TECHNOLOGY ADVANCES IN DESIGN AND OPERATION OF LARGE CONTAINER CARRIERS

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SUMMARY

Extra design efforts are imperative for enhancing structural integrity and safe operations of large container carriers that possess significant bow flare and low torsional rigidities due to the open deck structural configuration. Proper and rational classification assessment requires that first principles based engineering calculation methods be used to augment the standard classification review. The scope of the essential engineering assessment should encompass full-ship finite element analysis under non-linear sea loads, spectral fatigue analysis, transient and impact load analysis, finite element lashing analysis, parametric roll prevention and vibration analysis. Design and operation of large container carriers are beginning to benefit from technology advances in hydrodynamics and structural analysis. The design philosophy for large container carriers is first discussed to give an overview of the recent technology advances. Application examples of spectral fatigue analysis, finite element based lashing analysis and hull vibration are provided to illustrate the measures to enhancing structural integrity and safe operations of large container carriers.

1. INTRODUCTION

Sea-borne transportation of containers has been shaped by the desire for greater economic efficiency, an invisible hand in the market economy. Technology advances in design and operations of large container carriers have brought to realization many industry firsts. The increase in container carrier size reflects earlier advancements in that technological developments can indeed help shipowners gain operational and cost advantages.

Containerization emerged as a sea-borne transportation concept from the mid-1950s in the United States to reduce ship time at dockside, cut cargo handling costs and prevent petty pilferage. The American Bureau of Shipping (ABS) has been actively developing technology to support the classification of these industry firsts [1][2]:

- first full container carrier converted from the C2 vessel, the Ideal X in the mid-1950s,
- Sea-Land 7 class container carriers, the fastest ever built with twin engines/screws in 1969,
- American New York and 11 sisterships, the largest container carriers in 1984,
- American President Lines (APL) C-10 container carriers, the first post-Panamax size in 1988 (see Figure 1),
- M/S Emma Maersk, the largest container carrier over 11,000 TEU in 2006 (see Figure 2).

The technology advances that supported the classification of these industry firsts include the ABS Dynamic Loading Approach (DLA) – a “design-by-analysis” procedure for more accurate modelling of expected shiploads and dynamic stresses than traditional methods. With ample service experiences gained from the first generation of post-Panamax container carriers since 1988, compliance with the classification standards from a well recognized IACS member can ensure adequate long term structural integrity for container carriers of similar size. The ABS Classification Rules for container carriers were developed from a SafeHull dynamic load concept originated from the Dynamic Loading Approach, calibrated against in-service experience, scaled model tests and ship-borne real time measurements [3].
However, the current classification standards have not been fully calibrated for ultra large container carriers over 10,000 TEU, although real-time monitoring on board large container carriers will facilitate the understanding of structural performance [4]. Through a direct comparison of the principal dimensions of the first Post-Panamax container carrier (Figure 1) and the largest built (Figure 2), some intuitive observations on the design challenges can be revealing. The ultra large container carrier in Figure 2 possesses an inherently more slender hull structure, with greater L/D and L/B slenderness ratios than the Post-Panamax carrier in Figure 1. Between these two designs, the increase in container TEU capacity is more striking, over 140%. Considerable gain in TEU capacity is achieved through larger deck areas, hence significant bow and stern flare. The technical challenges posed by ultra large container carriers are discussed in the paper together with the technology solutions. With the next frontier for the industry moving towards the 18,000 TEU Malacca-Max design, the technology philosophy discussed in the paper is becoming more pertinent.

2. TECHNICAL CHALLENGES AND RATIONAL DESIGN PHILOSOPHY

Expansion of container carrier size is an on-going trend that is being dictated by container transportation infrastructure and manufacturing capability envelope such as port draft restrictions, container stack height limitations, container terminal cranes and main engine horsepower, all of which are being improved. For naval architects involved in design and operations of ultra large container carriers, careful consideration should be given to some specific technical challenges as highlighted in Figure 3.

