ΕΘΝΙΚΟ ΜΕΤΣΟΒΙΟ ΠΟΛΥΤΕΧΝΕΙΟ
ΣΧΟΛΗ ΝΑΥΠΗΓΩΝ ΜΗΧΑΝΟΛΟΓΩΝ ΜΗΧΑΝΙΚΩΝ
ΤΟΜΕΑΣ ΘΑΛΑΣΣΙΩΝ ΚΑΤΑΣΚΕΥΩΝ

Σύγχρονος σχεδιασμός της μεταλλικής κατασκευής πλοίων bulk carrier
(Modern structural design of bulk carrier vessels)

ΔΙΠΛΩΜΑΤΙΚΗ ΕΡΓΑΣΙΑ

ΠΑΝΤΕΛΕΗΜΩΝ Γ. ΣΠΑΝΟΛΙΟΣ

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Αθήνα, Σεπτέμβριος 2014

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Αθήνα, Σεπτέμβριος 2014

(Υπογραφή)

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ΣΠΑΝΟΛΙΟΣ ΠΑΝΤΕΛΕΗΜΩΝ

Φοιτητής Ναυπηγός Μηχανολόγος Μηχανικός Ε.Μ.Π.

Abstract

In the ***first chapter*** of this thesis an effort is being made to familiarize the reader on the concept of the bulk carrier design. At the beginning the term bulk carrier is defined as adopted by the IACS resolutions.

A brief text follows presenting the historical evolution of the bulk carrier. Beginning with the implementation of the ballast tanks under the cargo hold in the 1850's, to the consolidation of the modern cargo hold structure in the mid 1950's and the construction of the modern giant of the bulk carrier fleet, Vale Brazil.

Subsequently, the bulk carrier fleet is classified in various ways. According to the size of the bulk carrier or the commodity they are built to carry. Furthermore classification is made by the means that are used to load or discharge their cargo. Moreover reference is made to the assigned class notations according to the IACS. Afterwards, the typical single side skin bulk carrier structural configuration is presented, noting down the nomenclature of the main structural components of a cargo hold. Finally, a fleet analysis is conducted to help the reader understand the volume of the bulk carrier fleet, its age and the perspectives of the shipbuilding in this sector.

The ***second chapter*** of this thesis outlines the design principles followed in the structural design of the cargo spaces of a bulk carrier, starting from the structural design process, that is part of the design spiral. At this stage, a stepwise process determines the structural arrangements of the ship. Then the derivation of the hull scantlings is being made, followed by the assessment of the hull girder strength. Finally, the detail design of the components ends this part of the spiral.

Some issues concerning bulk carrier design are discussed in another part of this chapter. Following a brief description of the environment and the operational tasks that a ship has to cope with, the alteration of stresses imposed in the structure is outlined. Additionally, the loading patterns of the bulk carriers are presented, illustrating the effect they have on the shear forces and bending moments of the structure.

Subsequently, the net scantling approach is presented. That part describes a process for the determination of the minimum hull scantlings that should be maintained throughout the ship's life to satisfy structural strength requirements. Furthermore, the various limit states of the structure are listed. A limit state depicts a condition for which a particular structural member or the entire structure fails to perform the function that is expected of it.

The final part of this chapter focuses on the structural arrangement principles. Details are further given for the structure of the double bottom, the side structure and the bulkhead structure, areas of major significance in the construction of bulk carriers.

The ***third chapter*** outlines the determination of loads affecting the hull structure. The first loads assessed are the ones that are present in the still water conditions, meaning in conditions where the ship floats in calm water. The main components of this category are still water bending moments and shear forces. Those static loads should be supreimposed to the wave induced loads, in order to assess the total forces that result in negligible dynamic stress amplification of the structure. IACS formulas for the calculation of those loads are presented, whereas typical distributions of allowable and attained forces in specific cases are illustrated.

Subsequently, load cases as accepted by the IACS are presented. They describe situations where under specific regular waves, the long term response values of the load components considered being predominant to the structural members. Furthermore, the findings of a study concerning the design loads on primary structural members of a bulk carrier are discussed.

The distribution of the external pressures is the following step in the prescribed assessment procedure. This includes not only the definition of the hydrostatic pressure, but also the description of the hydrodynamic loads affecting the hull. Furthermore, attention is paid to the modification of the pressures that can arise from the avoidance of heavy weather situations, conducted by the crew of the vessel.

Moreover, the presence of internal pressures that are taken under consideration is discussed. The major load component of this category corresponds to the forces due to the bulk cargo and the liquids loaded onboard.

Finally, the loading conditions section defines a number of load cases which are likely to impose the most onerous local and global load regimes that are to be investigated in the structural analysis. The hold mass curves section outlines the use of such plots in the determination of the allowable mass of cargo as a function of draught.

In the ***fourth chapter*** of this thesis, the calculations for the structural components in the midship section area of a bulk carrier are presented. This procedure is part of the preliminary design of a bulk carrier as conducted by students in the ship design laboratory of the National Technical University of Athens, as part of the preliminary design lesson of the department of Naval Architecture and Marine Engineering. The following procedure is a translation from Greek of the tenth part of the work done by Antonis Dellis, whose kind permission was requested to reproduce the material in the present thesis, and was granted.

The ***fifth chapter*** of this study copes with the strength analysis of the hull structure. At first, the Finite Element Method analysis is presented, in order to assess the strength of longitudinal hull girder structural members, primary supporting structural members and bulkheads. Additionally, using this method, detailed stress levels in local structural details can be obtained, whereas the fatigue capacity of the structural details can be determined. The typical process of structural analysis using the finite element method is discussed, while the areas of concern in bulk carrier structures are displayed.

Subsequently, the procedure for direct strength analysis is explained. The yielding strength check is discussed, while buckling and ultimate hull girder strength assessment is further analyzed. Prone to buckling areas of bulk carriers are presented, while the findings of an ultimate hull girder strength assessment are represented.

Finally, the assessment of the fatigue life of the various structural members subject to fatigue failure is presented. The types of stresses that are considered for this type of assessment are discussed, while the selection of the correct S-N curve is mentioned. Last but not least, an example of a fatigue performance analysis of bulk carriers side frame structure is noted, whereas the findings of this study are featured.

The ***sixth chapter*** of this thesis copes with the overall design of the hold area in a bulk carrier, seen from an operational point of view. From the number of holds being required considering the size of the vessel and the density of cargo to be carried, to the definition of hold length and the transverse bulkheads to be fitted. Additionally, the purpose of topside tanks and hopper tanks presence is discussed, whereas the effects that ballast water management has on strength of the structure is also mentioned. Furthermore, the double bottom arrangement is presented, focusing on the effects of double bottom height on structural behavior of the ship.

Moreover, some fuel oil tank arrangements that are used on bulk carriers are assessed in the event of oil spill, and the probability of oil outflow is measured. Finally, the contribution that hatch covers have on the strength of the bulk carriers is cited. The main hatch cover types found on bulk carriers are presented, whereas an assessment of the collapse strength of specific hatch cover types is also featured.

The following part of this chapter focuses on the diversity of cargoes that a bulk carrier is set to carry, and the various aspects of structural design that each of them affects. Ore cargoes loading rates could influence the strength of the bulk carrier, whereas liquefaction phenomena could become a cause of bulk carrier loss. Additionally, carriage of certain types of ore cargoes, under specific circumstances, could result in spontaneous combustion of the cargo. Coal cargoes, if mixed with water onboard, are notable for their corrosivity. The main problem associated with grain carriage is its tendency to shift when the ship rolls, leading to loss of stability. Moreover, steel cargoes may lead to tanktop area exceeding the maximum permissible loads assigned by the classification society. Finally, hazards associated with timber cargoes are identified and measures for safe carriage of such cargoes are listed.

Finally, the ***seventh chapter*** outlines the main alternative designs that have been implemented last decades in bulk carrier structural design, whereas the major areas of concern for the design of the future are also listed.

At first, the implementation of double side skin configuration is discussed. The benefits arising from this introduction are listed and a comparison with the conventional single side design is made. Additionally, some alternative designs proposed for the side structure area are also discussed. Strength aspects such as collision resistance and the residual strength of the structure are mentioned, whereas the reliability levels of the proposed structure in comparison with the single side structure are also described.

Subsequently, the general characteristics of a Newcastlemax ore carrier (202,500 DWT) are presented, mainly by listing the structural arrangement of such a vessel and its advantages compared to a conventional bulk carrier. Moreover, a hybrid configuration (Hycon) bulk carrier is presented, demonstrating double sides in the fore and aftmost holds, whereas the other holds remain single sided. Furthermore, the Optimum 2000 is listed, a bulk carrier providing each cargo hold with a longitudinal bulkhead. This leads to advanced strength and stiffness of the structure.

Alternative designs are then presented. The curved inner bottom bulk carrier aims to reduce local stresses in the hold area by modifying the flat inner bottom and hopper tanks with an upside down arch plate. Non ballast seawater bulk carrier (NOBS) is further discussed, a design aiming to reduce the ballast seawater used by implementing an alternate hull shape. The Ecoship 2020 is a design listing a number of proposed innovations that can lead to more flexible, cost effective, energy efficient and environmental friendly structure. Mitsubishi air lubrication system (MALS) design is then discussed, a system aiming to reduce frictional resistance of the hull. Ecore ore carrier, a 250,000 DWT ore carrier is featured, implementing the use of one centre cargo hold, and alternative use of the wing tank areas.

Finally, the variable buoyancy ship is introduced, a bulk carrier adopting solutions aiming to eliminate the transportation of ballast water around the globe. This is achieved by having trunks that extend most of the length of the ship below the waterline, which are open when ship is at speed, leading to ballast water exchange.

Keywords: «bulk carrier, structural design, midship section, cargo hold, hull scantlings, strength analysis, loads, stresses, IACS, double side skin, alternative designs »

Περίληψη

Στο ***πρώτο κεφάλαιο*** της διπλωματικής με τίτλο «σύγχρονος σχεδιασμός της μεταλλικής κατασκευής πλοίων bulk carrier», γίνεται προσπάθεια να εξοικειωθεί ο αναγνώστης με την έννοια της σχεδίασης της μεταλλικής του bulk carrier. Αρχικά ορίζεται η έννοια του όρου bulk carrier, όπως αυτός έχει υιοθετηθεί στους κανονισμούς του IACS.

Στη συνέχεια ακολουθεί ένα σύντομο κείμενο όπου παρουσιάζεται η ιστορική εξέλιξη των πλοίων μεταφοράς ξηρού φορτίου χύδην. Ξεκινά από την εισαγωγή των δεξαμενών έρματος κάτω από την περιοχή του χώρου φορτίου, στα 1850, συνεχίζει στα μέσα της δεκαετίας του 1950, οπότε και καθιερώθηκε η σύγχρονη μορφή του αμπαριού και γενικά της εγκάρσιας τομής και ολοκληρώνεται στην κατασκευή των σύγχρονων γιγάντων της σημερινής εποχής, πλοίων όπως το Vale Brazil.

Ακολούθως, γίνεται κατηγοριοποίηση του στόλου των bulk carriers με διάφορους τρόπους. Ανάλογα με το μέγεθός τους, ανάλογα με το φορτίο που σχεδιάζονται να μεταφέρουν, ανάλογα με τα μέσα που χρησιμοποιούν για να φορτώσουν/εκφορτώσουν το φορτίο και φυσικά κατηγοριοποιούνται ανάλογα με τους κανονισμούς των νηογνωμόνων. Στη συνέχεια γίνεται παρουσίαση μιας τυπικής μεταλλικής κατασκευής bulk carrier, στο χώρο του αμπαριού, με την ονοματολογία από τα κυριότερα κατασκευαστικά στοιχεία από τα οποία αποτελείται. Τέλος γίνεται μια ανάλυση του παρόντος στόλου των bulk carriers, ώστε ο αναγνώστης να καταλάβει το μέγεθος του παγκόσμιου στόλου αυτού του τύπου πλοίων, τη διάρθρωση της ηλικίας του και τις προοπτικές για το μέλλον.

Το ***δεύτερο κεφάλαιο*** αποτυπώνει τις βασικές αρχές που ακολουθούνται στο σχεδιασμό της μεταλλικής κατασκευής των χώρων φορτίου, ξεκινώντας από την περιγραφή της διαδικασίας, που είναι υποσύνολο της σπειροειδούς διαδικασίας κατά τη μελέτη του πλοίου. Σε αυτό το στάδιο, μια βηματική διαδικασία καθορίζει τα κατασκευαστικά χαρακτηριστικά του πλοίου, ακολουθούμενη από τον υπολογισμό των παχών των ελασμάτων της γάστρας. Στη συνέχεια γίνεται η αξιολόγηση της αντοχής της κατασκευής, ενώ η σπειροειδής διαδικασία τερματίζεται με το λεπτομερή σχεδιασμό κάθε κατασκευαστικού στοιχείου.

Σε άλλο σημείο αυτού του κεφαλαίου, γίνεται λόγος για διάφορα πρακτικά θέματα που σχετίζονται με την σχεδίαση των bulk carriers. Επιπλέον, γίνεται μια σύντομη περιγραφή του περιβάλλοντος λειτουργίας του πλοίου και των επιχειρησιακών απαιτήσεων που έχει να αντιμετωπίσει κατά τις διάφορες φάσεις λειτουργίας του. Ακόμη, γίνεται λόγος για τους διάφορους τρόπους φόρτωσης των bulk carriers,και την επίδραση που αυτοί έχουν στις καμπτικές ροπές και τις διατμητικές δυνάμεις επί του πλοίου.

Στη συνέχεια, γίνεται περιγραφή της αρχής "net scantling approach", σύμφωνα με την οποία καθορίζονται τα ελάχιστα πάχη ελασμάτων που πρέπει να διατηρούνται σε όλη τη διάρκεια της ζωής του πλοίου, ώστε να ικανοποιούνται οι απαιτήσεις αντοχής, όπως περιγράφονται από τους νηογνώμονες. Επιπλέον, περιγράφονται και τα διάφορα "limit states" όπως καθορίζονται στους κανονισμούς, και αναφέρονται σε καταστάσεις κάτω από τις οποίες συγκεκριμένα κατασκευαστικά στοιχεία της μεταλλικής κατασκευής ή συνολικά η κατασκευή αποτυγχάνει να εκτελέσουν τη λειτουργία για την οποία έχουν σχεδιαστεί.

Στο τελευταίο μέρος του κεφαλαίου δίδονται περισσότερες πληροφορίες για τμήματα της κατασκευής των bulk carriers που θεωρούνται μείζονος σημασίας στη σχεδίαση, τμήματα όπως το διπύθμενο, οι εγκάρσιες φρακτές και η πλευρική κατασκευή στο χώρο του φορτίου.

Το ***τρίτο κεφάλαιο*** είναι αφιερωμένο στον προσδιορισμό των φορτίων που επηρεάζουν την κατασκευή του πλοίου. Τα πρώτα φορτία που αξιολογούνται είναι αυτά που είναι παρόντα στην κατάσταση όπου το πλοίο θεωρείται ότι ισορροπεί σε ήρεμο νερό. Τα κυριότερα είδη σε αυτή την κατηγορία είναι οι καμπτικές ροπές και οι διατμητικές δυνάμεις. Αυτές οι τιμές των μεγεθών σε ήρεμο νερό, πρέπει να προστεθούν στις φορτίσεις που λαμβάνονται σε κυματισμό, ώστε με τη φνώση πλέον των συνολικών δυνάμεων που ασκούνται στο πλοίο να αξιολογηθεί η αντοχή της κατασκευής σε φορτίσεις που οδηγούν σε σημαντικές καταπονήσεις. Επιπλέον, παρουσιάζονται οι τύποι που χρησιμοποιούνται από τον IACS για τον προσδιορισμό αυτών των φορτίσεων, ενώ παρουσιάζονται και διαγράμματα με τις επιτρεπόμενες τιμές και τις πραγματικές αυτών των μεγεθών, σε συγκεκριμένες καταστάσεις του πλοίου.

Στη συνέχεια αναφέρονται συγκεκριμένες καταστάσεις φορτίσεων που θεωρούνται κρίσιμες, κατά τον IACS, για διάφορα κατασκευαστικά στοιχεία, ενώ συζητούνται τα αποτελέσματα μελέτης που καταγράφει κάτω από ποιες καταστάσεις θαλάσσης υφίστανται τις μεγαλύτερες φορτίσεις συγκεκριμένα στοιχεία της κατασκευής των bulk carriers. Το επόμενο βήμα στη διαδικασία καταγραφής των φορτίων αποτελεί η κατανομή των εξωτερικών πιέσεων που ασκούνται στη γάστρα. Αυτή η διαδικασία δεν περιλαμβάνει μόνο την υδροστατική πίεση που υφίσταται το πλοίο, αλλά και την αναγνώριση των υδροδυναμικών φορτίων. Σε αυτό το μέρος καταγράφονται και καποιες διαφοροποιήσεις που έχουν μελετηθεί σε σχέση με όσα ορίζουν οι κανονισμοί, όταν το πλοίο μεταβάλλει την πορεία του για να αποφύγει δύσκολες καταστάσεις θάλασσας. Τέλος, η παρουσία των εσωτερικών πιέσεων που οφείλονται τόσο στο φορτίο όσο και σε υγρά που έχουν τοποθετηθεί σε δεξαμενές θα πρέπει να ληφθεί και αυτή υπόψη στο σχεδιασμό.

Στο τελευταίο κομμάτι αυτού του κεφαλαίου, γίνεται αναφορά στις διάφορες καταστάσεις φόρτωσης που καταγράφουν οι νηογνώμονες και πρέπει να εξεταστούν κατά την αξιολόγηση της κατασκευής, ενώ τέλος γίνεται μνεία και στις καμπύλες “hold mass curves”, που χρησιμοποιούνται για τον προσδιορισμό της επιτρεπόμενης ποσότητας φορτίου σε κάθε αμπάρι, σε συνάρτηση με το βύθισμα του πλοίου.

Στο ***τέταρτο κεφάλαιο*** της διπλωματικής, γίνεται η παρουσίαση των υπολογισμών, σύμφωνα με τους κανονισμούς του Αμερικανικού Νηογνώμονα (ABS), για τη διαστασιολόγηση των στοιχείων που απαρτίζουν τη μεταλλική κατασκευή του bulk carrier, στην περιοχή της μέσης τομής. Αυτή η διαδικασία είναι μέρος του θέματος μελέτης πλοίου, όπως αυτή πραγματοποιείται από τους φοιτητές του τμήματος Ναυπηγών Μηχανολόγων του ΕΜΠ, και η συγκεκριμένη εργασία αποτελεί το δέκατο κεφάλαιο της δουλειάς του φοιτητή Αντώνη Δελλή, του οποίου η άδεια παραχώρησης ζητήθηκε και δόθηκε.

Το ***πέμπτο κεφάλαιο*** της διπλωματικής ασχολείται με την ανάλυση της αντοχής του πλοίου. Αρχικά παρουσιάζεται η μέθοδος των πεπερασμένων στοιχείων (FEM), μια σημαντική διαδικασία για την αξιολόγηση της αντοχής όλων των σημαντικών στοιχείων της μεταλλικής κατασκευής, όπως τα διαμήκη ενισχυτικά, τα κύρια κατασκευαστικά στοιχεία της κατασκευής και οι εγκάρσιες φρακτές του πλοίου. Επιπλέον, με τη χρήση αυτής της μεθόδου είναι εύκολο να ληφθούν αναλυτικά δεδομένα για τις φορτίσεις σε τοπικό επίπεδο, ενώ μπορεί να γίνει αξιολόγηση και της κόπωσης της κατασκευής. Αφού παρουσιαστεί η τυπική μέθοδος για την αξιολόγηση με χρήση της μεθόδου πεπερασμένων στοιχείων, καταγράφονται οι περιοχές την μεταλλικής κατασκευής στα bulk carrier που αναμένεται να παρουσιάζουν καποιο πρόβλημα, και χρήζουν περαιτέρω παρακολούθησης.

Στη συνέχεια γίνεται εξήγηση της διαδικασίας που ακολουθείται από τους κανονισμούς για την ανάλυση της αντοχής, με έμφαση στον έλεγχο της αντοχής σε λυγισμό, διαρροής του υλικού κατασκευής και της συνολικής διαμήκους αντοχής της κατασκευής, ενώ συζητούνται και τα αποτελέσματα μελέτης πάνω σε αυτούς τους ελέγχους, πάντα σε σχέση με τα bulk carriers. Τελικά, γίνεται αξιολόγηση της διάρκειας ζωής των διαφόρων κατασκευαστικών στοιχείων που είναι επιρρεπή σε αστοχία λόγω κόπωσης. Ενώ αναφέρονται οι διάφοροι τύποι των φορτίων που λαμβάνονται υπόψη σε αυτούς τους υπολογισμούς, είναι απαραίτητη και η σωστή επιλογή καμπύλης S-N, ώστε να φτάσουμε σε ορθή αξιολόγηση. Τέλος, γίνεται αναφορά σε ένα παράδειγμα ανάλυσης της συμπεριφοράς σε κόπωση για τα πλευρικά ελάσματα ενός bulk carrier, και τα συμπεράσματα που βγήκαν από αυτή τη μελέτη.

Το ***έκτο κεφάλαιο*** της διπλωματικής επικεντρώνει στο συνολικό σχεδιασμό της μεταλλικής κατασκευής στην περιοχή των αμπαριών, όπως αυτή επηρεάζεται από τα λειτουργικά χαρακτηριστικά και τις ανάγκες που παρουσιάζονται κατά την επιχειρησιακή λειτουργία του πλοίου. Αρχικά εξετάζεται ο αριθμός των αμπαριών που απαιτούνται ανάλογα με το μέγεθος του πλοίου, το μήκος τους,τα χαρακτηριστικά του φορτίου που είναι να μεταφερθεί, και ο αριθμός των φρακτών που πρέπει να τοποθετηθούν. Επιπλέον, συζητείται ο λόγος ύπαρξης των άνω και κάτω πλευρικών δεξαμενών, ενώ καταγράφονται και οι συνέπειες που έχει η διαχείριση του έρματος στην αντοχή του πλοίου. Ακόμη, παρουσιάζεται η διάταξη του διπύθμενου, επικεντρώνοντας στις συνέπειες που έχει το ύψος του στην αντοχή του πλοίου.

Οι διάφορες διατάξεις των δεξαμενών πετρελαίου στα bulk carriers αποτελούν το θέμα στη συνέχεια, ενώ αξιολογείται και η συμπεριφορά τους σε ενδεχόμενο διάρρηξης τους, με τον υπολογισμό πιθανοτήτων εκροής ανάλογα με το σημείο που βρίσκεται η κάθε δεξαμενή. Τέλος, η συζητείται συνεισφορά των καπακιών των αμπαριών σε θέματα αντοχής, ενώ παρουσιάζονται και οι κυριότεροι τύποι καπακιών που συναντώνται στα υπό μελέτη πλοία.

Το επόμενο κομμάτι αυτού του κεφαλαίου επικεντρώνεται στην ποικιλομορφία που έχουν τα διάφορα φορτία που μεταφέρουν τα bulk carriers και τους διάφορους τομείς του σχεδιασμού του πλοίου που το καθένα επηρεάζει. Για παράδειγμα οι ρυθμοί φόρτωσης των ορυκτών μεταλλευμάτων μπορούν να επηρεάσουν την αντοχή του πλοίου, ενώ φαινόμενα ρευστοποίησης του φορτίου μπορούν να αποτελέσουν αιτία απώλειας του σκάφους. Επιπλέον, η μεταφορά συγκεκριμένων ειδών μεταλλευμάτων, κάτω από συγκεκριμένες συνθήκες, μπορεί να οδηγήσει σε αυτόματη ανάφλεξη του φορτίου. Τα φορτία γαιανθράκων, αν αναμειχθούν με νερό κατά τη μεταφορά τους, είναι γνωστά για την έντονη διάβρωση που προκαλούν. Τα φορτία σιτηρών, από την άλλη, είναι γνωστά για την τάση τους να μετακινούνται κατά την κίνηση του σκάφους σε κυματισμούς, κάτι που μπορεί να οδηγήσει στην απώλεια της ευστάθειας του πλοίου. Τα φορτία προϊόντων σιδήρου μπορεί να οδηγήσουν σε φόρτιση του εσωτερικού πυθμένα πάνω από τα όρια που επιτρέπουν οι νηογνώμονες, ενώ τέλος καταγράφονται και οι κίνδυνοι που παρουσιάζονται κατά τη μεταφορά προϊόντων ξύλου, και τα μέτρα προστασίας που πρέπει να λαμβάνονται.

Τέλος, στο ***έβδομο κεφάλαιο*** παρουσιάζονται όλα τα εναλλακτικά σχέδια της μορφής της μεταλλικής κατασκευής των bulk carrier που έχουν προταθεί τα περασμένα χρόνια, ενώ καταγράφονται και οι κυριότερες περιοχές καινοτομιών που μελετώνται και είναι υπό εξέταση η εφαρμογή τους στη λειτουργία του πλοίου.

Αρχικά, συζητούνται τα πλεονεκτήματα και τα μειονεκτήματα της εφαρμογής του διπλού τοιχώματος στα πλευρικά τοιχώματα του πλοίου, ένα θέμα που έχει απασχολήσει ιδιαίτερα τη ναυπηγική βιομηχανία τα τελευταία χρόνια. Επιπλέον, παρουσιάζονται καποια εναλλακτικά σχέδια που έχουν προταθεί για τη μορφή της ενίσχυσης στα πλευρικά τοιχώματα του κύτους. Αναφέρονται θέματα αντοχής των διατάξεων σε περιπτώσεις σύγκρουσης, ενώ καταγράφονται και συγκρίσεις της αξιοπιστίας της κατασκευής σε σχέση με τη συμβατική σχεδίαση.

Στη συνέχεια, περιγράφονται τα βασικά χαρακτηριστικά ενός μεταλλευματοφόρου πλοίου 202,500DWT, με το χαρακτηρισμό Newcastlemax, βασικά καταγράφοντας τη γενική του διάταξη και φέρνοντάς την σε σύγκριση με ένα συμβατικής σχεδίασης bulk carrier. Επιπλέον παρουσιάζεται ένα bulk carrier υβριδικού σχεδιασμού (Hycon), με το πρώτο και τελευταίο αμπάρι του να έχει διπλά τοιχώματα, ενώ τα ενδιάμεσα παραμένουν μονού τοιχώματος. Ακόμη, γίνεται αναφορά στο Optimum 2000, ένα bulk carrier που είναι εφοδιασμένο επιπλέον με μια διαμήκη φρακτή σε κάθε αμπάρι.

Το κεφάλαιο συνεχίζει με τα βασικότερα εναλλακτικά σενάρια της διαμόρφωσης πλοίων. Για παράδειγμα παρουσιάζεται το bulk carrier με κυρτό εσωτερικό πυθμένα (curved inner bottom bulk carrier) που έχει σαν σκοπό τη μείωση των τοπικών φορτίσεων στο χώρο του αμπαριού με την αντικατάσταση του επίπεδου εσωτερικού πυθμένα και της κάτω πλευρικής δεξαμενής από ένα έλασμα μορφής αντεστραμμένου τόξου. Στη συνέχεια βλέπουμε τη σχεδίαση για το Non ballast seawater bulk carrier (NOBS), όπου γίνεται προσπάθεια να μειωθεί το μεταφερόμενο έρμα του πλοίου με την υιοθέτηση ενός νέου σχήματος της γάστρας. Το Ecoship 2020 είναι ένα πρόγραμμα για σχεδίαση πλοίου που προτείνει σημαντικό αριθμό καινοτομιών, που μπορούν να οδηγήσουν σε μια πιο ευέλικτη, οικονομικά εύρωστη, ενεργειακά αποτελεσματική και περιβαλλοντολογικά φιλική κατασκευή. Στα πλαίσια αυτά καταγράφεται και το Mitsubishi air lubrication system (MALS), ένα σύστημα που αποσκοπεί στη μείωση της αντίστασης τριβής της γάστρας. Το Ecore, ένα μεταλλευματοφόρο πλοίο 250,000 DWT , παρουσιάζεται με τις καινοτόμες ιδέες της ύπαρξης μόνο ενός κεντρικού αμπαριού σε όλο το πλοίο και την εναλλακτική χρήση των πλευρικών δεξαμενών.

Τέλος, καταγράφεται και η μελέτη για το πλοίο μεταβλητής πλευστότητας (variable buoyancy ship) ένα που υιοθετεί λύσεις για την εξολόθρευση των μικροοργανισμών που μεταφέρονται με το έρμα των πλοίων παγκοσμίως. Αυτό επιτυγχάνεται με την ύπαρξη διαμήκων δεξαμενών που εκτείνονται σε μεγάλο μήκος του πλοίου, κάτω από την ίσαλο, οι οποίες είναι ανοικτές όταν το πλοίο ταξιδεύει με ταχύτητα, πράγμα που οδηγεί στην επιθυμητή ανταλλαγή έρματος.

Λέξεις κλειδιά: «bulk carrier, σχεδιασμός μεταλλικής κατασκευής, σχέδιο μέσης τομής, χώροι φορτίου, πάχη ελασμάτων γάστρας, ανάλυση αντοχής, φορτία, φορτίσεις, IACS, πλοία διπλού τοιχώματος, εναλλακτικά σχέδια »

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1

1.1 Introduction

In the first chapter of this thesis an effort is being made to familiarize the reader on the concept of the bulk carrier design. At the beginning the term bulk carrier is defined as adopted by the IACS resolutions.

A brief text follows presenting the historical evolution of the bulk carrier. Beginning with the implementation of the ballast tanks under the cargo hold in the 1850's, to the consolidation of the modern cargo hold structure in the mid 1950's and the construction of the modern giant of the bulk carrier fleet, Vale Brazil.

Subsequently, the bulk carrier fleet is classified in various ways. According to the size of the bulk carrier or the commodity they are built to carry. Furthermore classification is made by the means that are used to load or discharge their cargo. Moreover reference is made to the assigned class notations according to the IACS. Afterwards, the typical single side skin bulk carrier structural configuration is presented, noting down the nomenclature of the main structural components of a cargo hold.

Finally, a fleet analysis is conducted to help the reader understand the volume of the bulk carrier fleet, its age and the perspectives of the shipbuilding in this sector.

1.2 Definition of the term bulk carrier

According to the International Maritime Organization (IMO), as adopted by the International Convention for the Safety of Life at Sea (SOLAS)[1], bulk carrier means a ship which is constructed generally with single deck, topside tanks and hopper side tanks in cargo spaces, and is intended primarily to carry dry cargo in bulk. The definition includes such types as ore carriers and combination carriers.
Further clarification to the term bulk carrier was given by the IMO[2] by mentioning that the expression «primarily to carry dry cargo in bulk» means primarily designed to carry dry cargoes in bulk and to transport cargoes which are carried and loaded or discharged in bulk, and which occupy the ship's cargo spaces exclusively or predominantly. This definition excludes the woodchip carriers and the cement, fly ash and sugar carriers, provided that loading and unloading is not carried out by grabs heavier than 10 tones, power shovels and other means which frequently damage cargo hold structures.

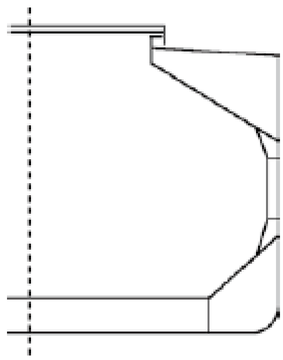

Figure 1.1
Typical single sided bulk carrier midship section [5]

1.3 Historical evolution

Of major significance in the evolution of the bulk carriers was the construction of the steamer S/S John Bowes[3] in 1852 on the river Tyne, UK. This 150 feet vessel was designed to deliver coal from the rivers Tyne and Wear to the Thames. The key feature which allowed John Bowes to complete so successfully with the much cheaper sailing ships was the facility to carry water ballast.

Until then, when a sailing collier arrived on the Thames and discharged its coal, it had to take onboard ballast in the form of sand or shingle so that it was stable for its voyage to the coal port. The ships had to queue to load ballast, pay for it and when they arrived at the coal port, pay for it to be shoveled out. This time consuming and expensive procedure was replaced by fitting tanks below the hold that could be filled with water once the ship was discharged. This water ballast maintained the screw collier's stability towards the next loading port. On arrival, a pump running off the engine simply emptied the water ballast tanks and the vessel was ready again to load coal cargo. This was a very elegant and relatively cheap solution to an age-old problem.

Until the middle part of the 20th century, the cargo holds of ships carrying dry cargo were generally partitioned into upper and lower holds. This was convenient for the carriage of cargo in boxes and in bags, and the partitioning deck itself contributed to the strength of the hull structure. Bulk carriers with topside tanks did not emerge until the 1950s. At the time, bulk cargo volumes were increasing and there was a growing need for ships that could carry loose, unpackaged dry cargoes.

The concept of a bulk carrier as used nowadays belongs to the shipbroker Ole Skaarup[4]. According to his experience, he considered that a functional design should have wide and without obstacles cargo holds. This implied that the engine room should be moved aft and there should be wide openings to the cargo holds, in order to facilitate loading and discharging of the bulk cargo. Additionally, the configuration of the cargo holds should eliminate the need for shifting boards. An also important development was the establishment of sloping ballast tanks in the upper side parts of the holds. This lead to the development of a self trimming hold where the bulk cargo is following its natural angle of response as being loaded and thus, eliminating the need for trimming of the cargo after the end of the loading. The first ship ever adopting those innovative ideas was the 19.000 dwt Cassiopeia, which was launched in Kockums shipyards in Sweden in 1954.

The same year, the first bulk carrier ever built in Japan[5] was delivered. The Nichiryu Maru was a twin engine, twin shaft ship with a length of 153metres, breadth of 21metres, depth of 11.5 metres and a deadweight of 15.368 tons. The ship was designed to carry iron ore as its main cargo.

In the year 2010, the biggest bulk carrier considering the cargo capacity was the 364.767 dwt ore carrier Berge Stahl[6], built in1986. The principal dimensions of this vessel are; length over all L_{OA} =342m, length between perpendiculars L_{PP}=328m, breadth extreme B=63.5m, Depth D=30.2m, and maximum draft T_{max}=23m.

According to some online shipping databases[7], in the period after the year 2011 (2011-2013) it is scheduled to have the deliveries of 35 bulk carrier vessels bigger than the Berge Stahl.Thus, the biggest ore carrier at present is the 402.347 dwt Vale Brazil[6] that was delivered by Daewoo Shipbuilding, S.Korea in the year 2011. The principal dimensions of this vessel are; length over all L_{OA} =362m, length between perpendiculars L_{PP}=350m, breadth extreme B=65m, Depth D=30.4m, and maximum draft T_{max}=23m.

1.4 Bulk carrier classification

Classifying bulk carriers in groups is a complicate case. The main feature that helps us categorize the bulk carries is their deadweight capacity, even though there are some categories that overlap each other or there are considered some minor groups in the same category, due to vague definition of the categories. On the other hand, the vessels can be grouped according to the commodity they are built to carry or furthermore by the means that are used to load or discharge their cargo. A brief description of the categories according to various sources[8&9] is given straight away.

1.4.1 Categories according to the capacity of the bulk carrier (dwt)

- **Mini bulkers**

Their capacity is considered less than 10.000 dwt. They are designed to carry cargoes to relatively sheltered waters, thus being used as feeders or transport limited amount of cargo to small and remote ports.

- **Handy** -*"The workhorses of the market"*

The handy sized bulker is so called because her comparatively modest dimensions permit her to enter a considerable number of ports worldwide. Such vessels are used in the many trades in which the loading or discharging port imposes a restriction upon the vessel's size, or where the quantity of cargo to be transported requires only a ship able to carry 40.000 tonnes or less. The bibliography defines a handy bulk carrier as a ship from 10.000 to 40.000 dwt. These ships carry a huge variety of cargoes. There are less handysize being built in recent years as economies of scale drive up parcel and ship sizes but they are still by far the most numerous of the size groups. Because less are being built, the age profile of the fleet is getting older.

A special category of handysize is the **Laker or Seawaymax**, a ship that is able to transit the locks of St. Lawrence Seaway and reach the Great Lakes of North America. The maximum dimensions are length over all L_{OA} =222.5m, beam B=23.77m, and maximum draft T_{max}=7.92m. The maximum deadweight is effectively about 28-32.000dwt. The trade is restricted to the ice free season of April to December. Ships that fail to sail out prior to the end of the season risk being frozen in.

It should be noted that these ships should not be confused with ships designated for Great Lakes trading only, which trade inside the Lakes as far as the St. Lawrence Seaway. With the beam limited to 75 feet and the draft to 27 feet, the only way to achieve satisfactory cargo capacity is to make L_{OA} much longer hence the curious very long narrow shape of these ships. There are other peculiar features to Great Lakes trading; The lack of severe waves and swell and hence stresses to the hull the lack of corrosive salt water and the lack of grabbed handling appliances at load and discharge ports means the ships tend to last much longer than the sea going vessels.

The long narrow shape and the nature of the handling appliances determine numerous hatch arrangements.

- **Handymax**

This term denotes the maximum size of handy sizes before reaching panamaxes. Handymaxes can be roughly categorized as 35.000-50.000dwt (other source[11] up to 60.000 dwt). The almost always use gears for loading and discharging whereas the panamaxes are usually gearless. This is a more modern fleet than the handysize fleet as this size has really replaced a great number of handy size trades as economies of scale have driven up parcel sizes.

- **Supramax**

Considered as a sub-category of the handies, their capacity usually ranges between 50.000 and 60.000dwt.

- **Panamax**

This is an exact term. It is the maximum size of vessel able to transit the Panama Canal. The maximum dimensions of the locks are; Length 289.5m, beam 32.3m and maximum draft at 12.04m. Panamaxes usually have deeper draft and thus do not load down to their marks or full draft when loading for a voyage via the Panama Canal. The size of this category is located between 60.000-80.000 dwt.

Panamax bulkers are extensively employed in the transport of large volume bulk cargoes such as coal, grain, bauxite and iron ore in the long haul voyages. The fact that most United States ports can accept no ships larger than panamax size is an important factor in their continued popularity.

The published extension to the Panama Canal is now playing a significant role to the design of the bulk carriers. The extended canal, expected to be operational in 2014 has lead to new designs named **New Panamax**[10] that will fully take advantage of the new available dimensions of the locks that measure 427m length, 55m wide and 18.5m in depth. These dimensions cannot be fully employed, since the use of locomotives for pulling the ships into the locks will be replaced by tugs and thus the available space will be reduced. The estimated new panamax limits have been specified as; length overall L_{OA} = 366m, Beam B= 49m and draft T= 15.2m.

- **Capesize**

These are vessels which being too large to transit the Panama Canal or the Suez Canal have to go from Atlantic to Pacific and vice versa via the Cape of Good Hope or the Cape Horn. Technically defined as larger than 32.2m, their capacity lies between 100.000-180.000dwt (other source[11] defines between 80.000-200.000dwt). The cargoes they carry are mainly coal and iron ore. Cape-sized vessels with loaded draft usually in excess of 17m, can be accepted fully laden at only a small number of ports worldwide and are engaged in the longhaul iron ore and coal trades. The range of ports which they visit is increased by the use of two port discharges, the ship being only part laden on reaching the second discharge port.

- **Very Large Bulk Carrier (VLBC)**

VLBC's are bulkers greater than 180.000 dwt. A number of these vessels are special types such as ore carriers, OBO, ore/oil carriers, categories that will be discussed below. Long contracts of affreightment enable these ships to be tailored to the intended load and discharge ports, and thus maximize economies of scale.

At this point, it would be useful to mention some other minor ship categories that can be found in the international bibliography. They are usually named after the seaway or the port that imposes the restrictions to some of the principal dimensions of the vessel.

- Kamsarmax

These are bulk carriers with size larger than a Panamax (about 82.000 dwt), that are suitable for berthing at the port of Kamsar (Equatorial Guinea), where the major loading terminal of bauxite is restricted to vessels less than 229m L_{OA}.

- Japanamax[12]

With an overall length of 225m which is the maximum size that can be accommodated at all of Japan's major grain terminals, their deadweight approaches the size of the kamsarmax. Other indicative principal dimensions;
L_{PP}xBxDxT / (221.5m)x(32.2m)x(19.99m)x(14.44m)

- Dunkirkmax

At a size of 175.000dwt, these ships are capable of loading and discharging cargo at the port of Dunkerque, France. Maximum length overall L_{OA} =289m, maximum beam B=45m

- Setouchmax

Sized approximately at 205.000dwt, the vessels of this category usually carry iron ore to the ports of Japan through the seaway at the Sea of Seto. The draft restriction in this case is limited to T_{max}=16.1m and the maximum overall length is L_{OA}=299.9m

- Newcastlemax

This category's name stems from the port of Newcastle, in the New South Wales, Australia. Mainly carrying coal, their size[13] lies between 203.000-208.000dwt. Maximum length overall L_{OA} =299.9m, beam B=50m and maximum draft T_{max}=18.3m

- Wozmax

Designed to satisfy the restrictions of the three major (Port Hedland, Port Walcott, Dampier) Western Australia's ports, these ore carriers are sized approximately 250.000dwt. Generally, they are beamer[14] and shallower in draught than the general run of giant ore carriers. Indicative principal dimensions of a 250.868dwt wozmax[15] ;
L_{OA}xL_{PP}xBxDxT / (329.95m)x(321m)x(57m)x(25.1m)x(18m)

- Unimax[16]

Iron ore carriers sized approximately 300.000dwt. This type of vessels has the versatility to enter major iron ore loading ports in Western Australia, while having a hull form most suitable for making the best use of very deep water of Villanueva port in Philippines and focusing on very deep water ports in Brazil, one of the largest places of iron ore loading. Delivered, thus named after the Universal Shipbuilding Corporation in Japan. Indicative principal dimensions of a 297.351dwt unimax; L_{OA}xBxT /(327m)x(55m)x(21.4m)

- Chinamax[17]

Tailor made ore carriers designed for trading between Brazil and Chinese ports. At a loading capacity of 380.000-400.000dwt, a chinamax carrier has L_{OA}xBxDxT /(360m)x(65m)x(30.4m)x(21.5m-max 24m).

