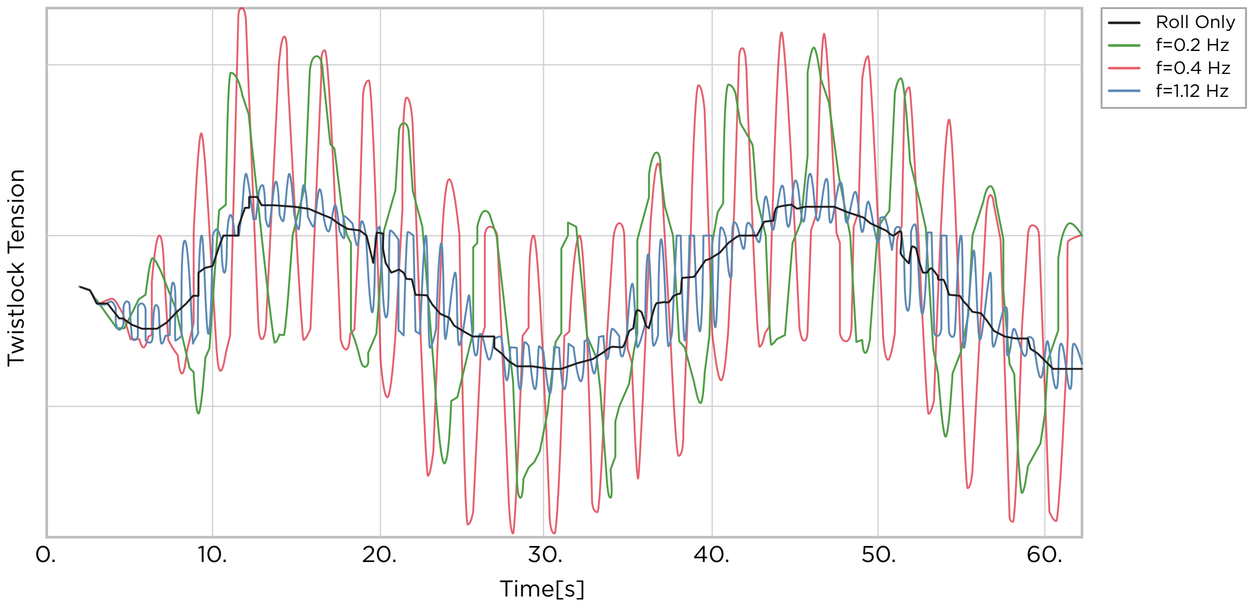



*Figure 17 – Stack resonance*

ABS currently participates in the TopTier Joint Industry Project (JIP) [22] with a number of industry partners. One of the objectives of the TopTier JIP is to use advanced simulation techniques and scaled model tests to predict stack forces during the resonance and to create the basis for future Rule developments.

## 7. ABS SUPPORT

ABS can assist owners, operators, shipbuilders and original equipment manufacturers with identification, assessment and mitigation of vibration issues found on board ships. Services offered include:

- Review of vibration measurement plan and report
- Statistical analysis of measured vibration data
- Identification of main structural vibration transmission paths
- Free and forced vibration analysis of full ship and local structural models
- Shaft axial, torsional and lateral (whirling) vibration calculations
- Noise analysis
- Propeller-induced dynamic load assessment using the computational fluid dynamics (CFD) analysis
- Container stack dynamic nonlinear analysis including the stack resonance analysis
- Vibration-induced structural fatigue analysis
- Technical assessment of different vibration mitigation measures (structural and machinery)
- Support for compliance with ABS Rule requirements and optional notations on habitability and noise
- Support for compliance with different industry standards regarding vibration criteria

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# 9. LIST OF ACRONYMS AND ABBREVIATIONS

| | |
|---|---|
| 3D | Three Dimensional |
| ABS | American Bureau of Shipping |
| ANSI | American National Standards Institute |
| CFD | Computational Fluid Dynamics |
| FEA | Finite Element Analysis |
| FFT | Fast Fourier Transform |
| FPSO | Floating Production Storage and Offloading |
| FRA | Frequency Response Analysis |
| FSI | Fluid Structure Interaction |
| IACS | International Association of Classification Societies |
| ILO | International Labor Organization |
| IMO | International Maritime Organization |
| ISO | International Standards Organization |
| JIP | Joined Industry Project |
| JWG | Joint Working Group |
| MLC | Maritime Labor Convention |
| MRA | Maximum Repetitive Amplitude |
| PRU | Power Related Unbalance |
| RCA | Root Cause Analysis |
| RMS | Root Mean Square |
| RPM | Rotations Per Minute |
| TDC | Top Dead Center |
| TRA | Transient Response Analysis |

# CONTACT INFORMATION


### NORTH AMERICA REGION
1701 City Plaza Dr.
Spring, Texas 77389, USA
Tel: +1-281-877-6000
Email: ABS-Amer@eagle.org

### SOUTH AMERICA REGION
Rua Acre, nº 15 - 11º floor, Centro
Rio de Janeiro 20081-000, Brazil
Tel: +55 21 2276-3535
Email: ABSRio@eagle.org

### EUROPE AND AFRICA REGION
111 Old Broad Street
London EC2N 1AP, UK
Tel: +44-20-7247-3255
Email: ABS-Eur@eagle.org

### MIDDLE EAST REGION
Al Joud Center, 1st floor, Suite # 111
Sheikh Zayed Road
P.O. Box 24860, Dubai, UAE
Tel: +971 4 330 6000
Email: ABSDubai@eagle.org

### GREATER CHINA REGION
World Trade Tower, 29F, Room 2906
500 Guangdong Road, Huangpu District,
Shanghai China 200000
Tel: +86 21 23270888
Email: ABSGreaterChina@eagle.org

### NORTH PACIFIC REGION
11th Floor, Kyobo Life Insurance Bldg.
7, Chungjang-daero, Jung-Gu
Busan 48939, Korea, Republic of
Tel: +82 51 460 4197
Email: ABSNorthPacific@eagle.org

### SOUTH PACIFIC REGION
438 Alexandra Road
#08-00 Alexandra Point, Singapore 119958
Tel: +65 6276 8700
Email: ABS-Pac@eagle.org





TX 06/22 22226