Dynamic Loading Approach for Structural Evaluation of Ultra Large Container Carriers

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The emerging global economic needs are driving the designs for the next generation of ocean going vessels. Current ultra-large container carrier (10,000 TEU plus) designs are considerably larger and more complex than any currently in service. Proper and rational classification assessment requires that first principles based direct calculation methods be used to augment the standard classification review. The design philosophy behind the ABS Dynamic Loading Approach enables comprehensive identification of potential failure mechanisms. The scope of the necessary engineering assessment encompass full-ship finite element analysis under non-linear sea loads, spectral fatigue analysis, finite element lashing analysis, free and forced vibration analysis, and transient and impact load analysis. This paper describes key aspects of the DLA design philosophy such as non-linear sea loads, load combinations, various applications derived from full-ship finite element analysis. Several examples are given to highlight some critical failure mechanisms to be considered for ultra-large container carriers.

KEY WORDS: full-ship finite element analysis; load combinations; non-linear dynamic sea loads; ultra-large container carriers; vibration analysis

INTRODUCTION

The designs for the next generation of ocean going vessels are largely being shaped by the demands for efficient, large-scale transportation of containers, dry bulk cargoes, oil and gas products around the globe. The challenges to meet these demands are being felt by the maritime industry as a whole. Central to these challenges is that the overall structural integrity standard that has to be maintained.

The structural designs of ultra-large vessels are also driven by a continual process of reducing the total production costs through design innovations, within which optimizing steel weight is one of the design objectives. Examples of design innovations are much wider spacing between transverse frames, fewer main supporting members and removal of secondary or tertiary stiffening members. Each and every one of these new structural features may potentially lead to in-service structural problems that are harder to identify at the design stage solely based on classification society rules.

Proper and responsible classification assessment requires that first principles based methods be used to rationalize the structural response, especially for new innovations in design. Inherent in this assessment must be the capability to practically integrate a seakeeping hydrodynamic analysis with a full-ship finite element analysis in a technically consistent manner. ABS has pioneered this approach to ship structure understanding by introducing the Dynamic Loading Approach (DLA) fifteen years ago (Liu, et al, 1992). Recently DLA has been enhanced to address the non-linear loading phenomena associated with ultra-large container carriers.

This paper first describes the design trends of ultra-large container carriers. The design philosophy for ultra-large container carriers is discussed to give an overview of the scope of the necessary engineering assessment. The inherent limitations of the traditional classification criteria and linear seakeeping analysis are highlighted by comparing non-linear sea loads in DLA based on a large amplitude motion computer program (LAMP) with the IACS wave hull girder loads and the
results from a linear seakeeping analysis. The uniqueness of DLA enhanced by LAMP to assess the structural integrity of ultra-large container carriers is explained within the context of identifying the non-linear, dynamic loading effects on structural response. Examples are given to highlight some critical failure mechanisms to be considered for ultra-large container carriers.

**DESIGN TRENDS**

Current ultra-large container carrier designs are considerably larger and more complex than any currently in service. The average container ship size has tripled since 1970. Ten years ago, a 4,000 TEU container ship represented the cutting edge of available technology. Today, vessels reaching twice that capacity are a reality and designs for 10,000 TEU or greater are being considered at every major shipyard in the world. As described below, these designs are pushing the boundaries of classification society rules and, in fact, in some cases are falling outside the application envelope. These ultra-large vessels behave very differently from their smaller counterparts. It is neither prudent nor technically responsible to simply extrapolate classification society rules to address these structures.

The design trends of container carriers can be characterized by the principal dimensions (length overall, breadth, depth and draft) and are driven primarily by the continual expansion of port facilities. Ultra-large container carriers are expected to be the workhorses for the huge container hubs that are strategically positioned across the globe.

Fig.1 shows vessel length overall plotted against design TEU capacity. For Panamax container carriers of 32.2 m in breadth with design capacity less than 5,000 TEU, the length overall is normally about 295 m. Post-Panamax container carriers extend the length overall from 275 m for 5,500 TEU to about 340 m for 9,500 TEU with breadth over 45.6 m. For ultra-large container carriers, the length overall seems to have been capped around 400 m, the length limit of some existing new-building docks but considerably longer than that of the largest oil carrier in service.

![Fig. 1 Vessel Length Overall and TEU](image1)

The hull girder natural frequencies of post-Panamax container carriers or larger, being inversely proportional to \(L^{3/2}\), can fall within the frequency range of bowflare slamming forces. Therefore whipping induced hull girder loads should be explicitly considered for these vessels. Length overall can also influence the positioning of the deckhouse in order to meet the SOLAS visibility requirements. For an ultra-large container carrier, the deckhouse is more likely to be shifted towards the midship region. This shift has design implications on the most critical hatch corners fore and aft the deckhouse. In addition, a slender deckhouse sitting on a flexible hull structure is more likely to resonate under vibratory forces from the main engine that provides the necessary shaft power to maintain the vessel’s service speed. Hence, forced vibration analysis should be an integral part of the engineering assessment.

Vessel breadth also changes in direct response to design TEU capacity. Fig. 2 illustrates vessel breadth and on-deck container rows varying with design TEU capacity. In addition to the breadth restrictions of Panama and Suez Canals, crane outreach at container ports is a limiting factor. Today, several large ports are being equipped with cranes with maximum outreach up to 64 m while a dozen or so ports can handle container carriers with over 20 on-deck container rows. It can be seen that crane outreach is no longer a design constraint on ship breadth from the viewpoint of container handling operations.