Within this category we could classify the **VLOC**'s of the Vale company, that have the commercial designation of Valemax, and are nowadays the largest bulk carriers sailing the seas of the world.

1.4.2 Categories according to the sort of cargo transported

- **Ore carrier**

Built for the carriage of ore only, they have a very small cubic capacity reflecting the low stowage factor of iron ore. Due to the high specific gravity of the cargo, cargo holds are relatively small and the side tanks are large. Designed for specific trade routes and loads, with long contracts of affreightment, these vessels achieve maximization of the economies of scale for the maritime industry.

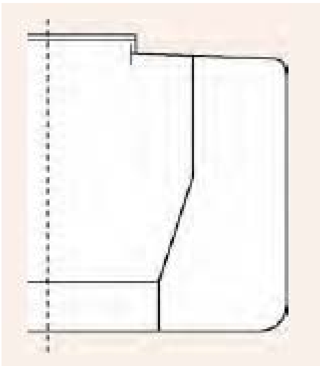

Figure 1.2
Typical ore carrier midship section [5]

- **Combination carrier (combo)**

They are able to load both dry and wet cargoes and became very popular in the 1960's and 1970's. The theory was that they could pay for the extra 15-20% building and running cost of a ship designed for dry cargo and oil by eliminating ballast legs in trading the ships. Practically this didn't work. On good tanker markets they traded oil and on good dry cargo markets they traded dry cargo. The major combo categories are the Ore/Oilers and the Ore/Bulk/Oilers, as presented below.

  - Ore/Oiler

A combination ship allowing the carriage of ore and oil, only having the small centre holds for carrying ore and the outer and centre holds for oil. When carrying oil both centre and side compartments can be used, whilst only the centre holds are used when carrying ore. The use of separate holds reduced the need for cleaning between oil and ore in the same cargo space but was wasteful and limited trading to ore and oil only.

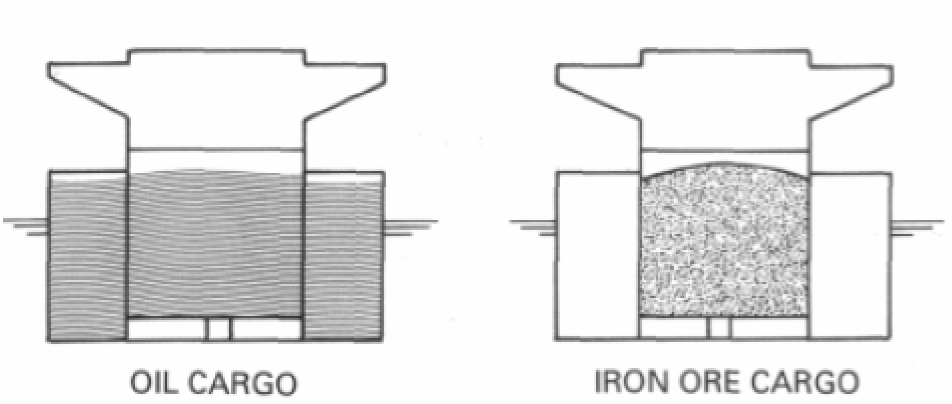

Figure 1.3
Loading combinations of an ore/oil carrier [9]

- Ore/Bulk/Oil (OBO)

A combination ship allowing oil and any dry bulk cargo in the same hold. The holds are strengthened for the carriage of ore. More popular than ore oilers due to their versatility.

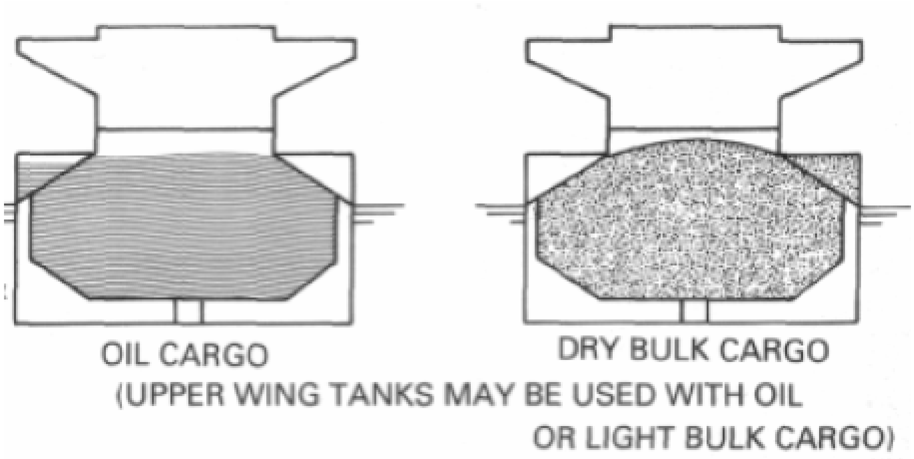

Figure 1.4
Loading combinations of an OBO carrier [9]

- **Belt self unloader**

Self unloaders are bulk carriers equipped with conveyor belt discharging systems with booms which can be swung out of the ship to discharge directly ashore, using the gravity feed system. The sides of the holds are angled down to grates which, when open allow the cargo to slide down by gravity onto a conveyor belt. This carries the cargo along to another conveyor belt which lifts the cargo to the height of the deck, where another belt on a boom will spew the cargo out at the place it is desired. Such systems are capable of achieving discharging rates similar to those of shore-based unloading equipment. This equipment is expensive to install and reduces the space available for cargo, but these disadvantages can be outweighed in the short sea trades by the ability to reduce time spent in port substantially.

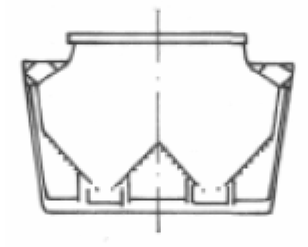

Figure 1.5
Typical belt self unloader midship section [4]

- **Bulk cement carrier**

Bulk carriers dedicated in carrying cement in bulk. The small fleet of specialized bulk cement carriers is incorporating pneumatic cargo pumping and handling gear with totally enclosed holds and moisture control systems, in order to prevent the presence of water or such conditions that would lead to cargo solidification.

- **Bulk In-Bags Out/ Bulk in-Bulk out ships (BIBO)**

BIBO ships[18] permit bulk loading and discharging of sugar or bagged discharge from the vessel's own bagging plant. The sugar is therefore very much better protected in transit, losses are minimalised and savings are made in time and cost. It should be made clear that BIBO ships are not considered as bulk carriers according to the definition of the term given at the beginning of this chapter.

- **Woodchip carrier**

Due to the low specific gravity of the cargo (wooden chips), cargo holds are deep and topside tanks are eliminated in order to increase cargo volume. Care is necessary for the reinforcement of the underside of the upper deck, since deck cranes and belt conveyors are generally fitted on deck. Lower ballast tanks are prone to corrosion due to the relatively higher temperature of the chip cargo. This type of carrier was once considered to be less versatile but its use has expanded to include carrying cargos such as soybean meal

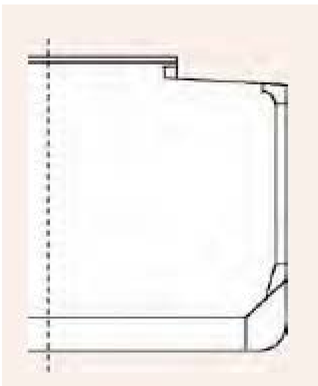

Figure 1.6
Typical woodchip carrier midship section [5]

- **Open hatch bulk carrier (conbulker)**

The open hatch bulk carrier offers a solution to the problem of access to the holds by having huge hatches witch cover the width of the ship. This ship type can therefore load timber, pipes, steel coils, packaged cargoes and containers more efficiently than a standard bulk carrier. This advantage is enhanced by having box shaped holds. This means that the holds are rectangular, the corners and sides not compromised by the slopes of the wing tanks. The cranes are large enough to load the heaviest containers or other heavy cargo.

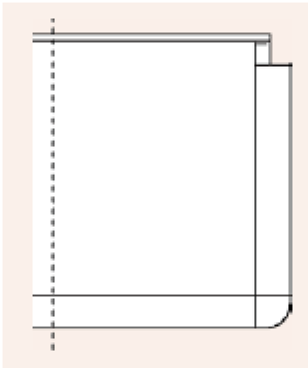

Figure 1.7
Typical conbulker midship section [5]

1.4.3 Class notations according to IACS[19]

This type of classification is useful for the structural analysis of the bulk carriers that will take place in the following chapters. Thus, a bulk carrier of length 150m or more is to be assigned one of the following additional service features;

- **BC-A**

Notation for bulk carriers designed to carry dry bulk cargoes of cargo density 1.0 t/m^3 and above with specified holds empty at maximum draught in addition to BC-B conditions.

- **BC-B**

Notation for bulk carriers designed to carry dry bulk cargoes of cargo density 1.0 t/m^3 and above with all cargo holds loaded in addition to BC-C conditions.

- **BC-C**

Notation for bulk carriers designed to carry dry bulk cargoes of cargo density less than 1.0 t/m^3.

1.5 Typical bulk carrier structural configuration

At this point it is useful to present the terminology used[20] in the structural configuration of a bulk carrier, for the readers' guidance. In figure 1.8 a single side skin bulk carrier typical cargo hold is presented.

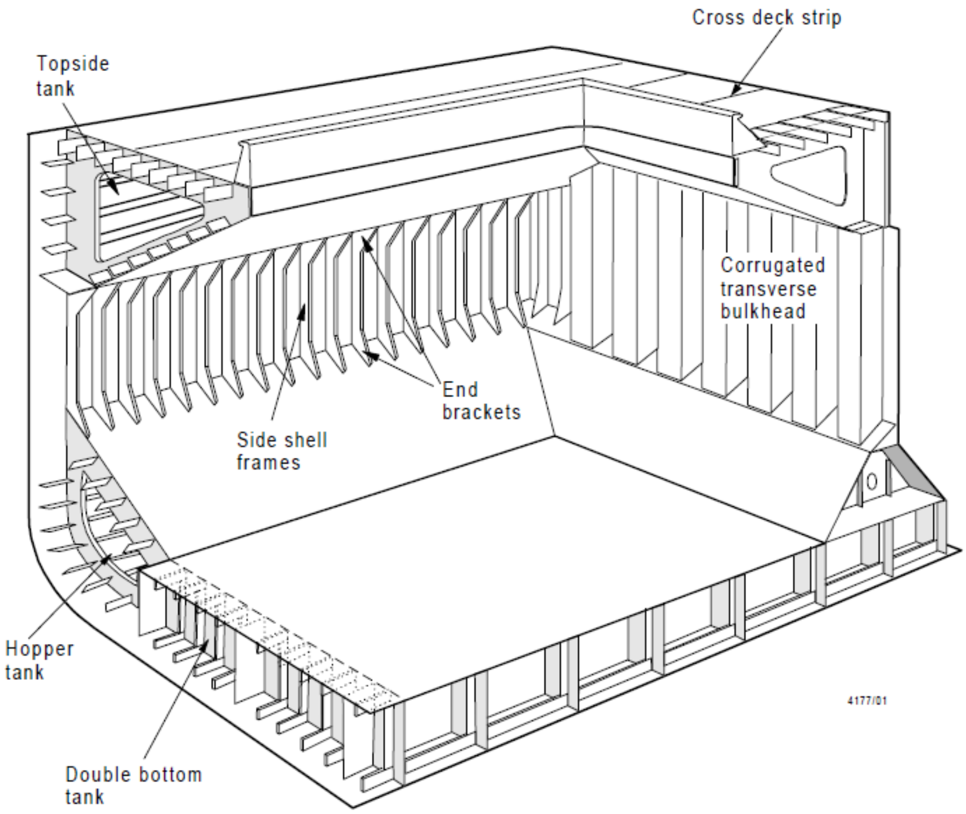

Figure 1.8
Typical cargo hold structural configuration [20]

Figure 1.9 presents the nomenclature used for the structural components of a cargo hold. Generally, the plating compromising structural items such as the side shell, bottom shell, strength deck, transverse bulkheads, inner bottom and topside and hopper tank sloping plating provides local boundaries of the structure and carries static and dynamic pressure loads exerted by the cargo, ballast, bunkers and the sea. This plating is supported by secondary stiffening members such as frames or longitudinals. These secondary members transfer the loads to primary structural members such as the double bottom floors and girders or the transverse web frames in topside and hopper tanks.

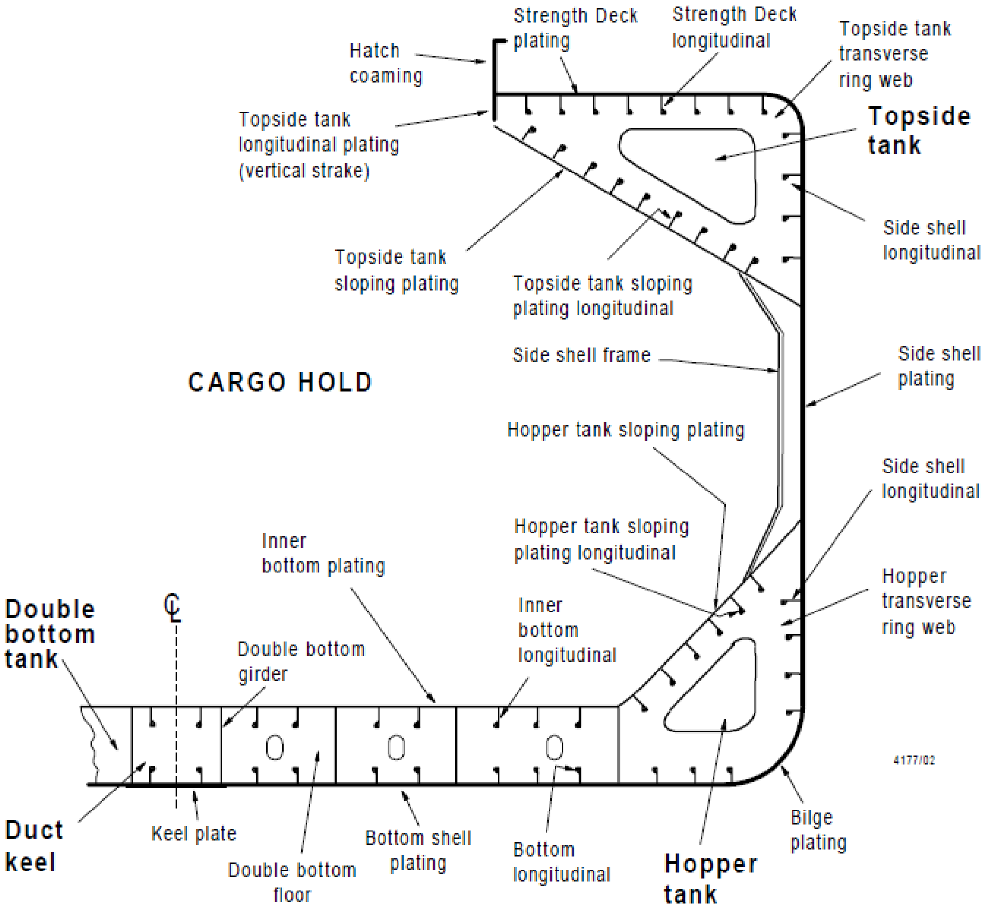

Figure 1.9

Nomenclature for typical transverse section in way of a cargo hold [20]

The transverse bulkhead structures, as presented in figure 1.10, including its upper and lower stools, together with the cross deck and the double bottom structures are the main structural members which provide the transverse strength of the ship to prevent the hull section from distorting. Additionally, if ingress of water into any one hold has occurred, the transverse watertight bulkheads prevent progressive flooding of other holds.

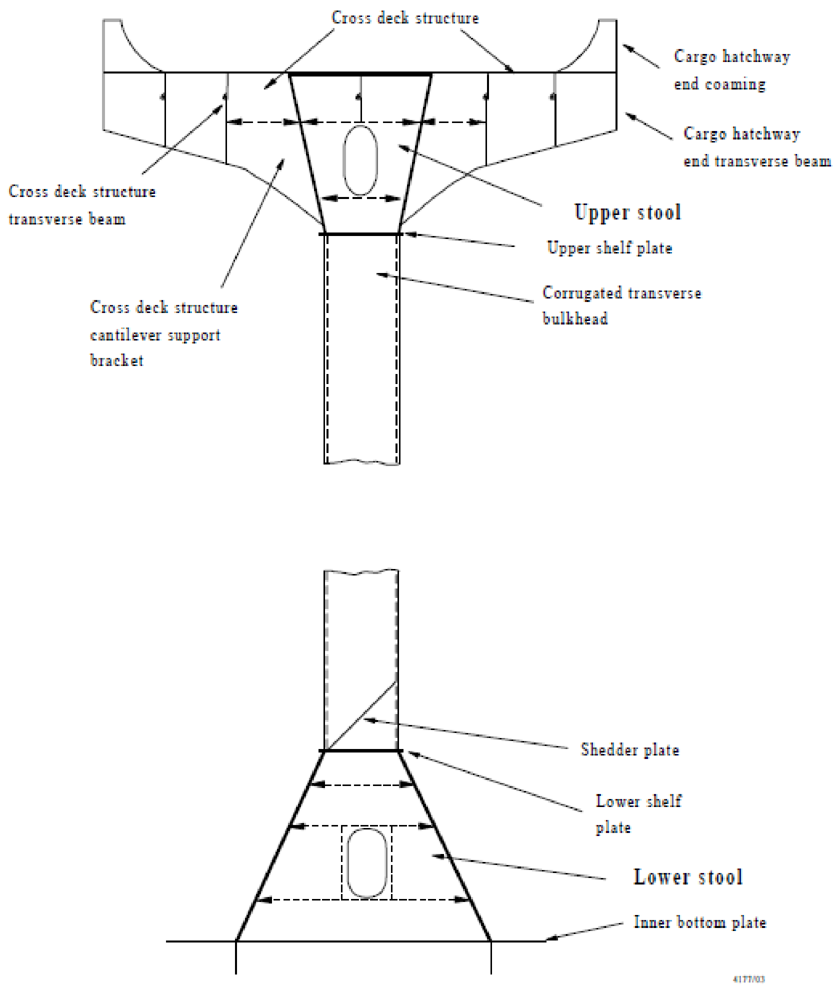

Figure 1.10
Nomenclature for typical transverse corrugated transverse watertight bulkhead [20]

1.6 Fleet analysis

According to data taken from the Lloyds[21], in April 2011 the number of bulk carrier fleet of existing or on order vessels was 11.936 ships. The available tonnage of the existing fleet is estimated in 547.3 million tones deadweight, whereas the on order tonnage reaches 244.4 million tones of deadweight.

Table 1.1 illustrates the evolution of the bulk carrier fleet tonnage for the last 30 years. The tonnage noted is the total dwt capacity estimated at the beginning of each year.

Table 1.1

Bulk carrier tonnage

Year	Tonnage (millions of dwt)
1980	186
1985	232
1990	235
1995	262
2000	276
2005	321
2010	457
2011	532
Source	UNCTAD, Review of maritime transport 2011, p. 36

Figure 1.11 illustrates the annual tonnage changes[22] for each year commencing from the year 1992. The interesting feature of this figure is the continuous growth of the world bulk carrier fleet noted the last years. This fact can be partly explained by the growing need of China for steel products (thus ore and coal shipments) and the worldwide needs for energy production (coal). Another factor contributing to this growth is the increase of the average age that bulkers reach the scrapyard to be dismantled, as can be clearly noted in table 1.2

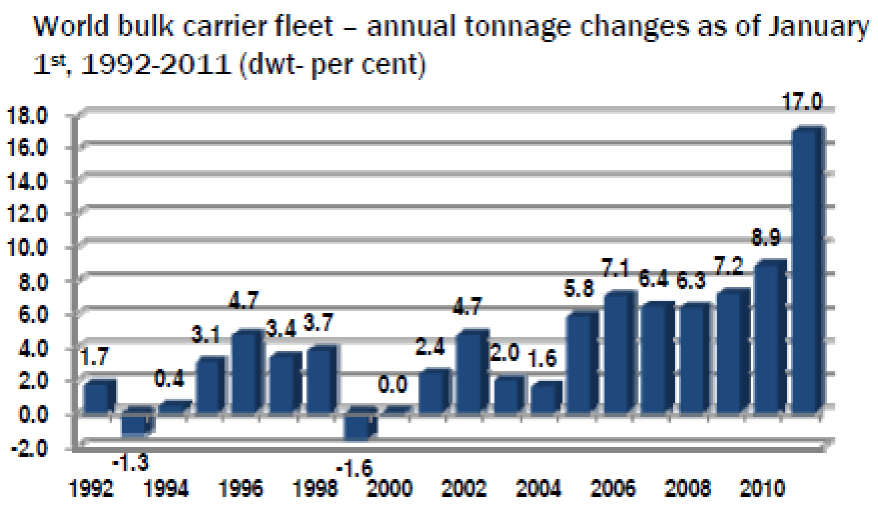

Figure 1.11
Bulk carrier annual tonnage changes (% dwt) [22]

Table 1.2	
Average age of broken up bulk carriers	
Year	**Average age**
1998	25.2
1999	25
2000	25.9
2001	26.7
2002	26.6
2003	26.5
2004	27.3
2005	28.1
2006	28.9
2007	29.1
2008	30.6
2009	30.6
2010	30.9
Source	UNCTAD, Review of maritime transport 2011, p. 55

Table 1.3 displays the age profile of the bulk carrier fleet and the corresponding tonnage listed to each age category, whereas in table 1.4 the size distribution of the current and on order fleet is presented. Table 1.5 highlights the overwhelming dominance of China in the bulk carrier shipbuilding industry.

Table 1.3		
Bulk carrier age profile		
Age	**No of ships**	**Tonnage (dwt)**
On order	2.854	244.327.102
0-5 years	2.704	207.833.992
5-10 years	1.289	88.417.918
10-15 years	1.175	73.427.910
15-20 years	823	63.144.468
20-25 years	802	43.408.705
25-30 years	1.263	53.841.308
30-35 years	516	12.769.763
35+ years	310	4.506.344
Source	Fairplay solutions magazine, April 2011,p32	

Table 1.4		
Bulk carrier size profile		
DWT	**In service**	**On order**
0-9.999	1.139	35
10.000-34.999	2.221	405
35.000-59.999	2.509	957
60.000-79.999	1.454	303
80.000-149.999	589	671
150.000-249.999	973	395
250.000+	97	88
Source	Fairplay solutions magazine, April 2011,p33	

Table 1.5		
Top bulker builders		
Country	**Ships on order**	**dwt on order**
China	1.478	124.884.180
Japan	616	53.176.769
South Korea	491	46.515.325
Philippines	85	11.283.692
India	85	3.472.000
Vietnam	71	2.692.086
Source	Fairplay solutions magazine, April 2011,p33	

[1] SOLAS 2004 , ch IX/1.6
[2] IMO resolution MSC.277(85)
[3]http://www.plimsoll.org/diversityofships/shipsofthesteamage/howthetrampsteamerevolved/default.asp#2
[4] Shipping Innovation (2009) Authors N.Wijnolst, T.Wergeland pp.206
[5] Bulk carrier, Class NK magazine p.6
[6] Det Norske Veritas ,DNV exchange fleet info online database
[7] World Shipping Register, http://www.world-ships.com/
[8] The essential guide to chartering and the dry freight market (2000). N.Collins
[9] Bulk Carrier Practice (1993). Captain J Isbester
[10] http://www.dnv.com/industry/maritime/shiptypes/bulkcarrier/panamacanalextension.asp
[11] fairplay solutions magazine, April 2011. page 33
[12] Oshima rolls out Japanamax. Motorship magazine, June 2005, page 17
[13] www.fairplay.co.uk – the swift rise of the Newcastlemax. 22April 2010
[14] Haul tracks of the sea, Rio Tinto review magazine, Dec.2008, page 13
[15] M/V WUGANG INNOVATION, Sea-Japan magazine, No 351/Feb-Mar 2012, page 3
[16] K line takes delivery of Unimax ore carrier. The motorship magazine, July-Aug 2008, page 5
[17] Chinamax-the ultimate workhorse of the seas. The motorship magazine, February 2009, page 48
[18] http://www.biboships.com/
[19] RINA, rules for the classification of ships, Common structural rules for bulk carriers, effective from 1 July 2012
[20]Guidance and information on bulk cargo loading and discharging to reduce the likelihood of overstressing the hull structure. IACS Bulk carriers. 1997
[21] Fairplay solutions magazine, April 2011,p32-33
[22] Source; shipping statistics and market review, Vol 55-No4 2011, page 5

2

2.1 Introduction

The second chapter of this thesis outlines the design principles followed in the structural design of the cargo spaces of a bulk carrier, starting from the structural design process, that is part of the design spiral. At this stage, a stepwise process determines the structural arrangements of the ship. Then the derivation of the hull scantlings is being made, followed by the assessment of the hull girder strength. Finally, the detail design of the components ends this part of the spiral.

Some issues concerning bulk carrier design are discussed in another part of this chapter. Following a brief description of the environment and the operational tasks that a ship has to cope with, the alteration of stresses imposed in the structure is outlined. Additionally, the loading patterns of the bulk carriers are presented, illustrating the effect they have on the shear forces and bending moments of the structure.

Subsequently, the net scantling approach is presented. That part describes a process for the determination of the minimum hull scantlings that should be maintained throughout the ship’s life to satisfy structural strength requirements. Furthermore, the various limit states of the structure are listed. A limit state depicts a condition for which a particular structural member or the entire structure fails to perform the function that is expected of it.

The final part of this chapter focuses on the structural arrangement principles. Details are further given for the structure of the double bottom, the side structure and the bulkhead structure, areas of major significance in the construction of bulk carriers.

2.2 Structural design process

The primary objective[1] of the structural design of every ship is the development of a structure that will be able to withstand all the forces acting on it, thus to avoid any structural failure on the vessel. The most important of these forces are the bending moments and shear forces that result from the waves encountered at sea and the loading applied by the cargo carried. As the structure must continue to meet these forces throughout the ship's life, the scantlings must include allowances for the corrosion and wear which can be expected.

On the other hand, "optimum design" is frequently assumed[2] to mean the minimum weight structure capable of performing the required service. While weight is always significant, cost, ease of fabrication and ease of maintenance are also important. Cost can increase rapidly if non-standard sections or special quality materials are used; fabrication is more difficult with some materials and, again, machining is expensive.

Structural failure[3] might occur in different degrees of severity. At the low end of the failure scale, there may be small cracks or deformations in minor structural members that do not jeopardize the basic ability of the structure to perform its function. Such minor failures may only have aesthetic consequences. At the other end of the scale is total catastrophic collapse of the structure, resulting in the loss of the ship. There are several different modes of failure between these extremes that may reduce the load-carrying ability of individual members or parts of the structure but, because of the highly redundant nature of ship structures, they do not lead to total collapse. Such failures are normally detected and repaired before their number and extent grow to the point of endangering the ship.

Structural design consists of a stepwise process in which the designer develops a structural configuration on the basis of experience, intuition, and imagination. He then performs an analysis of that structure to evaluate its performance. If necessary, the scantlings are revised until the design criteria are met. The resulting configuration is then modified in some way that is expected to lead to an improvement in performance or cost, and the analysis is then repeated to re-ensure that the improved configuration meets the design criteria. Thus, a key element in structural design is the process of analyzing the response of an assumed structure. The process of finding a structural configuration having the desired performance by synthesis is the inverse of analysis, and is not nearly so straightforward, especially in the case of complex structures. Consequently, it is only after completing several satisfactory design syntheses that the process of optimization can take place.

In summary, five key steps can be identified to characterize the structural design process, whether it be intuitive or mathematically rigorous:

- Development of the initial configuration and scantlings.
- Analysis of the performance of the assumed design.
- Comparison with performance criteria.
- Redesign the structure by changing both the configuration and scantlings in such a way as to effect an improvement.
- Repeat the above as necessary to approach an optimum.

Formally, the final optimization step consists of a search for the best attainable (usually minimum) value of some quantity such as structural weight, construction cost, overall required freight rate for the ship in its intended service or the so-called *total expected cost* of the structure. The last of these quantities, as proposed by Freudenthal (1969), consists of the sum of the initial cost of the ship (or other structure), the anticipated total cost of complete structural failure multiplied by its probability, and a summation of lifetime costs of repair of minor structural damages.

A further description of the procedure of the rule-based structural design is given by the Ship Structure Committee[4], which can be summarized in the flow chart included in figure 2.1 below.

Structural design follows the preliminary design. The first step in the structural design is the determination of the structural arrangements. As the figure indicates, there are a variety of factors that control the structural arrangement. These include designers' intentions, as well as requirements from multiple standards (e.g., IMO, Class, and National authorities). Following the structural arrangement, the usual next step is the determination of the scantlings. These are largely based on local strength requirements and primarily based (in most cases) on Class rules. The next step is to check and, if needed, to enhance the overall hull girder strength. This is again mainly

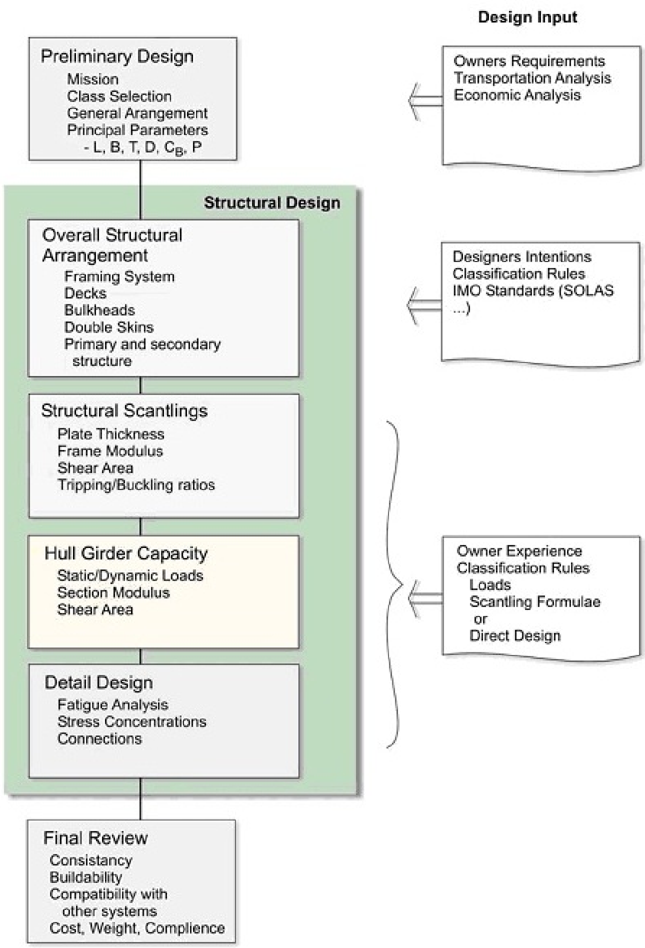

Figure 2.1

Rule-based ship structural design [4]

guided by Class rules. The final step is the design of details such as connections, openings and transitions. These details are guided by Class rules, general published guidance and by yard practices and experience. With this step completed, the structural drawings can be completed. There is a final step that can affect structural design. The structure must be reviewed for suitability in light of numerous other constraints. These include compatibility with other ship systems, produceability, maintainability, availability of materials and cost. Each step in the procedure described is part of a design spiral, and is repeated as necessary until a satisfactory result is achieved.

2.3 Bulk carrier structural design issues

Focusing on bulk carrier design, the nature of the bulker operation raises vital issues on the structural vulnerability of the vessel. A realistic operating scenariothat can help us focus on the diversity of the loads met on bulk carriers is presented by Caridis[5]. A bulk carrier is employed in the transportation of iron ore from South America to the Far East. Before entering the loading port, ballast water is discharged and transferred so as to trim the vessel and to achieve the correct draught for her to be ready for loading. During loading, excess loading may arise in way of bulkheads that separate empty from loaded holds, as the vertical shearing force there may increase substantially.

Modern conveyor belts in some ports are capable of loading at high rates (up to 16.000 tonnes/hr), with the cargo being dropped into the holds from heights of 20 metres or more. When the holds are empty, the mineral ore strikes the inner bottom, although generally no impact damage is experienced. In the case of large bulk carriers the loading rates may result in problems such as excessive global loading of the hull girder, excessive local loading and also in the synchronization of the de-ballasting operation which is performed at the same time as loading.

For a smaller vessel, her cargo handling equipment could possibly be used for loading, in which case the main deck would be subjected to loads of up to 40 metric tones, distributed over an area limited by adjacent hatch openings. As loading proceeds, the longitudinal bending moment and shearing force gradually change in tandem with local stress distributions, especially in way of load discontinuities.

On board the ship there are a number of tanks used for the storage of liquids (ballast sea-water, bunkers, marine diesel oil and fresh water). When the ship is at sea, these liquids are subjected to inertial accelerations, depending on the location of the respective tanks with respect to the ship's centre of gravity. Usually a number of tanks are located within the vicinity of the engine room. Others such as ballast tanks are located in the forepeak, the aft-peak, the double bottom and the topside tanks (in the case of bulk carriers). If the accelerations are sufficiently high, sloshing within the tanks will occur, thereby imposing loads on the vertical bulkheads of tanks. An operation that is carried out on a routine basis during the voyage is ballast water exchange, which involves the simultaneous de-ballasting of certain tanks and the ballasting of others, a procedure that is followed in order to avoid large stress changes. Ballast water exchange is also performed to prevent the spread of harmful species of micro-organisms, although another way of doing this is to treat ballast water on board.

It is possible that during the voyage the ship encounters heavy seas. In such a case, the longitudinal bending moment will fluctuate as the ship meets with irregular waves. If severe conditions are encountered, the ship's sides in way of the bow and the bow flare will be subjected to wave impact. As waves break against the sides, spray and water will strike the main deck forward. Furthermore, if emergence of the bottom forward takes place this will possibly be followed by slamming. Slamming is followed by whipping, a shudder of the hull girder, which is in effect a high frequency vibration that propagates from the region of the slam throughout the ship. In this way energy that is transferred to the hull girder during the slam is absorbed by the whole structure.

By the time the ship reaches the discharge port, the consumption of liquids such as bunkers and marine diesel oil will have brought about non-negligible changes to the longitudinal shearing force and bending moment distributions. When the ship berths, discharging can begin. Shore equipment is used and grabs are lowered into the holds, always in accordance with the masters instruction who consults the ship's loading plan. At the same time the ship takes on additional ballast in order to maintain proper trim and draught, and also so as to keep stresses at acceptable levels. When the discharging of individual holds nears completion, mechanical equipment such as bulldozers is used to collect the last traces of cargo. At this stage the grabs and the bulldozers used can strike the inner bottom and the lower parts of the hold thereby causing local dents and damage.

If the next voyage is in ballast condition, the ship will take on ballast water that can reach up to 50-60% of deadweight capacity. Usually two ballast conditions are specified: light ballast which is suitable for fair weather, during which 40-50% of deadweight capacity is transported and heavy ballast which corresponds to 50-65% of deadweight capacity. The latter is necessary when rough weather is encountered.

Concluding, the cycle of stresses[6] that the ship structure is exposed on the bulk carrier operations can be seen in figure 2.2

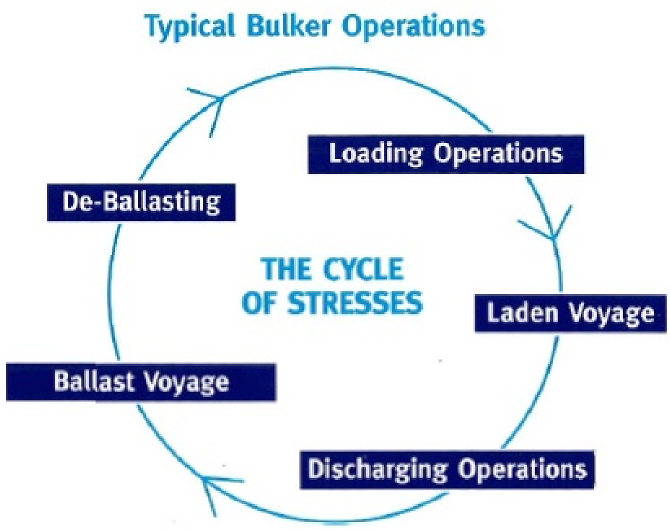

Figure 2.2

Typical bulker operations [6]

The distribution of cargo along the ship's length has a direct influence on both the global bending and shearing force of the hull girder and on the stresses in the localized hull structure. Three typical loading patterns[7] are utilized on bulk carriers: (a) homogeneous, (b) alternate hold, and (c) block loading. The pattern of each loading condition is shown on figure 2.3.

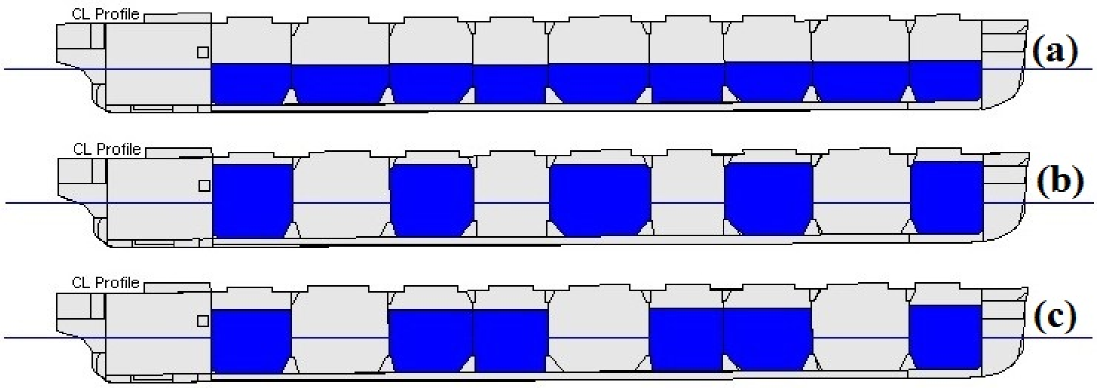

Figure 2.3

Loading patterns on bulk carriers [7]

(a) homogeneous, (b) alternate hold, (c) block loading

In the homogeneous loading condition, the cargo is evenly distributed in all available cargo holds. It is usually adopted for the carriage of low density cargoes, such as coal and grain, but when planning a homogeneous load, special care must be taken to mitigate the risk of cargo shifting.

The alternate hold loading is used when high density cargo is being transported to raise the center of gravity. If heavy cargo is loaded homogeneously, abrupt rolling can result from the low center of gravity. By loading the cargo twice as high in half as many holds, the extreme rolling can be mitigated. Alternate hold loading is something that must be considered in the design phase. Local structure – transverse bulkheads, tank top, and lower hoppers – must be adequately sized to accept the increased weight. In order to save steel weight and not over-design all the holds, only those holds that will be loaded in the alternate hold plan are reinforced. In addition to the local structure, this loading can induce high shear forces at the bulkheads where the loading switches from buoyancy-dominant to weight-dominant.

Finally, the block hold loading condition refers to the stowage of cargo in a block of two or more adjoining cargo holds with the cargo holds adjacent to the block of loaded cargo holds empty. This loading scheme is typically used when a vessel is partly loaded. When planning a block load it is very important to be mindful of the weight and buoyancy distribution over the cargo block. Loading manuals will often include charts indicating the amount of cargo that may be carried in a cargo hold at a given local draft. To enable cargoes to be carried in blocks, the cross deck and double bottom structure needs to be specially designed and reinforced. Block loading results in higher stresses in the localized structure in way of the cross deck and double bottom structures and higher shear stress in the transverse bulkheads between the block loaded holds.

It should be noted that regulations that have come into force of late, pose restrictions[8] on the loading flexibility of bulk carriers that could not withstand some flooding scenarios. The options given for alternate hold loading, were a) to limit the total amount of cargo carried to 90% of the ship's deadweight capacity, b) to sail with each hold loaded to at least 10% of its maximum allowable cargo mass if the deadweight exceeds 90% of the ship's deadweight capacity at the assigned freeboard and c) distributing cargo homogeneously if the deadweight exceeds 90% of the ship's deadweight capacity at the assigned freeboard.

In figures 2.4 and 2.5, a comparison can be made of the shear and bending moment distributions for the various loading patterns described above. Each pattern refers to the carriage of the same amount of cargo.