To address these design challenges, the essential structural assessment has to be a multifaceted process, as reliance on the structural requirements in classification society rules alone may not be sufficient. 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 [5]:

- **Full-ship finite element analysis integrated with non-linear 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. Whipping induced hull girder loads can be estimated from the non-linear motion calculations that take into consideration the hull flexural stiffness.
- **Spectral fatigue analysis.** Spectral fatigue analysis verifies the fatigue strength of critical structural details such as hatch corners, hatch coaming termination brackets and side shell longitudinal connections subject to the hot spot stress ranges.
- **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 with single deckhouse are often designed with engine room located closer to the midship region, resulting in a much longer propeller 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. The linking of these various analyses is accomplished through the Dynamic Loading Approach.
3. STRUCTURAL DESIGN CONSIDERATIONS

3.1 LOCATION OF DECKHOUSE AND ENGINE ROOM

Deckhouse and engine locations are typically co-located in most recent container carriers, even in the largest 11,000 TEU container carrier in service. Some ultra large container carrier design concepts (up to 13,000 TEU) sponsored by classification societies typically call for separation of these two spaces for two primary reasons:

- The Navigation Bridge can be brought forward to improve visibility.
- The length of open deck between forward and aft houses can be controlled to limit hatch opening distortions.

Although these reasons are theoretically valid, it is technically feasible to keep the traditional single engine/screw arrangement for ultra-large container carriers to a rather large size. This single island configuration circumvents possible extra costs associated with vessels built with twin engines/screws.

Nevertheless some less talked-about benefits can be gained from separation of deckhouse and engine room, which generally helps control hatch opening distortion and stress levels in critical areas. The 18,000 TEU Malacca-Max design is likely to feature the two-island deckhouse/engine room configuration.

3.2 OVERALL HULL GIRDER STRENGTH

The open deck configuration of ultra large container carriers presents challenges to structural engineers. The operational demands are pushing the designs into areas in which there is little direct service experience and many of the structural design features of ultra large container carriers fall outside the application envelope of the existing Rules of classification societies.

Especially pronounced for ultra large container carriers, large hatch openings leave relatively little deck area to provide for the main hull girder strength. For the 18,000 TEU Malacca-Max design, the final structural arrangement will be defined by combinations of thick deck plates of high strength steel, continuous hatch coamings and possible inboard longitudinal girders that result in less onerous field stress levels and hatch opening distortion.

3.3 LOCAL STRENGTH

The breadth of the 18,000 TEU Malacca-Max design is likely to approach 60 m. Given that the cargo bay length remains constant since it is largely governed by the standard length of containers, the aspect ratio of the double bottom within a cargo bay is becoming increasingly skewed toward a wider section with fewer floors and many longitudinal girders that intersect the vertical girders of the transverse bulkheads. The structural interaction of these major members should be properly evaluated for adequate strength against yielding, buckling and fatigue.

For ultra-large container carriers, it is not uncommon to have 10 tier container stacks in cargo holds. The lower portion of narrow side transverse webs is likely to be a critical area with increasing depth.

The structural design of ultra large container carriers is also driven by a continual process of reducing the total production costs through design innovations, in 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.

3.4 FATIGUE CRITICAL AREAS

In way of the forebody, midship and aft-body regions, hatch corners at the coaming top, main deck and lower decks can be critically stressed to the acceptance limits for yielding or fatigue, under combined action of vertical, horizontal and torsional moments. For ultra-large container carriers with the single island arrangement of deckhouse and engine room, some critical hatch corner inserts can be substantially thicker than what have been used in post-Panamax container carriers. With the two-island arrangement, the required thickness will be somewhat smaller due to better control of hatch opening distortion. However, all the hatch corners immediately

![Figure 3 Critical Areas in Ultra Large Container Carriers](image-url)
forward and aft of the deck house or engine room space will become critical from a fatigue standpoint.

For evaluation of the fatigue critical areas, the full spectral fatigue analysis in conjunction with a full ship finite element model and local fine mesh models has a distinct advantage over those spectral fatigue analysis methods involving some artificial boundary constraints, hull girder load effects and patch unit loads.

As shown in Figure 4, the forebody structure from the top of hatch coaming to the 2nd deck is modelled for fatigue strength evaluation of the critical areas such as: the hatch coaming termination brackets and hatch corners at the coaming top, main deck, intermediate deck and 2nd deck. This model allows rapid evaluation of the fatigue strength of more than 30 critical details.

The fatigue lives of the hatch coaming termination bracket and hatch corner highlighted in Figure 4 are slightly above the design limit of 20 years based on the North Atlantic wave environment. Many classification societies have specified some worldwide wave environments as the standard design basis for their spectral fatigue analysis methods. For these critical details, the fatigue lives calculated based on the worldwide wave environment are typically more than 40% longer than those based on the North Atlantic wave environment. The North Atlantic wave environment should be kept as the design principle for spectral fatigue analysis.