Ultra-large container carriers are being designed with a breadth over 50 m to maximize on-deck container carrying capacity. With a typical double side width less than 3 m, the open deck structure of an ultra-large container carrier is intrinsically more flexible than its smaller counterparts, resulting in higher hatch opening distortion. Reduced forebody deck area for the purpose of maximizing container capacity in forebody cargo holds can further aggravate this problem. Increased breadth for higher on-deck container capacity can lead to significant bow flare and an overhanging stern which are the contributing factors for non-linear motions and sea loads. Also with increased breadth, the double bottom structure becomes a greater load bearing member, and the outboard portion of double bottom floors can be critically stressed.

![Fig. 2 Vessel Breadth and TEU](image2)

The draft limits at some container hubs are being extended to receive ultra-large container carriers with a draft up to 16 m. The vessel depth is generally selected based on design in-hold stack weight and container
strength characteristics. For ultra-large container carriers, it is not uncommon to have 10 tier container stacks in cargo holds. As can be seen from Fig. 3, the change in depth is incremental. For qualitative comparison, the cross sections of some post-Panamax container carriers are also shown in Fig. 3. There is limited scope for further expanding ship depth. The lower portion of side transverse webs is likely to be a critical area with increasing depth.

**DESIGN PHILOSOPHY**

To address the design issues discussed in the previous section, the necessary structural assessment has to be a multifaceted process, as reliance on the structural requirements in classification society rules alone is not adequate. The scope of engineering assessment depends on a number of factors, such as, hull form, structural configuration and materials used. Typically, the recommended engineering analyses include:

- **Full-ship finite element analysis integrated with large amplitude seakeeping analysis.** The overall hull structural strength is assessed under non-linear sea loads for compliance with yielding and buckling strength criteria.

- **Bowflare slamming and stern slamming analysis.** The scantlings in the forebody and stern regions are explicitly strengthened against slamming pressures calculated in conjunction with seakeeping analysis.

- **Springing and whipping analysis.** Ultra-large container carriers, being inherently flexible, can be excited by steady state wave loads (springing) and transient bowflare slamming impacts (whipping) in two-node or higher modes. Springing generally presents no overall strength problems but contributes to fatigue damage. Strengthening against whipping induced damage in the forebody region is essential for long term structural integrity. Whipping induced hull girder loads can be estimated from the large amplitude motion calculations which take into consideration the hull flexural stiffness.

- **Spectral fatigue analysis.** Spectral fatigue analysis verifies the fatigue strength of critical structural details such as hatch corner inserts, hatch coaming termination brackets and side shell longitudinal connections subject to the hot spot stress ranges. The fatigue evaluation should also take into account additional fatigue loads due to springing.

- **Finite element lashing analysis:** When encountering oblique waves, hatch openings in ultra-large container carriers experience large distortions that can adversely affect container lashing. Such distortions are predicted from the full-ship finite element analysis and should be accounted for in the hatch cover and lashing bridge design.

- **Free and forced vibration analysis.** With increasing principal dimensions, the natural frequencies of the deckhouse and stern structures in ultra-large container carriers tend to be in the range of engine and propeller induced vibratory forces. The natural frequencies and forced vibration velocities can be determined from a full-ship finite element dynamic analysis.

- **Shaft alignment optimization.** Ultra-large container carriers are often to be designed with engine room located closer to the midship region, resulting in a much longer shaft. From the full-ship finite element analysis, it is possible to predict the deflection ranges of the local shaft foundation for optimizing shaft alignment.

From the above engineering analyses, potential structural design problems can be mitigated to avoid costly redesign. Fig. 4 shows the critical structural areas in an ultra-large container carrier that require special attention at the design stage. The linking of these various analyses is accomplished through the Dynamic Loading Approach.

**Dynamic Loading Approach**

The ABS Dynamic Loading Approach represents a first principles systematic dynamic loads and strength assessment procedure to evaluate ship structural strength under realistic dynamic load conditions. The concept was comprehensively described in Liu, *et al.* (1992) for tanker design and has been successfully used
for other ship types such as bulk carriers, LNG carriers and FPSOs. The dynamic loads are calculated by performing seakeeping analyses with differing wave headings and frequencies. In so doing, the phasing of the sea loads imposed on the hull structure due to the vessel motions can be rationally accounted for. These dynamic loads are then used in a finite element analysis of the entire hull structure. This analysis process places great emphasis on wave induced loading, ship motions, internal structural and cargo inertia loadings, and most importantly, load combinations. Ultra-large container carriers have a fine hull form, significant bow flare and overhanging stern, therefore wholesale application of the extreme-sea loads from a linear frequency domain seakeeping analysis may lead to unrealistic results. Non-linearities due to the bow and stern hull geometry, waves and motion response require that non-linear seakeeping analysis be performed. This is accomplished by using a large-amplitude motion computer program. This addition to DLA represents a positive step towards a better representation of individual load components and their instantaneous combination in realistic wave environments. Since this paper focuses on the further enhancement to DLA, the DLA analysis procedure is not further discussed.

**Large Amplitude Motion and Sea Load Prediction**

The limitations of applying a linear small amplitude seakeeping analysis in any first-principles based structural assessment can be attributed to its modeling of the 3-D hydrodynamic panel model of the hull surface to the still water line. For this reason, some methods are commonly used to semi-empirically correct non-linear sea loads. One frequently used method is to correct the total pressure field by removing the negative pressure around the still water line in the wave trough region and applying the positive pressure above the still water line in the wave crest region. In a large amplitude non-linear seakeeping analysis, the hull geometry is modeled to the deck edge and the pressure field below the wave profile is numerically defined.