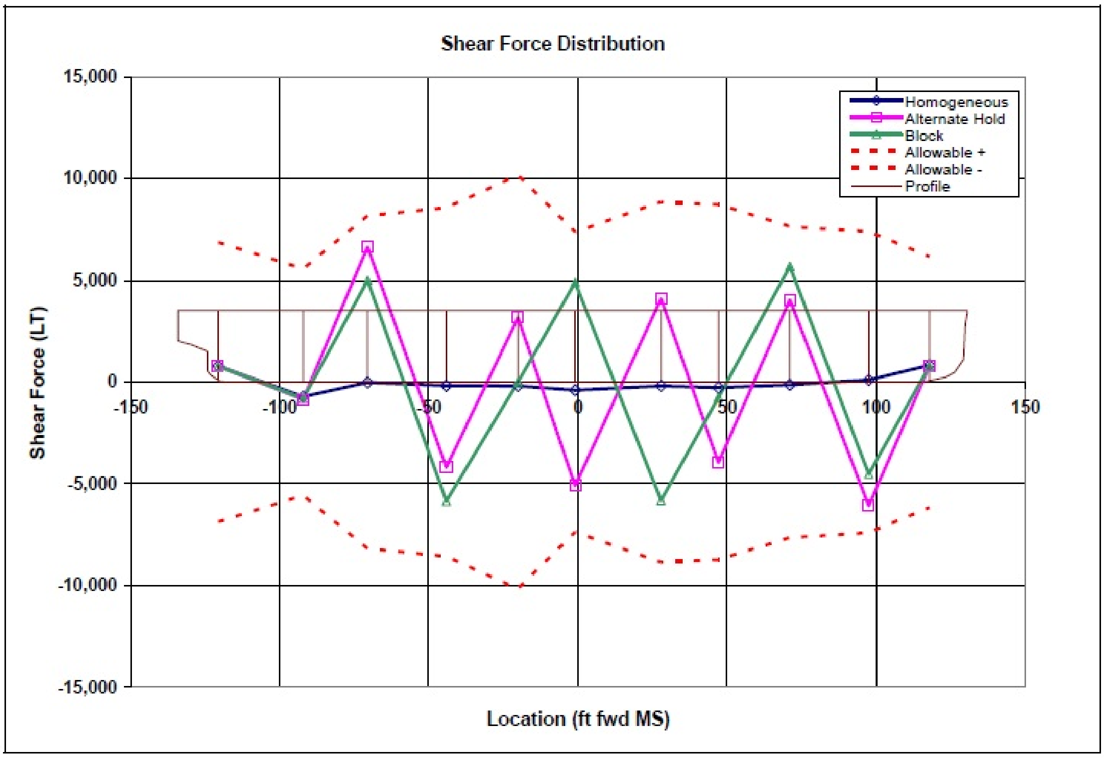

Figure 2.3

Shear force distribution for each loading condition [7]

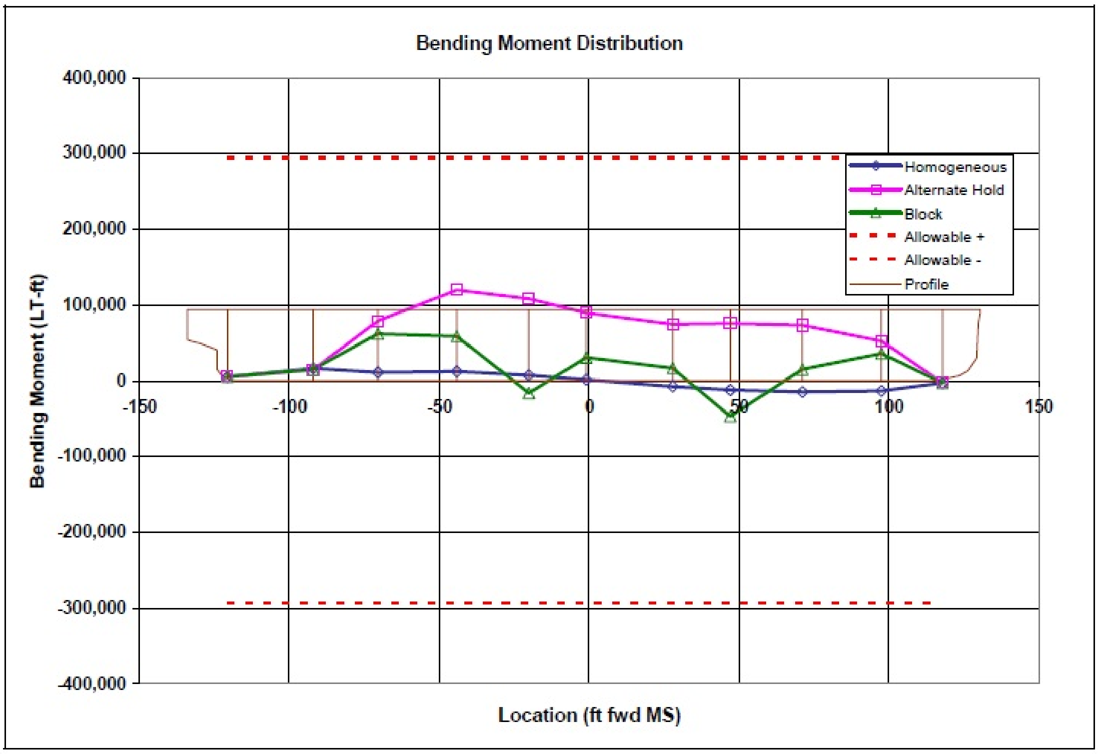

Figure 2.4

Bending moment distribution for each loading condition [7]

IACS describes briefly[9] the potential problems that can occur during the operational life of a bulk carrier. As mentioned above, the limitations described on the loading manual of the ship should not be exceeded, because a catastrophic failure of the hull structure may occur. When deviating from the cargo load conditions, it is necessary to ensure that both the global and local structural limits are not exceeded. It should also be noted that overstressing of local structural members can occur even when the hull girder still-water shear forces and bending moments are within their permissible limits.

The loading of cargo in shallow draught condition is another condition that can impose high stresses on the double bottom, cross deck and transverse bulkheads, if the cargo in the hold is not adequately supported by the buoyancy upthrust.

High loading rates can have an impact on the stresses imposed in the ship's structure. An overshoot of cargo for 5 extra minutes in two holds can cause the still water bending moments and shear forces to exceed the allowable limits. Additionally, at such high loading rates, possible inability of the vessels' pumps to discharge ballast water sufficiently, may result at high stresses in the hull.

The double bottom and the cross-deck structure are designed based upon a trimmed cargo distributed symmetrically in the hold space. Thus, any asymmetry of cargo distribution could result in the emergence of torsional loads acting on the hull girder. When heavy cargo is poured into a cargo space at one end of the cargo hold, the lateral cargo pressure acting on the transverse bulkhead, as a result of the cargo piling up at one end of the cargo space, will increase the loads carried by the transverse bulkhead structure and the magnitude of transverse compressive stresses in the cross deck.

When the same loading pattern is also adopted for the adjacent cargo hold, the lateral cargo pressure acting on the transverse bulkhead will be largely cancelled out. However, in this situation, a large proportion of the vertical forces on the double bottom is transferred to the bulkhead between the two loaded holds which could lead to shear buckling of the transverse bulkhead structure, compression buckling of the cross deck and increased SWBM in way of the transverse bulkhead. Cargo should always be stowed symmetrically in the longitudinal direction, and trimmed, as far as practical. Stowing cargo asymmetrically about the ship's centre line in a cargo space induces torsional loads into the structure which causes twisting of the hull girder. When the hull girder is subjected to torsion, warping of the hull section occurs which gives rise to shearing and bending of the cross deck structure.

In addition to the cargo holds, asymmetrical distribution of water ballast in the ballast tanks induces torsional loads, resulting in twisting of the hull girder. Torsional loading of the hull girder is considered to be an important contributory factor to recurring cracking at the hatch corners and to problems associated with hatch cover alignment and fittings. In extreme cases, this can lead to extensive buckling of the cross deck structure between the hatch openings. Where ballast holds, and in some instances ballast tanks, are partially filled, there is the likelihood of sloshing. Sloshing is the violent movement of the fluid's surface in partially filled tanks or holds resulting from the motion of the ship in a seaway. Sloshing will result in the magnification of dynamic internal pressures acting on the hold/tank boundaries. For any tank design, dimensions, internal stiffening and filling level, a natural period (frequency) of the fluid exists, which, if excited by the ship's motions, can result in very high pressure magnification (resonance) which can result in damage to the tank/hold's internal structure.

Last but not least, the use of cargo handling equipment during the operations at port, can inflict damage to the ship’s structure. The internal hold structure and protective coatings in the cargo hold and the adjacent double bottom are vulnerable to damage when the cargo is discharged using grabs. The weight of empty grabs can be as much as 35 tonnes. Other types of equipment employed to free and clear cargo, including hydraulic hammers fitted to extending arms of tractors and bulldozers can inflict further damage to the ship's structure, especially in way of the side shell and the associated frames and end brackets. Chipping (sharp indentations) and the local buckling or detachment of side frames at their lower connection could lead to cracking of the side shell plating which would allow the ingress of water in to the cargo space.

The corrosive nature of the cargo can deteriorate the protective coatings applied in the cargo hold, whereas the same effect can be caused by the carriage of high temperature cargoes, of the cargo settlement during the voyage and the abrasive action of the cargo. Where no protective coatings have been applied or the applied protective coatings have broken down, the rate of corrosion in that area will greatly increase, especially when carrying corrosive cargoes, such as coal. Corrosion will weaken the ship's structure and may, eventually, seriously affect the ship's structural integrity. The severity of the corrosion attained by a structural member may not be easily detected without closeup inspection or until the corrosion causes serious structural problems such as the collapse or detachment of hold frames resulting in cracks propagating in the side shell. Impact damage to the inner bottom plating or the hopper sloping plating will result in the breakdown of coatings in the adjacent water ballast tanks, thus intensifying the rate of structural deterioration.

2.4 Net scantling approach

By using the term scantling[10], we refer to the determination of the geometrical dimensions for a structural component/system. The initial scantling design is one of the most important and challenging tasks throughout the process of structural design.
The net scantling approach as described in the Common Structural Rules, assumes[11] that various rates of corrosion will occur to the structural members during the lifetime of the vessel. The net scantling approach sets out to determine and verify the minimum hull scantlings that are to be maintained from the new building stage throughout the ship's design life to satisfy the structural strength requirements. It clearly separates the net thickness from the thickness added for corrosion that is likely to occur during the ship in operation phase.

The main concept of the procedure is the application of a general, average global hull girder and primary support member wastage (wastage allowance) such that the overall strength of these large structural members is maintained. The strength of these large members is assessed using a lower average corrosion margin. However, these large members are made up from a composite of local members comprising local elementary plate panels and stiffeners. The strength of these local strength members is assessed using the full local corrosion margins. The strength of the members is assessed using the structural capacity in the wasted condition, or net thickness, while applying the expected extreme loads. This will ensure that the vessel will meet the minimum strength requirements even while in the defined extreme wasted condition. Since fatigue is a cumulative mode of failure that starts from the first day of service when the vessel is in the as-built condition up until the last days of service when the vessel could be in a fully corroded state, the net thickness associated with hull girder and local thickness for fatigue is averaged or taken as half of the full margins.

Concluding, and according to the IACS[12], the wastage allowance is the value of thickness diminution due to corrosion expected during the service life of the ship obtained by statistical analysis based on the thickness measurement data of ships and the steel renewal criteria which ensure that the net thickness is kept throughout the service life of the ship. Additionally, the value of the corrosion addition is obtained from wastage allowance by adding to the thickness diminution predicted till the next thickness measurement. The above mentioned principle is illustrated in figure 2.5

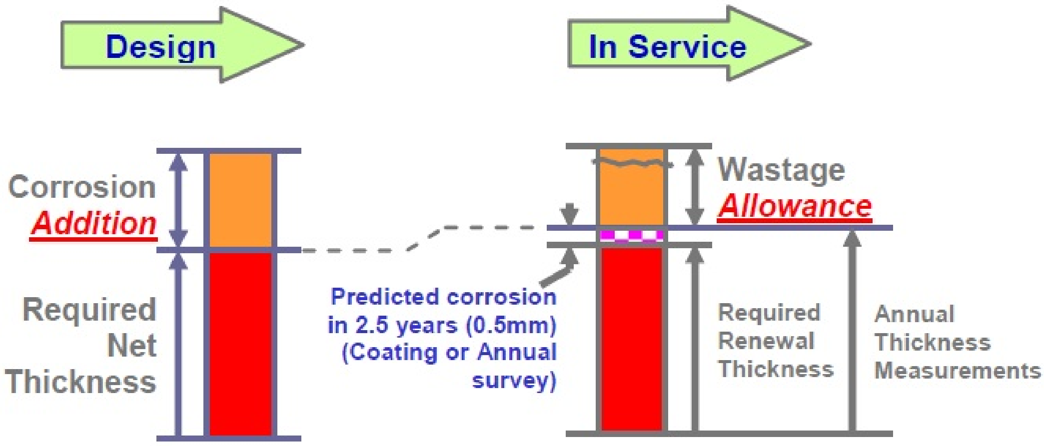

Figure 2.5

Net thickness principle [11]

According to Mansour et al.[13] , the net ship-based strength criteria has special value not only for design purposes but also in helping to formulate a maintenance strategy for the ship throughout its service life. The ship structure while in service is monitored for corrosion when thickness measurements or gaugings are taken during periodic surveys. When the thickness measurements indicate that the amount of corrosion wastage results in the thickness being equal to the net thickness, then renewal of the plate or member is required.

Focusing on the structure of the bulk carriers, the wastage allowance table that enables us to assess each structural member, is presented in table 2.1.

Table 2.1

One-side wastage allowance for bulk carriers[12]

<table>
<tr><th rowspan="2">Compartment type</th><th rowspan="2" colspan="2">Structural member</th><th colspan="2">One side wastage allowance in mm</th></tr>
<tr><th>BC-A or BC-B ships with $L \geq 150$ m</th><th>Other BC Ships</th></tr>
<tr><td rowspan="4">Ballast water tank(2) bilge tank, drain storage tank(7)</td><td rowspan="2">Face plate of primary members</td><td>Within 3m below the top of tank (3)</td><td colspan="2">2.0</td></tr>
<tr><td>Elsewhere</td><td colspan="2">1.5</td></tr>
<tr><td rowspan="2">Other members</td><td>Within 3 m below the top of tank (3)</td><td colspan="2">1.7</td></tr>
<tr><td>Elsewhere</td><td colspan="2">1.2</td></tr>
<tr><td rowspan="9">Dry bulk cargo hold (1)</td><td rowspan="3">Transverse bulkhead</td><td>Upper part (4)</td><td>2.4</td><td>1.0</td></tr>
<tr><td>Lower stool sloping plate, vertical plate and top plate (8)</td><td>5.2</td><td>2.6</td></tr>
<tr><td>Other parts</td><td>3.0</td><td>1.5</td></tr>
<tr><td rowspan="4">Other members</td><td>Upper part (4)</td><td rowspan="2">1.8</td><td rowspan="2">1.0</td></tr>
<tr><td>Webs and flanges of the upper end brackets of side frames of single side bulk carriers</td></tr>
<tr><td>Webs and flanges of lower brackets of side frames of single side bulk carriers</td><td>2.2</td><td>1.2</td></tr>
<tr><td>Other parts</td><td>2.0</td><td>1.2</td></tr>
<tr><td rowspan="2">Sloped plating of hopper tank, inner bottom plating</td><td>Continuous wooden ceiling</td><td>2.0</td><td>1.2</td></tr>
<tr><td>No continuous wooden ceiling</td><td>3.7</td><td>2.4</td></tr>
<tr><td rowspan="2">Exposed to atmosphere</td><td colspan="2">Weather deck plating</td><td colspan="2">1.7</td></tr>
<tr><td colspan="2">Other members</td><td colspan="2">1.0</td></tr>
<tr><td colspan="2">Exposed to sea water</td><td>Shell plating (6)</td><td colspan="2">1.0</td></tr>
<tr><td colspan="3">Fuel oil tanks and lube oil tank (2)</td><td colspan="2">0.7</td></tr>
<tr><td colspan="3">Fresh water tank</td><td colspan="2">0.7</td></tr>
<tr><td>Void spaces (5)</td><td colspan="2">Spaces not normally accessed, e.g. access only through bolted manholes openings, pipe tunnels, inner surface of stool space common with a dry bulk cargo hold or ballast hold, etc.</td><td colspan="2">0.7</td></tr>
<tr><td>Dry spaces</td><td colspan="2">Internal of machinery spaces, stores spaces, pump rooms, steering spaces, etc.</td><td colspan="2">0.5</td></tr>
</table>

Notes:

1. Dry bulk cargo hold includes holds, intended for the carriage of dry bulk cargoes, which may carry water ballast.
2. 0.7mm to be added to the plate surface exposed to ballast for plate boundary between water ballast and heated fuel oil or heated lube oil tanks. 0.3mm to be added to each surface of the web and face plate of a stiffener in a ballast tank and attached to the boundary between water ballast and heated cargo oil tanks. Heated oil tanks are defined as tanks arranged with any form of heating capability (most common type is heating coils).
3. This is only applicable to ballast tanks with weather deck as the tank top.. The 3 m distance is measured vertically from and parallel to the top of the tank.
4. Upper part of the cargo holds corresponds to an area above the connection between the top side and the inner hull or side shell. If there is no top side, the upper part corresponds to the upper one third of the cargo hold height (where a plane bulkhead is fitted in way of a dry cargo hold, the upper part of the bulkhead is defined in the same manner).
5. For the determination of corrosion addition of the outer shell plating, the pipe tunnel is considered as for a water ballast tank.
6. The thickness of the outer shell between the minimum design ballast draught amidship and the scantling draught waterline is to be increased by 0.5 mm.
7. 1.0mm is to be added to the plate surface within 3m above the upper surface of the chain locker bottom.
8. If there is no lower stool fitted (i.e. engine room bulkhead or fore peak bulkhead) or if a plane bulkhead is fitted, then this corrosion addition should be applied up to a height level with the opposing bulkhead stool in that hold. In the case where a stool is not fitted on the opposing bulkhead the vertical extent of this zone is to be from the inner bottom to a height level with the top of the adjacent hopper sloping plate.

2.5 Limit states

According to Paik et Thayamballi[14], limit state design is based on the explicit consideration of the various conditions under which the structure may cease to fulfil its intended function. For these conditions, the applicable capacity or strength is estimated and used in design as a limit for such behaviour. The load-carrying capacity of a structure is for this purpose normally evaluated using simplified design formulations or by using more refined computations such as nonlinear elastic–plastic large-deformation finite element analyses with appropriate modelling related to geometric/material properties, initial imperfections, boundary condition, load application, and finite element mesh sizes, as appropriate.

A limit state is formally defined by the description of a condition for which a particular structural member or an entire structure fails to perform the function that is expected of it. From the viewpoint of a structural designer, four categories of limit states are considered for steel structures, namely:

- Serviceability (or service) limit state (SLS)
- Ultimate limit state (ULS)
- Fatigue limit state (FLS)
- Accidental limit state (ALS)

SLS conventionally represents failure states for normal operations due to deterioration of routine functionality. SLS considerations in design may address:

(a) local damage which reduces the durability of the structure or affects the efficiency of structural elements
(b) unacceptable deformations which affect the efficient use of structural elements or the functioning of equipment supported by them
(c) excessive vibration or noise, which can cause discomfort to people or affect the proper functioning of equipment
(d) deformations and deflections that may spoil the aesthetic appearance of the structure.

ULS typically represents the collapse of the structure due to loss of structural stiffness and strength. Such loss of capacity may be related to:

(a) loss of equilibrium in part or of entire structure, often considered as a rigid body (e.g. overturning or capsizing)
(b) attainment of the maximum resistance of structural regions, members or connections by gross yielding, rupture or fracture
(c) instability in part or of the entire structure resulting from buckling and plastic collapse of plating, stiffened panels and support members.

FLS represents fatigue crack occurrence of structural details due to stress concentration and damage accumulation (crack growth) under the action of repeated loading.

Finally, ALS represents excessive structural damage as a consequence of accidents, e.g., collisions, grounding, explosion and fire, which affect the safety of the structure, environment and personnel.

A number of possible failure modes may be relevant for the various parts of the ship structure. For each failure mode, one or more limit states may be relevant. The failure modes to be considered for the assessment of ship structural safety with relation to the limit states, as proposed by the IACS[15], are shown in Table 2.2.

Table 2.2

Failure modes in relation to the limit states to be considered [15]

Possible failure modes to be considered	Limit states (1)			
	SLS	ULS	FLS	ALS
Yielding	Y	Y	-	Y
Plastic collapse	-	Y	-	Y
Buckling	Y	Y	-	Y
Rupture	-	Y	-	Y
Fatigue cracking	-	-	Y	-
Brittle fracture (2)	-	-	-	-

(1) "Y" indicates that the structural assessment is to be carried out.
(2) Controlled by the material rule requirement of steel grade.

2.6 Structural arrangement principles

In this part of the structural design process, the main principles affecting the hull structure configuration are presented. According to Lamb[16] the general arrangement designer should be aware of the impact of his decisions on the placement of the major structural components of the ship structure. Mainly, attention should always be paid, in order to maintain the structural continuity and the continuity of strength of the structure, and the avoidance of creating areas with high stress concentrations. The major principles are defined by the classification societies, whereas typical structural arrangements for each ship type have been established over the years. The major decisions a designer has to make in this step of the process have to do with the following components, as described in detail in the common structural rules booklets[17]:

- Stiffeners
- Primary supporting members (tripping brackets, end connections)
- The intersection area of stiffeners and the primary supporting members (cut outs, connections)
- Openings
- Pillars
- Deck structure
- Double bottom structure
- Side structure
- Bulkhead structure

The three last components are of major significance in the design process of bulk carriers, thus further detail is considered essential.

2.6.1 Double bottom structure

As defined in the CSR rules, for ships greater than 120 m in length, the bottomshell, the inner bottom and the sloped bulkheads of hopper tanks, if any, are to be longitudinally framed within the cargo hold region. Where it is not practicable to apply the longitudinal framing system to fore and aft parts of the cargo hold region due to the hull form, transverse framing may be accepted on a case-by-case basis subject to appropriate brackets and other arrangements being incorporated to provide structural continuity in way of changes to the framing system.

The height of double bottom in cargo area, d_{DB}, in m, measured from keel line at mid length of each cargo hold is not to be less than:

$$d_{DB} = 0.032B + 0.19\sqrt{T_{SC}}$$

,where B is the moulded breadth of the ship and T_{SC} the scantling draught (in meters).

A lower double bottom height may be accepted, provided all of the following requirements are satisfied:

• The spacing of adjacent girders is not to be greater than 4.6 m or 5 times the spacing of bottom or inner bottom stiffeners, whichever is the smaller.

• The spacing of floors is not to be greater than 3.5 m or 4 times the side frame spacing, whichever is the smaller. Where side frames are not transverse, the nominal frame spacing as specified by the designer is to be used.

Any variation in the height of the double bottom is generally to be made gradually and over an adequate length; the knuckles of inner bottom plating are to be located in way of plate floors. Where such arrangement is not possible, suitable longitudinal structures such as partial girders, longitudinal brackets, fitted across the knuckle are to be arranged.

In areas where a duct keel is arranged, the centre girder may be replaced by two girders spaced, no more than 3 m apart. Otherwise, for a spacing wider than 3 m, the two girders are to be provided with support of adjacent structure and subject to the Society's approval. The structures in way of the floors are to provide sufficient continuity of the latter. An illustration[18] of the various components of the bottom structure on a bulk carrier is shown in figure 2.6

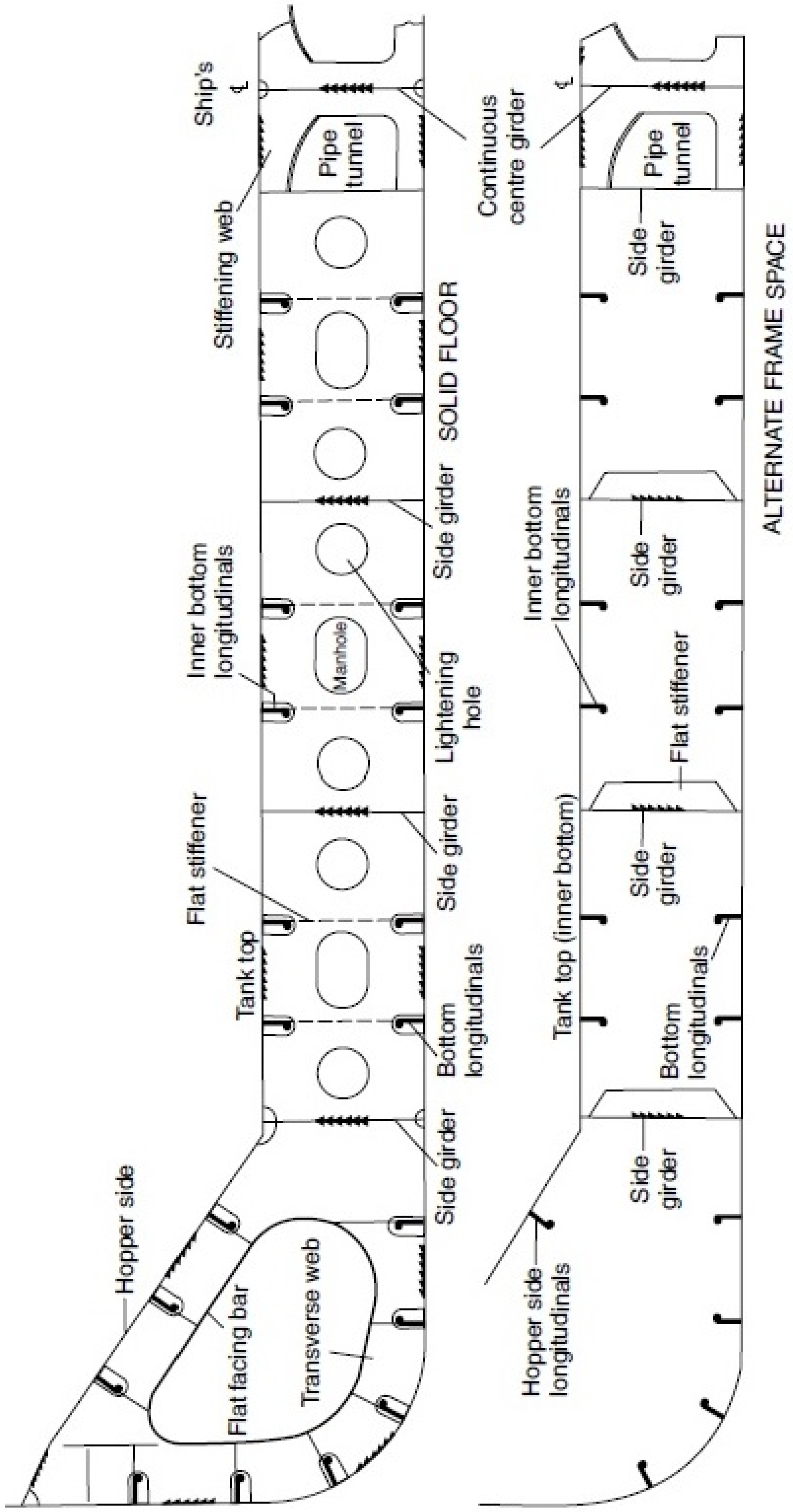

Figure 2.6

Bulk carrier double bottom construction [18]

Considering the girder spacing in bulk carrier structures, it is advised that the spacing of adjacent girders is generally not to be greater than 4.6 m or 5 times the spacing of bottom or inner bottom stiffeners, whichever is the smaller.

Finally, the spacing of floors is generally not to be greater than 3.5 m or 4 times the side frame spacing, whichever is the smaller. Further analysis of the rules develops principles on the construction of the keel plate, the stiffening of the floors and the bilge keel design.

2.6.2 Side structure

In conventional single-skin bulk carriers, the single side structure is supported by transverse or longitudinal primary supporting members. At every frame space a side frame[19] is to be arranged. Side frames are to be built-up symmetrical sections with integral upper and lower brackets and are to be arranged with soft toes. The side frame flange is to be curved (not knuckled) at the connection with the end brackets. The structural continuity with the lower and upper end connections of side frames is to be ensured within hopper and topside tanks by connecting brackets. An illustration of the above mentioned is shown in figures 2.7 & 2.8

In double-hull bulk carriers, the side shell, inner hull bulkheads and longitudinal bulkheads are generally to be longitudinally framed. Where the side shell is longitudinally framed, the inner hull bulkheads are to be longitudinally framed. Where the double side space of bulk carriers is void, the structural members bounding this space are to be structurally designed as a water ballast tank.

Double side web frames are to be fitted in line with web frames in hopper tanks. In addition, double side web frames are to be aligned with web frames or large brackets in topside tanks. Vertical primary supporting members are to be fitted in way of hatch end beams of bulk carriers or similar large deck opening supporting transverse structure.

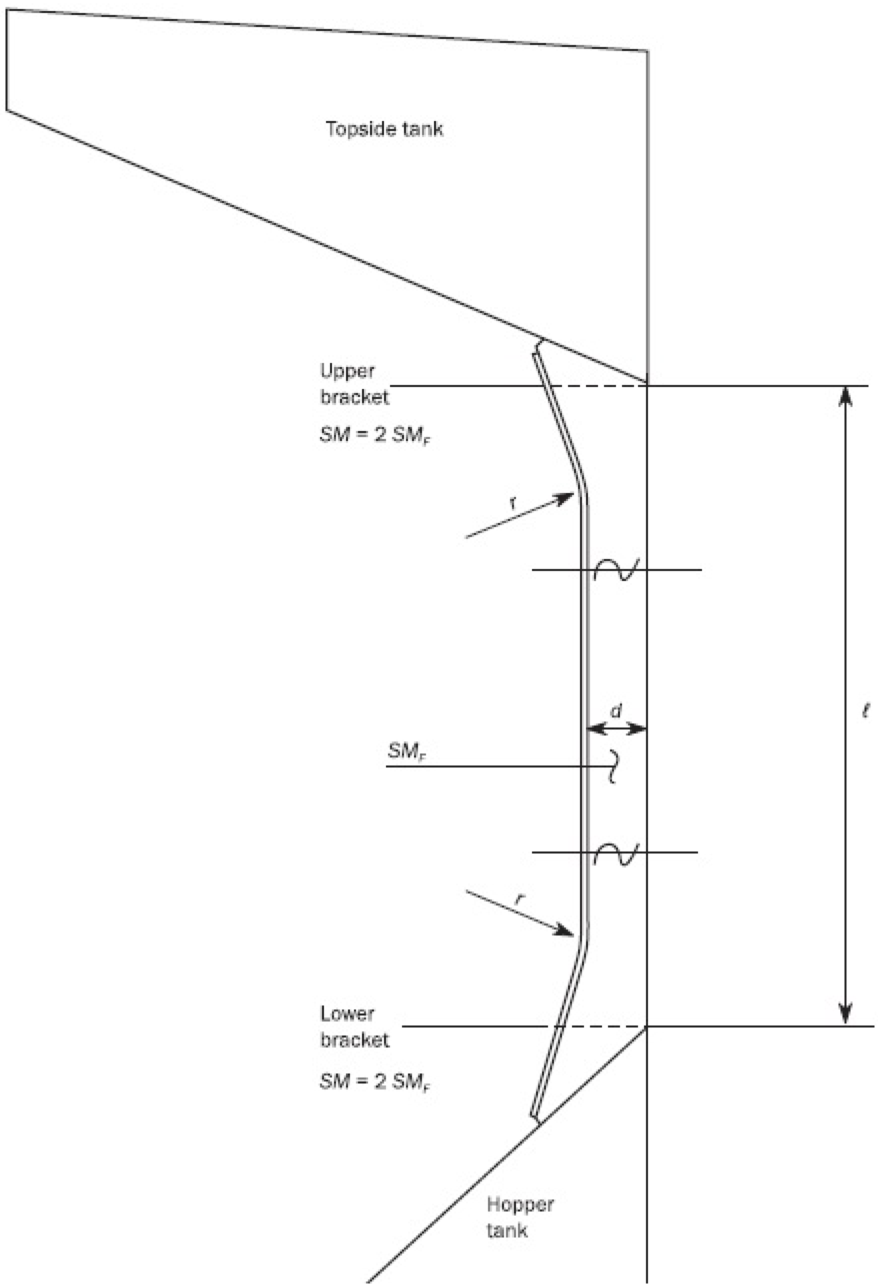

Figure 2.7

Side framing dimensioning [19]

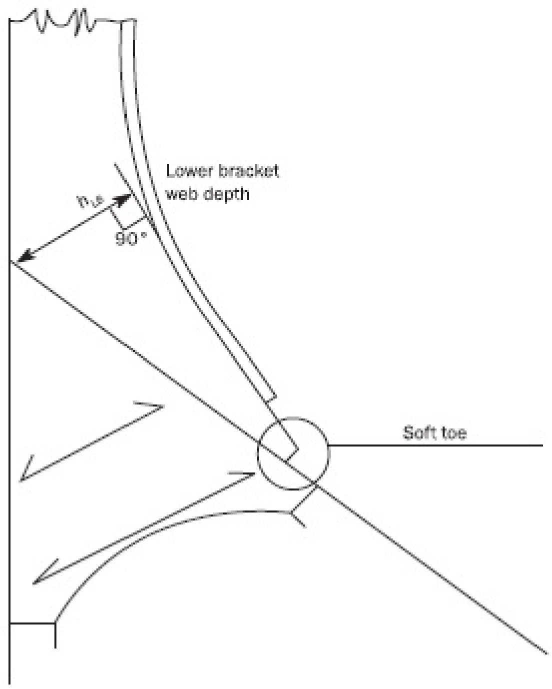

Figure 2.8

Example of support structure for lower end [19]

Transverse stiffeners on side shell and inner side, where fitted, are to be continuous or fitted with bracket end connections within the height of the double side. The transverse stiffeners are to be effectively connected to stringers. At their upper and lower ends, shell and inner side transverse stiffeners are to be connected by brackets to supporting stringer plates.

Longitudinal stiffeners on side shell and inner side, where fitted, are to be continuous within the length of the parallel part of the cargo hold region. They are to be fitted with soft toe brackets in way of transverse bulkheads aligned with cargo hold bulkheads and are to be effectively connected to transverse web frames of the double side structure.

Further design instructions are given for the connection of the inner hull plating and the inner bottom plating, and the configuration of the sheer strake.

2.6.3 Bulkhead structure

Bulkheads divide the ship into a number of watertight compartments, [Eyres[20]]. The main hull bulkheads of sufficient strength are made watertight in order that they may contain any flooding in the event of a compartment on one side of the bulkhead being bilged. Furthermore they serve as a hull strength member not only carrying some of the ship's vertical loading but also resisting any tendency for transverse deformation of the ship. As a rule, the strength of the transverse watertight bulkheads is maintained to the strength deck which may be above the freeboard deck. Finally each of the main hull bulkheads has often proved a very effective barrier to the spread of a hold or machinery space fire.

Two types of bulkheads are mainly used in ship structural design.

The *plane bulkheads*,that may be horizontally or vertically stiffened. Horizontally framed bulkheads are made of horizontal stiffeners supported by vertical primary supporting members. Vertically framed bulkheads are made of vertical stiffeners supported by horizontal stringers, if needed. The bulkhead stiffener webs of hopper and topside tank watertight bulkheads are to be aligned with the webs of longitudinal stiffeners of sloping plates of inner hull. Floors are to be fitted in the double bottom in line with the plane transverse bulkhead.

The main dimensions terminology of *corrugated bulkheads*[21] is presented in figure 2.9.

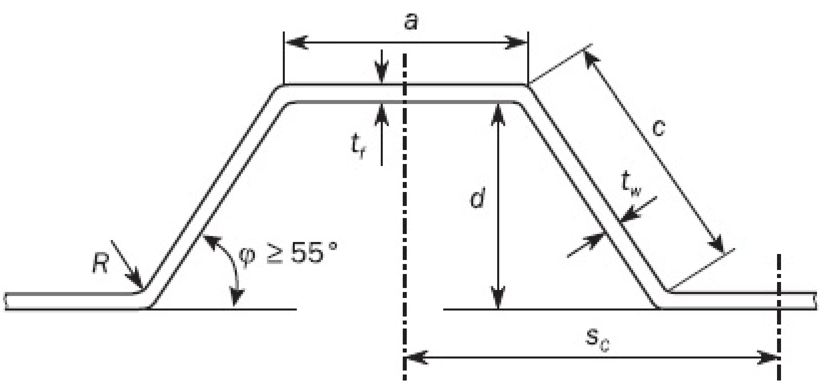

Figure 2.9

Corrugated bulkhead dimensioning [21]

The depth of the corrugation, d, in mm, is not to be less than:

$$d = \frac{1000l_c}{C}$$

where:

c : Mean span of considered corrugation, in m.

C : Coefficient to be taken as:

C = 15 for tank and water ballast cargo hold bulkheads.

C = 18 for dry cargo hold bulkheads.

Where a bulkhead is provided with a lower stool, floors or girders are to be fitted in line with both sides of the lower stool. Where a bulkhead is not provided with a lower stool, floors or girders are to be fitted in line with both flanges of the vertically corrugated transverse bulkhead. The supporting floors or girders are to be connected to each other by suitably designed shear plates. At deck, if no upper stool is fitted, transverse or longitudinal stiffeners are to be fitted in line with the corrugation flanges. A typical corrugated bulkhead has been illustrated in the previous chapter, in figure 1.10.

[1] Practical ship design (1998), David.G.M. Watson, pp 287
[2] The maritime engineering reference book (2008). Anthony F. Molland , pp 149
[3] Strength of ships and ocean structures (2008), Alaa Mansur & Donald Liu, pp 3
[4] Comparative study of ship structure design standards, SSC-446 (2007), Ship Structure Committee, page 9.
[5] Piero Caridis, book under publication
[6] Bulk Carries: handle with care, IACS , page 2
[7] Bulk Carriers: Design, Operation, and Maintenance Concerns for Structural Safety of Bulk Carriers, Ship Structure Committee, p 4
[8] Ban on alternate hold loading, The motorship, May 2006, page 30
[9] BULK CARRIERS: Guidance and Information on Bulk Cargo Loading and Discharging to Reduce the Likelihood of Over-stressing the Hull Structure, (1997), IACS.
[10] Marine structural design, (2003) Yong Bai, pp 71
[11] The development, implementation and maintenance of IACS common structural rules for bulk carriers and oil tankers, (2007) Gary Horn & Philippe Baumans.
[12] IACS Harmonised CSR –TB Report No: Pt1, Ch 3, Sec 3, Corrosion Additions and Wastage Allowances (2012), pp 3
[13] Strength of ships and ocean structures (2008), Alaa Mansour & Donald Liu, pp 98
[14] Ultimate limit state design of steel plated structures (2003), Jeom Kee Paik & Anil Kumar Thayamballi , pp2
[15] IACS, Common structural rules for bulk carriers and oil tankers, 01 Jan 2014, pp 99
[16] Ship design and construction, Vol I (2003), Thomas Lamb
[17] IACS, Common structural rules for bulk carriers and oil tankers, 01 Jan 2014
[18] Ship Construction, 6th edition (2007), D.J. Eyres, pp 171
[19] IACS, Common structural rules for bulk carriers and oil tankers, 01 Jan 2014, pp 737
[20] Ship Construction, 6th edition (2007), D.J. Eyres, pp 191
[21] IACS, Common structural rules for bulk carriers and oil tankers, 01 Jan 2014, pp 130

3

3.1 Introduction

The third chapter of this thesis outlines the determination of loads affecting the hull structure. The first loads assessed are the ones that are present in the still water conditions, meaning in conditions where the ship floats in calm water. The main components of this category are still water bending moments and shear forces. Those static loads should be supreimposed to the wave induced loads, in order to assess the total forces that result in negligible dynamic stress amplification of the structure. IACS formulas for the calculation of those loads are presented, whereas typical distributions of allowable and attained forces in specific cases are illustrated.

Subsequently, load cases as accepted by the IACS are presented. They describe situations where under specific regular waves, the long term response values of the load components considered being predominant to the structural members. Furthermore, the findings of a study concerning the design loads on primary structural members of a bulk carrier are discussed.

The distribution of the external pressures is the following step in the prescribed assessment procedure. This includes not only the definition of the hydrostatic pressure, but also the description of the hydrodynamic loads affecting the hull. Furthermore, attention is paid to the modification of the pressures that can arise from the avoidance of heavy weather situations, conducted by the crew of the vessel.

Moreover, the presence of internal pressures that are taken under consideration is discussed. The major load component of this category corresponds to the forces due to the bulk cargo and the liquids loaded onboard.

Finally, the loading conditions section defines a number of load cases which are likely to impose the most onerous local and global load regimes that are to be investigated in the structural analysis. The hold mass curves section outlines the use of such plots in the determination of the allowable mass of cargo as a function of draught.

3.2 Still water loads

The first part on the determination of the hull girder loads copes with the assignment of the loads that are present when the vessel is in the calm water condition. Those loads are mainly the still water shear force and the bending moment, that causes the ship either to lie on hogging or on sagging condition.

The primary response analysis[1] is carried out by hypothesizing that the entire hull of a ship behaves like a beam whose loading is given by the longitudinal distribution of weights and buoyancy over the hull. Since[2] cargo and ballast are being changed over time, the still water bending moment and shear force will also change over time. As in any beam stress computation, it is necessary first to integrate the loads to obtain the longitudinal distribution of the total shear force, and then to integrate them again to obtain the bending moment. The still water loads contribute an important part of the total shear and bending moment in most ships, to which wave induced effects must be added later. Considering a given longitudinal location, x, the shear force is the upward force that the left portion of the ship exerts on the portion to the right of this location. Similarly, the bending moment is the resultant moment exerted by the left portion on the portion of the ship to the right of location x. The conditions of static equilibrium require that the shear force and the bending moment be equal to zero at both ends of the ship.

The calculation of the maximum permissible still water shear force and bending moment, is a procedure described in detail in the Common Structural Rules by the IACS. Each classification society has different methods of assigning those values, that usually respond to different capacities calculated (examples for bulk carriers calculations can be found in the IACS Harmonized CSR TB Report, on Still water Bending moment, SWBM / Report No: Pt 1,Ch 9, Sec 3, July 2012, tables 1&2).