### 3.5 BOW AND STERN REGIONS

Bow and stern regions of container carriers are often exposed to significant transient loads resulting from bow flare or stern slamming impacts. These impact loads and subsequent whipping responses of the hull structure will be more pronounced for ultra-large container carriers. Special attention should be paid to slamming affected structural members such as shell plating, side shell stiffeners, main supporting members and bracket connections. In addition, whipping induced hull girder bending moments and shear forces may also induce buckling damage away from the immediate slamming impact areas. Hull structural response to whipping may also induce high accelerations that in turn may dislodge worn or corroded locking devices and lead to loss of containers overboard.

### 4. HATCH OPENING DISTORTIONS

For a given hatch opening, there are two common ways to measure hatch opening distortion caused by hull girder torsional moment. \( \Delta U \) measures the relative longitudinal displacement between longitudinal hatch coamings on the port and starboard sides. \( \Delta U \) is essentially an overall measure of hatch opening distortion, indicative of torsional rigidity and the magnitude of warping stress in way of hatch corners and cross deck boxes. Figure 5 graphically illustrates \( \Delta U \) and also shown in the same figure the plan view of the distorted hatch coamings of an ultra large container carrier. There are twenty cargo bay openings along the vessel’s length covered by hatch covers, four hatch cover pieces for each opening.

Another useful measure of hatch opening distortion is \( \Delta R \), the relative movement between the corner of the hatch cover and the coinciding point on the transverse hatch coaming. For container carriers without in-board longitudinal girders, an in-board hatch cover piece is normally rested on cross deck boxes while an outboard piece is additionally supported by longitudinal coamings. Hatch covers are attached to the hull structure through a one two-directional stopper (longitudinal and transverse) on one edge and one transverse stopper on the other. Therefore they are purposefully designed to be free from undesirable interference by hull structural deformation. Apart from friction between the contact pads fitted to both hatch covers and hatch coaming top, hatch cover pieces are expected to experience rigid body movement, sliding and rotating against the coaming top. The in-board hatch cover pieces tend to experience higher relative movements, as the cross deck boxes experiences local bending under hull girder torsional moment. Hatch covers and lashing bridges should be positioned and designed with due consideration to \( \Delta R \).
For smaller container carriers, hatch opening distortions do not normally present design and operational problems for hatch covers and lashing bridges. For example, the maximum $\Delta U$ for a typically Panamax container carrier is about 160 mm in the midship region. $\Delta U$ is gradually reduced to three quarters of that value in the forebody region and half of that in the aft-body region. $\Delta R$ depends on the number of hatch cover pieces fitted for each cargo bay. In the midship region, the maximum $\Delta R$ is about 40 mm with three hatch cover pieces. Fewer hatch cover pieces are installed to relatively narrower hatch openings in the forebody region. Therefore $\Delta R$ in the forebody region can experience the same level of relative movement.

For an ultra large container carrier over 10,000 TEU, the deformed hull structure together with hatch covers are shown in Figure 6. In this case, the maximum torsional moment occurs in the aft-body region. The horizontal axis denotes the cargo bay number starting from 1 for No. 1 cargo bay and ER stands for engine room. The maximum $\Delta U$ is 310 mm, representing 94% increase over the maximum value for a Panamax container carrier. Both the forebody (No. 3 Bay) and midship regions (No. 10 Bay) may experience a similar level of hatch opening distortion. Hatch corners and cross deck boxes in these cargo bays are more critical and should be examined in detail. Although the hatch opening distortions just forward and aft the deckhouse are 25% lower than the values in other regions, higher warping stresses are expected as a result of the extra rigidity exhibited by the closed main deck of the engine room/deckhouse.

There are two parameters that control hatch opening distortions: load and torsional rigidity. From the load side, torsional movement is directly proportional to vessel length and draft, as well as square of breadth. Regarding torsional rigidity, the cargo block length, vessel breadth and arrangement of deckhouse and engine room can have significant impacts on hatch opening distortion. To a lesser extent, depth/width of side structure and width of cross deck box also influence hatch opening distortion.

The relative movement for each cargo bay in the same vessel is given in Figure 8. The distribution of $\Delta R$ along the vessel length is largely consistent with that of $\Delta U$, with the distinct values in the midship and fore-body regions. The maximum value for $\Delta R$ is 62 mm if each cargo bay is fitted with four hatch cover pieces for each of the 20 cargo bays except the first bay (see Figure 5). The relative movement for each hatch cover piece would be substantially higher if three hatch cover pieces are fitted to each cargo bay. Since the cranes at some large container terminals are capable of lifting up to 60 tonnes, fitting three hatch cover pieces per cargo bay is technically feasible for container carriers with less than 10,000 TEU. However, consideration is to be given to stowage of these relatively large hatch cover pieces at container terminals.
For ultra large container carriers over 10,000 TEU, both hatch opening distortions $\Delta U$ and relative movements $\Delta R$ will be substantially higher, if the single island arrangement for deckhouse and engine space is adopted. For the 18,000 TEU Malacca-Max design, separation of deckhouse and engine room will create two relatively rigid blocks along the vessel length. The distribution patterns for $\Delta U$ and $\Delta R$ will be very different from those shown in Figures 7 and 8.