The means to achieve a better description of non-linear, dynamic sea loads is through a three-dimensional large amplitude motion program (LAMP Version 4) (Shin, et al., 1997). The pressure distribution of the wetted hull surface under the wave profile is directly calculated and, therefore, more realistically accounted for than any linear seakeeping program. It can accurately predict non-linear motions, hull-girder sea loads, whipping-induced hull girder loads and pressure distributions over the wetted hull surface. The sea loads on ultra-large container carriers are calculated using LAMP time domain simulations with regular waves as well as random waves.

Within DLA, the regular wave characteristics used in the LAMP time domain simulation are identified by a linear seakeeping theory and associated with the maximum Dominant Load Parameters (DLP) in the vessel’s design service life, which is assumed operating in the North Atlantic. Typical DLPs can, for example, be vertical wave bending moment, vertical acceleration (inertia load), wave-induced torsional moment and horizontal bending moment. The regular waves are defined by wave amplitudes, frequencies and headings. The required input data for LAMP are similar to those for a linear seakeeping analysis program. The LAMP time domain simulation is performed for the regular waves associated with each DLP. The motions, accelerations and pressures from LAMP define the input for the hull finite element analysis at the instance when the DLP is maximized.

Instead of defining regular waves for DLPs, LAMP time domain random simulations generate non-linear sea loads acting on the vessel in a number of extreme sea states. Extreme sea states and headings can be determined based on time domain simulation. To do so would require numerous and lengthy time domain simulation runs for all headings and sea states. In practice, however, the extreme sea states and headings for strength verification are selected from the short term responses predicted by a linear seakeeping analysis for each sea state in the scatter diagram. The most representative response for each sea state is the significant response, i.e., the average of the 1/3 highest response peaks. Time domain simulations are then performed to determine the short-term extreme values of the targeted DLP. The design sea loads are linked to 10th probability of exceedance by assuming that the tail of the long term probability of exceedance curve can be approximated by the weighted sum of the short term probabilities for the most critical sea states.

**NON-LINEAR SEA LOADS**

Because of the assumption of wall-sideness, it has been known that a linear seakeeping theory is not suitable for predicting motions and sea loads for ocean going vessels with large bow flare, overhanging stern and smaller block coefficient. This section describes individual load components that exhibit significant non-linearities and their impacts on hull structural response of ultra-large container carriers. The non-linear sea loads are compared with the IACS wave hull girder loads and the results from a linear seakeeping analysis. Also discussed in this section is the load combination concept in the context of full-ship finite element analysis.

**Vertical Wave Bending Moment**

The ratio between wave induced sagging bending moment and hogging moment is often used to gauge the non-linearities of vertical wave bending moment. A number of recent studies examine in detail these non-linear effects. Fonseca, et al. (2005) estimated that the maximum sagging bending moments for a 175 m container carrier under some measured abnormal waves are approximately twice the maximum hogging moments. As illustrated in Fig. 5 for an ultra-large container carrier, the linear seakeeping analysis predicts
the same wave hogging and sagging bending moments. It is known that under the same design wave condition, the linear seakeeping approach tends to over-estimate the wave hogging bending moment and under-estimate the wave sagging bending moment. Also plotted in Fig. 5 are the vertical wave bending moments specified in the International Association of Classification Societies Unified Requirement S11 (IACS URS11) “Longitudinal Strength Standard” that was developed for full form vessel types such as tankers and bulk carriers. Although the IACS URS11 formula recognizes and partially reflects the said non-linear phenomenon, the limitations of applying it to container carriers with small block coefficient, large bow flare and overhanging stern are clearly evident. In the example given in Fig. 5, both the linear seakeeping analysis and IACS URS11 under-predict the design wave sagging bending moment by a significant amount. The hogging bending moment predicted by LAMP is reasonably close to the design value from IACS URS11 while the linear seakeeping approach substantially over-estimates the hogging bending moment. For full form vessel types such as tankers and bulk carriers, the linear seakeeping approach generally predicts higher hogging and sagging bending moments. The study on two unconventional oil product tankers by Parunov, et al. (2005) shows that for the North Atlantic wave environment, the linear strip theory predicts the maximum wave bending moments that are 20–30 percent higher than the design values from IACS URS11. However, in general, there still lacks an agreed upon measure to quantify the non-linear effects for wave induced vertical bending moment.

Fig. 5 Non-linear Effects on Wave Vertical Bending Moment

Block coefficient is used as an indicator of the non-linear motions and sea loads in IACS URS11 (Nitta, et al., 1992). From Fig. 6, which shows the ratio of wave sagging and hogging bending moments against block coefficient, it is clear that use of block coefficient alone is not sufficient. The data points in this figure are from the independent studies by Nitta, et al. (1992) and Jensen, et al. (1995) as well as from ABS studies using LAMP non-linear sea load data. There is considerable scatter of the bending moment ratio in this figure. This is due partly to the fact that vessels with small block coefficients do not necessarily have large bow flare. The IACS URS11 formula provides a lower bound on the ratio of wave sagging bending moment to the hogging moment. However, this ratio for ultra-large container carriers can be considerably higher.

In Fig. 6, the ratio of sagging wave bending moment to hogging wave bending moment for the seven vessels from Jensen, et al. (1995) are plotted against block coefficient. The sagging and hogging wave bending moments were calculated using a quadratic strip theory which is most appropriate for small to moderate sea states. The main limitation of this approach is associated with the perturbation procedure taking into consideration only the linear and quadratic response transfer functions. However, it is more computationally efficient than a non-linear time domain seakeeping analysis and can take advantage of some well established extreme value prediction methods.