Focusing on the IACS calculations, the minimum still water bending moment[3] in the hogging condition (in kNm) derives from a simple equation:

$$M_{SW-h-\min} = f_{SW}(171 C_W L^2 B (C_B + 0.7) 10^{-3} - M_{WV-h-mid})$$

,where

- f_{sw} is a distribution factor along the ship's length,
- and $M_{WV\text{-}h\text{-}mid}$ is the vertical wave bending moment for strength assessment in hogging condition. The vertical wave bending moment at any longitudinal position is calculated by an equation of similar shape, $M_{WV-h} = 0.19 f_{nl-vh} f_m f_p C_W L^2 B C_B$

, where f_i are the various coefficients considering the nonlinear effects applied to hogging, the mode of assessment and the distribution along the ship's length.

Further consideration is given on the minimum bending moments, corresponding to other situations, such as the seagoing condition, the harbor condition or a tank testing condition and the flooded scenarios at sea. Equations of similar form are used for the determination of the minimum still water bending moments on sagging condition. As a result, an envelope noting the permissible bending moments can be created, as illustrated[4] in figure 3.1

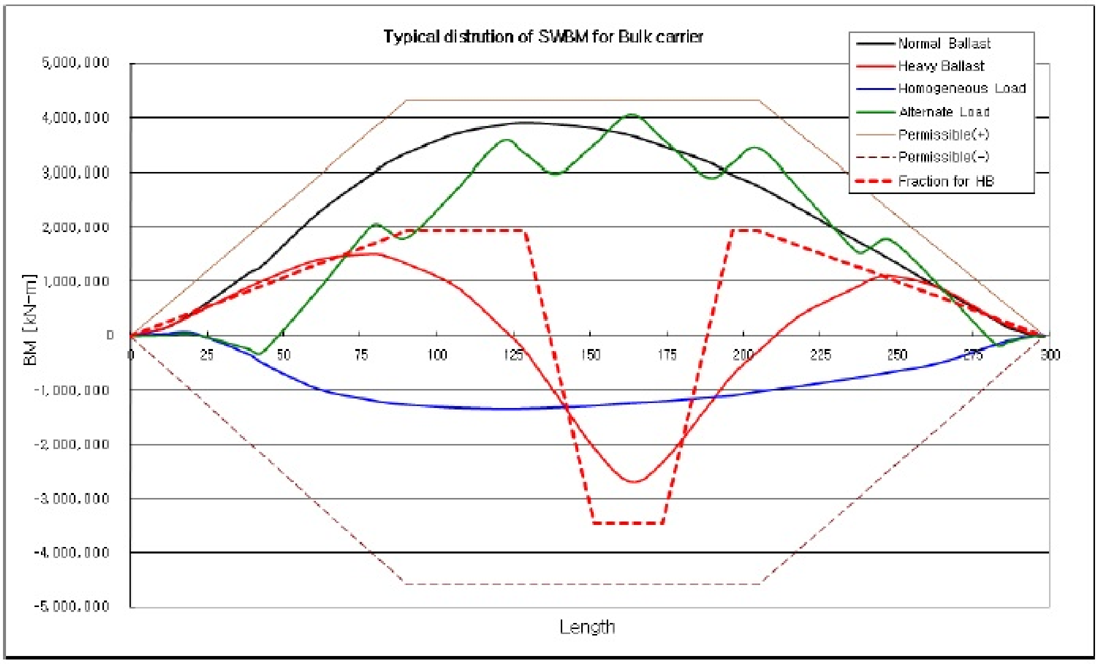

Figure 3.1

Typical distribution of SWBM for bulk carrier [4]

The shearing force[5] at any position of the ship's length is that force which tends to move one part of the ship vertically relative to the adjacent portion. The procedure for the derivation of the minimum hull girder positive and negative shear forces in various seagoing conditions or in sheltered waters is once again described in the CSR rules. The variation in the still water shear stress distribution along the ship's length for different flooding scenarios has been depicted by Paik et al.[6] in figure 3.2. For a capsize bulk carrier, seven scenarios where taken under consideration, namely

L1: Ballast condition / Intact

L2: Alternate ore load condition / Intact

L3: Alternate ore load condition / Hold No 5 flooded

L4: Alternate ore load condition / Hold Nos 4&5 flooded

L5: Alternate ore load condition / Hold Nos 4,5&6 flooded

L6: Alternate ore load condition / Hold No1 flooded

L7: Alternate ore load condition / Hold Nos 1&2 flooded

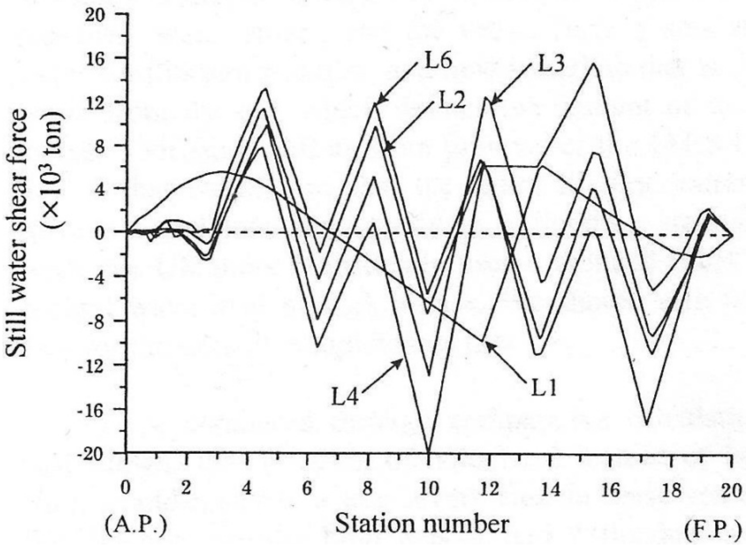

Figure 3.2

Distribution of still water shear force in the assumed flooding scenarios [6]

It is evident that some cases of hold flooding can amplify the magnitude of the hull girder loads occasionally beyond the design values. Decrease of the residual hull girder strength together with the increase of the hull girder loads may possibly lead to hull girder breakage.

For the flooded conditions L5 & L7, in which three holds amidships or two forward holds are flooded, respectively, it is seen that the vessel can founder since the draught exceeds the ship depth resulting from loss of the reserve buoyancy. It can be said that if more than two cargo holds are flooded in laden condition, the possibility of foundering could be significant. This is of course not unexpected because bulk carriers are ordinarily not designed to such a (2 or 3 compartment) standard for flooding or damage stability purposes.

For the flooded conditions L4 & L6, in which two holds amidships or one forward hold are flooded, respectively, it is seen that the magnitude of extreme hull loads could become very large, potentially leading to hull girder collapse. In this particular case, it is this evident that if more than one cargo hold particularly forward part is flooded the possibility of foundering due to hull girder collapse could be large even if the survival buoyancy is sufficient in the beginning stages of flooding. Moreover, once flooding occurs in one hold, particularly forward part, progressive flooding into the adjacent holds by collapse of transverse bulkheads is possible if the bulkheads had not been previously specifically designed to withstand such accidental flooding conditions.

3.3 Wave induced loads

The ability to predict the behavior of the ship in waves represents a key point to the quantification of global and local loads acting on the ship. The principal wave loads are those referred to as low-frequency dynamic loads or loads involving ship and wave motions that result in negligible dynamic stress amplification. Once these quasi-static loads are determined, the structural response in terms of stress or deflection can be computed by methods of static structural analysis. According to Mansour et al.[7], four procedures of varying degrees of sophistication may be used to estimate the wave-induced loads and their resultant bending moments and shear forces.

Approximate methods are used for making early estimation of the hull structural loading. Such methods include the use of semi-empirical formulations and quasi static computations. The main concept of the procedure is that the ship lies in a state of static equilibrium, on either the crest or trough of a wave, whose length is equal to the ship's length between perpendiculars (L), and whose height is L/20. Other sources have proposed different standard wave height, such as $0.6L^{0.6}$ for the ABS and $1.1L^{0.5}$ from the US Navy. In practice, such methods are proven to be overestimating the wave induced bending moments.

On the other hand, the IACS[8] has proposed formulas for the assignment of the maximum vertical and horizontal wave bending moments, in intact and flooding condition, the vertical wave shear force and the wave torsional moment. For the vertical wave bending moment, the mode of equation used has been mentioned in the still water loads analysis just above.

$$M_{WV-h} = 0.19 f_{nl-vh} f_m f_p C_W L^2 B C_B$$

$$M_{WV-s} = -0.19 f_{nl-vs} f_m f_p C_W L^2 B C_B$$

,where f_i are the various coefficients considering the nonlinear effects applied to hogging or sagging, the mode of assessment and the distribution along the ship's length.

For the vertical wave shear forces, the equations used are:

$$Q_{WV-pos} = 0.52 f_{q-pos} f_p C_W LBC_B$$

$$Q_{WV-neg} = -0.52 f_{q-neg} f_p C_W LBC_B$$

,where f_i are the various coefficients considering the nonlinear effects, the mode of assessment and the distribution of the shear force along the ship's length.

Figure 3.3 illustrates the distribution of the wave induced shear force[9] across the bulk carrier's length, for the various intact and flooded conditions mentioned in the still water loads presentation.

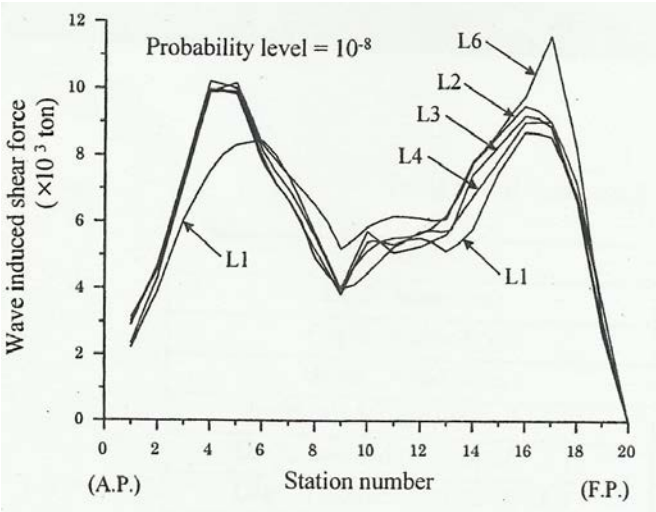

Figure 3.3

Distribution of wave induced shear force in the assumed flooding scenarios [9]

Finally, according to the IACS CSR, the wave torsional moment at any longitudinal position is to be taken (in kNm):

$M_{wt} = f_p (M_{wt1} + M_{wt2})$, where

$$M_{wt1} = 0.4 f_{t1} C_W \sqrt{\frac{L}{T_{LC}}} B^2 D C_B$$

$$M_{wt2} = 0.22 f_{t2} C_W L B^2 C_B$$

f_{t1} and f_{t2} are distribution factors across the ship's length

f_p is a coefficient depending on the type of assessment (strength or fatigue)

Another method for assessing wave induced loads is taking *strain and pressure measurements on actual ships*. As one can suggest, this method applies in existing ships and is not applicable for new ship designs. The principal value of full scale load response (stress or strain) measurements lies in the development of long term statistical trends of seaway induced hull loads from measurements carried out over a multiyear period. Full-scale monitoring designed mainly as a decision support for ship maneuvering, can also be used to monitor stresses in ships as predicted by numerical calculations.

The third method mentioned by Mansour, is the *measurement of loads on laboratory models*. In this procedure, a model geometrically and dynamically similar to the ship is equipped with instruments that measure vertical or horizontal shear and bending moment, or torsional moment, amidships and at other sections. This may be accomplished by recording the forces or deflections between several segments produced by transverse cuts through the model. Impact loads can also be determined by recording pressures at several points distributed over the model surface. The experiments are conducted in a towing tank that is equipped to produce either regular or random waves. As expected, this is a time consuming and expensive procedure, thus nowadays, the principal use for model testing is to provide verification on computer aided techniques.

Last but not least method is the direct *computation of the wave induced fluid load*. In this procedure, appropriate hydrodynamic theories used to calculate ship motions in waves are applied to compute the pressure forces caused by the waves and ship motion in response to those waves. The total structural loading at any instant is expected as the sum of the wave pressure forces, the ship motion induced pressures, and the reaction loads due to the acceleration of the ship masses. Various theories are used for this assessment (such as the frequency linear strip theory, the linear three dimensional theory, quadratic strip theory, the time domain strip theory, etc), but the analysis of them overcomes the scope of this thesis.

3.4 Load cases

In order to generate the dynamic load cases for structural assessment (strength and fatigue), a variety of Equivalent Design Waves (EDW) is used. The term EDW refers to regular waves that generate response values equivalent to the long term response values of the load components considered being predominant to the structural members.

The latest IACS CSR rules[10] designate the following EDW's. Numbers 1 and 2 denote the maximum or the minimum dominate load component for each EDW, whereas P and S denote that the weather side is on port side and on starboard side respectively.

- *HSM load cases*/ HSM-1 and HSM-2: Head sea EDWs that minimise and maximise the vertical wave bending moment amidships respectively.
- *HSA load cases*/ HSA-1 and HSA-2: Head sea EDWs that maximise and minimise the head sea vertical acceleration at FP respectively.
- *FSM load cases*/ FSM-1 and FSM-2: Following sea EDWs that minimise and maximise the vertical wave bending moment amidships respectively.
- *BSR load cases*/ BSR-1P and BSR-2P: Beam sea EDWs that minimise and maximise the roll motion downward and upward on the port side respectively with waves from the port side.
  BSR-1S and BSR-2S: Beam sea EDWs that maximise and minimise the roll motion downward and upward on the starboard side respectively with waves from the starboard side.
- *BSP load cases* / BSP-1P and BSP-2P: Beam sea EDWs that maximise and minimise the hydrodynamic pressure at the waterline amidships on the port side respectively.
  BSP-1S and BSP-2S: Beam sea EDWs that maximise and minimise the hydrodynamic pressure at the waterline amidships on the starboard side respectively.

- *OST load cases*/ OST-1P and OST-2P: Oblique sea EDWs that minimise and maximise the torsional moment at 0.25L from the AE with waves from the port side respectively.
  OST-1S and OST-2S: Oblique sea EDWs that maximise and minimise the torsional moment at 0.25L from the AE with waves from the starboard side respectively.
- *OSA load cases*/ OSA-1P and OSA-2P: Oblique sea EDWs that maximise and minimise the pitch acceleration with waves from the port side respectively.
  OSA-1S and OSA-2S: Oblique sea EDWs that maximise and minimise the pitch acceleration with waves from the starboard side respectively.

Following the determination of the EDW's, for each situation, the ship motions responses are described in tables by the society (reference tables 1-3, pages 166-168 of the IACS CSR abovementioned booklet) and the global loads corresponding to each dynamic load case to be considered are mentioned for the strength assessment.

Finally, the reference value of the global loads and the inertia load components (hull girder loads, longitudinal/transverse/vertical accelerations) is to be multiplied by a relevant Load Combination Factor (LCF), in order to achieve the desirable assessment.

Focusing on the bulk carrier structures, a study carried by Zhu & Shigemi[11] evaluated the design loads on primary structural members (figure 3.4) of bulk carriers and came up with the sea states having the maximum effect on structural strength. The dominant sea states were the:

- Vertical bending moment at head sea (L-180)
- Vertical bending moment at following sea (L-0)
- Roll (R)
- Hydrodynamic pressure at waterline (P)

The results of the analysis showed that in *homogenous loading condition*, the design regular wave L-0 or P under which the resultant pressure of the external and internal pressure is relatively large at the midship section is the dominant wave condition for the primary structural members of double bottom that is directly affected by the resultant pressure, and the hold frames that is affected by the deformations of the double bottom. Moreover, the design regular wave L-180 is the dominant wave condition for the transverse bulkheads, lower stools and girders near the lower stools, which are easily influenced by loads in the longitudinal direction of the ship. While the design regular wave R is not the dominant wave condition for the examined primary structural members in the homogenous load condition.

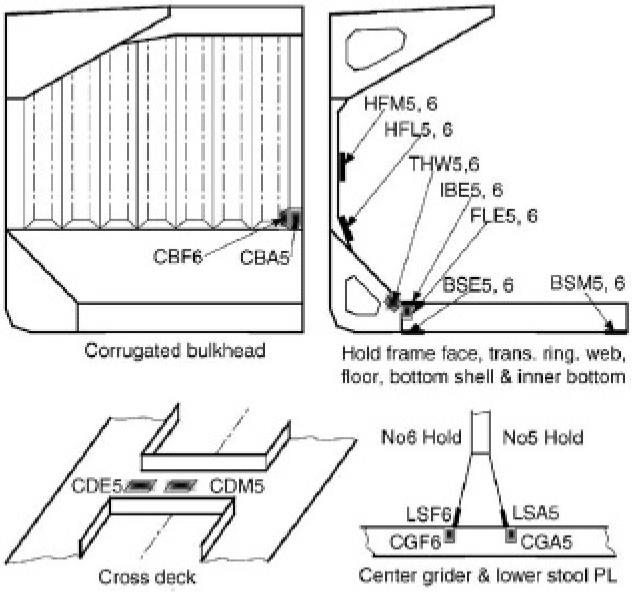

Figure 3.4

Locations of primary structural members used in the study [11]

Considering the *alternate load condition*, the results have the same character as those obtained for the homogenous load condition. However, the design regular wave L-180 and P have larger influence on the strength of the primary structural members, compared with the homogenous load condition. The responses of the primary structural members have the same tendency for both the full load conditions, as the internal dynamic pressure does not exist in the design regular wave L-0, and hydrodynamic pressure is the only dynamic load. On the other hand, the primary structural members mainly in the loaded hold are largely influenced by the inertial forces due to cargo in the design regular waves L-180 and P, in the alternate load condition. The stresses of the primary structural members in the holds near the bow and stern are relatively large in the design regular wave L-180 due to the effect of pitching motion compared with the stresses in the design regular waves L-0,P and R under which the load distribution along the longitudinal direction of the ship is almost the same. The absolute values of the stresses are larger in alternate load condition than in homogenous load condition in general.

Finally, in the *heavy ballast condition*, the design regular waves L-0 and P are the dominant wave condition for most of the primary structural members. However, the design regular wave L-180 has relatively large effect on the stresses of the primary structural members compares with both full load conditions, because the hydrodynamic pressure and the inertial forces of ballast water due to vertical acceleration are overlapped near the midship section. Furthermore, the regular design wave R is the dominant wave for the hold frames at the ballast hold (No 6 hold), as the inertial forces of ballast water due to transverse acceleration is large. But the magnitudes of the stresses are generally smaller compared with that in both the full load conditions.

3.5 External pressures

The determination of external loads affecting the hull is a complicated process, because the external pressure is influenced by a large number of parameters, such as hull form, wave motion characteristics, ship speed, heading angles etc. According to Bai[12], the various methods used for the determination of the external pressure on a ship are usually based on a number of assumptions, thus the values calculated should be used with caution.

The external loads that act as local transverse loads for the hull plating and the supporting structure consist of two components, one static and one dynamic. The static pressure[13] is the hydrostatic pressure P_S that is related to the vertical distance between the free surface and the load point. According to the IACS[14] , the hydrostatic pressure (in kN/m^2) derives from the formula $P_S = \rho g (T_{LC} - z)$. An illustration of the shape and the terminology used in this type of pressure is given in figure 3.5.

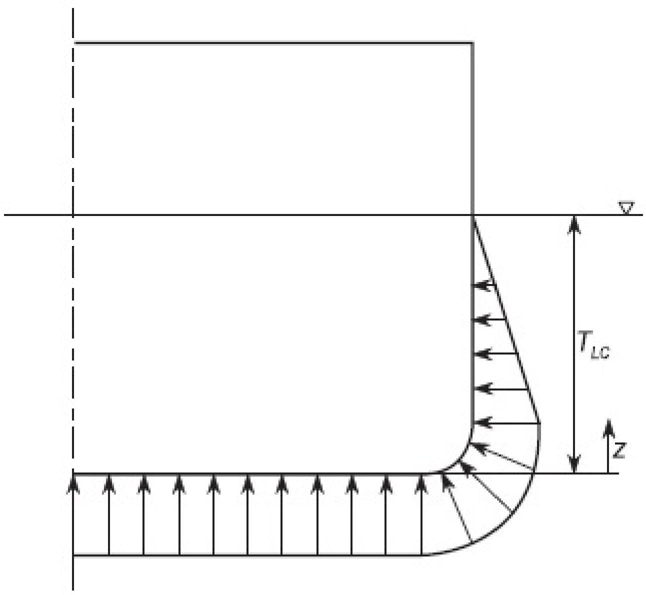

Figure 3.5

Hydrostatic pressure P_S [14]

Moving towards the more complex determination of hydrodynamic loads, for each load case described in the previous section of this chapter, the position of the waterline in comparison with the still wave situation is different, thus the distribution of the pressure is significantly modified. At this point, and exclusively for illustration purposes, we present the calculations needed for the hydrodynamic pressure P_W, just for one load case scenario, taking under consideration the BSP load case. A situation with beam sea equivalent design waves that maximise and minimise the hydrodynamic pressure at the waterline amidships,on the port and on starboard side, respectively. The hydrodynamic pressure distribution is shown in figure 3.6, whereas the value of P_W derives from the formulas in table 3.1. Various coefficients and other parameters are used, such as girth distribution coefficients, coefficients for non-linear effects and ballast water exchange scenarios. It is easy for everyone to trace back the full procedure, at any scenario, in the IACS rules.

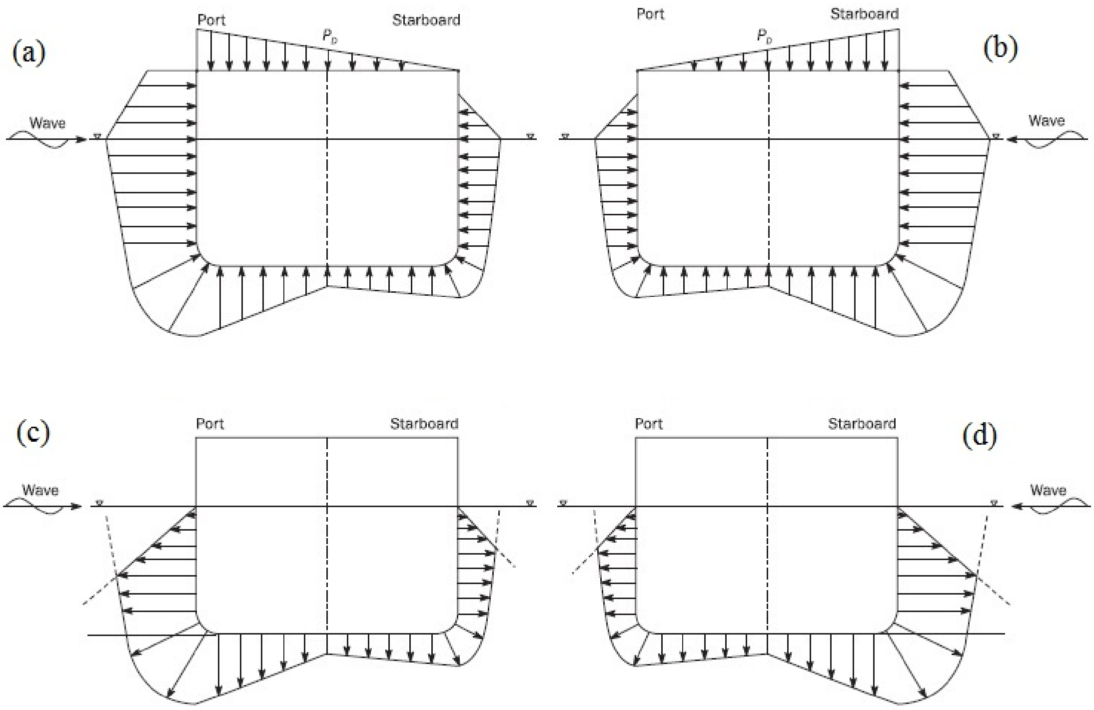

Figure 3.6

Transverse distribution of dynamic pressure for (a) BSP-1P, (b) BSP-1S, (c) BSP-2P and (d) BSP-2S. [14]

Table 3.1

Hydrodynamic pressures for BSP load cases [14]

	Wave pressure, in kN/m²		
Load case	$z \le T_{LC}$	$T_{LC} < z \le h_W + T_{LC}$	$z > h_W + T_{LC}$
BSP-1P	$P_W = \max(P_{BSP}, \rho g (z - T_{LC}))$	$P_W = P_{W,WL} - \rho g (z - T_{LC})$	$P_W = 0.0$
BSP-2P	$P_W = \max(- P_{BSP}, \rho g (z - T_{LC}))$		
BSP-1S	$P_W = \max(P_{BSP}, \rho g (z - T_{LC}))$		
BSP-2S	$P_W = \max(- P_{BSP}, \rho g (z - T_{LC}))$		

The IACS CSR rules provide further guidance on other external pressures that might contribute in the strength assessment of the structure. Pressures on exposed decks such as the green seas phenomena and the presence of distributed or unit loads are described thoroughly. Furthermore, external impact pressures for the bow area are discussed, focusing on the bottom slamming and the bow impact scenarios. Finally, the hydrodynamic pressures affecting the superstructure, the deckhouses and the hatch covers are mentioned.

A study by Shu & Moan[15] assessed the effect of heavy weather avoidance on the wave pressure distribution along the midship transverse section of a bulk carrier. When sailing on a seaway, the shipmasters will in general try to avoid severe sea states by adopting actions such as reducing speed, changing course or both of the formers according to certain limiting operational criteria relating to the safety and comfort of passengers and crew, to the safety and capacity of the vessel or to operational considerations.

Since severe sea states are avoided, the occurrences of actual sea states encountered by ships during its service life must be different from those given by the scatter diagrams for the geographical area in which the ship is operating. This especially must be true for high sea states. It is also known that high sea states usually contribute most to the long term prediction values.

The comparison of the wave pressure distribution envelope along the midship transverse section of a fully laden capsize bulk carrier, obtained by simplified rule formulas and that obtained by long term prediction with different roll damping at exceedance probability level of 10^{-8} is shown in figure 3.7.

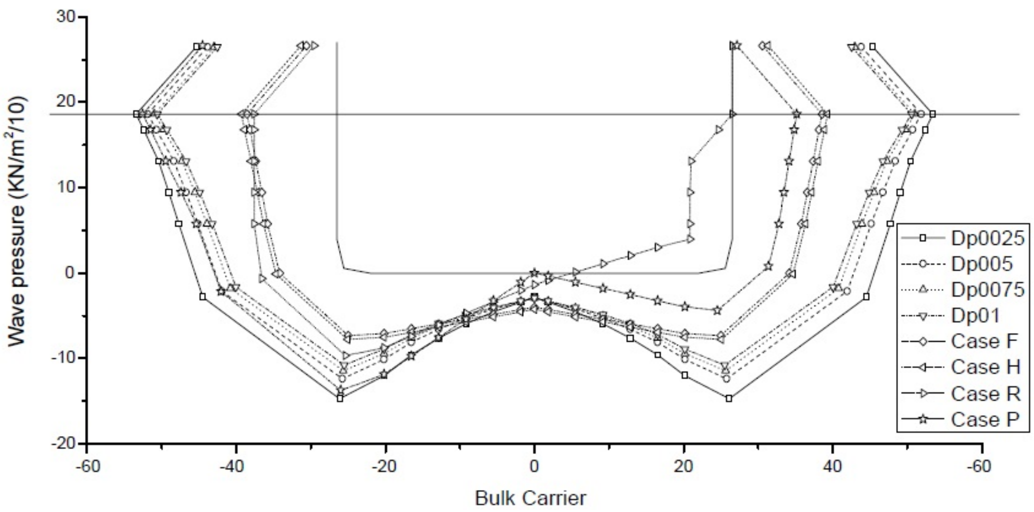

Figure 3.7

Wave pressure distributions of a bulk carrier in the full load condition (IACS wave data). Dp0025, Dp005, Dp0075 and Dp01 represent 2.5%, 5%, 7.5% and 10% of critical damping, respectively. [15]

The results of the study showed that the roll damping is of significant importance for the long term prediction values of wave pressures along the midship transverse section of the bulk carrier, especially for the area between the bilge and the centre bottom. The extreme value of wave pressure at the centre bottom is not affected by the roll damping.

Compared with other (OCEANOR) wave data, which are believed to describe the real wave condition on sea, the IACS wave data usually yields lower extreme values of wave pressures along the midship transverse section, which indicate to some extent that the IACS wave data has implicitly included the effect of heavy weather avoidance. Finally, the influence of heavy weather avoidance on the extreme values of wave pressure along the midship transverse section is dependent on how the heavy weather avoidance is accounted for.

Other comparison efforts[16] between the IACS formulas and long term predictions on the hydrodynamic pressure distribution at midship section of a bulk carrier observed (figure 3.8) that the maximum values among the hydrodynamic pressures under the seven EDWs (HSM, HSA, FSM, BSR, BSP, OST and OSA) obtained by the simplified formulae are almost equivalent to the long-term prediction value of the hydrodynamic pressure at any point of the cross section of the ship.

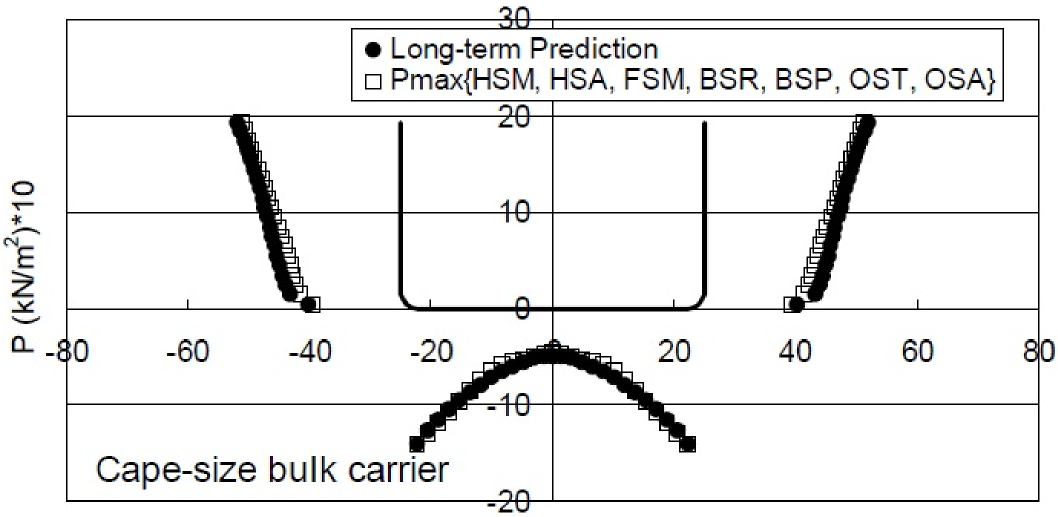

Figure 3.8

Comparison of hydrodynamic pressures at midship section obtained by long-term prediction and those obtained by simplified formulae for a Capesize bulk carrier. [16]

3.6 Internal pressures

Having determined the external pressures acting on the hull, the identification of the internal pressures that affect the structure is the next complex step. The major load component of this category corresponds to the forces due to the bulk cargo and the liquids loaded onboard.

The bulk cargo load consists of two parts[17]: static and dynamic which correspond to the weight of cargo and the inertial load of cargo due to accelerations of the vessel, respectively. The bulk cargo pressure is further decomposed into normal and tangential components at the cargo hold boundary. According to Lamb[18], internal friction forces arise within the cargo itself and between the cargo and the walls of the hold. As a result, the component normal to the wall has a different distribution from the load corresponding to a liquid cargo of the same density.

For the determination of the lateral pressure of dry bulk cargo, for both the static and dynamic cases, empirical formulations based on the material frictional characteristics such as the angle of repose and the slope of the wall are usually utilized. A definition of the angle of repose is given by Rhodes[19]. When solid bulk cargoes such as grain are loaded, they are usually poured into the ship's hold. If they are poured onto one spot, a conical shaped pile will form, which will have a certain slope profile. The angle of repose is the maximum slope angle of non-cohesive (free flowing) granular material. It is the angle between a horizontal plane and the cone slope of such a material. The angle of repose is governed by the shape and surface of the individual particles of cargo within a particular stow and cargo moisture content. The less the angle of repose of the cargo; the greater the ease with which the cargo will shift. In the event of the absence of precise data for the cargo transported, the angle of repose can be taken at 30^o in general cases, 35^o for iron ore and 25^o for cement cargoes.

For fully and partially filled cargo holds, the Common Structural Rules[20] define the shape of the upper surface of the cargo. When the cargo hold is loaded to the top of the hatch coaming, the upper surface of the dry bulk cargo is an equivalent horizontal surface at h_C, in m, above inner bottom at centerline as shown in figure 3.9.

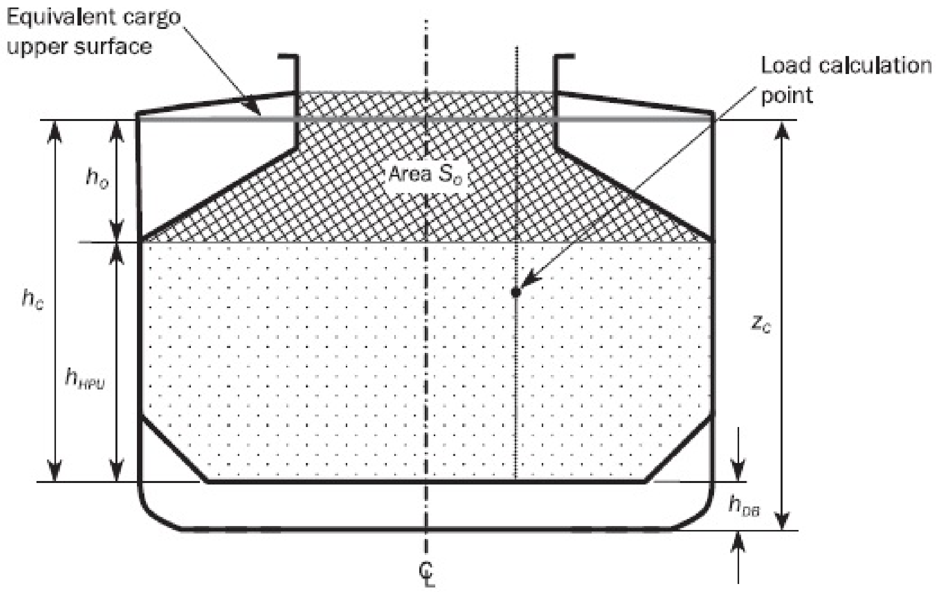

Figure 3.9

Definition of effective upper surface of cargo for a full cargo hold [20]

On the other hand, for heavier cargoes where the cargo hold is not loaded up to the upper deck, the effective upper surface of the cargo is to be made as below mentioned (figure 3.10): One central horizontal surface of breadth $B_H/2$, in m, at a height $h_{C\text{-}CL}$, in m, above the inner bottom. A sloped surface at each side with an angle ψ/2, in degrees, (where ψ the angle of repose) between the central horizontal surface, and the side shell or inner hull, or the hopper plating, as the case may be.

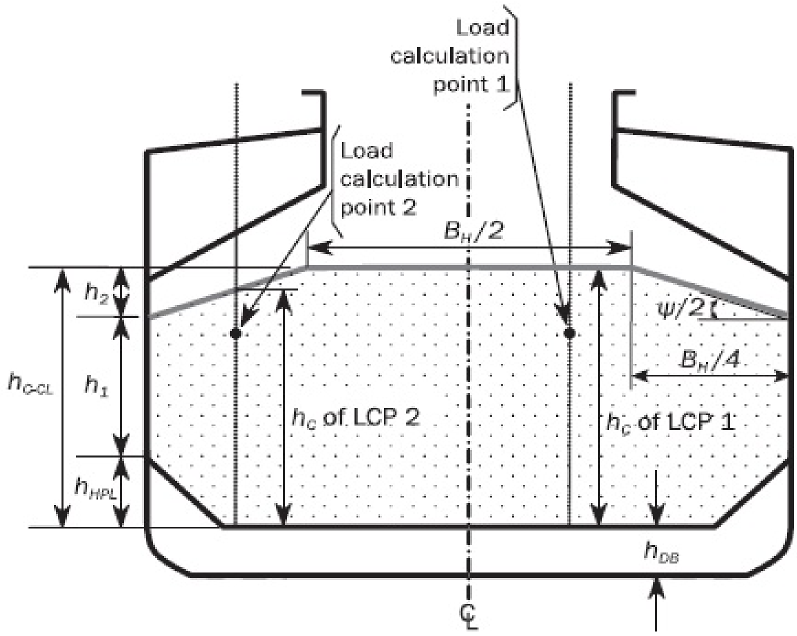

Figure 3.10

Definition of the effective upper surface of cargo for a partially filled cargo hold [20]

Finally, the total pressure due to dry bulk cargo acting on any load point of a cargo hold boundary, in kN/m^2, is to be taken as:

$P_{in} = P_{bs} + P_{bd}$, where P_{bs} the static pressure due to dry bulk cargo and P_{bd} the dynamic inertial pressure.

The *static pressure* is calculated using the following formula:

$$P_{bs} = \rho_C g K_C (z_C - z)$$

, where ρ_C the density of the cargo

K_C a coefficient according to the position of the area assessed and

z_C the height of the upper surface of the cargo above the baseline in way of the load point.

The *dynamic pressure* on the other hand, is calculated using the formula that follows:

$$P_{bd} = f_\beta \rho_C \left[0.25 a_X (x_G - x) + 0.25 a_Y (y_G - y) + f_{dc} h_C a_Z (z_C - z)\right]$$

,where f_β a heading correction factor,

a_x, a_y, a_z the longitudinal, transverse and vertical accelerations as calculated by the rules,

and f_{dc} a factor depending on the mode of assessment.

In the case of a unit cargo[21], the local translational accelerations at the centre of gravity are applied to the mass to obtain a distribution of inertial forces. Such forces are transferred to the structure in different ways, depending on the number and extension of contact areas and on typology and geometry of the lashing or supporting systems. Generally, this kind of load is modeled by one or more concentrated forces or by a uniform load applied on the contact area with the structure. Special provisions are applied for the carriage of steel coils in cargo holds of bulk carriers, where the arrangement and dunnage within the hold is fully prescribed.

Considering the pressures due to liquid, the assessment involves the internal pressure generated by liquids at any loading point of the structure for static and dynamic scenarios. The Common Structural Rules provide formulas for the computation of the static liquid pressure for scenarios on normal operations at sea, for harbor conditions and during ballast exchange procedures. Moreover, the dynamic liquid pressure can be determined for tanks and ballast holds on each loading case prescribed by the rules.

According to Bai[22], the internal pressure in a tank, which carries liquids consists of three parts. The hydrostatic pressure that is equivalent to ρgh, the changes in pressure head that are due to the pitching and rolling motions of the ship, and the inertial force of the liquid column due to the accelerations caused by the motion of the ship. For completely full tanks, fluid inertial velocities relative to the tank walls are small and the acceleration in the fluid is considered as corresponding to the global ship acceleration. On the other hand, where the tank is partially filled, significant fluid internal velocities can arise in the longitudinal and on transversal directions, producing additional pressure loads. Thus, impacts can occur on horizontal or sub-horizontal plates of the upper part of the tank walls for high filling ratios and in vertical or sub-vertical plates of the lowest part of the tank, at low filling levels.

A comparison made by Amlashi et al.[23] for a capesize bulk carrier under alternate hold loading condition (AHL), considering the external and internal pressures calculated according to the CSR-BC rules and DNV rules, obtained the following results, as also depicted in figure 3.11:

- The design cargo pressure calculated according to DNV rules at inboard part of the inner bottom is 67% larger than the CSR-BC rule values for AHL condition with heavy cargo. This is due to the higher cargo level and inclusion of vertical acceleration of the cargo in DNV rules. When a more relevant acceleration coefficient for hogging condition is used, the cargo pressure according to DNV rules is significantly reduced as a result of the negative inertial forces. Still the cargo pressure in DNV rules (345Kpa) is higher than in CSR-BC (308Kpa), due to the higher cargo level used in DNV rules.

- It is the difference between downward (cargo) pressures and upward (sea) pressures of the loaded cargo hold which is of importance for the double bottom bending. For the present vessel, the difference between the downward and the upward pressure for the heavy cargo AHL according to DNV rules is 4.6 times larger than the CSR-BC rules at inboard part of the double bottom.

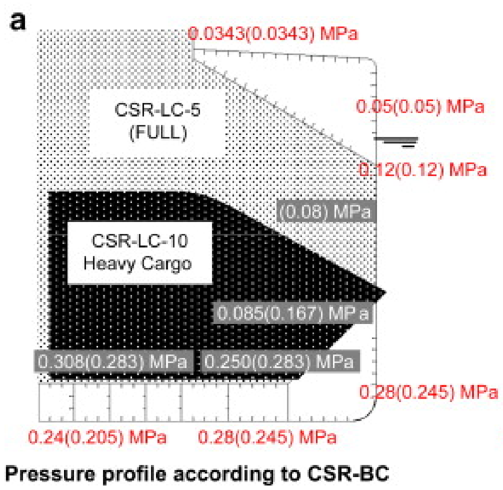

Figure 3.11

Pressure profiles for alternate hold loading conditions according to CSR-BC (a) and DNV (b); the values outside the parenthesis are applicable for heavy cargo AHL, while those in parenthesis are relevant for fully loaded cargo AHL. [23]

3.7 Loading conditions

The determination of local and global loads on the hull structure is to be followed by the assessment of strength in the cargo hold region. This procedure defines a number of load cases which are likely to impose the most onerous local and global load regimes that are to be investigated in the structural analysis. According to the Lloyd's Register[24], the various load cases are summarized as follows:

- Load cases based on homogenous loading applicable to all notations.
- Load cases of ballast loading applicable to all notations.
- Load cases during loading/unloading in harbour.
- Load cases of loading/unloading in multiple ports based on homogenous loading applicable to all notations, except when notation {no MP} is assigned.
- Load cases of alternate ore loading applicable to BC-A notation only.
- Load cases of block ore loading applicable only to BC-A with "any hold may be empty" notation or any other "hold … may be empty" notation which allows two adjacent holds loaded with specified holds empty.
- Load cases of heavy grain loading with slack or empty hold applicable to all notations.