5. SAFE, EFFICIENT CONTAINER STOWAGE

Large scale sea-borne transportation of containers has a proven track record. For smaller post-Panamax container carriers, container securing systems deployed in holds and on deck can be satisfactorily reviewed using the standards from classification societies. During the approval of a container securing manual, container stacks in hold and on deck are systematically evaluated against various limit states such as container corner post collapse, container racking, twistlock breaking, corner casting failures and others. Load cases for roll/heave and pitch/heave are considered in the standard review.

Care should be exercised when applying the same approach to container stowage arrangements for ultra-large container carriers. A number of design challenges that should be taken into consideration are:

- Excessive hatch opening distortions discussed in the previous section.
- Prevention of container losses due to parametric roll. The industry is beginning to understand this relatively new phenomenon which is considered responsible for damages or losses of on-deck containers.
- Stowage of 20’ containers in 40’ cell guide and on deck. OSHA requirements in the U.S. stipulate that stevedores are prohibited to work on a container, and that securing 20’ containers using longitudinal stacking cones in 40’ container bays is not allowed in U.S. ports unless fall protection is provided.
- Higher lashing bridges introduced to secure containers stowed at higher tiers. These lashing bridges are more flexible, and have to cope with extra loads exerted as a result of excessive hatch opening distortions.
- Economic advantages of pursuing higher tier container stack weight. Attempts have been made to examine container stowage on deck up to 10 tiers.
A finite element based lashing analysis has been developed by ABS to address the above technical challenges and to augment the standard review of container securing systems. Such an approach considers all the flexibility properties in the lashing systems (see Figure 9).

Each container is modeled by beams and plate elements with due consideration to its stiffness and weight distribution. Lashing rods are explicitly modeled as non-linear rod elements which feature tensional stiffness and zero compressive stiffness. Also included in the overall model are twistlocks with a predefined gap of separation under tension. The hatch covers and lashing bridges are explicitly modeled with relative fine mesh plate elements. However the hatch cover assembly is attached to the hull structure to allow its movement against the hull structure. From the DLA-based full ship finite element analysis, the hatch opening distortion, vessel motions and inertia forces are the input parameters defined for evaluation of the lashing system.

The evaluation results for the container assembly identify the high forces in the container stacks that rest on the edges of the in-board hatch cover pieces. Higher tensions are found in the lashing rods in the vicinity of the corners of a hatch cover piece that move away from the lashing bridge. When the corners of a hatch cover move towards the lashing bridges, the lashing rods in the vicinity have much lower or zero lashing forces. Generally relative movements between hatch covers and hatch coaming top are absorbed by the short cross lashing rods. The long cross lashing rods are relatively less effective against this form of relative movement. Because of the uneven distribution of stiffness, some adjacent container stacks are likely to be in contact with each other (see Figure 10).

**6. HULL VIBRATION**

Ultra-large container carriers are inherently more flexible and may exhibit significant dynamic responses induced by impacting waves and vibratory forces from the main engine and propeller. The said flexibility may be attributed to:
- Flexible hull structure of an open deck configuration associated with vessel’s principal dimensions,
- Tall and slender deckhouse structure coupled with flexible hull structure,
- Lighter scantlings as a result of striving for least weight or production costs.

Excessive shipboard vibration can affect habitability, safety and functionality of these vessels. 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, buckling and fatigue damage to the immediate areas and beyond.

ABS has developed a vibration guide to provide users, specifically shipowners, shipyards and naval architects, with practical guidance on the design, analysis and measurement procedures to vibration-acceptable design solutions [6]. Optional classification notations for passenger comfort and crew habitability can be assigned to vessels based on the ABS Guide for Crew Habitability on Ships [7].