Jensen, et al. (1995) proposed a measure of the non-linear sea loads using a coefficient for bow flare and forward speed $C_{FV}$:

$$C_{FV} = 0.5 \times F_n + C_f$$

where $F_n$ is the Froude number at the vessel’s sustainable speed in extreme sea conditions and $C_f$ is the bowflare coefficient which is defined as the ratio of the forward area difference between the upper deck plane and waterline plane divided by freeboard height and vessel length $L$. The technical merit of such a definition is that the trend of the non-linear wave bending moment can be better presented.

For a number of ocean going vessels, two of which are ultra-large container carriers, the ratio of wave sagging bending moment to hogging moment calculated by LAMP is also included in Fig. 6. These ultra-large container carriers have block coefficients less than 0.70 and bow flare and speed coefficients greater than 0.40.

Fig. 7 shows the ratio of sagging bending moment to hogging bending moment from Jensen, et al. (1995) and LAMP plotted against the bow flare and speed coefficient $C_{FV}$. For oil carriers and other full form vessel types, the bow flare and speed coefficient is normally less than 0.15. The corresponding ratio of wave sagging bending moment to hogging moment is expected to be less than 1.20. For LNG carriers with a
block coefficient over 0.7, the flare and speed coefficient is less than 0.3 and the bending moment ratio is expected to be about 1.4 to 1.5. For container carriers, this ratio is between 1.8 and 2.0.

For ultra-large container carriers, the forebody hull girder strength is normally governed by the maximum hogging still water bending moment. Designers routinely use the total hogging bending moment to control hull girder section modulus. If the non-linear sagging bending moment is included in the calculation, the total sagging moment may also be quite significant. Hence, the upper deck structure may be subject to considerable compression. This normally does not cause any problems to hatch coaming and upper deck that are of substantial construction. However, the 2nd or 3rd decks may need to be strengthened to prevent buckling damages.

Figs. 8 and 9 illustrate the effects of higher vertical wave sagging bending moment on hull structures. In Fig. 8, the envelope curves of still water hogging bending moment (SWBM) for three container carriers are normalized by the maximum still water hogging bending moment values (SWBM_{max}). Comparison of these three designs suggests that the envelope curve for Design A tapers to zero more quickly in the forebody than the envelope curves for Designs B and C.

Typically, the maximum wave hogging bending moment is of the same magnitude as the still water component in the midship region. If the sectional properties at the forebody region are sized based on the IACS URS11 wave loads in conjunction with the sharply tapered still water hogging bending moment curve as was the case for Design A, the sagging bending moment may become a very relevant load parameter for the forebody region. It is noteworthy that the forebody hull form in a container carrier also changes considerably towards the forward perpendicular. The effective strength deck area is sometimes reduced to accommodate additional forebody cargo in-hold TEU capacity. The net effects of all these design considerations are summarized in Fig. 9, with the ratio of the offered section modulus SM_{off'd} to the required section modulus SM_{req'd} at deck for Design A approaching 1.0. Therefore, it is imperative that the forebody upper deck region of Design A be explicitly checked for compliance with the buckling criteria. The curves with triangles and squares in Fig. 9 represents the maximum and minimum envelope curves for still water bending moment that are normalized by the maximum still water hogging bending moment (SWBM_{max}), respectively.

Wave-Induced Vertical Shear Force

The non-linear effects on wave-induced vertical shear force can also be significant for vessels with large bowflare. In general, the non-linear phenomenon on vertical wave shear force is less well understood. However, it is known that the predicted wave-induced vertical shear forces from a linear seakeeping analysis are higher than the design values specified in IACS URS11. IACS is working to increase the URS11 wave shear forces.

Due to the large bowflare and overhanging stern in a container carrier, the vertical wave induced shear forces due to sagging waves are much higher than the wave shear forces due to hogging waves. This is because in sagging waves, the wave crests are situated at the bow and stern, creating large upward pressures at
the two ends and resulting in much higher vertical shear forces. In hogging waves, wave troughs are positioned at both ends and the wave-induced vertical shear forces increase slightly when compared to the IACS vertical shear forces.

As illustrated in Fig. 10, LAMP predicts the known non-linear effects. In this figure, the vertical wave shear force predicted by LAMP is normalized by the maximum vertical wave shear forces from IACS URS11. Linear seakeeping theory gives positive and negative wave-induced shear forces of equal magnitude. Thus linear seakeeping forces due to hogging waves overestimate the negative wave shear force in the forebody region and the positive wave shear force in the afterbody.

The container still water loading conditions commonly have a negative still water shear force in the forebody region and a positive still water shear force in the aftbody region. The structural requirements due to vertical shear force in regular waves are largely controlled by the negative shear force in the forebody region and the positive shear force in the aftbody.

For ultra-large container carriers, wave-induced torsional moment is one of the key load parameters which can greatly affect the accuracy of the full-ship finite element analysis. Comparison of the wave-induced torsional moments from classification society rules and LAMP can demonstrate the technical merits of the latter. Most rule formulae for wave-induced torsional moment were developed through extensive calibration of the design load and strength criteria, generally using 2-D strip theory seakeeping programs. Fig. 11 shows a comparison of the wave-induced torsional moments calculated from the ABS rule formula and LAMP. The maximum wave-induced torsional moments at five selected locations along the vessel length were predicted using LAMP time domain random wave simulation. The wave-induced torsional moment curves for three most critical load cases were also calculated using LAMP time domain regular wave simulation. The predicted torsional moments from these two methods show good correlation both in magnitude and distribution pattern. The results also confirm that the maximum wave induced torsional moment in the forebody region is about 30 percent less than the maximum at the aftbody region. The envelope curve from the ABS Rules correctly captures this trend. From Fig. 11, it can be seen that the maximum torsional moments calculated by LAMP are about 30 percent higher than the rule values. This difference can be attributed to the non-linear effects due to the fine hull shape, large bow flare and overhanging stern of ultra-large container carriers.