It should be noted that in every case, the presence or absence of liquids in the cargo hold or the ballast/fuel tanks in the double bottom in way of cargo holds is taken under consideration and the addition of those loads arising from the liquids is mandatory. Assessment is made not only for the midship cargo hold area, but additionally for the foremost hold, and the aftmost hold area. For the assessment in each load case, the relative values of the still water bending moment and the still water shear force are to be used. IACS[25] provides tables (an abstract is presented in table 3.2) for each load case to be evaluated, giving instructions on the local and global loads that should be used.

Table 3.2

FE Load combinations applicable to loaded hold in alternate condition of BC-A (FA) - midship cargo hold region (Abstract from [25])

No.	Description	Loading pattern Aft Mid Fore	Draught	$C_{BM\text{-}LC}$: % of perm. SWBM	$C_{SF\text{-}LC}$: % of perm. SWSF	Dynamic load case
Seagoing conditions						
1	Full load	M_H M_H M_H	T_{SC}	50% (sag.)	100%	BSP-1P/S OST-1P/S
2	Full load item a	M_{Full} M_{Full} M_{Full}	T_{SC}	50% (sag.)	100%	BSP-1P/S
3	Slack load item b	M_{Full} 50% M_H M_{Full}	T_{SC}	0%	100%	BSP-1P/S
4	Deepest ballast item c		$T_{BAL\text{-}H}$	100% (hog.)	100%	FSM-2 BSR-1P/S OST-2P/S
				100% (sag.)	100%	BSP-1P/S BSR-1P/S OST-1P/S

3.8 Hold mass curves

The hold mass curves are included in the loading manual. They denote[26] the maximum allowable and the minimum required mass of cargo in each cargo hold as a function of draught in seagoing condition as well as during loading and unloading in harbor. Additionally, they may provide the maximum allowable and minimum required mass of cargo and double bottom contents of any two adjacent holds as a function of mean draught in way of these holds. Chatzitolios et al[27] give a simplified illustration of the hold mass curves on a cargo hold (figure 3.12), whereas a description of what each curve limits is presented.

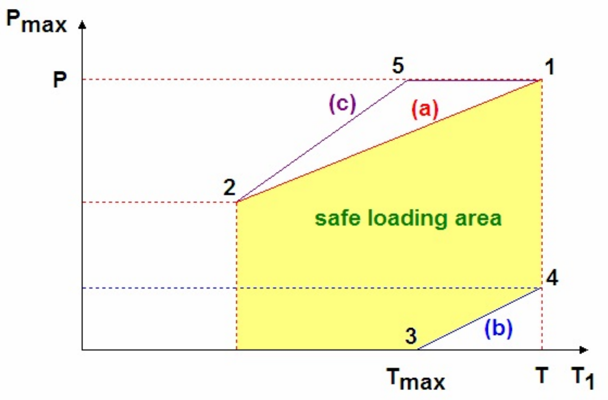

Figure 3.12

Example of hold mass curves [27]

Curve (a) connects the approved loading conditions 1 (maximum cargo mass P at scantling draught T) and 2 (part load condition), denoting the maximum permissible cargo mass. Curve (b) connects the approved loading conditions 3 (loading condition at the maximum permissible draught T_{max} at which the considered hold may be empty) and 4 (minimum required cargo mass at scantling draught). The enclosed (shaded) area is considered to be the safe loading area in which the net resulting load on the double bottom is within acceptable limits.

It is to be noted that the approach is rather conservative, as curve (a) suggests that the maximum permissible cargo mass which can be taken in the hold can only be loaded when sailing at the scantling draught. Most designs, however, have sufficient margin to sail with the maximum cargo mass at a draught less than the scantling draught. In that case curve (a) is replaced by the two segmented curve (c), thus enlarging the loading flexibility of the ship, thus enlarging the loading flexibility of the ship. It is also to be noted that the hold mass curves are not necessarily straight lines.

The abovementioned paper also presented a procedure aiming to calculate the maximum permissible draught in way of the empty ore hold of a capesize bulk carrier (the hold designed to carry heavy cargo but operating empty), as a function of the hogging SWBM. In such conditions, there is a significant risk of buckling at the bottom plating due to the combination of local and global compression stresses. For No 5 cargo hold, the sensitivity of the bottom plating buckling strength to the static bending moment is depicted in figure 3.13.

Figure 3.13

Sensitivity to the hogging SWBM [27]

[1] Strength of ships and ocean structures (2008), Alaa Mansour & Donald Liu, pp 6
[2] Structural reliability analysis of marine structures, (July 1992), DNV classification notes No 30.6
[3] IACS Common Structural rules for Bulk Carriers and Oil Tankers (01 Jan 2014) pp 185
[4] IACS Harmonized CSR TB Report, on Still water Bending moment, SWBM / Report No: Pt 1,Ch 9, Sec 3, July 2012 , pp 8
[5] The maritime engineering reference book (2008), Anthony Molland, pp 126
[6] The strength and reliability of bulk carrier structures subject to age and accidental flooding (1998), Jeom Paik & Anil Thayamballi
[7] Strength of ships and ocean structures (2008), Alaa Mansour & Donald Liu, pp 8
[8] IACS Common Structural rules for Bulk Carriers and Oil Tankers (01 Jan 2014) pp 188
[9] The strength and reliability of bulk carrier structures subject to age and accidental flooding (1998), Jeom Paik & Anil Thayamballi
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[11] Practical estimation method of the design loads for primary structural members of bulk carriers (2003), Tingyao Zhu & Toshiyuki Shigemi
[12] Marine Structural Design (2003), Yong Bai, pp 34
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[14] IACS Common Structural rules for Bulk Carriers and Oil Tankers (01 Jan 2014) pp 194
[15] A study on the effect of heavy weather avoidance on the wave pressure distribution along the midship transverse section of a VLCC and a bulk carrier, Z. Shu & T. Moan
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[17] Bulk carrier safety, Karen Frystock & Jack Spencer. Marine technology, Vol. 33,No 4, Oct. 1996, pp 309-318
[18] Ship design and construction Vol I (2003), Thomas Lamb, pp 18-15
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[23] Ultimate strength analysis of a bulk carrier hull girder under alternate hold loading condition- A case study. Part 1: Nonlinear finite element modelling and ultimate hull girder capacity (2008), Hadi Amlashi & Torgeir Moan
[24] Structural Design Assessment. Primary Structure of Bulk Carriers. Guidance on direct calculations. May 2004, Lloyd's Register, pp 17
[25] IACS Common Structural rules for Bulk Carriers and Oil Tankers (01 Jan 2014) pp 279
[26] IACS Common Structural rules for Bulk Carriers and Oil Tankers (01 Jan 2014) pp 315
[27] A practical assessment of existing bulk carrier local structural strength in relation to the allowable hold mass curves. K.Chatzitolios, G. de Jong, JE Kokarakis, pp3

4

4.1 Introduction

In the fourth chapter of this thesis, the calculations for the structural components in the midship section area of a bulk carrier are presented. This procedure is part of the preliminary design of a bulk carrier as conducted by students in the ship design laboratory of the National Technical University of Athens, as part of the preliminary design lesson of the department of Naval Architecture and Marine Engineering. The following procedure is a translation from Greek of the tenth part of the work done by Antonis Dellis[1], whose kind permission was requested to reproduce the material in the present thesis, and was granted.

4.2 Principal dimensions

In order to calculate the various dimensions of the components in the midship section area, it is essential to define the principal dimensions of the vessel, as dictated by the rules.

Length

ABS defines that the scantling length, L, is to be: $96\%L_{SWL} \leq L \leq 97\%L_{SWL}$

,where L_{SWL} the summer loadline length.

Using the preliminary design drawing lines, we measure that L_{OA}=236.554m, L_{BP}=225.90m and L_{SWL}=230.251m. As a result, as scantling length we use the average:

$$L = \frac{96\% \cdot L_{SWL} + 97\% \cdot L_{SWL}}{2} = 222.192m$$

Breadth

Breadth, B, is considered as the greatest molded breadth in meters, thus B= 36.40m

Depth

D is the molded depth at side in meters measured at the middle of L from the molded base line to the top of the freeboard-deck beams. In our study D= 18.85m

Scantling depth

The depth D_S for use with scantling requirements is the distance in meters from the molded base line to the strength deck, thus excluding the plate thickness. As a result, D_S=18.868m

Draft

d is the molded draft, and is the distance in meters (feet) from the molded base line to the summer load line, thus d=13.182m

Molded displacement

Δ is the molded displacement of the vessel in metric tons, excluding appendages, taken at the summer load line. The calculations in the present study show that

Δ= 95967.89t

Block coefficient

The calculations of the preliminary design result to C_B=0.86

4.3 Loads

4.3.1 Wave Bending Moment

The wave bending moment, expressed in kN-m, may be obtained from the following equations:

$M_{ws} = -K_1 \cdot C_1 \cdot L^2 \cdot B \cdot (C_B + 0.7) \cdot 10^{-3}$, sagging moment

$M_{wh} = +K_2 \cdot C_1 \cdot L^2 \cdot B \cdot C_B \cdot 10^{-3}$, hogging moment, where

K_1=110, K_2=190

$$C_1 = 10.75\left(\frac{300-L}{100}\right)^{1.5} = 10.75\left(\frac{300-222.192}{100}\right)^{1.5} = 10.0637$$

L=222.192m

B=36.4m

C_B=0.86

By replacing the numbers to the abovementioned equations, we may calculate the wave bending moments:

$M_{ws} = -110 \cdot 10.0637 \cdot (222.192)^2 \cdot 36.40m \cdot (0.86 + 0.7) \cdot 10^{-3} = -3097388.442$ kN-m

$M_{wh} = +190 \cdot 10.0637 \cdot (222.192m)^2 \cdot 36.40m \cdot 0.86 \cdot 10^{-3}$=+2944752.496 kN-m

4.3.2 Still water bending moment

The value of still water bending moment derives from the following equation:

$M_{sw} = C_{ST} \cdot L^{2.5} \cdot B \cdot (C_B + 0.5)$, where

C_{ST} =0.00544 ,for 210m<L<250m

$M_{sw} = 0.00544 \cdot (222.192m)^{2.5} \cdot 36.40m \cdot (0.86 + 0.5)$=204581.1278tm or

M_{sw}=2006246.099kN-m

Concluding, the total bending moment is the maximum algebraic sum of still-water bending moment and wave-induced bending moment, for each situation, thus:

Sagging: $M_{TS} = M_{sw} + M_{ws}$ =2006246.099kN-m+3097388.442kN-m$\Rightarrow$

M_{TS} =+5103634.541kN-m

Hogging: $M_{TH} = M_{sw} + M_{wh}$ =2006246.099kN-m+2944752.496kN-m$\Rightarrow$

M_{TH} =+4950998.595kN-m

The results of the bending moments it the three loading conditions are presented in table 4.1

Table 4.1

Bending moments calculation results [1]

Bending moments	
condition	**Value (kN-m)**
Still water	2006246.099
sagging	+5103634.541
hogging	+4950998.595

4.3.3 Bending strength standard

Hull Girder Section Modulus. The required hull girder section modulus for 0.4L amidships is to be the greater of the values obtained from the following equations:

- Section modulus

$SM = \frac{M_T}{f_p}$,where

M_T=+5103634.541kN-m,the total bending moment as calculated above

f_p =17.84kN/cm^2 ,the nominal permissible bending stress

$$SM = \frac{M_T}{f_p} = \frac{5103634.541kN - m}{17.84kN/cm^2} = 270859.1693 \text{ cm}^2\text{m}$$

- Minimum Section Modulus. The minimum hull girder section modulus amidships is not to be less than obtained from the following equation:

$SM = C_1 \cdot C_2 \cdot L^2 \cdot B \cdot (C_B + 0.7)$,where

C_1=10.0637 as calculated above

C_2=0.01

L=222.192m

B=36.40m

C_B=0.86

As a result, $SM = C_1 \cdot C_2 \cdot L^2 \cdot B \cdot (C_B + 0.7) \Rightarrow$

SM=$10.0637 \cdot 0.01 \cdot (222.192m)^2 \cdot 36.40 \cdot (0.86 + 0.7)$=245411.112 cm^{2}m

The comparison of the two Section Modulus values denotes that the section modulus of the structure should be greater than **SM_{req}=270859.1693 cm^{2}m**

Hull Girder Moment of Inertia

The hull girder moment of inertia, I, amidships, is to be not less than:

$I = L \cdot \frac{SM}{33.3} = 222.192m \cdot \frac{270859.1693cm^2m}{33.3}$ =1807289.506cm^{2}m

I_{req}=1807289.506cm^{2}m

4.4 Midship section dimensions

Subsequently, according to the ABS rules, the dimensioning of the major midship section elements takes place. This procedure enables calculations for the proper sizing of elements such as:

- Side shell plating
- Sheer strake
- Bottom shell plating amidships
- Main deck plating
- Plate flat keel
- Bilge keels
- Stringer plate
- Center girder
- Side girders
- Solid floors
- Docking brackets
- Double bottom shell
- Hopper tank bulkhead
- Topside tank bulkhead
- Transverses
- Hatch coaming

In the present study, a corrosion margin of 4 mm is used for each plate.

4.4.1 Side shell plating

The minimum thickness, *t*, of the side shell plating throughout the amidship 0.4*L*, for the vessel, is to be obtained from the following equation:

$$t = \frac{s_b}{645} \cdot \sqrt{(L - 15.2) \cdot \left(\frac{d}{D_S}\right)} + 2.5mm$$, where

s_b =860mm, the spacing of transverse frames

L= 222.192m, length of vessel as previously defined

d=13.182m, the molded draft

D_S=18.868m, the molded depth, as previously defined

$$t = \frac{860}{645} \cdot \sqrt{(222.192 - 15.2) \cdot \left(\frac{13.182}{18.868}\right)} + 2.5mm = 18.534\text{mm}$$

Finally, we select **SIDE SHELL PLATING t=23mm**

4.4.2 Sheer strake

The minimum width, b, of the sheer strake throughout the amidship 0.4L for ships with length 200m<L<427m, is considered b=1800mm, according to the rules.

In general, the thickness of the sheer strake is not to be less than the thickness of the adjacent side shell plating, thus we define the thickness of the **sheer strake t=23mm**

4.4.3 Bottom shell plating amidships

The term "bottom plating amidships" refers to the bottom shell plating from the keel to the upper turn of the bilge, extending over the amidships 0.4L.The thickness, t, of the bottom plating amidships is not to be less than obtained from the following equation. In our case, for vessels with longitudinally-framed bottoms,

$$t = \frac{s_b}{508} \cdot \sqrt{(L - 62.5) \cdot \left(\frac{d}{D_S}\right)} + 2.5mm$$, where

s_b =860mm, the spacing of transverse frames

L= 222.192m, length of vessel as previously defined

d=13.182m, the molded draft

D_S=18.868m, the molded depth, as previously defined

$$t = \frac{860}{508} \cdot \sqrt{(222.192 - 62.5) \cdot \left(\frac{13.182}{18.868}\right)} + 2.5mm = 20.3815mm$$

Finally, we select **BOTTOM SHELL PLATING AMIDSHIP t=21mm**

4.4.4 Main deck plating

The thickness, t, of the main deck plating is not to be less than obtained from the following equation:

$$t = \frac{24.38 \cdot s_b}{1615.4 - 1.1L}$$, where s_b=780mm, spacing of deck beams

$$t = \frac{24.38 \cdot 780}{1615.4 - 1.1 \cdot 222.192} = 13.87mm$$

Thus, we select **MAIN DECK PLATING t=18mm**

4.4.5 Plate flat keel

The thickness of the flat plate keel is to be 1.5 mm greater than that required for the bottom shell plating at the location under consideration, thus we select **PLATE FLAT KEEL t=22.5mm**

To calculate the width, b, of the plate, we use the product $70 \cdot B$ in mm, where B=36.40m, as defined previously. Thus, b=2548mm, and finally we define **b=2600mm**

4.4.6 Bilge keels

The thickness of the bilge keels should be equal to the thickness of the bottom shell plating, thus we define **BILGE KEEL thickness t=21mm**

4.4.7 Stringer plate

The thickness of the stringer plate should be equal to the thickness of the main deck plating, thus we define **STRINGER PLATE thickness t=18mm**

4.4.8 Center girder

The thickness of the center girder should not be less than the one defined by the following equation:

$t = 0.056 \cdot L + 5.5mm = 17.94mm$

Thus, we define **CENTER GIRDER thickness t=20mm**

4.4.9 Side girders

The rules define the minimum thickness of the side girders by using the following equation:

$t = 0.036 \cdot L + 4.7 + c$ mm, where c=0 for side girders

Thus, $t = 0.036 \cdot 222.192 + 4.7 + 0$ =12.7mm

Finally, we select the thickness of the **SIDE GIRDERS t=15mm**

Amidships, side girders of the thickness obtained from the previous equation are to be so arranged that the distance from the center girder to the first side girder, the distance between the girders, and the distance from the outboard girder to the center of the margin plate does not exceed 4.57 m. In the present study, a side girder is placed every 4 longitudinals.

4.4.10 Solid floors

Solid floors are to be fitted on every frame under machinery and transverse boiler bearers, under the outer ends of bulkhead stiffener brackets and at the forward end. Elsewhere, they may have a maximum spacing of 3.66 m in association with intermediate open floors, or longitudinal framing of the bottom or inner bottom plating. Thus, in our study we choose to use a solid floor every four frames,

$4 \cdot 0.860$ =3.440m< 3.66m

The minimum thickness ot the solid floors is taken using the following equation, $t = 0.036 \cdot L + 4.7 + c$ mm, where c=1.5mm for floors where the bottom shell and inner bottom are longitudinally framed

Thus, $t = 0.036 \cdot 222.192 + 4.7 + 1.5$ =14.2mm

Finally, we select the thickness of the **SOLID FLOORS t=18mm**

4.4.11 Docking brackets

Docking brackets are to be provided on the center girder where the spacing of the floors exceeds 2.28 m, unless calculations are submitted to verify that the girder provides sufficient stiffness and strength for docking loads. In the vessel under study, the docking brackets are positioned beneath the stringer plate in every frame between the sold floors. The thickness of the brackets is to be taken equal to the thickness of the floors, thus **DOCKING BRACKETS t=18mm**

4.4.12 Double bottom shell

The thickness amidships for the double bottom shell is given by the following equation:

$t = 56 \cdot L \cdot 10^{-3} + 5.5mm = 56 \cdot 222.192 \cdot 10^{-3} + 5.5mm$ =17.94mm

We select **DOUBLE BOTTOM SHELL t=20mm**

4.4.13 Hopper tank bulkhead

The hopper tank bulkhead thickness should be adequate to cover the regulations for the deep tank bulkhead plating and furthermore, the thickness suggested for the platform deck in enclosed cargo space.

As a tank-end floor, the minimum thickness results from the following equation:

$t = \left(sk\sqrt{qh}/254\right) + 2.5mm$, where

s=780mm the spacing of stiffeners

k= 1 where the aspect ratio of the panel α>2

Y=24 $kg/mm^2 = 235.36N/mm^2$

q=235/Y=1 (for Y= $235.36N/mm^2$)

h=D-h_{DB}=17.018m

Thus, t_{min}=15.63mm

As a platform deck in enclosed cargo space, the thickness is given by the following equation:

$t = K \cdot s_b\sqrt{h} + 1.5mm$, where

s=780mm the spacing of stiffeners

K=0.00394

h=17.018m

The calculation gives a minimum thickness t_{min} =14.18mm

As a result, we choose the thickness of the **HOPPER TANK BULKHEAD t=18mm**

4.4.14 Topside tank bulkhead

The minimum thickness results from the following equation:

$t = \left(sk\sqrt{qh}/c\right) + 1.5mm$, where

s=780mm the spacing of stiffeners

k= 1 where the aspect ratio of the panel α>2

Y=24 $kg/mm^2 = 235.36N/mm^2$

q=235/Y=1 (for Y= $235.36 N/mm^2$)

h=D-h_{DB}=17.018m

c=290

Thus, t_{min}=12.59mm

Finally, we select the thickness of the **TOPSIDE TANK BULKHEAD t=15mm**

4.4.15 Transverses

The thickness of the plate for the transverses amidships is given by the following equation:

$t = 0.036 \cdot L + 6.2 = 0.036 \cdot 222.192 + 6.2$ =14.19mm

Thus, we choose the thickness of **TRANSVERSES t=15mm**

4.4.16 Hatch coaming

The height of coamings of hatchways secured weathertight by tarpaulins and battening devices is to be at least 600 mm for the bulk carrier under consideration, whereas coaming plates are not to be less than 11 mm thick. Finally, according to the drawings of the parental vessel, the height of the hatch coaming is considered h=1.2m and the thickness of the plating is taken for **HATCH COAMING t=15mm**

4.5 Stiffeners

4.5.1 Bottom and inner bottom longitudinals

Each bottom longitudinal frame, in association with the plating to which it is attached, is to have a section modulus *SM* not less than that obtained from the following equation:

$SM = 7.8 \cdot C \cdot h \cdot l^2 \cdot s_b$,where

C= 1.3 (without struts)

h=13.182m ,the distance from the keel to the load line

l=3.12m, distance between the supports

s_b=780mm spacing of longitudinals

As a result, $SM = 7.8 \cdot 1.3 \cdot 13.182 \cdot (3.12)^2 \cdot 780$ =1014.899cm^3

The spacing of the adjoining plate is 780mm (frame spacing), whereas its thickness is 21mm. In order to withstand those demands, we choose to use a Bulb profile 340x14HP longitudinal, that presents a section modulus of SM=1065.23cm^3.

The inner-bottom longitudinals are to have values of *SM* at least 85% of that required for the bottom longitudinal, thus SM_{REQ}=85%$\cdot$1014.899 =862.665cm^3.

For the adjoining plate spacing of 780mm (frame spacing), with a thickness of 20mm, we choose to use a Bulb profile 340x14HP longitudinal, that presents a section modulus of SM=1059.62cm^3.

4.5.2 Longitudinal frames (side)

The section modulus *SM* of each longitudinal side frame is to be not less than obtained from the following equation:

$SM = 7.8 \cdot c \cdot h \cdot s \cdot l^2$,where

c= 0.95

h is *(a)* above 0.5*D* from the keel, the vertical distance, in m, from the longitudinal frame to the bulkhead or freeboard deck, but is not to be taken as less than 2.13 m, and *(b)* at and below 0.5*D* from the keel, 0.75 times the vertical distance, in m, from the longitudinal frame to the bulkhead or freeboard deck, but not less than 0.5*D*.

s the spacing of longitudinal frames, in m

l the unsupported span, in m

The vessel under design has longitudinal frames inside the topside tanks and the bottom wing tank sides, thus a calculation has to be done for both positions.

- Top side tanks

h=12.75m>2.13m$\Rightarrow$h=12.75m

s=0.800m

l=2.65m

As a result, SM=530.77cm^3. For the adjoining plate spacing of 800mm, with a thickness of 23mm, we choose to use a Bulb profile 370x15HP longitudinal, that presents a section modulus of SM=1366.38cm^3.

- Bottom wing tanks

In a similar way,

h= $0.750 \cdot 13.15 = 9.863m$ >9.425m $= 0.5 \cdot D \Rightarrow$ h=9.863m
s=0.800m
l=2.65m
As a result, SM=410.59cm^3. For the adjoining plate spacing of 800mm, with a thickness of 23mm, we choose to use a Bulb profile 300x12HP longitudinal, that presents a section modulus of SM=739.20cm^3.

4.5.3 Longitudinal frames (deck)

The section modulus *SM* of each longitudinal frame is to be not less than obtained from the following equation:

$SM = 7.8 \cdot c \cdot h \cdot s \cdot l^2$,where

c= 0.945

h=2.55m spacing of longitudinal frames,

s=780mm the spacing of longitudinal frames

l=7.28m, the unsupported span, (greater than $0.2 \cdot B = 6.44m$)

Finally, $SM = 7.8 \cdot 0.945 \cdot 2.55 \cdot 780 \cdot (7.28)^2$ =777.005cm^3. For the adjoining plate spacing of 780mm, with a thickness of 18mm, we choose to use a Bulb profile 340x14HP longitudinal, that presents a section modulus of SM=1047.69cm^3. For simplification reasons, we are going to use the same longitudinals for the inclined surfaces of those tanks. As a result, we use the 370x14HP longitudinal in the topside tank, and the 300x12HP longitudinal for the bottom wing tank. The bottom girders are selected the same as the bottom longitudinals, 340x14HP.

Concluding, the stiffeners selected are presented in table 4.2

Table 4.2

Stiffeners selection [1]

Position	**Stiffener**	**Minimum SM (cm^3)**	**Calculated SM (cm^3)**
Bottom	HP 340x14	1014.899	1065.23
Inner bottom	HP 340x14	862.665	1059.62
Topside tank	HP 370x15	530.77	1366.30
Bottom wing tank	HP 300x12	410.59	739.20
Main deck	HP 340x14	777.005	1047.69
Girders	HP 340x14	-	1027.34

Since the values of the resulting section modulus of the midship section and the moment of inertia is not enough when compared to the ones extracted on previous charter of the preliminary design, it is essential to increase the size of stiffeners. The final selection of stiffeners is presented in table 4.3

Table 4.3

Stiffeners final selection [1]

Position	Stiffener
Bottom	HP 340x14
Inner bottom	HP 340x14
Topside tank	HP 370x15
Bottom wing tank	HP 300x12
Topside side shell	HP 370x15
Wing tank side shell	HP 300x12
Main deck	HP 340x14
Girders	HP 340x14

The midship section drawing as determined by the present work can be seen in figure 4.1, at the end of this chapter.

4.6 Midship Section Modulus

In order to calculate the midship section modulus, we create table 4.4 that includes all the structural components that contribute to this calculation. The following expressions are used:

l: length of element

b: width of element

t: thickness

Y: vertical distance of the neutral axis of the element from the midship section neutral axis

h: vertical projection of each element

A:surface area of element

z_i: neutral axis distance of each element

The position of the neutral axis of midship section is determined by applying the moments of inertia method:

$$NA = \frac{\sum A \cdot Z_i'}{\sum A}\text{, where}$$

Z_i' : distance of the neutral axis of each element from Baseline

A: surface area of each element

The moment of inertia of the midship section is calculated by using the following sum.

$$I = 2 \cdot \left[\sum \left(\frac{A \cdot h^2}{12} \right) + \sum \left(A \cdot Y^2 \right) \right]$$

Finally, the section modulus of the midship section can be calculated as follows:

$$SM = \frac{I}{D - N.A.}$$

Table 4.4

Section modulus calculation [1]

	Section modulus calculation								
	Component (number)	**b (cm)**	**t (cm)**	**Area (cm^2)**	**Y (m)**	**AY ($m*cm^2$)**	**AY^2 (m^2*cm^2)**	**h (cm)**	**$Ah^2/12$ (m^2*cm^2)**
Plate	**Main deck**	1746.1	1.8	3142.98	19.22	60408.0756	422990	-	148.17
	Side shell (2)	3410	2.3	7843	7.78	61018.54	203	3410	759993.24
	Sheer strake (2)	360	2.3	828	17.95	14862.6	88372	360	894.24
	Inner bottom	2640	2	5280	1.86	9820.8	175119	2	0.18
	bottom	2696	2.1	5661.6	-0.01	-56.616	329517	2.1	0.21
	Flat keel	260	2.25	585	-0.01	-5.85	34048	2.25	0.02
	Center girder	185	2	370	0.935	345.95	16530	185	105.53
	Side girders (6)	1110	1.5	1665	0.935	1556.775	74386	1110	17095.39
	Topside tank sloping (2)	2067.14	1.5	3100.71	16.29	50510.5659	233129	-	4298.00
	Hopper tank sloping (2)	1232.2	1.8	2217.96	3.65	8095.554	34940	-	2909.00
	Hatch coaming (2)	240	1.5	360	20.58	7408.8	60475	240	172.80
	Bilge keel (2)	-	2.4	104	0.56	58.24	5182	-	0.00
	Stringer (2)	500	1.8	900	18.87	16983	113926	-	4.00
Stiffener	**Main deck (20)**	34	1.4	1312.2	19.279	25297.9038	178400	-	15.08
	Bottom (36)	34	1.4	2361.96	0.21	496.0116	129657	-	27.14
	Inner bottom (28)	34	1.4	1837.08	1.84	3380.2272	61353	-	21.11
	Topside sloping (24)	40	1.4	1954.8	18.07	35323.236	213509	-	31.20
	Hopper tank sloping (12)	30	1.2	498.5	3.1291787	1559.895599	10049	-	6.76
	Topside side (12)	40	1.4	977.4	16.951576	16568.47086	85128	-	0.29
	Hopper tank side (8)	30	1.2	398.8	4.4830218	1787.829075	3922	-	0
	Girders (12)	34	1.4	1180.98	0.8460077	999.1181664	54176	-	8
	TOTAL			**41399.0**		**315420.0**	**2270836**		**785722**
	Midship section area						**41399.0**		**cm^2**
	Neutral axis distance from BL						**7.619**		**m**
	Moment of inertia I_{yy}						**305655.3**		**m^2cm^2**
	Z_{max}(=D-N.A.)						**11.231**		**m**
	Section Modulus						**272154**		**m*cm2**

The moment of inertia amidships, is calculated $\mathbf{I_{yy}=305.66m^4> I_{req}=180.73\ m^4}$
Additionally, the Section Modulus calculated is $\mathbf{SM=272.154\ m^4 >SMreq=270.6\ m^4}$
Since the calculated values of the moment of inertia and section modulus are greater than the ones required from the regulations, the dimensions set are considered adequate and the process is considered ended.

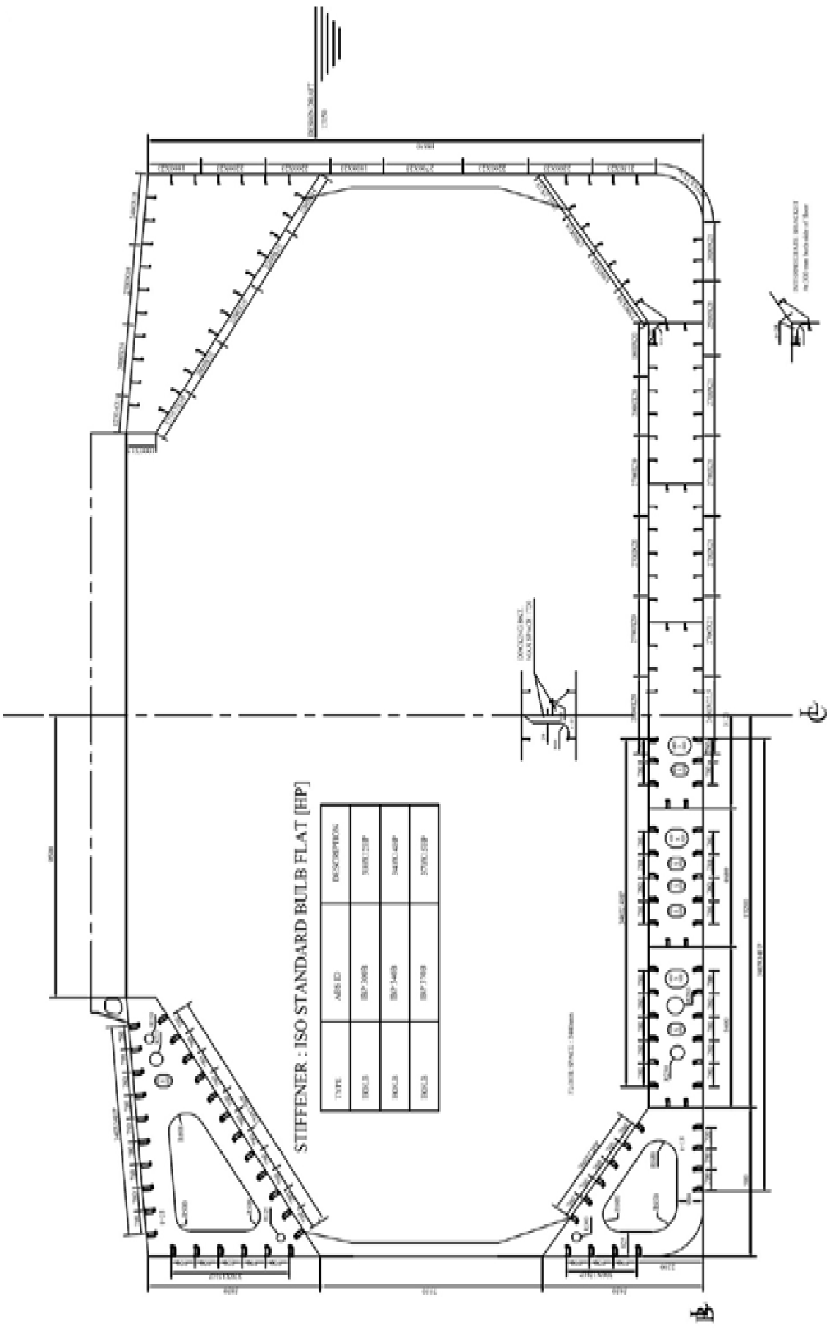


Table 4.1

Midship section drawing [1]

[1] Preliminary design of “M/V The Flying Duchman”, part 10. National Technical University of Athens, department of Naval Architecture and Marine Engineering.(Athens,2012) Antonis Dellis.

5

5.1 Introduction

The fifth chapter of this study copes with the strength analysis of the hull structure. At first, the Finite Element Method analysis is presented, in order to assess the strength of longitudinal hull girder structural members, primary supporting structural members and bulkheads. Additionally, using this method, detailed stress levels in local structural details can be obtained, whereas the fatigue capacity of the structural details can be determined. The typical process of structural analysis using the finite element method is discussed, while the areas of concern in bulk carrier structures are displayed.

Subsequently, the procedure for direct strength analysis is explained. The yielding strength check is discussed, while buckling and ultimate hull girder strength assessment is further analyzed. Prone to buckling areas of bulk carriers are presented, while the findings of an ultimate hull girder strength assessment are represented.

Finally, the assessment of the fatigue life of the various structural members subject to fatigue failure is presented. The types of stresses that are considered for this type of assessment are discussed, while the selection of the correct S-N curve is mentioned. Last but not least, an example of a fatigue performance analysis of bulk carriers side frame structure is noted, whereas the findings of this study are featured.

5.2 Finite Element analysis

According to IACS[1], the finite element analysis consists of three parts: (a) Cargo hold analysis to assess the strength of longitudinal hull girder structural members, primary supporting structural members and bulkheads. (b) Fine mesh analysis to assess detailed stress levels in local structural details, and (c) Very fine mesh analysis to assess the fatigue capacity of the structural details.

Melchers et al.[2], suggest that the response of ship structures under applied ballast/cargo loading and sea conditions may be classified into the following five levels:

- global structure (or hull girder)
- cargo hold
- grillage
- frame and girder, and
- local structure

For each case, the resulting action effects are calculated by Finite Element Modeling (FEM), and the response at each level provides the boundary conditions for the next lower level analysis. Figure 5.1 shows the flow diagram of finite element analysis, as proposed by the IACS.

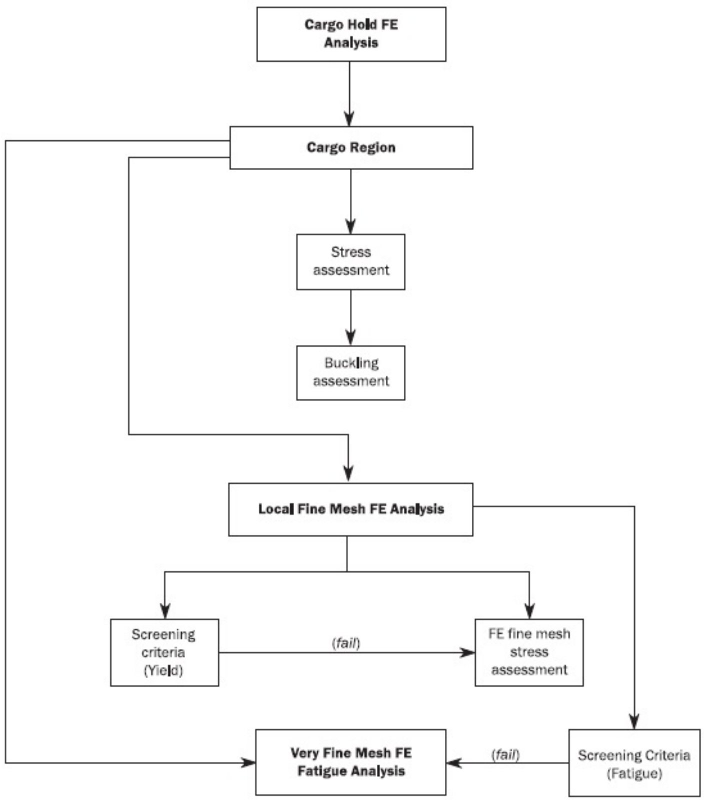


Figure 5.1

Flow diagram of finite element analysis [1]

The types and numbers of finite elements must be selected so that they will be able to accurately portray the stiffness and stresses of the structure to be analyzed. The general types of finite elements to be used in the finite element analysis, as suggested by the IACS[3], are:

- *Rod (or truss) element.* Line element with axial stiffness only and constant cross sectional area along the length of the element.
- *Beam element.* Line element with axial, torsional and bi-directional shear and bending stiffness and with constant properties along the length of the element.
- *Shell (or plate) element.*Shell element with in-plane stiffness and out-of-plane bending stiffness with constant thickness.

Melchers et al[4] mention some additional types:

- *Membrane (or plane) stress element.* A 2D element with membrane stiffness in the plane, but without out-of-plane bending stiffness.
- *Solid element,* which is a 3D element.
- *Boundary and spring element.*
- *Point (or mass) element.*

A finite element model usually involves several types of elements. All primary longitudinal and transverse members are best modeled by quadrilateral plate-shell elements. Support members that do not involve a deep web may be modeled by beam or truss elements. Stiffened panels and grillages may be modeled as an assembly of plate-shell elements and beam elements.

The typical process of structural analysis using the finite element method, as proposed by Okumoto et al[5]. is illustrated in figure 5.2. After choosing an appropriate analysis program for the specified problem, the modeling is done by determining the appropriate size of the structure. It may be possible to reduce the size of the model by defining suitable boundary and loading conditions. The next step is to prepare the geometrical data of the finite elements with visual checking of the validity of the input data through the computer display. The loading data as well as the boundary conditions of the structure are then added. After executing the program, the calculated result must be assessed to check whether there could be some error in the input data in view of the calculated deformation, stress etc. If there was a mistake, the procedure is to be repeated from the beginning. The resulting deformations and stresses can be assessed using various color graphics techniques. In this way, areas where the deformations or stresses exceed permissible limits are identified and can be further analyzed.

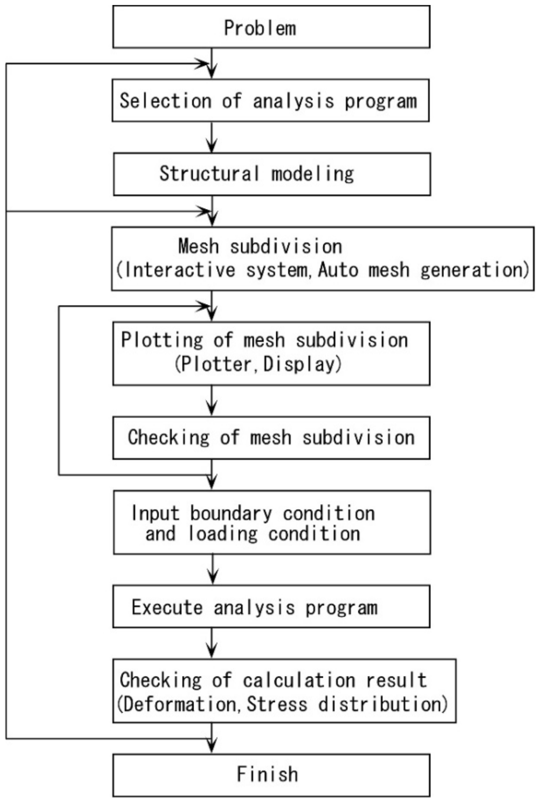


Figure 5.2

Procedure of FEM structure analysis. [5]

Generally, the finite element model shall provide results suitable for evaluating the strength of the girder system and for performing buckling analysis of plate flanges and girder webs. This may be done by using a 3D finite element model of the midship area. Several approaches may be applied; ranging from a detailed 3D model of the cargo holds to a coarse mesh 3D-model, supported by finer mesh sub models. Coarse mesh models can be used for calculating deformations and stresses typically suited for buckling control. The deformations may be applied as boundary conditions on sub models for finding the stress level in more detail. The same principles may normally be used on structures outside the midship area but within the cargo area, provided special precautions are taken regarding model extent and boundary conditions.