Some well-known empirical formulae for estimating hull girder natural frequencies can provide an initial indication of possible vibration problems for ultra large container carriers. However the results from these formulae are of limited use for avoidance of possible vibration problems. The accuracy of these formulae deteriorates for high frequency symmetric or asymmetric vibration modes. If the forced vibration levels of some local structures are to be predicted with a reasonable level of confidence, a full-ship finite element analysis approach has to be adopted.
6.1 WHIPPING AND SPRINGING

The natural frequencies of overall hull girder structures and local structures should be carefully checked against the frequencies of vibratory forces. Figure 11 illustrate 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. 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 fore- and aftbody structures against these dynamic loads. For ultra large container carriers, the lowest hull girder vibration mode is often associated with torsion.

Ultra large container carriers possess hull girder natural frequencies in asymmetric and symmetric mode shapes as low as 0.40 Hz (or 2.5 seconds in natural period). For the 18,000 TEU Malacca-Max design, lower natural frequencies are expected. The duration of a bowflare slamming pressure pulse may be very close to the lowest vertical hull girder natural period. As indicated in Figure 11, 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. The buckling strength of the forebody region needs to be verified against whipping-induced sagging bending moment and positive shear force. The critical areas in the forebody region are the deck and longitudinal bulkhead members. Similar whipping strength verification should be extended to the hull structure aft of the deckhouse. However, stern slamming loads are likely to induce resonance associated with high order vibration modes.

Springing is a steady state vibratory response of hull girder structure in response to cyclic 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 Figure 11). Additional hull girder loads induced by springing represent a relatively low percentage of the total and hull structures are expected to 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.

6.2 ENGINE AND PROPELLOR INDUCED VIBRATION

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 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. 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.
of the ship. Large container carriers have slender and tall deckhouses that are prone to excessive vibration. Proper design consideration should therefore be given to the shear 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. The natural frequencies of the deckhouse and stern structures are heavily influenced by the inherent flexibility of the hull structure. The vibration modes of these local structures interact with the hull girder vibration modes. Therefore the accuracy of the vibration assessment depends critically on the quality of the full ship finite element model.

The engine-induced vibratory forces are generally well defined. If an engine is installed in accordance with the manufacturers’ specifications, unbalanced external forces are insignificant in magnitude. Most unbalanced external moments are also relatively low, except the lateral H-Moments and X-Moments of the 3rd and 4th orders, depending on the engine types. Therefore the vibration level due to these unbalanced moments should be closely predicted and verified during seatrials. Some unbalanced forces are not defined by engine manufacturers and have to be determined through calibration of the predicted values against the measurements. For example, thrust moment as shown in Figure 12 can be the results of unbalanced forces from the assembly of shaft line, engine flywheel and engine crank shaft.

The source of propeller induced vibratory forces is propeller cavitation, i.e. unsteady cavities attached to rotating propeller blades. Rapid change of the cavity volume can radiate intensive, fluctuating pressure impact on the stern shell plating. Large diameter, heavy load propellers for large container carriers are particularly vulnerable to cavitations. Central to the estimation of unsteady propeller loads, cavitation, and propeller-induced hull pressure is the prediction of stern flow. The calculation process involves computational fluid dynamics and fluid cavitation dynamics.

Through the forced vibration analysis, the predicted single amplitude velocities at some vibration sensitive locations are verified for compliance with the applicable acceptance criteria for navigation deck and crew accommodation space, machinery equipment, vibration-induced fatigue limit, and owner’s specifications. Figure 13 shows the predicted single amplitude velocities at the bridge wing (port side) under the 4th order X-moment (see Figure 12 for the unbalanced moment exerted on the main engine). The predicted values in the longitudinal, transverse and vertical directions are calculated for a range of engine revolution per minute from 80–115 RPM. The maximum velocity in the transverse direction occurs at 111 RPM, in the close vicinity of the natural frequency of the deckhouse. 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. Therefore, it is crucial that forced vibration analysis be included in the design and assessment of ultra-large container carriers.

7. CONCLUSIONS

To ensure structural integrity and safe operations of ultra-large container carriers, the application of standard Classification Rules must be augmented by the first principles based engineering analysis methods. These methods have been further developed and matured by ABS over the last two decades through many large container carrier development projects. There exist many technical challenges that have been and will continuously be encountered by naval architects during the development of ultra-large container carriers.

With the future possibility of ultra large container carriers growing to the 18,000 TEU Malacca-Max design, the necessary scope of the critical engineering assessment should encompass full-ship finite element analysis under non-linear sea loads, spectral fatigue analysis, transient and impact load analysis, finite element lashing analysis, parametric roll preventions and vibration analysis. Application examples such as spectral fatigue analysis of critical details, finite element based lashing analysis and hull vibration are provided in the paper to illustrate the measures to achieve safe design and operations of ultra-large container carriers.
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