**Load Combinations for Structural Analysis**

The combination of individual load components with due consideration to load phasing is one of the most important elements of the Dynamic Loading Approach. Within DLA, the loading parameters for each load case are external pressure (hydrostatic and hydrodynamic) over the actual wetted surface, vessel motions and accelerations defined by the equivalent wave. Internal pressure (static and dynamic) and inertia forces can be directly calculated from the instantaneous vessel motions and accelerations.

![Fig. 10 Envelope Curves for Hull Girder Wave Shear Forces](image)

![Fig. 11 Envelope Curves for Wave Induced Torsional Moment](image)
artificial forces to a full-ship finite element model to simulate the instantaneous torsional moment distribution along the ship, the basic load parameters are defined by external pressure, internal pressure, inertia forces, motions and accelerations. The hull girder loads such as bending moment, shear force and torsional moment are the derived load effects from these basic load parameters and computed solely for the purpose of verifying the adequacy of individual load components and their instantaneous combinations.

For head sea load cases in DLA, a combination of wave-induced vertical bending moment and shear force can be checked. As an example, three key elements of a DLA load case for wave hogging moment are shown in Fig. 12. First, the full-ship finite element model in a design equivalent wave (see Fig. 12a) is in dynamic equilibrium. The wave length is about 95 percent of the length between perpendiculars; the wave crest is located amidships with a wave height 2.35 percent of the wave length. Second, the external pressure distribution on the shell plate shown in Fig. 12b is mapped onto the external shell plating with a maximum pressure amidships under a hogging wave. Last but not the least, the wave-induced vertical bending moment and shear force are calculated from the external pressure, internal pressure and inertia forces. From the magnitudes of the individual load components that are non-dimensionalized by the IACS design loads, the adequacy of the individual load components and their combination can be confirmed. Further visual check can be made based on the following two points:

- The maximum vertical wave bending moment occurs at zero vertical wave shear force;
- The vertical wave shear forces and bending moments at two extreme ends of the vessel approach zero.

For oblique sea load cases, five wave load components are vertical bending moment (VBM), vertical shear force (VSF), horizontal bending moment (HBM), horizontal shear force (HSF) and torsional moment (TM). The distribution curves along the vessel length for these load components are normalized by their respective rule values. For each load case, these curves represent the instantaneous combination of these load components.

In oblique waves, the full-ship finite element model is in dynamic equilibrium in a design equivalent wave that is very different from that for the maximum vertical wave bending moment (see Fig. 13a). The wave length is about 45 percent of the length between perpendiculars; the waves approach the vessel from the starboard side at an angle of 60 degrees from the stern. Viewed from the starboard side, the two wave crests are located at the stern and some distance from the fore perpendicular with a wave height 3.42 percent of the wave length. The external pressure distribution on the hull surface shown in Fig. 13b is mapped onto the external shell plating with a maximum pressure located just forward of the midship region.

All the wave-induced hull girder loads shown in Fig. 13c are calculated from the external pressure, internal pressure and inertia forces at an instant in time domain when the wave induced torsional moment is maximum at 0.3L from the after perpendicular. From the magnitudes of the load components that are non-dimensionalized by the rule wave-induced hull girder
loads, the adequacy of the individual load components and their combination can be confirmed. From these curves, the following points can be observed:
- The maximum horizontal wave bending moment occurs at zero horizontal wave-induced shear force;
- The horizontal wave shear forces and bending moments at two extreme ends approach zero;
- The wave-induced torsional moment is distributed along the vessel length approximately the same manner as the horizontal wave shear force.

Where $M_X^b$ is the wave-induced torsional moment of the cross section at the baseline, $P_Y$ is the horizontal wave shear force, and $z_s$ is the vertical distance from the shear center to the baseline. The above equation was discussed by Pedersen (1985). $M_X^b$ consists of a hydrostatic contribution caused by an asymmetric static load distribution and a hydrodynamic one due to waves. The plausible explanation for similar distribution curves for wave-induced horizontal shear force and torsion moment is that $M_X^b$ is relatively small and $M_X$ is dominated by the term $P_Yz_s$.

The instantaneous load combination curves that result when the 60 degree waves shown in Fig. 13a move pass the vessel, are plotted in Fig. 14 with a minimum wave-induced torsional moment in the aft-body. It should be noted that the wave-induced hull girder loads in Figs. 13a and 14 have opposite signs. This occurs because the vessel is under a wave hogging bending moment in this case.

**STRENGTH ASSESSMENT BASED ON DYNAMIC LOADING APPROACH**

This section describes the strength assessment of a recently designed ultra-large container carrier under non-linear sea loads described in the previous section. The strength adequacy of the hull structure is examined by a finite element method using a three-dimensional (3-D) global model representing the whole ship and finer mesh models for critical areas such as hatch corners and termination brackets. The nodal displacements from the 3-D global model are used as the boundary load effects in the subsequent fine-mesh analyses of the local structures.

The full-ship finite element analysis provides the necessary stress and displacement data for yielding and buckling evaluation of all the primary hull structural members. The allowable von-Mises stress is taken as 95 percent of the minimum material yield stress. The plating and stiffeners of tank/hold boundaries are also checked against the buckling criteria with due consideration to nominal corrosion design values. For the main supporting members, yielding and buckling are the two failure modes that should be examined. For this purpose, the finite element mesh size for modeling the
main supporting members is of one longitudinal spacing or finer.