On choosing the appropriate element type and mesh size, Amlashi et al[6]. note that even today, rapid increases in computer processing power and memory have not eliminated computational cost and time constraints. This is due to the constant increase in the required mesh density to converge to the most reliable solution. An increase in mesh density (fine mesh) through the model is theoretically possible but not in practice due to significant efforts and computational costs. Therefore, a balance between required accuracy and efforts is needed.

The extend of analysis is decided such that the actual stress conditions of the ship can be reproduced by considering the arrangement of cargo oil and ballast tanks, the loading pattern and the arrangement of members near the bulkhead. The model longitudinally extends at three cargo hold lengths, and at full depth and breadth. The size of the mesh is selected considering the stress condition in the model and the meshing of elements is performed rationally so as to avoid meshes with large aspect ratios. The standard size on an element in the stress evaluation area is decided by taking one side of the element as approximately equal to the spacing of the nearby stiffeners. A typical finite element model[7] representing the midship cargo hold region of a bulk carrier can be seen in figure 5.3, whereas a transverse section model is presented in figure 5.4

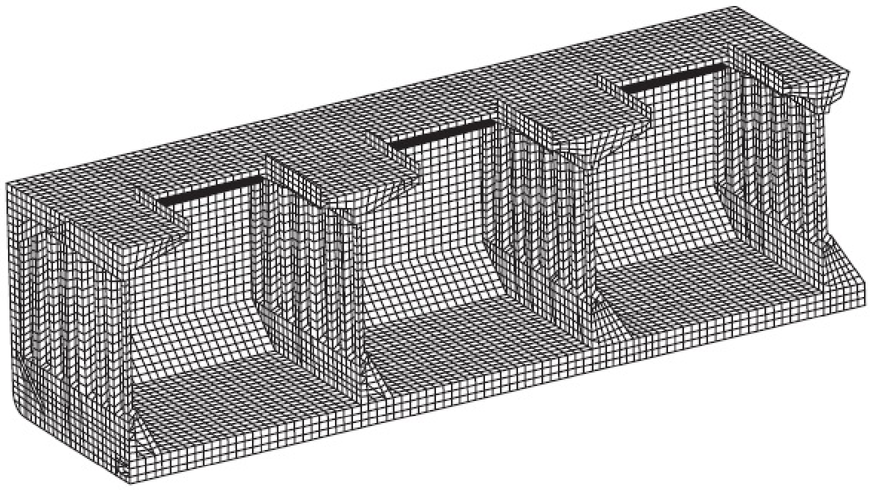


Figure 5.3

Example of structural model of bulk carrier [3]

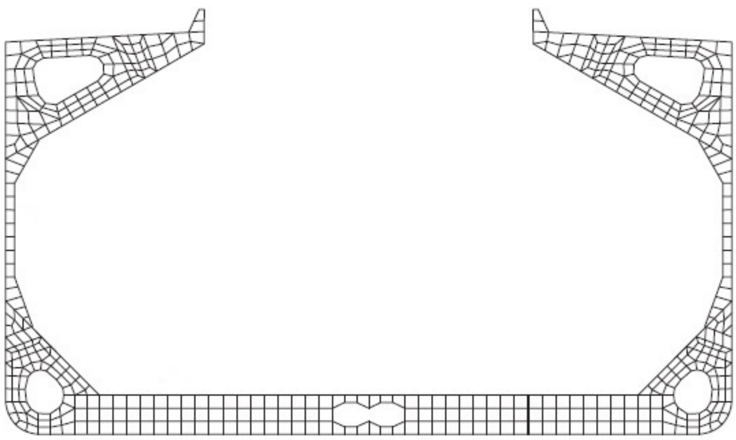


Figure 5.4

Typical bulk carrier transverse section FE model [3]

Table 5.1 provides the areas of a bulk carrier to be assessed with fine meshes, as proposed in the Common Structural Rules[8].

Table 5.1
Typical details to be refined [8]

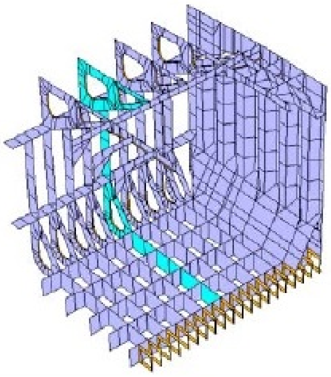
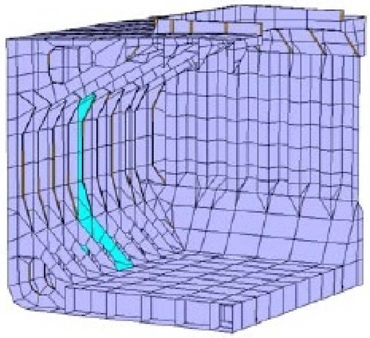
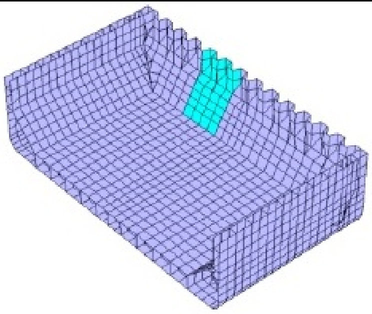
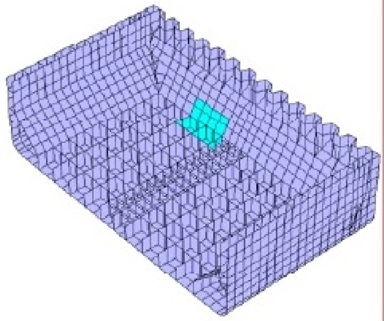
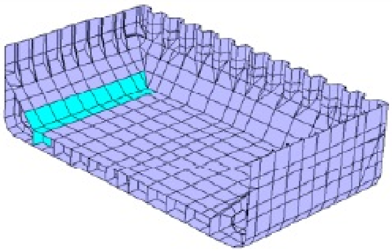
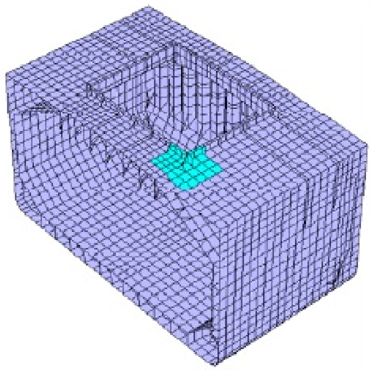
Structural member	Area of interest	Additional specifications	Description
Primary supporting member	Most stressed transverse primary supporting member for double side skin constructions	Refining of the most stressed transverse primary supporting members located in: • double bottom • hopper tank • double skin side • topside tank	
	Most stressed transverse primary supporting member for single side skin constructions	Refining of the most stressed transverse primary supporting members located in: • double bottom • hopper tank • topside tank side shell frame with end brackets and connections to hopper tank and topside tank	
Transverse bulkhead and its associated lower stool	Most stressed connection of the corrugations with the lower stool	High stressed elements, including the diaphragm(s) of the lower stool, are to be modeled	
	Most stressed connection of the lower stool with the inner bottom	High stressed elements are to be modeled	

Table 5.1 (cont.)

Typical details to be refined [8]

Structural member	Area of interest	Additional specifications	Description
Inner bottom and hopper sloping plates with their associated supporting members	Most stressed connection of the inner bottom with the hopper sloping plate	Refining of the most stressed following members: • inner bottom • hopper sloping plate • floor • girder	
Deck plating	Deck plating in way of the most stressed hatch corners	High stressed elements are to be modeled	

For the hot spot stress analysis, it is suggested[9] that the mesh size is to be gradually changed from very fine mesh to fine mesh through the transition areas as shown in Fig 5.5. All structural members, including brackets, stiffeners, longitudinals and faces of transverse rings, etc., within transition areas are to be modeled by shell elements with bending and membrane properties.

Figure 5.5

Mesh size transition for hotspot stress analysis [9]

5.3 Strength analysis

The determination of loads and loading conditions has to be followed by the evaluation of the suitability of the initial design established, based on the strength assessment against specified acceptance criteria. The probable failure modes of the hull structure to be assessed are yielding, buckling, fatigue, and ultimate hull girder strength. According to Class NK[10], direct strength analysis involves the evaluation of yielding strength and buckling strength of primary structural members, considering the corrosion deduction amount of bulk carriers to be used for direct stress analysis.

The overview of the procedure for evaluation, as proposed by Class NK[11], is given in figure 5.6

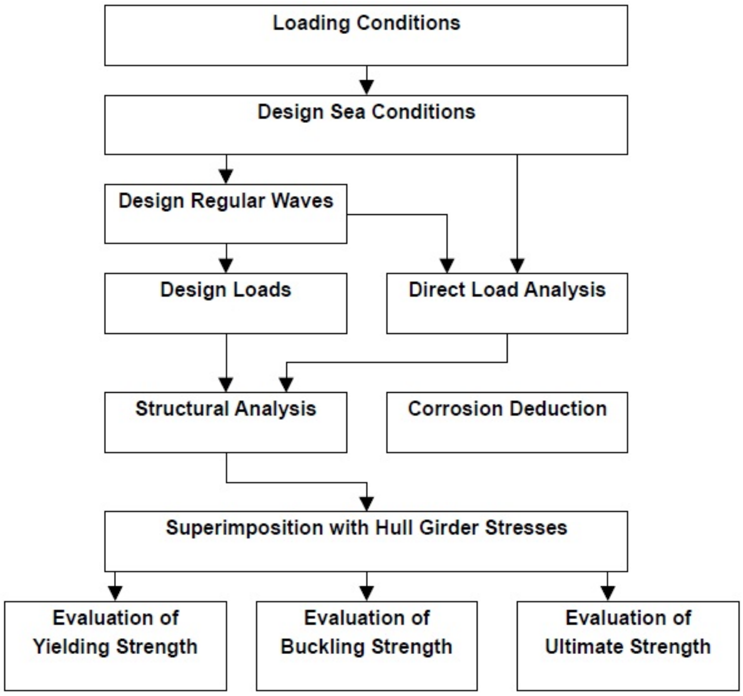


Figure 5.6

Evaluation procedure for direct strength analysis [11]

Lehmann et al.[12] mention that the scope of strength analysis in bulk carrier design should cover the following aspects:

- Global hull girder strength with particular view to bending and shear stresses in the hull girder.
- Strength of the double bottom grillage, particularly in case of heavy cargo and/or empty holds, considering supporting effects by the lower wing tanks and/or bulkhead stools.
- Strength of the bulkheads, taking into account interaction effects especially with the bulkhead stools and double bottom.

- Local strength of structural details considering stress concentrations and fatigue. Particular attention has to be paid to knuckles in the upper and lower wing tanks, connections between the stools and the bulkhead plating and/or inner bottom, end connections of side frames, hatch corners, terminations of coamings and transitions at the ends of the hold area.

5.3.1 Yielding

Allowable stress intensities based on the material yield strength of various steel grades, appropriately adjusted based on service experience, are used[13] to assess structural members against material yield failure modes. In general, total direct stresses and Von Mises stresses are considered for beam elements and plate elements respectively, including primary, secondary and local bending and shear stresses. For plating elements, all influential stress components, including tertiary stresses, are included in stress intensity calculations.

Von Mises equivalent stress is given by the following formula:

$$\sigma_{eq} = \sqrt{\sigma_x^{\,2} - \sigma_x\sigma_y + \sigma_y^{\,2} + 3\tau_{xy}^{\,2}} \quad \text{,where}$$

σ_x, σ_y : element normal stresses, in N/mm^2

τ_{xy} : element shear stresses, in N/mm^2

The reference stresses in FE model that do not include orthotropic elements are not to exceed 235/k N/mm^2, where k is the material factor. For a FE model that includes orthotropic elements the reference stresses are not to exceed 205/k N/mm^2. For the case of bi-axial stress in plate elements, as mentioned by Mansour et al.[14], a specific combination of stresses, rather than the maximum normal stress, constitutes the limiting condition. In this regard, the yielding criteria is that the Hencky-von Mises stress, is not to exceed 95 percent of the yield stress of the material, f_y, thus:

$$\sigma_{eq} = \sqrt{\sigma_x^{\,2} - \sigma_x\sigma_y + \sigma_y^{\,2} + 3\tau_{xy}^{\,2}} \le 0.95 f_y$$

5.3.2 Buckling and ultimate hull girder strength

In assessing the buckling and ultimate strength of plate and stiffened panels, an interaction equation is used to consider the combined effect of simultaneously acting bi-axial compression, lateral pressure and shear. The equation is given in terms of a stress ratio of computed stress/critical stress for each independent stress component. The interaction value is not to exceed unity. Mansour et al[15]. refer to this process as “unity check”. The criteria are given in terms of ratios between the calculated nominal stress and the critical buckling stress for each independent stress component, with the sum of these stress ratios squared not to exceed unity. This together with buckling strength assessment for stiffeners and stiffened panels based on established analytical or empirical formulas suitable to the hull structure are used. For longitudinals and other stiffeners, both column and torsional/flexural buckling are considered. The critical buckling strength of longitudinals and stiffeners is considered as the ultimate strength in order to account for the beam column behavior. It is worth noting that this mode of instability often turns out to be the weakest for some asymmetric longitudinal stiffener designs.

Considering the buckling failure mode on bulk carriers, particular attention should be paid to the following areas, according to Lloyd's Register[16] :

- Double bottom holds, especially at mid-hold length.
- Double bottom girders, especially at ends of holds adjacent to bulkheads or stools, at first plate opening from bulkheads or stool, at mid-hold.
- Bottom shell and inner bottom plating, especially at ends of holds adjacent to bulkheads or stools, at mid-hold.
- Hopper side tank transverse ring web.
- Hopper sloped plate.
- Topside tank transverse ring web in way of ballast hold.
- Transverse bulkhead and ring structures in topside and hopper side tank in way of the hold transverse bulkhead.
- Topside tank sloped plate in way of the transverse bulkhead.
- Bulkhead and stool plating, especially at mid-span of bulkhead and adjacent to stool, in stool shelf plate outboard.
- Cross deck plate and upper stool.

The calculated equivalent buckling stress is to be based on a "corroded thickness" of plating, t_{corr}, calculated as follows:

$$t_{corr} = t - t_c$$

, where t is the modeled thickness and t_c is the standard thickness deduction for corrosion. t_c is considered 1,0mm in every position, except from the water ballast tanks (within 1,5m of weather deck when the two sides are exposed to water ballast), where t_c is taken as 2,0mm.

For the analysis of the buckling requirements based on local loads, the equivalent applied stress is to be calculated by increasing the stress result from the FE model in proportion to the modeled thickness of the plating, divided by the corroded thickness: $\sigma_a = \sigma_{LOCAL} x \dfrac{t}{t_c}$

, where σ_a is the equivalent applied stress and σ_{LOCAL} is the stress from FE model.

For the analysis of the buckling requirements based on combined local and global loads, the equivalent applied stress is to be calculated by adding the local stress corrected as above calculated to the global stress result:

$$\sigma_a = \sigma_{LOCAL} x \frac{t}{t_c} + \sigma_{GLOBAL}$$

,where σ_{GLOBAL} is the stress resulting from the application of the hull girder bending moment.

Finally, when the critical equivalent elastic buckling stress exceeds 50 percent of the specified minimum yield stress, then the buckling stress is to be adjusted for the effects of plasticity using the Johnson-Ostenfeld formula:

$$\sigma_{cr} = \sigma_o (1 - \sigma_o / 4\sigma_c)$$

,where σ_{cr} is the critical equivalent buckling stress corrected for plasticity effects

σ_c is the critical equivalent elastic buckling stress

σ_o is the specified minimum yield stress

According to IACS[17], the vertical hull girder ultimate bending capacity at any hull transverse section is to satisfy the following criteria:

$M \leq \frac{M_U}{\gamma_R}$, where

- M is the vertical bending moment, in kNm
- M_U is the vertical hull girder ultimate bending capacity, in kNm
- γ_R is partial safety factor for the vertical hull girder ultimate bending capacity to be taken equal to $\gamma_R = \gamma_{\mathrm{M}} \gamma_{DB}$
- γ_M is a partial safety factor for the vertical hull girder ultimate bending capacity, covering material, geometric and strength prediction uncertainties; in general, to be considered γ_M =1.1
- γ_{DB} is a partial safety factor for the vertical hull girder ultimate bending capacity, covering the effect of double bottom bending, to be taken equal to:

  γ_{DB} =1.0 for sagging condition

γ_{DB} =1.10 for BC-B and BC-C bulk carriers, and loaded cargo holds in alternate condition of BC-A bulk carriers.

γ_{DB} =1.25 for empty cargo holds in alternate condition of BC-A bulk carriers

The vertical hull girder bending moment, *M* in hogging and sagging conditions, to be considered in the ultimate strength check is to be taken as:

$M = \gamma_S M_{SW-U} + \gamma_W M_{WV}$, where

- M_{SW-U} the permissible still water bending moment, in kNm, in hogging and sagging conditions at the hull transverse section
- M_{WV} the vertical wave bending moment, in kNm, in hogging and sagging conditions at the hull transverse section
- γ_S is a partial safety factor for the still water bending moment, γ_S =1 for bulk carriers
- γ_W is a partial safety factor for the vertical wave bending moment, γ_W =1.2 for bulk carriers

The ultimate bending moment capacities of a hull girder transverse section, in hogging and sagging conditions, are defined as the maximum values of the curve of bending moment capacity versus the curvature of the transverse section considered (see Figure 5.7). The curvature is positive for hogging condition and negative for sagging condition.

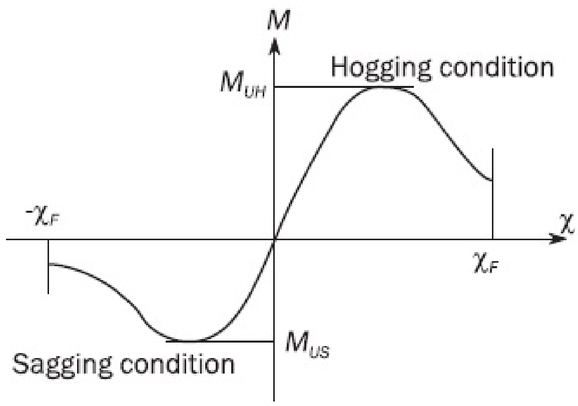


Figure 5.7
Bending moment capacity versus curvature χ [17]

As an example of ultimate hull girder assessment, the progressive collapse behavior of a 170.000 DWT bulk carrier under vertical hogging or sagging moment, as presented by Paik et al.[18] is illustrated in figure 5.8. Some selected typical failure events are also represented in the figure.

In bulk carriers, the spacing of transverse frame (or floor) at the bottom part is different from that at deck or at side shells. The tension flange (i.e., bottom plates) of the bulk carrier hull under sagging moment yields prior to buckling collapse of the compression flange (i.e., deck plates). In hogging condition, however, buckling collapse of the compression flange (i.e., bottom plates) takes place prior to yielding of the tension flange (i.e.,deck plates). This is because the deck panels of bulk carrier structures are typically much sturdier than bottom panels. Regardless of this, the section modulus at bottom is of course much larger than that at deck because bulk carriers have large deck openings. It is however less consistent with the normally expected ultimate strength characteristics of usual ship designs since the ultimate hogging moment of bulk carriers is smaller than the ultimate sagging moment.

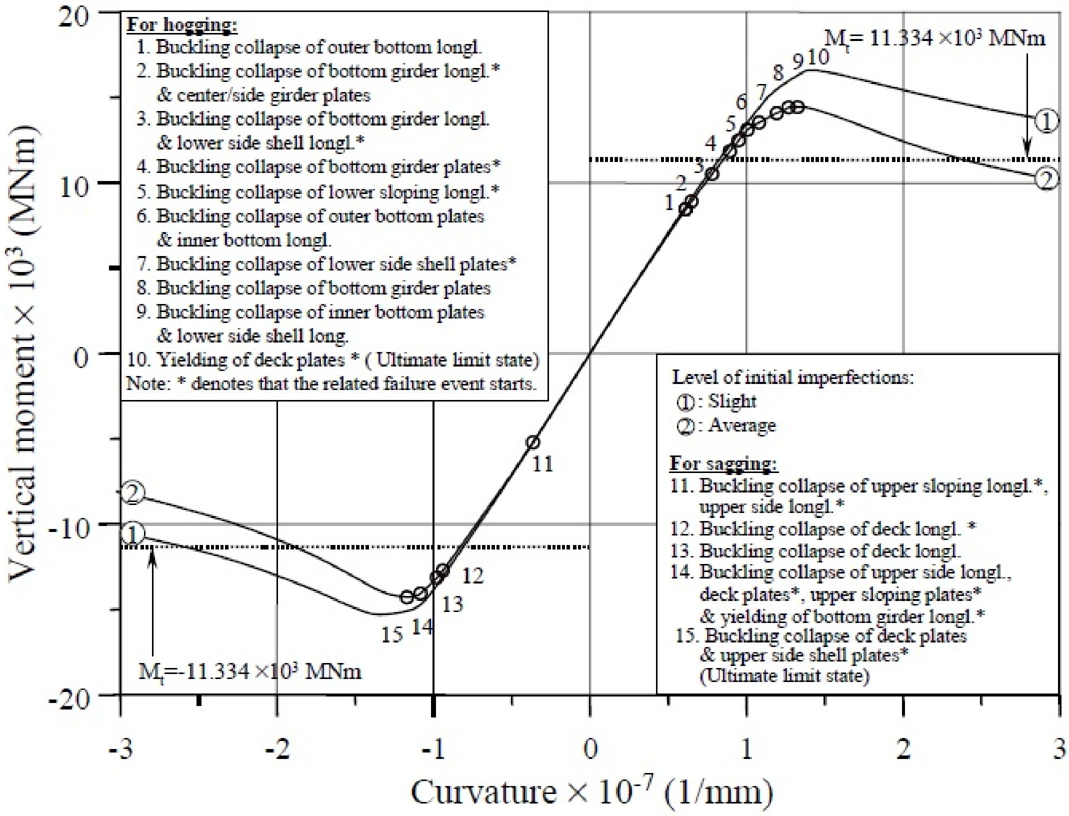


Figure 5.8

Progressive collapse behavior of the 170,000 DWT single sided bulk carrier under vertical moment varying the level of initial imperfections [18]

5.3.3 Fatigue strength assessment

Mansour et al.[19] mention that fatigue constitutes a major source of local damage in ships and other marine structures. This occurs because the most important loading on the structure, the wave-induced loading, consists of large numbers of load cycles of alternating sign. The effects of fatigue are especially severe in locations of high stress concentration, and fatigue cracks have sometimes proven to be the triggering mechanism for brittle fracture. The prevention of fatigue failure in ship structures is strongly dependent on proper attention to the design and fabrication of structural details to reduce stress concentrations. This must be followed by thorough and regular inspection of the structure in service to detect and repair any fatigue cracks that do occur before they can grow to such size that the structure is endangered.

The fatigue assessment is required to verify that the fatigue life of critical structural details is adequate. A simplified fatigue requirement is applied to details such as end connections of longitudinal stiffeners using stress concentration factors (SCF) to account the actual detail geometry. A fatigue assessment procedure using finite element analysis for determining the actual hot spot stress of the geometric detail is applied to selected details. Generally, fatigue assessment is performed for structural details located in the ship's cargo hold region in order to prevent the following types of fatigue failure:

- Fatigue cracks initiating from the toe of the weld and propagating into the plate.
- Fatigue cracks initiating from free edge of non-welded details.

The following assumptions are made in the fatigue assessment, according to IACS[20]:

- A linear cumulative damage model, i.e. Palmgren-Miner's Rule, is to be used, in connection with the design S-N curves.
- Design fatigue life, T_{DF}, is taken not less than 25 years.
- Rule quasi-static wave induced loads are based on North Atlantic wave environment. They are determined at 10^{-2} probability level of exceedance by the Equivalent Design Wave (EDW) concept.
- Net thickness approach is used.
- Type of stress used for crack initiating at the weld toe is the hot spot stress. Type of stress used for crack initiating at free edge of non-welded details is local stress at free edge.
- Fatigue stress range $\Delta\sigma_{FS}$ may be calculated by simplified stress analysis or by finite element stress analysis for details with more complex geometry.
- Long term distribution of stress range of a structural detail is assumed to follow a two-parameter Weibull distribution. Weibull shape parameter ξ is equal to 1 and the fatigue stress range $\Delta\sigma_{FS}$ is given at the reference probability level of exceedance equal to 10^{-2}.
- The acceptance criteria for fatigue checking are the total fatigue damage D to be less than 1 for the design fatigue life.

The members and locations of bulk carrier structures to be assessed for fatigue strength are listed by Class NK[21].

- *Inner bottom plating.* Intersection of sloping plate of lower stool, girder, floor plate and inner bottom plating. Intersection of sloping plate of bilge hopper tanks, girder, floor plate and inner bottom plating.
- *Sloping plate of bilge hopper tanks.* Intersection of lower end of hold frame and sloping plate of bilge hopper tank. Intersection of inner bottom plating and sloping plate of bilge hopper tanks.
- *Transverse bulkhead.* Intersection of sloping plate of lower stool and transverse bulkhead. Intersection of sloping plate of upper stool and upper part of transverse bulkhead. Intersection of slant plating of topside tanks and upper part of transverse bulkhead.
- *Sloping plate of topside tank.* Intersection of upper end of hold frame and sloping plate of topside tanks. Intersection of end of hatch coaming and sloping plate of topside tanks.
- *Sloping plate of lower stool.* Intersection of inner bottom plate and sloping plate of lower stool.

An illustration[22] of the abovementioned areas is shown in figure 5.9.

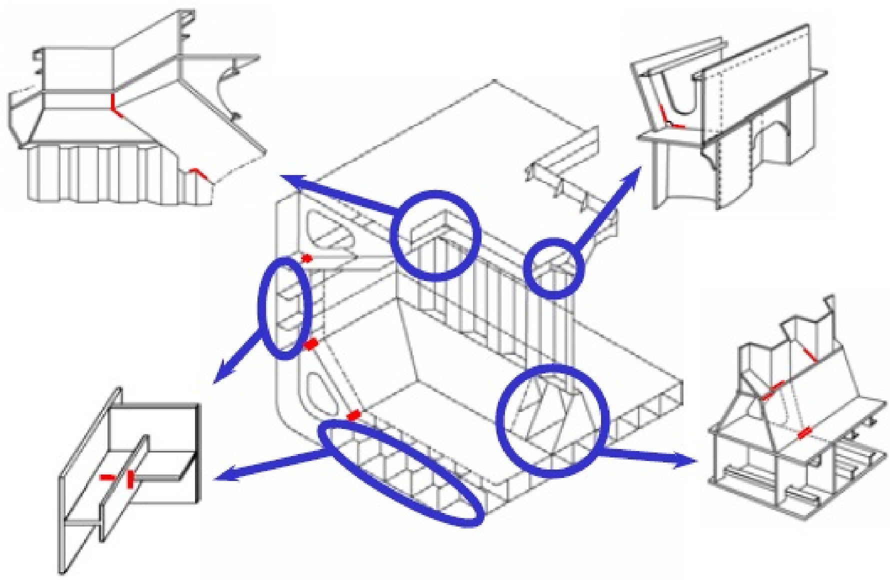


Figure 5.9

Bulk carrier details to be checked in fatigue [22]

Assessment of the fatigue strength of structural members includes the following three steps:

- Calculation of stress ranges.
- Selection of the design S-N curve.
- Calculation of the cumulative damage and the fatigue life calculation.

According to IACS[23], several types of stresses are used in fatigue assessment:
Nominal stress. A general stress in a structural component calculated by beam theory based on the applied loads and the sectional properties of the component. The sectional properties are determined at the section considered (i.e. the hot spot location) by taking into account the gross geometric changes of the detail (e.g. cutouts, tapers, haunches, brackets, changes of scantlings, misalignments, etc.). The nominal stress can also be calculated using a coarse mesh FE analysis or analytical approach.
Hot-Spot Stress. A local stress at the hot spot (a critical point) where cracks may be initiated. The hot-spot stress takes into account the influence of structural discontinuities due to the geometry of the connection but excludes the effects of welds.
Notch Stress. A peak stress at the root of a weld or notch taking into account stress concentrations due to the effects of structural geometry as well as the presence of welds.

The calculated fatigue life, T_F, is to be greater than the design fatigue life T_{DF}, thus:

$$T_F \geq T_{DF}$$

The selection of the design S-N curve is determined by the nature of the environment that the area is exposed (in-air or corrosive) and the status of the structural member under assessment (welded joint or free edge). As an example[24], basic design curves for corrosive environment are shown in figure 5.10, and can be represented by linear relationships between log(Δσ) and log(N) as follows:

$\log(N) = \log(K_2) - m \cdot \log(\Delta\sigma)$, where

N , predicted number of cycles to failure under stress range $\Delta\sigma$.

K_2, constant related to design S-N curve, as given in table 5.2

Table 5.2

Basic S-N curve data, corrosive environment [24]

Class	K_2	m	Design stress range at 2×10^6 cycles, N/mm^2
B_{corr}	2.246E12	3.0	103.9
C_{corr}	1.267E12	3.0	85.9
D_{corr}	7.600E11	3.0	72.4

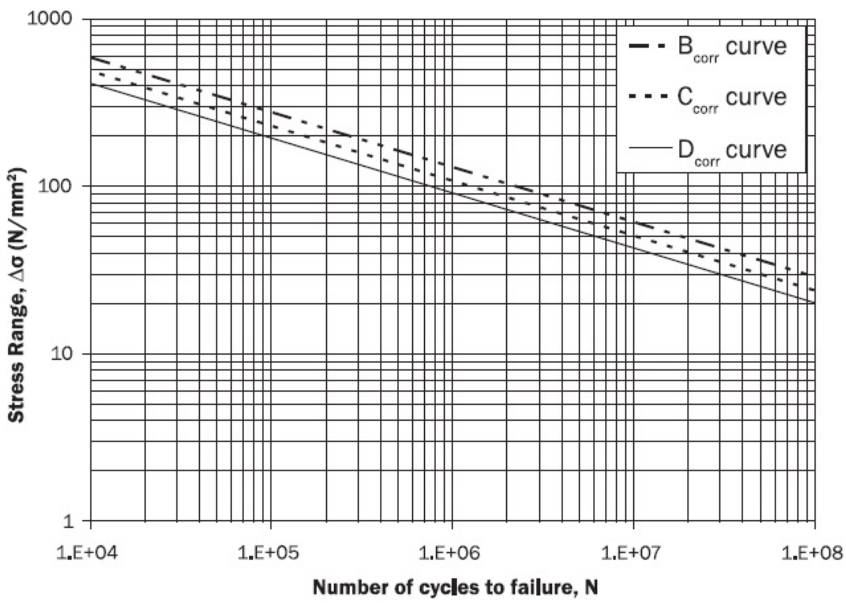


Figure 5.10

Basic design S-N curves, corrosive environment [24]

The fatigue assessment of the structure is based on the application of the Palmgren-Miner cumulative damage D taken as:

$D = \sum_{i=1}^{n_{tot}} \frac{n_i}{N_i}$, where

n_i , the number of cycles at stress range $\Delta\sigma_i$.

N_i ,the number of cycles to failure at stress range $\Delta\sigma_i$.

n_{tot} , the total number of stress range blocks.

i, stress range block index.

An example of fatigue performance of bulk carrier side frame structure is mentioned by Paik et al.[25], as side shell failure is a potential mode of water ingress into a cargo hold. Based on the dynamic pressure load study, side shell fatigue is checked in hold No 1, on a Panamax and a Capesize bulk carrier, in the vicinity of the aft transverse bulkhead of the hold. Specifically, the side frame lower connection details at the intersection of the side shell and the sloping plate of the lower hopper tank are of interest, because that location is affected by wave profile changes in both the laden and ballast conditions. The fatigue process at that location can be considered to be driven primarily by external dynamic pressure if one assumes that the ore in the laden condition is left as poured, although in our fatigue analysis we have included the additional effect of stresses due to secondary bending of the double bottom structure arising from the differential (internal minus external) double bottom pressures as well. In our case, three structural conditions are considered. For a nominal 20 year life of the structure, a gross structure is considered (i.e., no corrosion in 20 years), a 10% corroded structure (i.e., 10% corrosion in 20 years), and a 20% corroded structure (i.e., 20% corrosion in 20 years).

The resulting damage estimates (by Miner's rule) are shown in table 5.3, and the conclusions of the fatigue calculations are directly listed:

Table 5.3

Comparative fatigue damage estimates [25]

Vessel	Corrosion	Stress Range (kg/mm^2)	20 Year Miner Sum	Fatigue Life (Years)
Capesize	Gross	29.69	0.61	20+
	10%	32.99	1.47	14.3
	20%	37.12	2.24	12
Panamax	Gross	21.09	0.18	20+
	10%	23.43	0.46	20+
	20%	26.37	0.72	20+

- Fatigue estimates for the particular Panamax are acceptable. Contrary, for the Capesize considered, the fatigue estimates may be unacceptable for corrosion levels exceeding 10%.
- For these two particular vessels, fatigue was the only failure mode that indicated lower safety margins for the Capesize in relation to the Panamax. Yielding, buckling and ultimate strength were also checked in the same study, but in those failure modes this particular Capesize actually fared better than the Panamax.
- The Capesize pressure force is about 15% higher than the Panamax. This, together with the added effect of the difference in unsupported span, gives rise to fixed end moments at the Capesize side frame that are nearly 50% greater compared to the Panamax (71.81ton-m versus 44.79 ton-m). For identical end details, the fatigue stress range is proportional to the fixed end moment and the fatigue damage varies as the stress cubed. Hence for identical end details one would expect the Capesize side frame fatigue damage to be more than three times that of the Panamax. In these particular vessels, the difference is even greater because the end details are not identical.
- In the two vessels/locations considered, the side frame end connection details are not similar. The features of the end bracket are quite important to fatigue performance. In general, an integral bracket with an effective

flange plate (such as that used in the Panamax) is far superior in terms of fatigue performance than the non-integral bracket (which is used in the Capesize considered).

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[23] Fatigue assessment of ship structures. IACS rec. No 56 (1999) IACS, pp 56-10
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[25] The strength and reliability of bulk carrier structures subject to age and accidental flooding (1998), Jeom Paik & Anil Thayamballi, pp27

6

6.1 Introduction

The sixth chapter of this thesis copes with the overall design of the hold area in a bulk carrier, seen from an operational point of view. From the number of holds being required considering the size of the vessel and the density of cargo to be carried, to the definition of hold length and the transverse bulkheads to be fitted. Additionally, the purpose of topside tanks and hopper tanks presence is discussed, whereas the effects that ballast water management has on strength of the structure is also mentioned. Furthermore, the double bottom arrangement is presented, focusing on the effects of double bottom height on structural behavior of the ship.

Moreover, some fuel oil tank arrangements that are used on bulk carriers are assessed in the event of oil spill, and the probability of oil outflow is measured. Finally, the contribution that hatch covers have on the strength of the bulk carriers is cited. The main hatch cover types found on bulk carriers are presented, whereas an assessment of the collapse strength of specific hatch cover types is also featured.

The following part of this chapter focuses on the diversity of cargoes that a bulk carrier is set to carry, and the various aspects of structural design that each of them affects. Ore cargoes loading rates could influence the strength of the bulk carrier, whereas liquefaction phenomena could become a cause of bulk carrier loss. Additionally, carriage of certain types of ore cargoes, under specific circumstances, could result in spontaneous combustion of the cargo. Coal cargoes, if mixed with water onboard, are notable for their corrosivity. The main problem associated with grain carriage is its tendency to shift when the ship rolls, leading to loss of stability. Moreover, steel cargoes may lead to tanktop area exceeding the maximum permissible loads assigned by the classification society. Finally, hazards associated with timber cargoes are identified and measures for safe carriage of such cargoes are listed.

6.2 Hold design

Starting from scratch, one has to identify weather the intended bulk carrier structure is weight or volume critical, in order to assess the design requirements that might become dominant. According to Watson[1], weight is considered the critical factor when the cargo to be carried is heavy in relation to the space provided for it. As an example, iron ore loaded in alternate holds, and therefore using less than half the available space will take a bulk carrier down to its maximum draft. Contrary, volume becomes the critical criterion when the cargo to be carried is light, thus the bulk carrier will be loaded at the full available cargo space without reaching the maximum draft. A formula to determine the critical cargo density for a given bulk carrier is presented in the abovementioned book:

$$Cargo\,S.G. = \frac{cargo\,dwt}{cargo\,vol} = \frac{\frac{cargo\,dwt}{total\,dwt} \times \frac{total\,dwt}{displt}}{\frac{cargo\,vol}{total\,vol} \times \frac{total\,vol}{displt}}$$

Using some assumptions on several dimensional ratios, Watson achieves to result the critical Cargo density (S.G.) = 0.77 or $1.29\,m^3/tonne$. That means that the ship under consideration will be weight critical if the cargo that is designed to carry has a cargo density of more than 0.77 or stows at less than $1.29\,m^3/tonne$, and volume critical if the cargo is lighter.

Taggart[2] mentions that the density of the anticipated cargo controls the location of the inner bottom. For dense cargoes, it is advised that the hold should be narrow at the top, in order to prevent problems of cargo shifting. Furthermore, to prevent violent motions that would result from excessive metacentric height, it is desirable that the centre of gravity of the cargo should be relatively high. Those considerations lead to a configuration with high inner bottom and large wing tanks.

Contrary, low density bulk carrier needs much more volume to carry the cargo, that results in a lower inner bottom. This leads to the configuration that includes the high slopping inner bottom at the bilge and the topside tanks. Variations may enable the omission of topside tanks or the presence of inner side shell that makes easier the cleanup of cargo and provides extra space for water ballast.

Taggart[3] describes in detail the four principal requirements to be met on the general arrangement determination stage.

- Watertight subdivision and integrity
- Adequate stability
- Structural integrity

- Adequate provision for access

The first approach is based on limited information that might include:

- Required volume of cargo spaces, based on type and amount of cargo.
- Method of stowing cargo and cargo handling system.
- Required volume of tankage, mainly fuel and ballast for a specific range.
- Required standard of subdivision and limitation of main transverse bulkhead spacing.

Considering the bulk carrier hold design, additional considerations should include:

- Minimum interferences or obstructions inside the hold, for rapid discharge of cargo and minimum cleaning needs.
- Shape for self-trimming to the point of operation of the discharge equipment.
- Self loading with minimum hand trimming from the point of discharge of the loading equipment.
- Hatches of size and location to suit type of cargo.
- Distribution of cargo to limit the longitudinal bending moment of the hull girder.
- Assignment of ballast spaces for proper distribution when ship is light.

For bulk carriers that carry heavy cargoes, the number of holds according to Papanikolaou[4] must be odd, in order to achieve the alternate hold loading of the cargo, for stability and strength purposes. Thus, the demand for odd number of cargo holds is considered as a significant factor for the total length of the vessel.

Okumoto et al[5] note that the transverse strength of the double bottom and the side frame on bulk carriers is retained by the torsional rigidity of hopper and shoulder tanks, supported by transverse bulkheads. The torsional rigidity depends on the distance between transverse bulkheads, namely the hold length. Thus, for the bulk carrier design, hold length is a very important factor. They also suggest that a 5 holds arrangement can be applied up to 70,000 DWT, 7 holds up to 150,000 DWT and 9 or 11 holds for ships bigger than 150,000 DWT. Also there exists an opinion that a double hull side construction will be applied for ships bigger than 150,000 DWT.

Figure 6.1 illustrates the relationship between ship length and maximum hold length, as presented by Paik et al.[6]

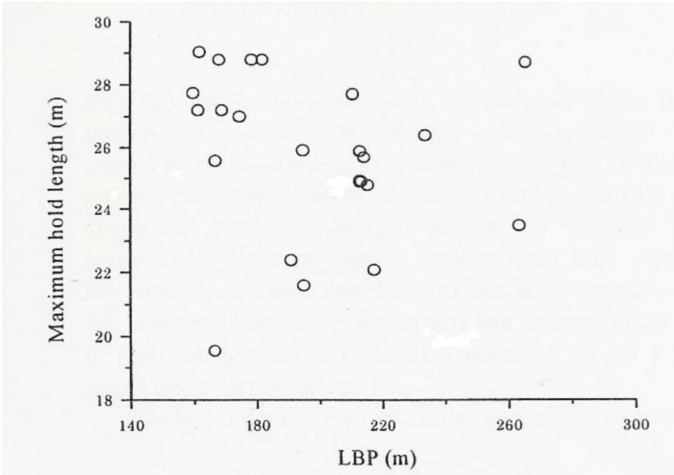

Figure 6.1

Ship length to maximum hold length [6]

Contraros et al[7] presented an interesting illustration (figure 6.2) of the cargo hold evolution for handymax and panamax bulk carriers for vessels built from 70s to 2000s. New designs produce bulk carriers with reduced double bottom height, reduced number of bottom girders (widely spaced), and increased double bottom width, due to reduced width of the bilge hopper box girder tank. Considering the cargo hold's length has remained almost constant, this practice alters the width to length aspect ratio of the double bottom, resulting to appreciably reduced stiffness due to reduced height of the double bottom.