For each critical structural area, a local finite element model of progressively finer meshing in way of the fatigue hot spots is required. The mesh size of plate thickness dimension is necessary to capture the hot spot stresses and stress ranges. For every critical hot spot, both the total stress level and accumulative fatigue damage should be controlled. This can be achieved through a fine-mesh finite element analysis under non-linear sea loads and a spectral fatigue analysis. The allowable stress limit is a function of the mesh size that is required to predict the hot spot stress distribution.

**Full-ship Finite Element Analysis**

For ultra-large container carriers, the following loading conditions are typically examined:

- Heavy container full load condition;
- Light container full load condition;
- Heavy container alternate load condition.

The Dominant Load Parameters (DLP) for these loading conditions are vertical bending moment, horizontal bending moment, forward vertical acceleration, roll amplitude and torsional moments at several locations along the vessel length. For each dominant load parameter, two load cases are analyzed to examine the structural responses corresponding to the maximum and minimum DLP values.

**Fig. 15 Deformed Hull Structure (Head Seas)**

The deformed full-ship finite element model shown in Fig. 15 is for the heavy container full load case with the maximum vertical hogging bending moment as a DLP, the corresponding load combination curves for this load case are shown in Fig. 12. The results from this load case are used for strength assessment of the bottom structure under biaxial compression due to vertical hogging bending moment and external pressure. It can be a critical load case for examining the web frames under external pressure amidships. For this load case, the shear strength of the side shell and inner longitudinal bulkhead may also be critical at the quarter distances from the extreme ends. The full-ship finite element model under sagging waves is also be analyzed for assessing the buckling strength of the upper deck region, especially in the fore-body region.

The deformed full-ship finite element model shown in Fig. 16 is for the heavy container full load case with the minimum torsional moment as a DLP, the corresponding load combination curves for this load case are given in Fig. 14. Verification of the structural response for this load case can be performed by checking the load combination curves against the deformed finite element model (Fig. 16a) as well as the deformed centerline profile (Fig. 16b) and the main deck (Fig. 16c). As the vessel’s loading condition has a still water hogging bending moment, the deformed centerline profile exhibits the correct deformation pattern under a total hogging bending moment (still water plus wave components). The same can be said about the deformed main deck under a negative horizontal bending moment. Although classification society rules often specify some load combination factors for wave-induced vertical bending moment, horizontal bending moment and torsional moment, the load combination concept in the Dynamic Loading Approach is considered most comprehensive. This load case allows the accurate strength assessment of a number of critical structural members such as the upper deck transverse box structure and the hatch corners forward and aft of the engine room.

Hatch opening distortion is another critical parameter to be considered in the extreme torsional moment condition. The extreme distortion for each hatch opening should be accounted for when designing hatch covers, lashing bridges and container securing system. Hatch opening distortion can be measured in various ways. The sum of the absolute diagonal changes of a hatch opening at the hatch coaming top level is one commonly used measurement. The relative displacement $\Delta U$ as defined in Fig. 16d offers an alternative measure. The greater the hatch opening distortion, the more critical the hatch corners. Moreover, the buckling strength of the cross-deck box structures should also be checked. The high compressive stresses in these structures can be induced by the combined effects of the still-water and wave-torsional moments in conjunction with other active load components. The relative movement between the corner of a hatch cover and the coinciding hatch coaming point is the most conventional way for determining the required tolerances for hatch covers.

**Finite Element Lashing Analysis**

As the number of in-hold container rows grows from 16 for a typical post-Panamax container carrier to over 20 for an ultra-large container carrier, the hatch opening distortion $\Delta U$ for the former is about 20 cm and can increase by as much as 70 percent for the latter. Excessive hatch opening distortion is not only indicative of possible strength problems at hatch corners, hatch termination brackets and cross deck box structures but may also cause operational problems for hatch covers and container securing. Use of the hatch opening distortion results from the DLA full-ship finite element analysis is critical for examining the functionality of the container securing system. For the purpose of illustrating the concept, five container stacks sitting on
one hatch cover and lashed to the lashing bridges are shown in Fig. 17a. The deformation of the hull structure and container stacks in response to the maximum relative movement between the hatch cover and coaming are shown in Fig. 17b.

![Deformed Hull Structure](image)

**Fig. 16 Deformed Hull Structure (Oblique Seas)**

In the finite element lashing analysis, all the container stacks for one entire deck bay are modeled and the loads applied to the model are the forced displacements from the full-ship finite element analysis and the vessel motions and accelerations from the seakeeping analysis. Unlike conventional lashing calculation methods, the finite element lashing analysis explicitly model the stiffness parameters of the containers, lashing bridges, lashing rods, hatch covers and hull structure. Lashing rods are assumed to have zero stiffness when subject to compression. The finite element lashing analysis allows the detailed examination of some typical failure modes associated with the lashing system such as container racking, topping, corner post collapse, separation of structural components and breaking of lashing rods.

**Shaft Alignment Optimization**

Optimization of shaft alignment is a derived application of the DLA full-ship finite element analysis. For an ultra-large container carrier, the design trend is to locate the engine room and deckhouse closer to the midship region due to visibility considerations, resulting in a longer shafting assembly. The relative deformation of the double bottom structure supporting the main engine and shaft assembly can be accurately calculated by the DLA full-ship finite element analysis and directly used to investigate the shaft bearing reactions, tail shaft bearing to tail shaft contact and shafting stresses.