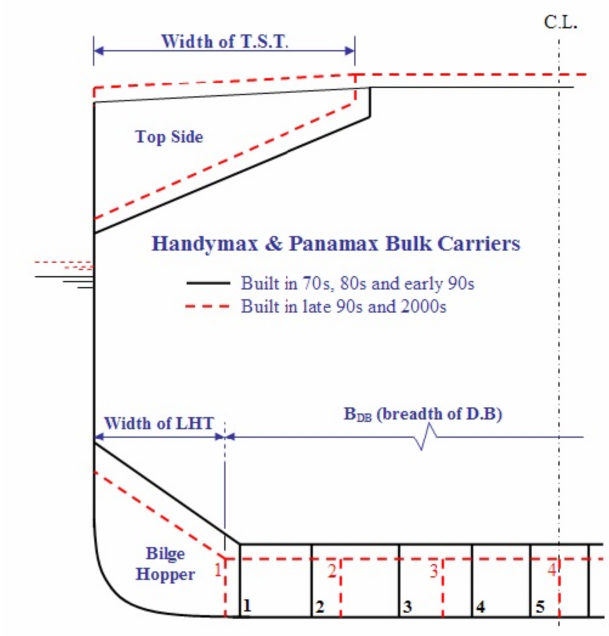

Figure 6.2

Evolution of Handymax and Panamax bulk carriers built in 70s until today [7]

6.2.1 Transverse bulkheads

Considering the transverse watertight bulkheads, every ship, according to the rules[8] must have one collision bulkhead, one aft peak bulkhead and one bulkhead at each end of the machinery space. Furthermore, the number and disposition of bulkheads are to be arranged to suit the requirements for subdivision, floodability and damage stability. For smaller bulk carriers (less than 150 m in length not required to comply with subdivision requirements), the number of bulkheads to be selected should not be less than the one mentioned in table 6.1

Table 6.1

Number of bulkheads for bulk carriers less than 150m in length [8]

Length in m	Number of bulkheads for ships with aft machinery (1)
$90 \le L < 105$	4
$105 \le L < 120$	5
$120 \le L < 145$	6
$145 \le L < 150$	7
(1)	Aft peak bulkhead and aft machinery bulkhead are the same.

IACS suggests that the bulkheads in the cargo hold region are to be spaced at uniform intervals as far as practicable. This, apart from the standard structural blocks to be considered on ship construction, enables a constant cargo hold length that leads to standard hatch cover sizes. Contrary, Taggart[9] mentions that for shallow draft bulk carriers that carry heavy cargoes, the arrangement enables alternatively long and short holds in order to achieve an acceptable metacentric height. This distribution creates very high vertical shear forces near the bulkheads, that may lead to the need for increases in the shell plate thickness.

Additionally, an extensive study on the optimum positioning of bulkheads in a Panamax bulk carrier, in order to meet the goal of increased payload capacity in addition to lowering fuel oil consumption, was conducted by Deltamarin[10]. Deltamarin has recognised that since scantlings are mainly determined by analyzing structural response to global hull girder loads, the key to structural optimization is the minimization of these loads. The most important global load effect in bulk carriers is the vertical hull girder bending moment. While the wave component of hull girder bending is typically based on rule values for unrestricted ocean service, the still water component depends on the distribution of weight (cargo, water ballast, etc.) along the ship's length. Thus, the company customizes software tools to optimize the general arrangement of a hull for minimum still water bending.

In the case of the bulk carrier, the optimization task was formulated in such a way that the variables were the locations of transverse watertight cargo hold bulkheads in the ship's stability and hydrostatics model. The objectives were the positive (hogging) and negative (sagging) hull girder bending moments under relevant loading conditions, including light and heavy ballast conditions as well as homogeneous and alternate cargo conditions. As a result of the procedure, a design variation (i.e. a compartmentation layout) that yields the least severe design bending moment can be identified and implemented as a basis for general arrangement. In the case of a Panamax design (figure 6.3), the optimization of bulkhead positions lead to a 5% decrease of the hull girder bending moment. Additionally, lighter hull scantlings were achieved as a result.

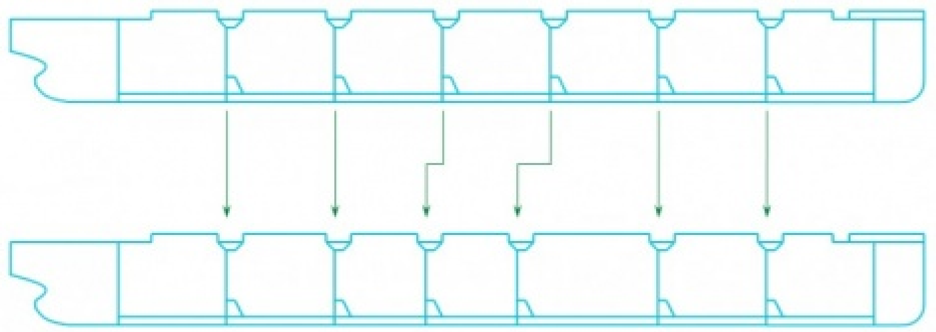

Figure 6.3

Optimization of bulkhead position, conducted by Deltamarin [10]

As discussed in a previous chapter, according to Paik et al[11], the typical corrugated transverse bulkhead is vertically corrugated and without horizontal girders. Lower and upper stools with different proportions from each other are normally located at its ends. Most corrugated bulkheads have shedder plates on top of horizontal lower stool plates. The corrugation span as illustrated in figure 6.4, normally increases as the vessel becomes larger (eg corrugation span 12m for a Panamax and 16m for a Capesize). Statistics on the corrugation angle showed that it ranges from 55 to 90 degrees. The corrugation shape in the ballast holds is typically rectangular, whereas in ore or light holds is trapezoidal. Additionally, it was found that the corrugation section modulus increases as the vessel size becomes larger.

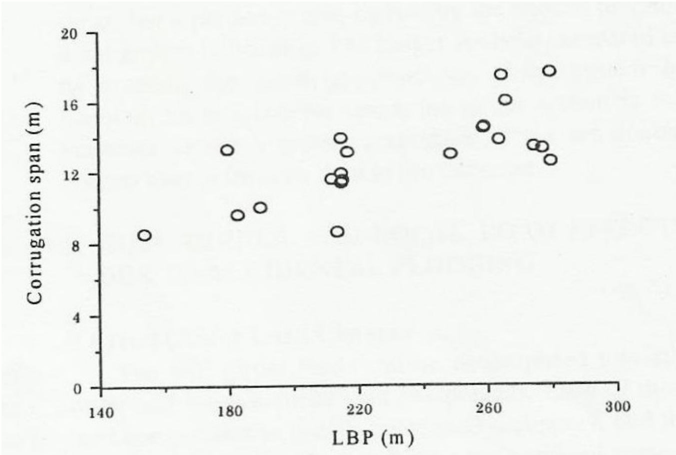

Figure 6.4

Ship length to corrugation span [11]

According to Paik et al.[12], transverse bulkheads in dry cargo holds are usually designed to withstand three load components:

- Lateral pressure due to dry cargo and/or flooding water
- Carry-over bending moment, resulting from overall double bottom bending, which is important in alternate hold loading situations
- In-plane axial force due to the net double bottom pressure.

The two first components are mainly related to cargo mass, whereas the latter is a function of cargo mass and draft. Frystock et al.[13], finally, note that the vertical bending moment acting on the transverse bulkhead is a function of torsional rigidity of the upper and lower stools, and the stiffness of the double bottom structures, with additional bending moment components transmitted from the double bottom to the bulkhead.

Okumoto et al[14] presented a formula for calculating the shearing strain S on the transverse bulkhead of a bulk carrier in an empty hold and ballast condition situation. They mention that shear deformation on the transverse bulkhead is caused by the vertical load on the transverse bulkhead. The S value as calculated is non dimensional and the formula is based only in the principal dimensions of the hold area:

$$S = \frac{B^2 dl}{722D(0.7B+l)(3\sqrt{D}+3.5)\left(0.55+0.01\frac{l}{s}\right)}, \text{ where}$$

- B breadth of the cargo hold, in m
- d full load draft, in m
- l length of hold, in m

- D depth of the ship, in m
- s floor space, in m

The calculated S values for a 30.000 DWT ship with 5 holds were 0.051 and for a bulk carrier of 130.000 DWT with 9 holds 0.064. As a result, the authors selected S=0.07 as a criterion. Considering a 250.000 DWT bulk carrier with 9 and 11 holds, the S values were 0.104 and 0.096 respectively.

6.2.2 Topside tanks and hopper tanks

Topside (wing) tanks are mainly used for water ballast, according to Eyres[15] and in special occasions can be used for the carriage of light grains. The thickness of the sloping bulkhead of this tank is determined in a similar manner to that of the deep tank bulkheads. The topside tank is stiffened internally by longitudinal framing supported by transverses. Transverses are arranged in line with the end of the main cargo hatchways; and in large ships, a fore and aft diaphragm may be fitted at half the width of the tank, between the deck and the sloping plating. Furthermore, Taggart[16] notes that the required ballast on a bulk carrier must be distributed properly along the length of the ship to reduce the probability of excessive bending moments on the ship girder.

Okumoto et al[17] state that the wing tank and the hopper tank are connected by the side shell construction and have a strong resistance against vertical forces. Additionally, against the horizontal forces, the hopper tanks connected to the double bottom also present strong resistance. The vertical force on the double bottom, such as water pressure on the bottom and the cargo weight, and the vertical force on the side shell, causes torsional moments in the fixed parts of the hopper and shoulder tanks. The torsional rigidity of these parts is important in resisting the torsional moments.

The bulk carrier configuration with inclined upper and lower wing tanks, according to Taggart[18], allows:

- A small area for clean up under the square of the hatch once most of the cargo has been discharged, as the remaining cargo slides down to canted sides. This also allows discharging gear to reach all areas, as the tank top breadth is roughly equal to the hatch opening breadth.
- Stowage free of shifting boards or other temporary devices to prevent the load from shifting to one side. Thus the upper wing tank configuration presents minimum free surface when the bulk cargo is stowed to the top of the hold. Furthermore, Isbester[19] mentions that the upper hopper thanks occupy space into which bulk cargo would never flow, a valuable feature for grain trades.

Some topside tanks are simply joined to the adjacent lower hopper and double bottom by trunking. This system considers the topside tank as an extension of the lower hopper and double bottom tank, thus it can only be filled when the lower tank is full, while the DB tank cannot be emptied until all the ballast has drained from the topside tank. Finally, some topside tanks can be used to carry grain, whereas it is not advised, because of the amount of cleaning work required before and after the loading.

A completely different equilibrium of forces acts on the wing tanks[20] of ore carriers. In laden condition, the loads from the cargo and the sea act by compressing the transverse section on the empty wing tank. Contrary, in ballast condition where the wing tank is full, there are no forces from cargo and the sea loads are significantly reduced, since the draft of the vessel is lesser. Thus, tension is present to the structural members of the wing tanks (figure 6.5).

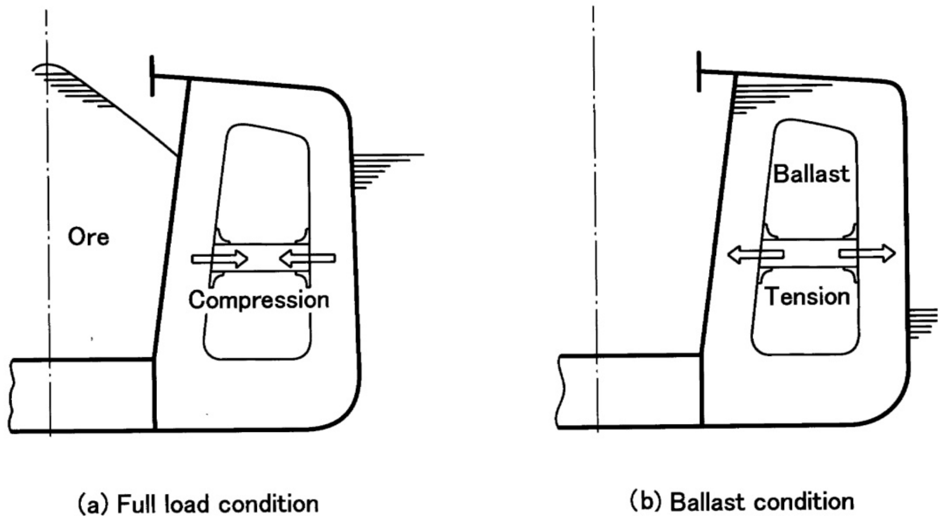

Figure 6.5

Load on wing tank of ore carrier [20]

For ore carriers, in a cargo hold length there may be an additional watertight transverse bulkheads[21] to separate wing ballast tanks or side void spaces (figure 6.6). Therefore in ballast loading conditions, it may happen that, for the two side tanks separated by such a transverse bulkhead, one is empty and the other is loaded with ballast. Consequently, hull girder nominal shear force in the loading manual is not a straight line in such a cargo hold length, since there will be also a peak nominal shear force at the additional watertight transverse bulkhead.

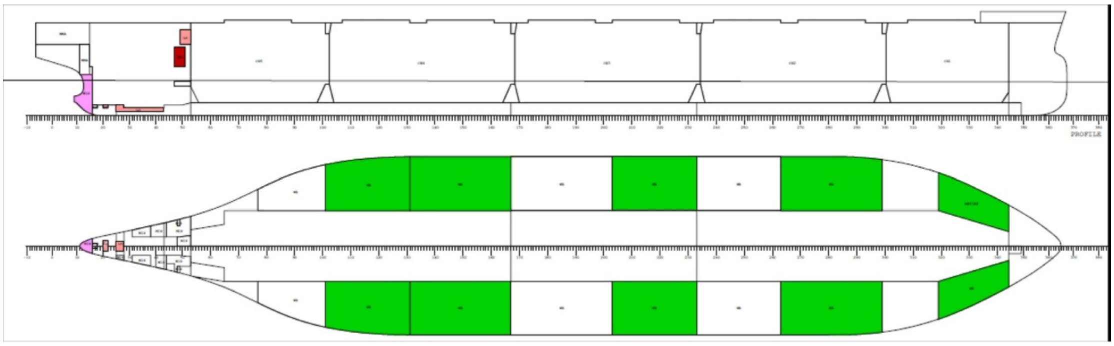

Figure 6.6

Additional watertight transverse bulkhead arranged in side tanks in each cargo hold length [21]

Last decades, an enormous effort has been made to minimize the introduction of unwanted organisms from the discharge of ballast water in any marine environment. Since bulk carriers utilize large amounts of ballast water, new design features were considered mandatory for the success of this effort. Ballast water management analysis conducted on several bulk carrier types (Handy, Panamax, Capesize) by ABS[22] reached the following results:

- Sequences of the bulk carriers are quite complex, requiring many steps to maintain drafts and longitudinal strength within acceptable limits.
- Bending moments approach the 100% allowable value of each bulk carrier exchange sequences. These ships were not designed to have ballast tanks emptied during the course of the voyage and therefore, careful planning is necessary to ensure that bending moments are maintained within acceptable levels.
- For all designs, it is difficult to exchange ballast in the cargo hold while maintaining compliance with forward draft, shear force and bending moment criteria.
- The cargo holds are generally not designed to withstand loads induced by resonant sloshing experienced during partial filling conditions. This precludes exchanging ballast in the holds during severe weather conditions.

IMO describes[23] three ballast water exchange methods that are currently in use:

Sequential method. At this process the ballast tank intended for the carriage of ballast water is first emptied and then re-filled with replacement ballast water to achieve at least 95% volumetric exchange. The use of this method requires that particular attention should be given to the ballast tank layout, total ballast capacity, individual tank configuration and hull girder strength. If the plan requires simultaneously empting and refilling closely matched diagonal tanks, then consequential torsional stresses should be considered. Still water bending moments, shear forces and stability are to remain within safe limits.

Flow-through method. A process by which the replacement ballast water is pumped into a ballast tank intended for the carriage of ballast water, allowing water to flow through overflow or other arrangements. Adequate provision should be made to avoid the risk of over pressurization of ballast tanks or ballast pipes. This method eliminates concerns of exceeding shear force and bending moment limits.

Dilution method. A process by which replacement ballast water is filled through the top of the ballast tank intended for the carriage of ballast water with simultaneous discharge from the bottom at the same flow rate and maintaining a constant level in the tank throughout the ballast exchange system. Adequate provision should also be made for avoiding the risk of over pressurization of the tanks. The hydrodynamic performance of the ballast tank is crucial to ensure full water exchange and sediment scouring.

6.2.3 Double bottom arrangement

Taggart[24] lists the advantages of double bottom structure. It results in a strong bottom that is well adapted to withstand the upward pressure of the sea as well as the longitudinal hull girder bending stresses, especially the compression resulting from hogging stresses. It provides tankage for liquids such as fuel oil, fresh water and ballast, thus using space that is unsuitable for other purposes. It results in a structure which can withstand a considerable amount of bottom damage caused by grounding without flooding of the holds or machinery spaces, provided the inner bottom remains intact. Additionally, a smooth inner hull free of stiffening structure is produced, which provides easier cleaning accessibility.

The effect of double bottom height on the structural behavior of bulk carriers was assessed by Contraros et al.[25] Using a Panamax bulk carrier as a model ship, they considered a structure with five different double bottom heights, and they studied the effects of this variation on the structural behavior of the vessel, in several loading conditions. They found that the shear stress decreases considerably as the double bottom height increases. This is due to the additional shear area available by the corresponding increased height of girders and floors. They also mention that the dominant loading condition for the double bottom grillage to produce the maximum stress values, is the oblique sea conditions. Additionally, they state that the present IACS CSR formulation (d_{DB}=B/20 or 2m, whichever is lesser) for the double bottom height requires urgent revision, and the formula that controls the DB height should include parameters related to the draught of the vessel, the aspect ratio of the double bottom (i.e. width of the double bottom between hopper tanks, over length of the cargo hold, in relation to vessel length). Furthermore, a more realistic spacing of the double bottom floors and girders is to be adopted, to assure double bottom support and accurate transmission of more balanced shear forces to the transverse bulkheads.

Table 6.2 that is part of the conclusions of the abovementioned paper provides a comparison between the current and old Rule formulations on the given bulk carrier, as well as values for a proposed interim formula for the establishing of a minimum acceptable double bottom height, based on the proposed spacing of the double bottom floors and girders.

Table 6.2

Double bottom height, spacing of floors and girders comparison [25]

Items considered	Values of "As Designed" (mm)	Values as calculated by FE (mm)	IACS CSR	IACS CSR Requirements (mm)	Proposed Formulations	Proposed Formulation (mm)
DB Height	1680	>>1900	whichever is lesser B/20 or 2 m	1610	$d_{DB} = 45B_{DB}+80\sqrt{d} + (L+240)$	1772
Spacing of DB Floors	2580 (Frame Sp. 860)	2580	whichever is lesser 3.5 m or 4 frames spacing	3240	whichever is lesser 3.0 m or 3 frames spacing	2580
Spacing of DB Girders	4050 (Sp.of longs 810)	3240	whichever is lesser 4.6 m or 5 spacing of longs	4050	whichever is lesser 3.0 m or 4 spacing of longs	3000

Paik et al[26] note that the double bottom height and also the width of flat part of inner bottom in conventional bulk carriers increase remarkably as the vessel becomes larger, as represented in figure 6.7.

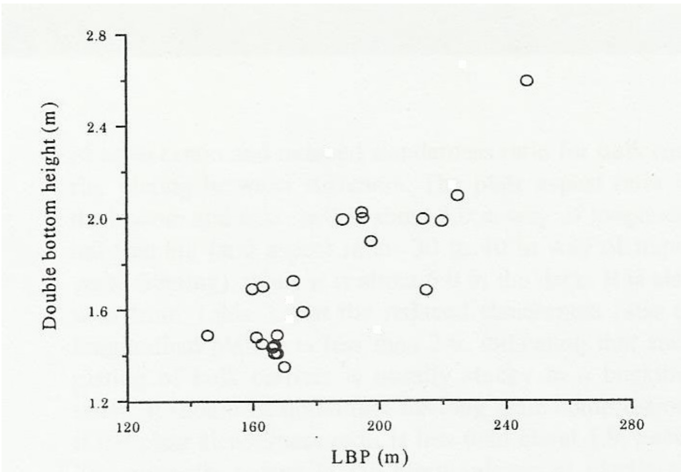

Figure 6.7

Ship length to double bottom height [26]

Capesize bulk carriers usually carry their fuel oil in engine room wing tanks. For smaller Handysize or Panamax ships, heavy fuel oil (HFO) is most commonly allocated to center double bottom tanks. Alternatively, bulk carriers may have HFO in the outboard double bottom/wing tanks, or arranged in deep tanks at the fore part of the vessel together with the engine room tanks.

Considering the tank arrangements, various studies have been made to assess the influence of the arrangement and location of bunker tanks to the oil outflow from collision and grounding. IMO proposed a probabilistic based procedure for assessing oil outflow performance, by using the *probability of zero outflow* P_O, that represents the likelihood that no oil will be released into the environment, given a collision or grounding casualty which breaches the outer hull. Additionally, the *mean outflow parameter* O_M is the nondimensionalized mean or expected outflow. The five bunker tank arrangements evaluated by Michel et al[27], for a Panamax vessel with HFO capacity of 2200m^3 are illustrated in figure 6.8. The projected outflow for the five configurations can be found in table 6.3.

- *B1.* HFO arranged in a pair of deep tanks forward of No 1 hold and a pair of engine room wing tanks. A double bottom is arranged under the forward deep tanks.
- *B2.* Similar to B1 configuration, except that 2m wide void spaces are arranged outboard of all fuel tanks.
- *B3.* All HFO is allocated to two pairs of engine room wing tanks.
- *B4.* HFO is allocated to three centerline double bottom tanks.
- *B5.* HFO is allocated to three parts of double bottom/wing ballast tanks.

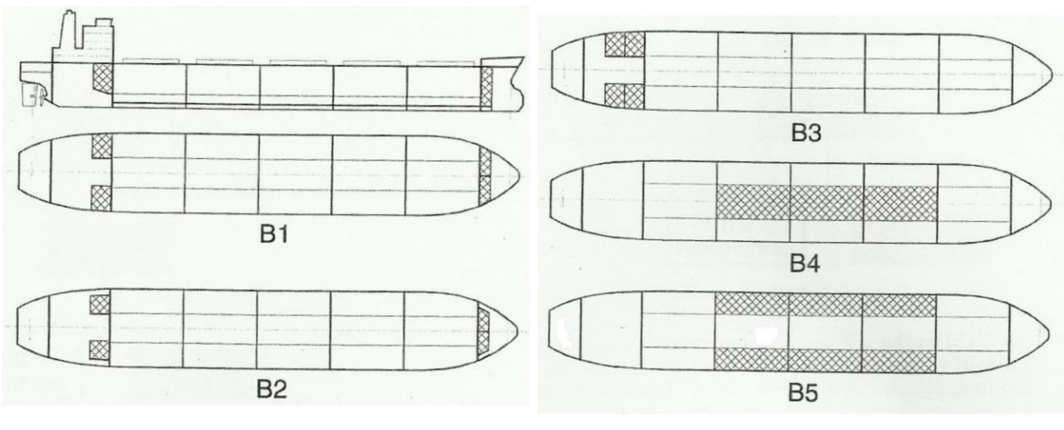

Figure 6.8

Bulk carrier tank configurations [27]

Table 6.3

Bulk carrier outflow parameters [27]

	Probability of Zero Outflow			Mean Outflow (m^3)		
	Collision	Grounding	Combined	Collision	Grounding	Combined
B1	0.852	0.914	0.889	40	5	19
B2	0.968	0.912	0.934	9	5	7
B3	0.921	0.990	0.962	32	1	13
B4	0.999	0.514	0.708	1	77	46
B5	0.968	0.514	0.696	25	42	35

Concluding, the findings related to outflow for bulk carriers are:

- Configuration B3 provides good outflow performance. The tanks located in the engine room, confined to a short length of the ship reduce the probability of penetration in collisions. Breaching the tanks in a grounding scenario is very unlikely, since they are located aft and above the inner bottom.
- The forward deep tanks in configuration B1 are susceptible to damage from both collisions and groundings. When double hull protection is arranged outboard of the bunker tanks (configuration B2), the mean outflow is significantly reduced.
- The double bottom configurations (B4 & B5) have the poorest outflow performance. B4 condition has a slightly lower mean outflow compared to B5. The large center double bottom tanks in B4 have a high probability of damage and because of their size, thus spill more oil than the small wing tanks of configuration B5.

Finally, Barone et al[28] note that the evaluation of the total O_M given by the tank longitudinal subdivision of a Panamax bulk carrier, shows that forward positions of bottom tanks lead to greater contributions to the total oil outflow. Therefore, the influence of the tank longitudinal position was investigated over the whole ship body, considering a 300m^3 tank and moving it forward along the cargo area. The results are shown in figure 6.9, where the x values represent the aft tank boundary position as a percentage of L and the y values is the increase of O_M expressed as percentage of the value relative to the aftermost tank position. The increase of O_M is noticeable mainly due to greater probability of forward tanks to be involved in grounding effects.

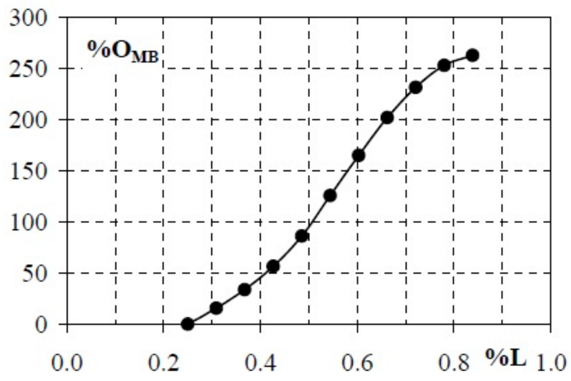

Figure 6.9

Influence of O_M due to longitudinal position of a 300m^3 DB tank [28]

6.2.4 Hatch cover design

Hatch covers are formed[29] by several steel panels which rest horizontally across the hatchway, sealing the hatch opening. Each panel consists of an upper surface constructed of steel plate, reinforced and supported on the underside by steel beams or stiffeners. The panel may be of open construction, or may be a sealed unit closed on its underside by plating similar to that on its upper sides, and treated inside with a rust inhibitor. Since the ship at a seaway moves and flexes, the different conditions of loading lead the vessel to hog or sag, leading to large changes in the size and shape of the hatch opening. As a consequence of the rigidity of the hatch covers and the flexibility of the ship's hull, elastic joints are necessary between hatch covers and hatch coaming. MacGregor[30], a company that manufactures hatch covers, adds that for stable and smooth operation of the hatch covers on board, the panels should be stiff. However, to maintain weathertightness at sea, the steel structure of a hatch cover, as well as the bearing pad and sealing arrangements must adapt to the varying shape of the coaming top while the hull is working and flexing at sea. The optimal stiffness of the steel structure of a hatch cover panel is a compromise between the above issues.

Mac Gregor further states[31] that the large size of the hatches reduces the torsional stiffness of the hull and causes twisting and diagonal changes in the hatchway, as well as warping of the deck plane in rough seas. The longitudinal bending of the hull or hogging/ sagging causes considerable changes in the hatch length. The third major type of flexible deformation is bending of the sides inwards and outwards. This not only occurs at sea but also in port when the draught changes due to variations in loading. In winter conditions the pressure of ice contributes to the flexible deformations of the hull.

The main types of hatch covers used in bulk carrier structures are the folding type, the side sliding type and the piggy back hatch covers, whereas other types (lift away hatch covers, stacking hatch covers) are used in a more limited basis.

Folding type hatch cover. Two pairs of panels cover the hold area. One pair of panels folds to the fore end of the hatch and the other to the after end. More complex systems have three or four folding panels in a set (figure 6.10). The system can be wire operated when cranes or derricks are available or can be hydraulically powered by external or internal hydraulic cylinders

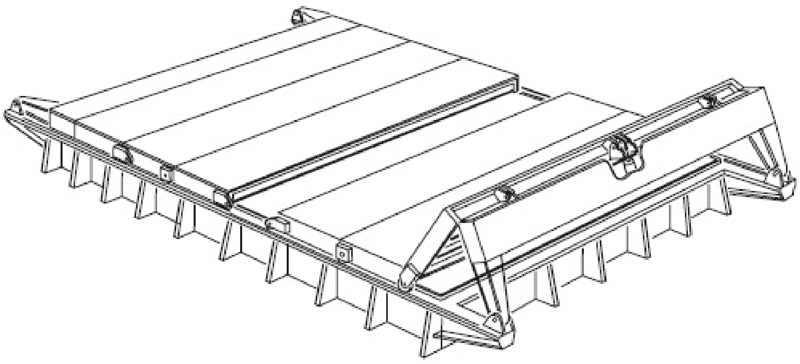

Figure 6.10

Folding type[32] hatch cover [32]

Sliding type. The traditional side-rolling cover[33] consists of two panels per hatch, each panel rolling sideways (figure 6.11) on a pair of transverse ramps, thus presenting a minimum obstacle when loading. In some cases both panels can be stowed together on one side to further enhance access when loading and unloading. This alternative reduces daylight opening by approximately 50%. A single-panel type where the panel stows transversally or longitudinally is mainly used on very large ore carriers (VLOC's), with sufficient free deck area. The covers open by lifting to the rolling position and rolling out by the drive mechanism.

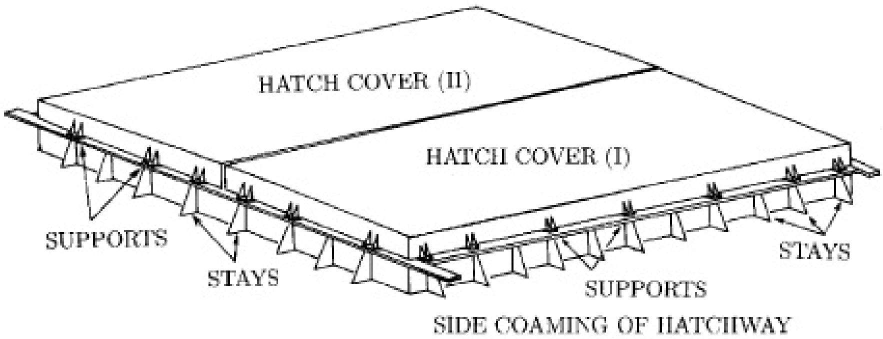

Figure 6.11

Side-sliding type hatch cover [34]

Piggy-back hatch covers [33] are used on bulk carriers when the available deck space is insufficient to accommodate folding, side-rolling or end-rolling covers. This system always comprises two panels (figure 6.12), with one panel being raised high enough for the other to roll underneath and to support the lifted panel on to its 'back'. Both panels can then be rolled back and forth. The system can either be applied to a pair of hatches or to the two panels of a single hatch.

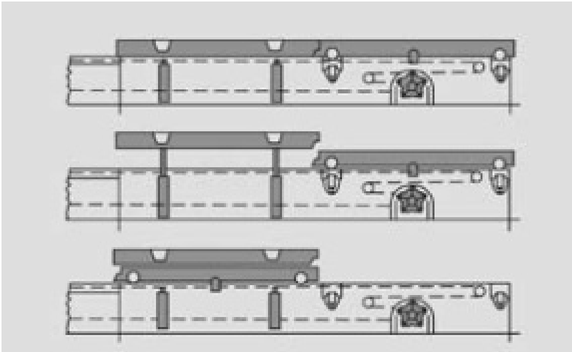

Figure 6.12

Piggy-back hatch cover [33]

Yao et al.[34] assessed the collapse strength of two types of hatch covers under lateral pressure load, using FEM analysis. Additionally, simple methods to evaluate the strength were used.The folding type hatch cover was modeled as a beam of which both ends were simply supported. The hatch cover of side sliding type was modeled as an orthotropic plate of which three edges were simply supported while the remaining edge was considered free. The major findings of the comparison of the two designs are summarized below:

- Hatch cover cannot sustain a fully plastic load because of the occurrence of local buckling collapse of a top panel as a stiffened plate under combined lateral pressure and thrust loads.
- Except the hatch covers of Handysize and Capesize bulk carriers designed in accordance with the IACS rule, the top panel undergoes local buckling between stiffeners before its overall collapse as a stiffened plate starts to take place.
- The overall buckling collapse of the top panel as a stiffened plate becomes a trigger for the overall collapse of a whole hatch cover.
- In case of hatch covers of Panamax and Capesize bulk carriers designed with the IACS rule, some strength surplus can be expected even after the corrosion margin has been wasted.

6.3 Cargo type diversity

Since bulk carriers are employed to transport a variety of dry cargo types, it would be useful to focus on the special considerations each cargo type would put on the structural design. Capaitzis[35] states that the basic parameter for dry cargo is the stowage factor, expressed in m^3/ton, or its inverse specific gravity, expressed in ton/m^3. Specific gravity values range significantly, from approximate figures of 3.5 for iron ore, 1.25 for bauxite, 1.10 for phosphates, 0.80 for coal, 0.75 for grain, while other commodities cover this range and all the way down to 0.35 ton/m^3. Some examples of the bulk carrier specialization needs could consider the introduction of self unloading equipment for bulk cement, special arrangement and equipment for timber carriers, open hatches for paper products, smaller hatches for grain traders etc. Other elements could be the cargo inflammability and gases necessitating adequate ventilation, fluidity and angles of repose for grain, strengthened double bottoms and tanktops for steel coils or grab discharge and corrosive or abrasive cargoes.

The International Maritime Safety Bulk Cargo Code (IMSBC Code), mentions the main special hazards associated with solid cargoes in bulk when they are shipped: structural damage due to improper cargo distribution, liquefaction of cargo and chemical reaction of cargoes.

6.3.1 Ore

IMO[36] suggests that in order to avoid the overstressing of the hull when loading high density cargoes (stowage factor about 0.56 m^3/ton or lower), or when detailed information is not available for high density bulk materials, the following precautions are recommended.

- The maximum number of tones of material loaded in any cargo space should not exceed $0.9 \cdot L \cdot B \cdot D$ tones, where
  L = length of the hold in m.
  B = average breadth of the hold in m.
  D = summer load draught in m.
- Where material is untrimmed or only partially trimmed, the corresponding height of material pile peak above the cargo space floor should not exceed 1.1xDxStowage Factor, where the S.F. is given in m^3/ton

Current loading rates for iron ore carriers can be[37] in excess of 16.000 t/hr. Some shipowners and operators are of the opinion that these loading rates are already pushing the limits for the safe loading and operation of such vessels. There is real concern as to whether current bulk carriers and ore carriers have adequate local and global structural strength to withstand the consequences of the highest cargo loading rates, particularly pertinent for older vessels. Kokarakis et al.[38] describe the effects of overload that could occur by an overshoot of the cargo in the hold, on the strength of the vessel (table 6.4). A 10% overload could increase the SWBM by up to 80%, and shear force by up to 26%.

Table 6.4

Load variation due to 5% cargo overshoot error [38]

Vessel Size	Bending Moment Variation %	Shear Force Variation %
Capesize	79	18
Panamax	60	26
Handymax	23	19
Handysize	24	14

Another vital issue on ore carriers could be the local strength of inner bottom plating and stiffeners, which are experiencing significant impact loads during cargo loading at high loading rates. Dry bulk cargoes are typically loaded by conveyors and may be dropped from height levels above the main deck with consequent high impact loads on the inner bottom, in particular at the start of loading with high density cargoes.

Liquefaction is another major phenomenon that bulk carriers may encounter when carrying some types of ores (mainly nickel ore). The procedure is well described in the Standard Cargo magazine[39] : Such cargoes normally contain a degree of moisture within the particles. If the cargo has laid in piles at the mine, having been transported to the terminal in open barges or trucks and loaded onto the terminal stockpiles during heavy rain, there may be a dramatic increase in moisture levels. When the cargo is subject to recurring cycles or cyclic forces, such as the movement of the ship (rolling/pitching/slamming), it could reach its flow moisture point. Then, the cargo enters a stage of transition whereby it begins to react like a fluid because of the loss of friction between the particles. This process is called liquefaction. The cargo tends to undergo a progressive shift in one direction with the ship's rolling and does not return to the centre (figure[40] 6.13). With further rolling, the ship gradually acquires more weight of cargo to one side and develops an increasing list. This dangerous situation leads to further loss of ship stability and potentially capsizing.

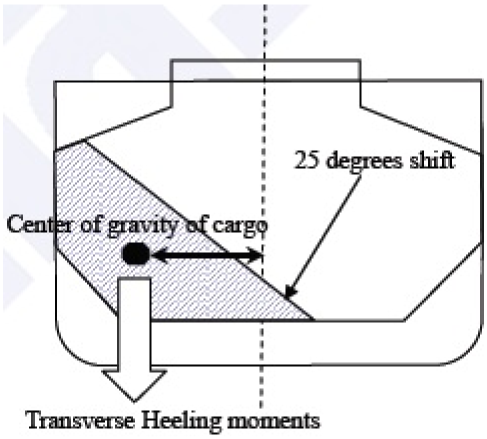

Figure 6.13

Assumed cargo shift for stability evaluation [40]

Specially constructed cargo ships[41] to carry nickel ore cargoes (figure 6.14) go far beyond the structural configuration of the conventional bulk carriers, by implementing cargo holds that would allow a small quantity of cargo shifting due to liquefaction, and a relatively small free surface area.

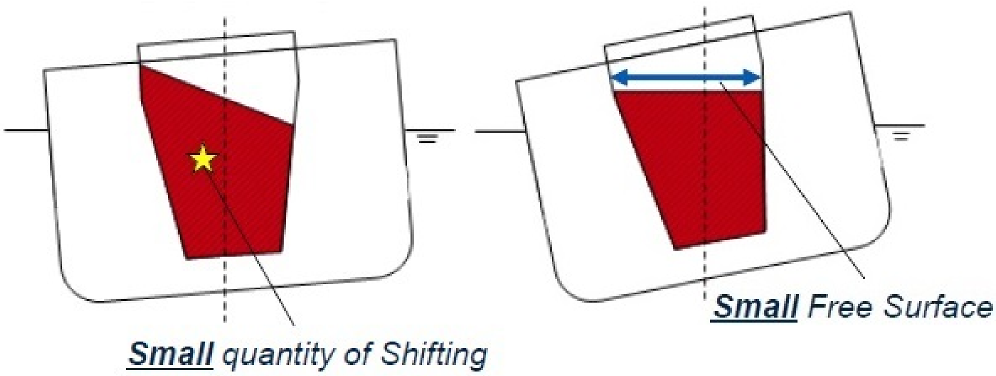

Figure 6.14

Ore carrier advantages on liquefaction phenomena [41]

Whilst iron ore is not naturally corrosive, it is abrasive[42] due to its high density and hardness. The abrasive nature of iron ore restricts the use of protective coatings on the tank top plating and lower stools, whereas premature wear of coating has been observed at the lower bracket connections to the side shell frames. The unloading of iron ore also requires the use of large grabs that can weigh up to thirty tones. The robust use of the grabs, bulldozers and vibration hammers to remove loose cargo provides a risk of sustaining mechanical damage to members inside the cargo hold.

The carriage of a specialized type of iron ore, the Direct Reduced Iron (DRI) can cause additional problems to the structural integrity of the bulk carrier. Harrison in his paper[43] presents the potential hazards. The direct reduction process is a means to increase the iron content of ore through the removal of oxygen. Iron ore pellets are placed in a reactor through which a reducing gas is passed at high temperature, leading to the removal of oxygen from the ore, leaving behind the iron in a free metallic form, thus increasing the Fe content from about 65% to almost 85%. The difficulties in the carriage are due to this procedure being revisable (the DRI can re-oxidize). In doing so, it releases energy as heat. It is possible for this to lead to thermal runaway leading to burning of the iron, by reaching 1000^{o}C. This process is accelerated by contact with water, especially sea water, since salt acts as a catalyst in the process. In addition hydrogen is liberated and when mixed with air in the hold forms an explosive atmosphere.

As a result, special precautions are to be taken when DRI is loaded in cargo holds. The principal issue is the ability to inert the holds. Another is to be able to monitor the temperature and take readings of oxygen and hydrogen. Since bulk carriers are not designed to have systems for inerting cargo holds, special arrangements have to be made by fitting pipes across the tank top, connected to a pipe led to the deck, in order to carry this type of cargo.

6.3.2 Coal

The major effect that influences the hold structure when carrying coal is corrosivity of coal due to moisture formed in the cargo hold (sweat) whereas liquefaction phenomena have also been reported due to presence of water in the cargo. Additionally, coal cargoes may release methane and hydrogen, both of which are flammable gases, which can make an explosive mixture with air. Some coals are liable to spontaneous heating, which can cause fire.

Isbester[44] names two types of sweat (figure 6.15). Cargo sweat consists of condensation which forms on the surface of cold cargo when warm moist air comes in contact with it. Ship’s sweat is the condensation which occurs when warm moist air in the hold comes into contact with the cold steelwork which forms the deck and shell plating of the ship. The magnitude of the two phenomena can be amplified when loading from a cold region and sailing to hot areas or vise versa, thus if the thermal differences between the cargo and the environment are considerable.