**Fine-mesh Fatigue Strength Analysis**

The mesh size of the full-ship finite element model is generally not sufficiently refined to accurately predict high stress concentrations but the stress results can be used to screen some critical areas for subsequent local fine-mesh finite element analyses. If the geometric shapes of some critical areas cannot be sufficiently represented in the full-ship finite element model, as is the case for hatch corners in the upper deck region, the global- or coarse-element stresses should be kept well below the minimum material yield stress. In so doing, the local hot spot stresses from the subsequent fine-mesh finite element models can be effectively controlled.

Some critical details in ultra-large container carriers are well established. The main deck hatch corners forward and aft of the deckhouse should always be analyzed in detail. The hatch corners amidships at the hatch coaming, main deck and 2nd deck should also receive sufficient attention during the design process. The hull girder load combination curves discussed in the previous sections help identify the most significant load component for each critical structural detail.

In many cases, the stresses in some critical details due to torsional moment are not obvious to design engineers unless the results from both the full-ship and fine-mesh finite element models are available. The critical structural detail in Fig. 18 represents the hatch corner in a partial deck aft of the deckhouse. This partial deck is below the 2nd deck level. At this hatch corner, the torsional moment is the only significant load component that causes high stress concentrations.
Hatch corner insert plates with recessed double curvatures offer a practical way to reduce the hot spot stresses.

![Fig. 18 Hatch Corner in Partial Deck](image)

Since there exists a strong correlation between dynamic stress and fatigue life, the first step toward improving the fatigue strength of a critical detail is to control the fine-mesh stress level below the minimum material tensile strength. The subsequent spectral fatigue analysis can give a direct confirmation of the fatigue strength. The predicted fatigue life is to be a minimum of 25 years for the North Atlantic wave environment. In the spectral fatigue analysis, the vessel speed is assumed to be 75 percent of the design speed. Three representative container loading conditions are used to calculate the cumulative fatigue damage. For container carriers, full load heavy TEU, full load light TEU and alternate loading conditions are often selected. The wave-induced cyclic loads for spectral fatigue analysis are calculated by a 3-D seakeeping program with correction to the side shell areas exposed to intermittent waves.

![Fig. 19 Fatigue Life under Different Wave Headings](image)

Fig. 19 shows a polar diagram of the predicted fatigue lives for various wave headings at 30 degree intervals. Loading Condition A — a full load heavy TEU condition with the deepest draft contributes most to the cumulative fatigue damage ratio. Loading Condition B — a full load light TEU condition is also moderately critical. Loading Condition C — a partial loaded condition causes the least cumulative fatigue damage. The fatigue lives are shown in the polar diagram at 200 year intervals. Wave headings of 120, 60, 240 and 300 degrees result in low fatigue lives due to high wave-induced torsional moments.

The spectral fatigue analysis results can also be presented in terms of cumulative fatigue damage due to a range of sea states. As can be seen from Fig. 20, moderate sea states with significant wave height of 3.5–7.5 m contribute most to the cumulative fatigue damage; the contribution from sea states of 7.5–11.5 m is also significant. The cumulative fatigue damage due to very benign and extremely severe sea states is relatively small. Figs. 19 and 20 show useful intermediate results that can be used to verify the correctness of the spectral fatigue analysis. The example for the hatch corner demonstrates how the fine-mesh fatigue analysis methodology is integrated in the Dynamic Loading Approach.

![Fig. 20 Fatigue Damage in Various Sea States](image)

**Dynamic Responses Due to Wave Impact Loads and Vibratory Forces**

Ultra-large container carriers, being inherently flexible, may exhibit significant dynamic responses induced by impacting waves and vibratory forces from the main engine and propeller. In many cases, it is unavoidable to have the dynamic loading frequencies fall within ±20 percent of the structural natural frequencies. Excessive dynamic responses can pose habitability and equipment serviceability problems; for example, high vibratory velocities at the navigation deck may render some equipment inoperable or cause irritations to the crew. Wave impact loads can lead to yielding and buckling damage to the immediate areas and beyond. This section discusses free- and forced-vibration analysis, and hull girder springing and whipping analysis.

Some empirical formulae for estimating hull girder natural frequencies can provide an initial indication of possible vibration problems for ultra-large container carriers. However, the information that can be extracted from these empirical formulae is of limited use for avoidance of possible vibration problems, and the
The natural frequencies of overall hull girder structures and local structures should be carefully checked against the frequencies of vibratory forces. Fig. 21 shows the relationship between the natural frequencies for a typical ultra-large container carrier and the frequencies of some significant vibratory forces. The diagonal line in the figure indicates the coincidence of both natural frequency and frequency of vibratory force, signifying structural resonance. As described below, avoidance of structural resonance due to some vibratory forces such as whipping and springing is not always possible. The practical measure to minimize possible structural damages is to strengthen forebody structures against these dynamic loads.

**Dynamic Whipping and Springing Responses**

The lowest hull girder natural frequencies for asymmetric and symmetric mode shapes may be at or below 0.40 Hz (or 2.5 seconds in natural period) for ultra-large container carriers. The duration of a bowflare slamming pressure pulse can be close to the lowest vertical hull girder natural period. As indicated in Fig. 21, whipping causes resonance at the lowest hull girder natural frequencies. The hull girder vibration in response to bow flare slamming impact on the forebody structure can induce a high sagging bending moment and positive shear force at the instant of impact, with reduced peaks of bending moment and shear force in subsequent response cycles due to damping. If the scantlings are not explicitly designed with due consideration for whipping, high sagging bending moment and positive shear force may cause buckling and yielding in the forebody deck region and longitudinal bulkheads.

Springing is a steady state vibratory response of hull girder structure under wave loads in moderate sea states. For ultra-large container carriers, the encountered wave frequencies are also likely to be close to the lowest hull girder natural frequencies (see Fig. 21). It is found that additional hull girder loads induced by springing are relatively low and hull structures have sufficient strength to withstand the combined effects of hull girder loads due to regular waves and springing. The primary concern in this case is the possibility for cumulative fatigue damage to deck structural members under many springing induced stress cycles. This cumulative damage should be explicitly accounted for in the fatigue strength assessment.

**Engine and Propeller Induced Forced Vibrations**

Because of the engine power and propeller thrust required to maintain the design service speed which is usually higher than 24 knots, the vibratory forces from the main engines and propellers of ultra-large container carriers are expected to be significantly higher than the forces experienced by their smaller counterparts. From data sheets provided by engine manufacturers as well as on-board vibration measurements, vibratory forces of some orders can be ignored while other orders need to be included in a forced vibration analysis. In the example illustrated in Fig. 21, the 1st and 2nd order engine vibratory forces may be within ±25 percent of the hull girder natural frequencies but their force magnitudes are negligibly small. As a result, the forced vibration levels are insignificant. The frequency range of vibratory forces of the propeller 2nd blade rate is well outside the hull structural natural frequencies. Therefore, the main focus of the forced vibration analysis for some ultra-large container carriers is the vibration levels due to the 3rd and 4th order engine induced vibratory forces and the 1st blade rate propeller induced vibratory forces.

The deckhouses of some ultra-large container carriers are relatively slender and may also be elevated higher to meet the SOLAS visibility requirements. Proper design consideration should therefore be given to the stiffness of the deckhouse by installing longitudinal bulkheads aligned with the hull structural members below and providing adequate support to restrain movement of the bridge wings.
Fig. 22 shows the predicted single amplitude velocity at the bridge wing (port side) under the 3rd order engine induced X-moment (see Fig.22 for the unbalanced moment exerted on the main engine). The maximum velocity in the longitudinal direction occurs at some engine RPM. The acceptance criteria used in the figure is defined by ISO 6954 “Mechanical Vibration and Shock Guidelines for the Overall Evaluation of Vibration in Merchant Ships”. In many cases, ultra-large container carriers are to be designed to a higher vibration standard. The predicted velocities due to other vibratory forces can also be very high. Therefore, it is crucial that forced vibration analysis be included in the design and assessment of ultra-large container carriers.

![Fig. 22 Single Amplitude Velocity at Port Side Bridge Wing under 3rd Order Engine Induced X Moment](image)

### CONCLUSION

Ultra-large container carriers present engineering challenges to the marine industry in terms of maintaining overall structural integrity standards. The sheer size of these vessels and many production-friendly innovations are outside the application envelope of classification society rules. It is also not prudent to rely on linear seakeeping and static balanced finite element analysis approaches.

Some current ultra-large container carriers (10,000 TEU plus) can be characterized by a length overall over 350 m, a breadth and a depth to accommodate over 18 rows and 10 tiers in hold. The hull has a small block coefficient and is designed with a large bowflare and overhanging stern to maximize the on-deck container loading capability.

To achieve the desired accuracy, the sea loads acting on ultra-large container carriers have to be predicted by a large amplitude motion program. It is not sufficient to analyze a full-ship finite element model without a proper description of highly non-linear sea loads such as vertical wave bending moment, shear force and torsional moment.

The wave sagging bending moment for a typical large container carrier is expected to be twice the wave hogging moment. The ratio of sagging to hogging moment is 1.4 to 1.5 for LNG carriers and less than 1.2 for fuller form vessel types. Under sagging waves, the maximum vertical wave shear force can be more than 50 percent higher than the IACS design wave shear force while under hogging waves, the wave shear force is about 25 percent higher. The maximum torsional moment predicted from the Large Amplitude Motion Program (LAMP) can be 30 percent higher than the results from linear seakeeping analysis.

For a full-ship finite element analysis, proper combination of all the components with due consideration to the component phase relationships is the key to ensure that the strength evaluation is rational and free from artificially imposed forces. The results from a full-ship finite element analysis can be used to:

- Verify the strength of the hull structure against buckling and yielding;
- Identify critical structural areas for subsequent fine-mesh finite element analyses for strength and fatigue evaluation;
- Optimize shaft alignment;
- Facilitate the design of lashing bridges and hatch covers;
- Analyze on-deck container securing system under maximum hatch opening distortion.

In addition to the full-ship finite element analysis, hatch corners and hatch termination brackets on the deck structure and some partial deck should be evaluated by spectral fatigue analysis. For ultra-large container carriers, these details cannot be easily designed solely based on experience and many of them require hatch corner insert plates with recessed double curvatures.

The full-ship finite element model should also be utilized to evaluate the dynamic characteristics of the hull structure against the vibratory forces from transient or impact wave loads, main engine and propeller. Ultra-large container carriers, being inherently more flexible than their smaller counterparts, may become vulnerable to:

- Buckling and yielding under whipping induced bending moment and shear force,
- Additional cumulative fatigue damages due to springing induced stress cycles;
- Excessively high vibrations in deckhouse under 3rd and 4th order engine-induced vibratory forces;
- Excessively high vibrations in stern structure under 1st blade rate propeller-induced vibratory forces.

Proper and responsible classification assessment requires that a first principles based methodology be used to rationalize the structural response of a dynamic nature. The Dynamic Loading Approach presented in this paper offers a rational design philosophy for ultra-large container carriers.
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