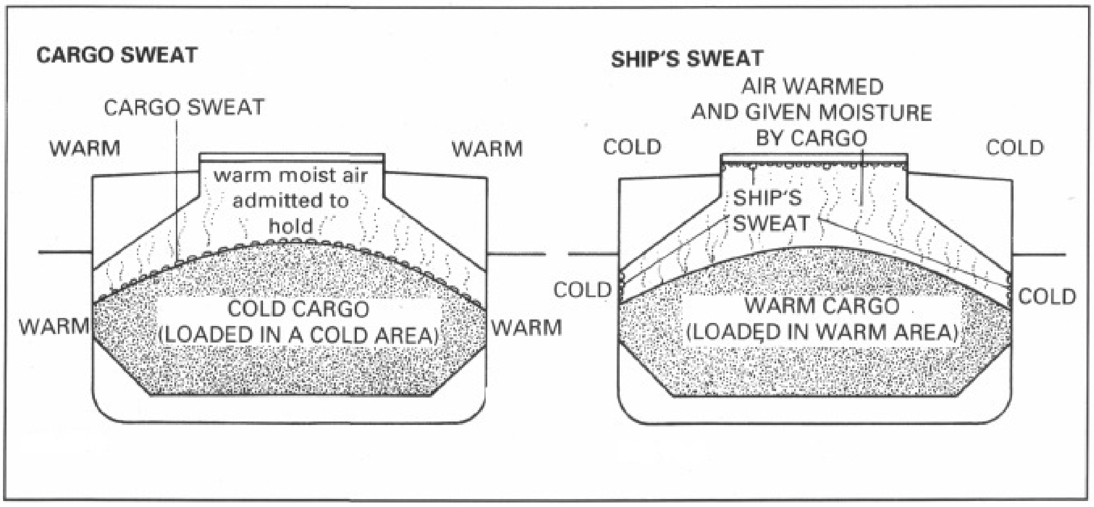

Figure 6.15

Cargo sweat and ship's sweat [44]

Additionally, when carrying coal[45] (figure 6.16), the moisture generated in the holds due to differences in the temperature in the hold and surrounding seawater will dissolve the sulphur in the coal causing a chemical reaction that will lead to the development of corrosion. As a result, the areas adjacent on the side shell and the plating of the cargo hold are prone to such corrosive phenomena. The process[46] of side shell sweating and the presence of impurities in coal (such as sulphur) are two reasons why corrosion rates in bulk carrier cargo holds are so variable. Their presence determines how often and to what extent corrosion is accelerated.

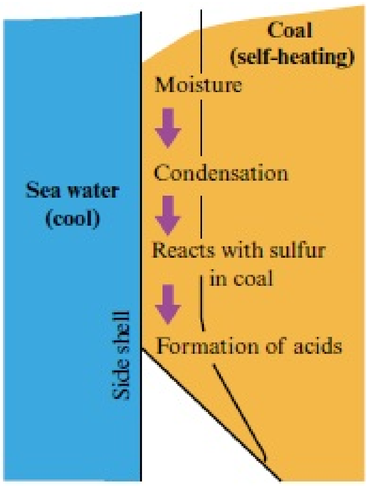

Figure 6.16

Acid production process [45]

Coals may emit methane[47], a flammable gas. A methane/air mixture containing between 5% and 16% methane constitutes an explosive atmosphere which can be ignited by sparks or naked flame. Methane is lighter than air and may, therefore, accumulate in the upper region of the cargo space or other enclosed spaces. If the cargo space boundaries are not tight, methane can seep through into spaces adjacent to the cargo space. As a design precaution, to minimize the risk of spontaneous combustion, the coal should not be stowed adjacent to hot areas. Hot areas are considered the areas of cargo hold in contact with the cargo having a temperature consistently greater than 55^{o}C during carriage of the cargo, such as can sometimes be experienced when heated fuel oil service and settling tanks have a common boundary with the cargo hold. Additionally, hatches should be able to seal the cargo area, not allowing ventilation of cargo, thus reducing the oxygen in the atmosphere and the possibility of coal self igniting.

6.3.3 Grain

The main feature of grain and other agricultural products is their ability to flow freely. Their tendency to shift when loaded in bulk carriers and exposed to ship's motions, endanger ship's stability and for this reason special regulations governing grain carriage are into force (International Grain Code). Isbester[48] describes in detail the process of grain cargo shift on bulk carriers:

The free flowing characteristic of grain reduces the stability of any ship which carries it. Grain in a partially filled cargo compartment displays a free surface effect similar to that of a liquid in a partially filled tank. If the ship rolls the grain is likely to flow to one side of the compartment, where it will cause the ship to list or to capsize. Conventional bulk carriers are well suited to the carriage of grain, as their design reduces some of the adverse effects of bulk grain upon stability. The design of the holds of bulk carriers has been developed to create compartments which can be filled to near 100 per cent of capacity without trimming, except for spout trimming by the shiploader. The upper wing tanks occupy spaces into which cargo would not flow, thereby greatly improving the self trimming character of the conventional bulker hold.

The area within the hatch coaming on a conventional bulk carrier is much smaller than the hold area below, so that the free surface of the cargo is much reduced when the hold is filled with cargo to the top of the coaming. The coaming, formed of deep vertical plating, acts as a feeder from which cargo will flow down to fill any spaces remaining within the hold as the cargo settles during the voyage.

It is evident[49] on bulk carrier design that the conventional configuration with hopper shaped holds reduce the extent to which cargo can shift (figure 6.17), as well as providing tank space which allows adjustments in the stability by ballasting. Moreover, topside tanks may be loaded with grain, optimizing the amount of cargo carried, although this practice is not common, since after unloading the cargo, excessive efforts are needed to clean the confined wing tank areas. Additionally, since grain is to be consumed by human or animals, cleanliness of cargo holds is considered as prerequisite, whereas any contamination by water or oil, or contact with sources of heat is to be avoided.

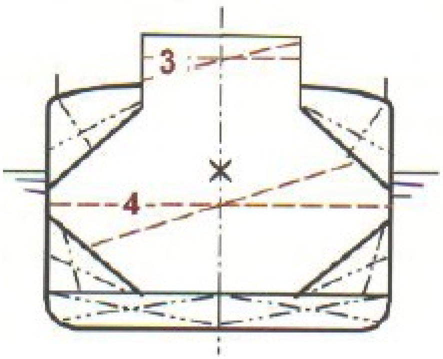

Figure 6.17

Cargo shift constrained by hatchway (level 3) or wing tanks (level 4) [49]

Contrary, a disadvantage of the presence of hopper tanks could be encountered when the need for two or more parcel carriage in the same hold at the same time arises, as described by UK P&I Club[50]. Although separation material is used to avoid admixtures, the level of separation between two parcels should not be located in the vicinity of the upper ballast tank hoppers (figure 6.18, situations 3&4). This ensures that when the inevitable settling of cargo occurs, during the voyage, the surface area of the separation material will remain adequate and prevent admixture. This problem does not arise in the vicinity of the lower hopper tanks (figure 6.18, situations 1&2).

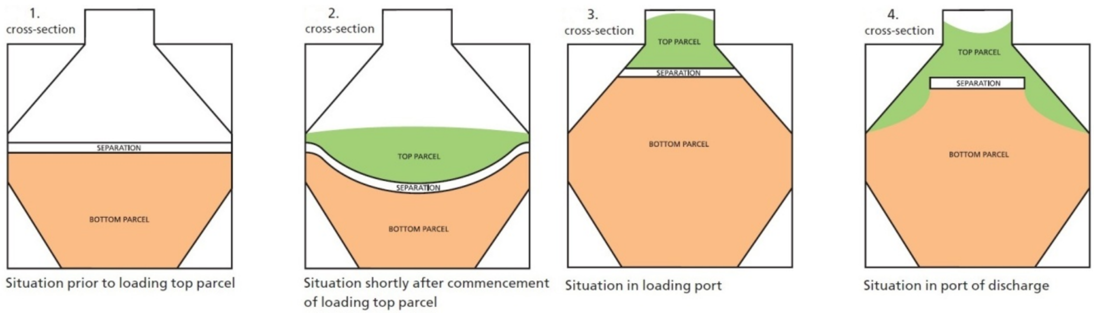

Figure 6.18

Separation of two parcels in bulk carrier hold [50]

6.3.4 Steel cargoes

Isbester[51] notes that steel coils do not constitute a homogeneous bulk cargo, and that the main problem arising from this is that ship's stated maximum permissible loading per square meter for tanktop loading does not apply to cargoes which apply patch loads.

UK P&I Club[52] provides specific examples of this effect. When loading steel coils it is usual to load not more than three tiers high with individual coils weighing up to 10 tonnes. If the unit weight is more than 10 tonnes, only two tiers are loaded (figure 6.19) and if more than 15 tonnes then only one tier is loaded. Usually two lines of double dunnage measuring 6"x 1" are laid between the coil and the tanktop. The pressure exerted over the small bearing surface of the lowest coil is about 30 tonnes. Without due care, the customary dunnage may not be sufficient to effectively spread this weight and there is a risk that the tank top will be overloaded beneath each unit.

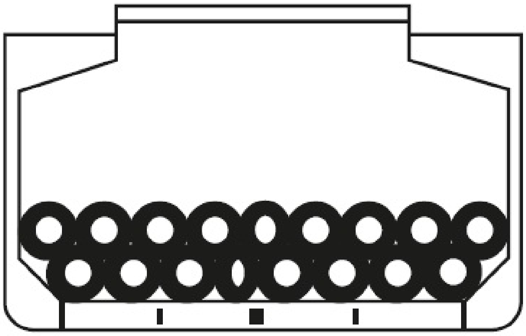

Figure 6.19

Two tier steel coil loading[53]in a double side skin bulk carrier []

The stowage of steel slabs poses similar problems. A typical slab may measure 6 m x 1.25m x 0.25m and weigh 14.75 tonnes. The area of such a slab is 7.5m and when stacked 7 high, there would be 103 tonnes bearing down on the tank top. Assuming the slabs were stowed flat, this would indicate a load of 13.74 tonnes per square metre – 14.5% in excess of a 12 tonne permissible limit. However the lowest slab is likely to be supported by three or four baulks of timber in order to facilitate handling by forklift truck. This means that the entire stack is supported on a maximum of four points, resulting in a tremendous concentration of weight on a small area. Unless larger dunnage is utilized (figure 6.20), thereby spreading the load to within satisfactory limits, the tank top is likely to be overloaded when such cargo is loaded in the manner described.

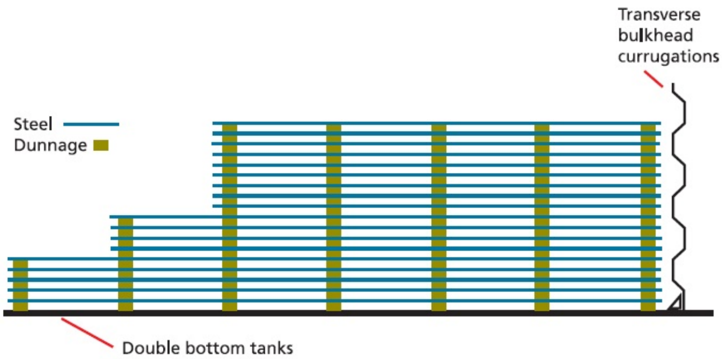

Figure 6.20

Dunnaging constructional steel[54] [54]

Bearing in mind the manner in which steel billets and slabs are usually dunnaged and stowed, it should be realised that little or no weight of that stowage will be distributed to the sloping tank sides unless special dunnaging arrangements are constructed to do so. Additionally, if fork lift trucks are to be used in the hold for the positioning of heavy items such as steel coils in the stow, it will be necessary to confirm that the weight of the loaded fork lift truck does not exceed the maximum permitted tanktop load.

6.3.5 Timber

Timber and forest products (such as logs) are all relatively light and bulky cargoes[55] which fill ship's cargo compartments long before she is down to her marks. In order to carry the maximum cargo it is normal to carry additional cargo on deck, provided that the configuration of the ship and the nature of the cargo permit, thus, its weight shall not exceed the designed maximum permissible loading on weather decks and hatchcovers. In this particular case, it is evident that specialized carriers with box shaped holds would be better in log carriage than conventional bulkers with hopper and topside tanks, spaces that possess valuable volume.

Since timber has the physical feature to absorb water, it is evident that special calculations are to be made on stability issues. Additionally, when loading timber, further attention should be paid to avoid mechanical damage on side frames and bottom plates, whereas proper and adequate lashing equipment is to be used on deck lashings, since ship's motions causes notable accelerations on deck.

Adequate balance between the maximum permitted volume of cargo on deck and improving the stability can be achieved by carrying ballast or additional bunkers. The first step towards achieving adequate stability is to reduce free surface effect to a minimum. Next, as much ballast should be carried as the limiting draft permits. Thirdly, extra bunkers can be carried. In this manner, Clark[56] lists the requirements for a vessel to be loaded on the special "lumber" load marks, as mentioned in the "lumber regulations". Listed below are the ones of structural interest:

- The deck cargo is protected by sea by a raised forecastle, and if under 100m in length, a raised superstructure aft.
- The ship is built with additional longitudinal subdivision in the midships double bottom tanks, in order to minimize the loss of stability through free surface effects due to slack tanks.
- The timber stow extends over the entire effective length of the weather deck. This ensures that the reserve buoyancy of the stow is evenly distributed along the ship's length and that there is no trimming effect due to the immersion of a partial stow, either near the bow or stern, occurring at the ends of a roll.

- The deck stow of timber is adequately secured and built up evenly to a height sufficient to provide reserve buoyancy but is not excessive for the voyage weather conditions.

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7

7.1 Introduction

The seventh chapter of this thesis outlines the main alternative designs that have been implemented last decades in bulk carrier structural design, whereas the major areas of concern for the design of the future are also listed.

At first, the implementation of double side skin configuration is discussed. The benefits arising from this introduction are listed and a comparison with the conventional single side design is made. Additionally, some alternative designs proposed for the side structure area are also discussed. Strength aspects such as collision resistance and the residual strength of the structure are mentioned, whereas the reliability levels of the proposed structure in comparison with the single side structure are also described.

Subsequently, the general characteristics of a Newcastlemax ore carrier (202,500 DWT) are presented, mainly by listing the structural arrangement of such a vessel and its advantages compared to a conventional bulk carrier. Moreover, a hybrid configuration (Hycon) bulk carrier is presented, demonstrating double sides in the fore and aftmost holds, whereas the other holds remain single sided. Furthermore, the Optimum 2000 is listed, a bulk carrier providing each cargo hold with a longitudinal bulkhead. This leads to advanced strength and stiffness of the structure.

Alternative designs are then presented. The curved inner bottom bulk carrier aims to reduce local stresses in the hold area by modifying the flat inner bottom and hopper tanks with an upside down arch plate. Non ballast seawater bulk carrier (NOBS) is further discussed, a design aiming to reduce the ballast seawater used by implementing an alternate hull shape. The Ecoship 2020 is a design listing a number of proposed innovations that can lead to more flexible, cost effective, energy efficient and environmental friendly structure. Mitsubishi air lubrication system (MALS) design is then discussed, a system aiming to reduce frictional resistance of the hull. Ecore ore carrier, a 250,000 DWT ore carrier is featured, implementing the use of one centre cargo hold, and alternative use of the wing tank areas.

Finally, the variable buoyancy ship is introduced, a bulk carrier adopting solutions aiming to eliminate the transportation of ballast water around the globe. This is achieved by having trunks that extend most of the length of the ship below the waterline, which are open when ship is at speed, leading to ballast water exchange.

7.2 Double side bulk carriers

IACS rules[1] define the double side skin on bulk carriers as a configuration (figure 7.1) where each ship side is constructed by the side shell and a longitudinal bulkhead connecting the double bottom and the deck. Hopper side tanks and topside tanks may, where fitted, be integral parts of the double side skin configuration. The minimum double side width, W_{ds}, is suggested not to be less than 1 m measured perpendicular to the side shell.

According to ABS[2], double sides on bulk carriers enhance protection of the primary structural members against cargo related corrosion and mechanical damage, as well as provide a barrier against extensive flooding due to low-impact side shell damage. The exposure of damage-prone transverse frames of conventional bulk carriers is eliminated on double side bulkers, whereas the creation of stiffer side structure eliminates the flexing or fatigue of conventional side frame structures. From an operational point of view, the damage per ton of cargo discharged can be six times lower than the conventional bulk carriers and the time required for cargo discharge is decreased, due to the smooth hold sides. The advantages and disadvantages of the double side skin (DSS) in comparison with the single side skin (SSS) bulk carriers have been summarized by ABS in table 7.1, which examines various aspects of the design, such as corrosion, flooding, mechanical damage, maintenance and steel weight.

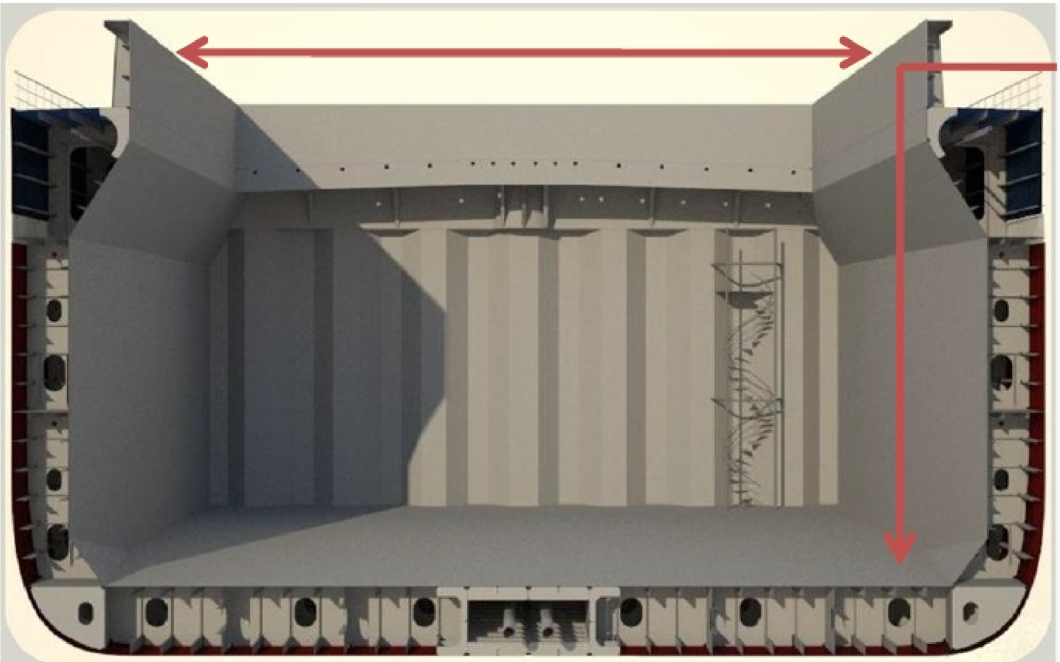

Figure 7.1

Modern[3] double side bulk carrier [3]

Table 7.1

Comparison of DSS bulk carriers to SSS bulkers [2]

	DSS Bulk Carriers	SSS Bulk Carriers
Perception	Pros: • Safer in structure • Flexible in operation	Pros: • Commercially competitive
	Cons: • Loss of grain capacity (for handymax vessels)	Cons: • Vulnerable to side structure failure • Effect of regulations yet to be evaluated
Corrosion	Pros: • High corrosion resistance, only when double hull is left void	Pros: • Easy blasting, re-coating and renewing of side structure if necessary
	Cons: • Extensive corrosion is envisioned if the hull space were used for ballast	Cons: • Hold frames are exposed to cargoes with high corrosion rates
Flooding resulting from damage to side structure	Pros: • Improved resistance against low energy collision resulting in holds flooding	Pros: • Hold structure and hull girder are strengthened against one hold flooding, and easily maintained
		Cons: • If side shell integrity were breached, one hold flooding may lead to a progressive flooding and loss of the ship
Mechanical Damage	Pros: • Hold side structure is protected from possible mechanical damage	Pros: • Hold frames are easily accessible for repairs
	Cons: • Repair work of DSS structure may require hot work in confined space – both outer/inner hull	Cons: • Hold frames are vulnerable to mechanical damage during unloading

Table 7.1 (continued)

Comparison of DSS bulk carriers to SSS bulkers [2]

	DSS Bulk Carriers	SSS Bulk Carriers
Inspection and Maintenance	Pros: • Access to DSS spaces will be facilitated using the hull structure – in the absence of ballast	Pros: • Hold structure and hull girder are strengthened against one hold flooding and easily maintained
	Cons: • Maintenance work could be more challenging due to DSS spaces being confined	Cons: • Special means of access is necessary (permanent means of access is not feasible)
Steel Weight	Pros: • Small difference as long as the strengthening for hold flooding is exempted in SOLAS XII	Pros: • Lighter than the same size for DSS BCs
	Cons: • Heavier than the same size of SSS BCs – such effect may become larger of smaller BC	

Spyrou et al.[4] present the modifications proposed on the midship configuration of a double side Panamax by Oshima shipyard to compensate for the reduced cargo hold space. Instead of modifying the principal hull dimensions, larger inner bottom area was considered and as a result, smaller topside and hopper tanks were used, as shown in figure 7.2.

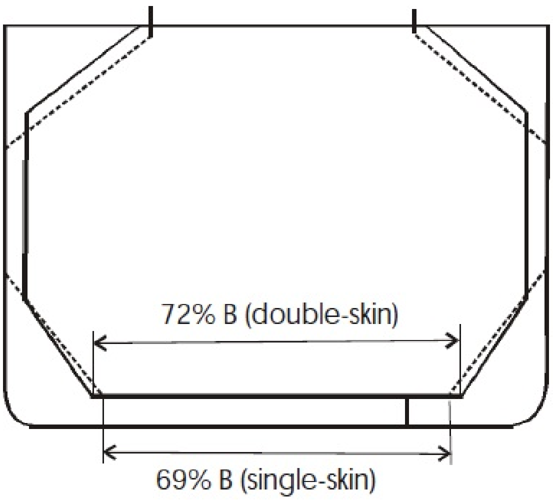

Figure 7.2

Layout of a typical cargo hold of a Panamax for DSS and SSS construction [4]

The typical structure adopted for the double sides of large bulk carrier ships consists of longitudinal stiffening with transverse webs (*alternative 1*, figure 7.3). Alternative designs were proposed and assessed by Fricke et al.[5] *Alternative 2* included arrangement of transverse webs in the sides at each frame location. In this way the width of the double side can be reduced because the space is not affected by longitudinals. The transverse web structure is heavier than that with longitudinals, because additional plates are arranged there and because increased plate thicknesses are required in the upper part of the side shell and longitudinal bulkhead to achieve satisfactory buckling strength.

Alternative 3 is a mixed design with longitudinals on the longitudinal bulkhead and transverse frames at the side shell as possible. Their support is provided by transverse webs and by stringers which are arranged at a distance of three frame spacings. The advantage compared with alternative 1 is more space in the double hull, which allows a reduced width to be realized. Contrary, the transverse webs require an increased thickness in the lower part, due to high shear forces and the necessary openings at this area.

Finally, *alternative 4* provides a curved shell for the inner skin. By forming the inner skin from an unstiffened curved shell, its increased buckling strength is utilized. The whole curved part is only stiffened by transverse webs and a few stringers. This design provides buckling strength to global hull girder stresses as well as to local pressure forces and bending of the transverses.

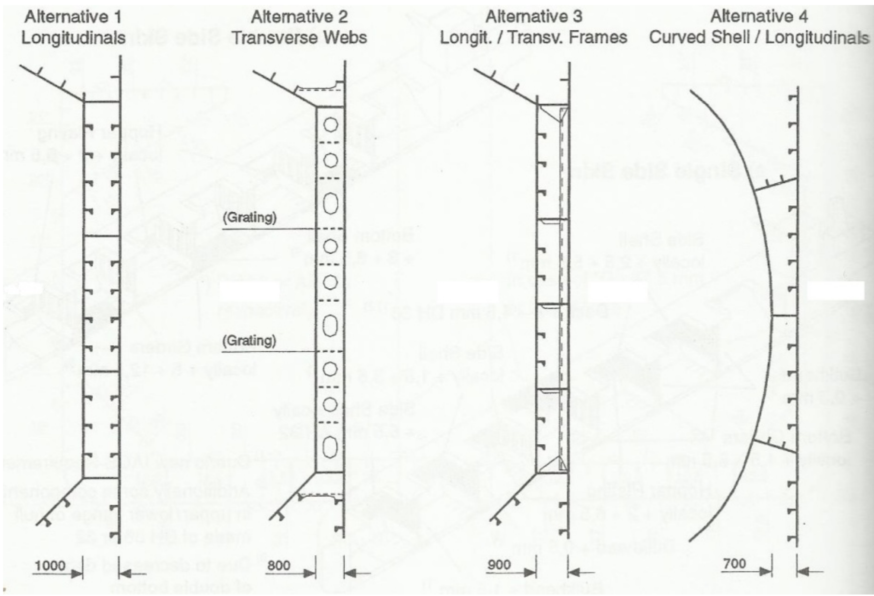

Figure 7.3

Double side design alternatives [5]

Hsu et al[6] studied the strength aspects of double side skin bulk carriers. They state that DSS in a bulk carrier is initially designed to support the shearing force, especially for bulk carrier with alternate hold loading. Additionally, compared to SSS design, DSS proved also very good transverse performance. Hsu evaluated the strength performance of two identical ships, the one with single side skin and the other provided with double side skin. Considering *shear strength*, the shear stress levels of shell were found quite different. For a 1000 KN vertical shear force, the maximum shear stress of the SSS ship is 11.75 N/mm^2, but only 6.78 N/mm^2 for the DSS ship. This means that the shell plate thickness of the SSS ship is dominated by shear stress, while the shear strength of outer shell for the DSS ship has safety margin of up to 40%. The study on the *transverse strength* revealed that the transverse webs in DSS design with the highest stress level are prevented by longitudinal bulkhead and inner bottom from exposing to a high corrosive cargo environment. However, the stress intensive areas of side frames in SSS design are totally exposed to the cargoes. Taking under consideration the operational aspects of the design, the total steel weight of DSS ship is increased 3.55% compared to SSS, whereas the available cargo hold volume is reduced 3.54%.

Soares et al[7] assessed the reliability levels of a conventional single hull bulk carrier compared to a double hull bulk carrier. It was found that double hull design has a higher level of reliability, whereas it maintains the same safety level for both sagging and hogging conditions. Contrary, failure has a higher probability in sagging than in hogging in single hull ships.

The analysis of the stress distribution near collapse concluded that the behavior of the two bulk carriers is very similar for both sagging and hogging bending moment. Under sagging collapse bending moment, most of the deck has already collapsed as well as the intersection of the side shell with the bottom of the wing tank. For hogging, the collapse bending moment is achieved with buckling of the bottom plating. However the inner bottom longitudinals and the bottom girders have already collapsed.

Additionally, the corrosion progress taken under consideration, sagging case was found always to be the dominant mode of failure for the single hull. However, for the double hull bulk carrier, the hogging case become the dominant one at 5,10 and 15 years. Figure 7.4 shows the time dependent probability of failure normalized by the initial value (for as built thickness), for the two designs assessed. BDH stands for the double hull bulk carrier and BSH is the single hull bulk carrier.

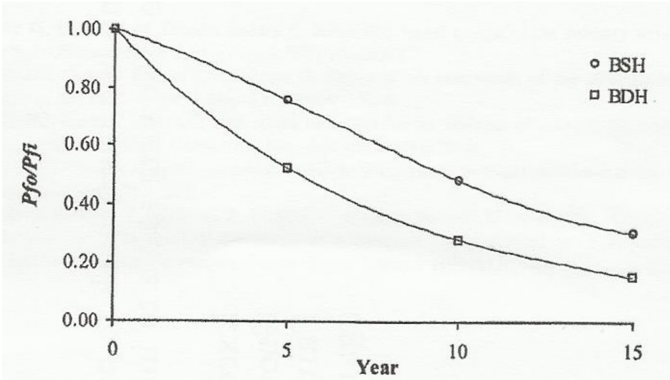

Figure 7.4

Time dependent probability of failure [7]

Ozguc et al[8] studied the collision resistance and residual strength of single side skin and double side skin bulk carriers subject to collision damage. The main results of the study considering various collision cares are given below:

- The ship structural design has very significant influence on the collision resistance. The collision energy absorption capability depends on the thickness of outer shell, inner shell, side stringers, transverse webs, width of the side ballast tank and width of lower and upper wing tanks.
- Energy absorption when rupture of the outer shell of DSS occurs is approximately 10% less than the energy absorbed by the outer shell of SSS. However, the maximum energy absorbed, i.e. the energy absorbed when the skin of cargo hold (inner shell for DSS, outer shell for SSS) ruptures, in 2.2 times more for DSS than for the SSS.
- For all cases, DSS has higher rupture energy than SSS.
- DSS bulk carriers have higher safety index than the SSS bulk carriers in hogging and in sagging conditions under similar collision damage scenarios, and this index value is greater in the hogging case compared to that in the sagging case.
- Ultimate sagging moments of resistance in intact and damaged hulls are considerably less than ultimate hogging moment.

7.3 Newcastlemax bulk carrier (202,500 DWT)

A bulk carrier designed[9] by CSBC Corporation, Taiwan, destined to transport mainly ore and coal products in specific routes between Australia and China. As discussed in the first chapter, the port of Newcastle, Australia, poses limitations (length less than 300m and width less than 50m) to the principal dimensions of such vessels, and therefore has led to the development of the Newcastlemaxes.

The midship section (figure 7.5) is a double hull form with vertical longitudinal bulkheads similar to container vessel's hull. While satisfying the owner's requirements, it provides a safety structure with ecological consideration. Nine cargo holds were proposed, considering economical hull weight design could provide enough longitudinal strength. The main feature of the ship's design lines is a shallow draft hull form with breadth-draft ratio (B/T) greater than 3. Considering the structural configuration, the proposed design has vertical longitudinal bulkheads that form the double side, thus no hopper side tanks and topside tanks are fitted. This feature excludes the structure from the definition of the Common Structural Rules, thus their implementation on the intended structure can be waived. As a result, steel weight of the structure can be reduced since the corrosive margin of thickness addition requested by CSR can be dismissed. Furthermore, saved steel weight means more cargoes could be transported during the operational life of the vessel.

For cargo hold arrangement, No 1,3,5,7&9 holds are reinforced for the carriage of ore. The double bottom/double side ballast tank arrangement without using one cargo hold as deep tank could provide enough draft for encountering the heavy weather comparing to other common design, while using one cargo hold as ballast tank could be risky of hold structure damage due to sloshing loads and could increase the cost and time in ballast water management and treatment.

The double side space was designed with longitudinal framing system and supported with open type transverse webs. Four horizontal stringers were arranged in the double side space to make the inspection and maintenance easier. To achieve compliance with the rules and consider sustainable longitudinal and double bottom strength, the height of double bottom was more than 2.5 meters. The longitudinal framing system was used for double bottom ballast tanks structure, whereas the pipe duct in double bottom was designed by transverse framing system. Transverse corrugated bulkheads with upper and lower stools were arranged, while the side transverse web spacing is twice as the double bottom structure, for supporting the longitudinal members.

Concluding, the following advantages of the special structural design can be listed:

- The hull has enough breadth to arrange clear passway when the hatch covers are opened. With reduced camber and coaming height, the operations become safer and maintenance costs are lower.
- The wider double hull design provides the adequate longitudinal strength and the capacity to bear the shear force in the alternative loading condition.
- The open hatch type design with the vertical longitudinal bulkheads provides clean hold shape and beneficial for faster cargo loading and discharging in comparison of the conventional design.

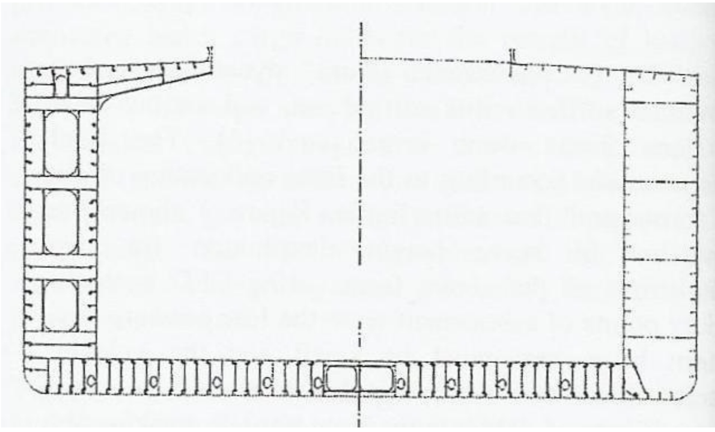

Figure 7.5

Midship section of 202,500 DWT double hull bulk carrier [9]

7.4 Hycon bulk carrier

Hybrid configuration[10] (Hycon) bulk carrier design provides double sides in the fore and aftermost hold of a single hull bulk carrier (figure 7.6). This design increases[11] efficiency of cargo unloading, as well as hold cleaning work. Additionally, it improves structural safety by eliminating hold frame corrosion, damage and reducing flooding risk drastically. Wide hatch covers provide easy access to cargo hold area.

Moreover, the weight of the extra steel used for the inner skin in the fore and aft holds is counterbalanced, since no extra steel is needed for the deck. Finally, protection has been added where the wave action is the most severe. The structural safety of the hybrid design brings structural stiffness by reducing flexing and fatigue from wave loads at the fore end of the side structure.

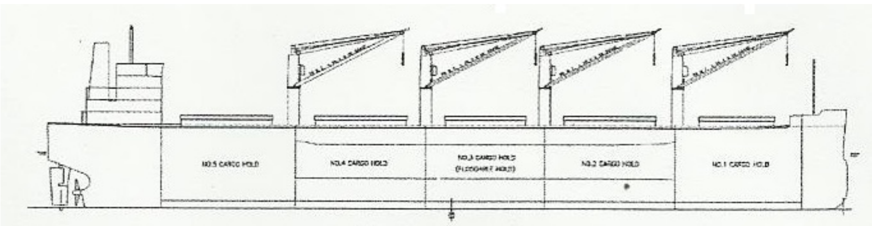

Figure 7.6

Handymax Hycon bulk carrier [11]

7.5 Optimum 2000 bulk carrier

Norwegian shipbroker O-J Libaek & Partners[12] produced a new design (figure 7.7), designated the Optimum 2000 Tri-Cargo Carrier (TCC). The invention[13] comprises transverse bulkheads and at least one longitudinal centreline bulkhead intersecting the transverse bulkheads of a bulk carrier. The transverse bulkheads and the longitudinal centreline bulkhead form longitudinal cargo holds. The longitudinal form of the cargo holds and hatches facilitate unloading by the grab of a gantry crane located above the hatch, and this is also advantageous with respect to strength and stiffness of the vessel.

The centreline bulkhead strengthens the deck, and enabled by this, the cargo holds are provided with large hatch openings and single piece hatch covers. The width of the hatch openings of the vessel according to the invention is essentially the same as the width or beam of the tank tops, thus creating an "open hatch" which improves the trimming of bulk cargoes significantly. It also gives the discharging equipment, such as grabs a better access to the holds during discharging of bulk cargoes and thereby reduces the risk of stevedore damages.

The division of the cargo holds into port and starboard holds by the centreline bulkhead strongly reduces the sloshing of the cargo, which gives the design according to the invention, a much better stability than known designs. Additionally, the centreline bulkhead increases the strength of the vessel hull, which is an added advantage when loading heavy gravity cargoes such as ore. The centerline bulkhead also provides a stiff support for the deck and the hatch coamings located along the centreline of the ship, which means that compared to OBO or bulk carrier designs according to prior art, which have only a girder or no support at all for the centreline coamings, the deflection and bending problems related to the coamings are significantly reduced. In addition the ship side being of the double hull type further increases the strength of the vessel.

The longitudinal walls of the upper wing tanks are preferably continuous in the longitudinal direction of the vessel. These walls thereby form longitudinal girders which contribute to the structural integrity of the vessel. A similar contribution have the longitudinal walls of the upper tanks of the centreline bulkhead being continuous. The girders formed by the continuity of the longitudinal tank walls should preferably at least extend through the central portion of the vessel, as this is the portion of the vessel which is most subjected to bending. Together with the longitudinal walls of the ballast tanks these girders provide torsional and bending stiffness and strength to the vessel, which from a constructional point of view is very important. The continuity of the longitudinal walls of the upper wing tanks and the upper tanks of the centreline bulkhead compensates for lack of continuity in the vessel deck due to the width of the hatches being almost identical to the width of the deck, and the continuity of the longitudinal walls of these tanks is therefore an important feature of the new bulk carrier design.

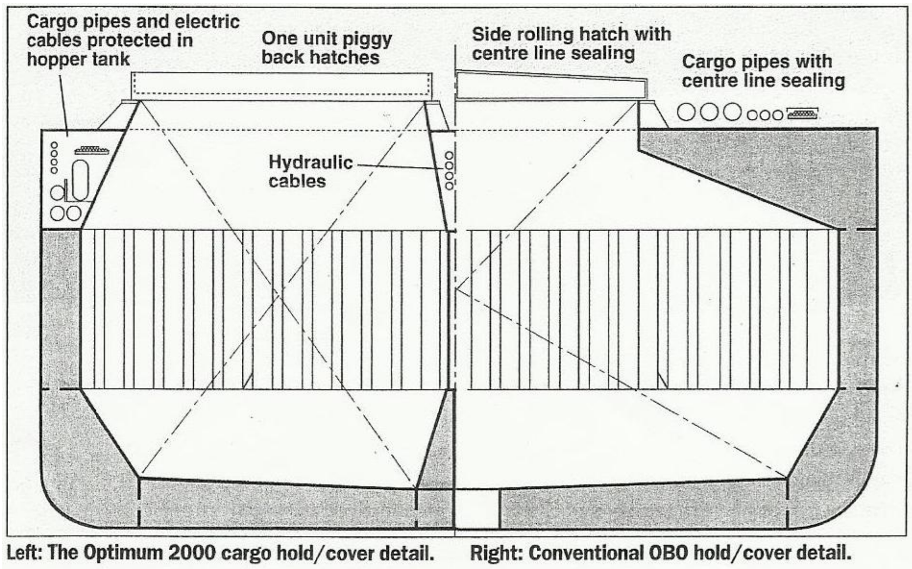

Figure 7.7

Optimum 2000 design versus conventional OBO hold configuration [12]

7.6 Curved inner bottom bulk carrier

An alternative transverse structural configuration in cargo hold region was proposed by Haggag[14]. The main target is to reduce as possible the local stresses on hull structure without any reduction of scantlings, mainly at side frame lower end bracket, giving it the ability to receive more dynamic loads without failure. The proposed modification (figure 7.8) is achieved by the replacement of inner bottom and hopper plates with an upside down arch plate, and this only in cargo holds region, in places where the load of cargo acts. The common points of this configuration with conventional bulk carriers are the intersection point of hopper plates with side shell and the double bottom height at ship's center line, thus the span of side frame remains the same. It is noted that the moment of inertia of midship section will slightly increase, thus its effects on longitudinal shear force and bending moment are considered negligible.

The arch equation was proposed as a standard parabola. Considering the ship's center line as the Y-axis in a Cartesian coordinate system, the equation is: $y = (0.6B)x^2$, where χ is the horizontal coordinate of any point on the arch in breadth direction and y is the vertical coordinate corresponding to χ measured from the most lower point of the arch at ship's center line. B is the breadth of the hold area.

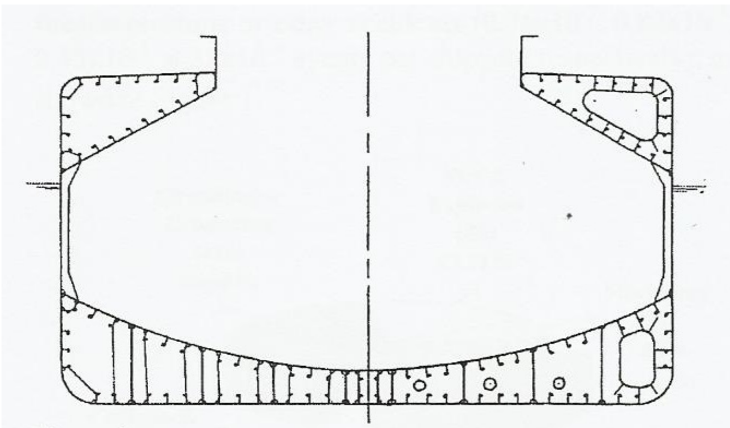

Figure 7.8

Curved inner bottom bulk carrier midship configuration [14]

The advantages of the curved inner bottom structure compared to conventional bulk carrier structures are:

- Static stresses on local hull structure reduced about 22% for the same loads and scantlings.
- Deflections are considerably reduced.

- For side frame lower end bracket, the stress range reduced about 40%, so the elementary fatigue life for each loading case increased about 140%.
- The cumulative damage for proposed model is considerably smaller than the one of conventional design.
- After considering 10% corrosion on side frame, the new design still has structural safety margin greater than of current design.

From an operational point of view, the following effects of the proposed structure are mentioned:

- The hold volume is reduced due to the lost volume under the arch (this volume is added to ballast water tanks in double bottom). The average volume reduction is about 5.7%.
- The net average increment in steel weight is 2.9%, due to the increase of floor depth, increase of bottom girders and their stiffeners, even though there is s reduction of inner bottom plating.
- Since the center of gravity of cargo moves upward, at about 8% of the old design, the total ship’s center of gravity rises about 4%. So, the metacentric height of ship is reduced, leading to the minimization of the risk for cargo shifting due to high angular velocity of rolling.
- At the discharging operation, the arch with its continuous gradual slop will allow cargo to slide down under gravity, reducing the time required for clean-up operation of the remaining cargo. This function substitutes and expands the use of hopper plate in the entire breadth of the hold.

7.7 NOBS bulk carrier

NOBS is the symbol and abbreviation of “Non Ballast Seawater”. NOBS bulk carrier has been under preliminary research by Guangzhou XED Ship design co. Ltd.[15] The preliminary research shows that, adopting upper U and lower V design for ship hull molded line, as illustrated in figure 7.9, can effectively avoid the technical requirements of IMO’s BWM (International Convention for the Control and Management of Ships' Ballast Water and Sediment) and PSPC (Performance Standard for Protective Coatings), and improve the resistance performance. Compared with traditional ships, NOBS has better performance in saving ship building, operation and repairing cost. These advantages can be added to the elimination of water pollution of alien sea caused by the ship carried seawater, since NOBS is “essentially environment protected”, the effect of environment protection is brought during the whole life circle of a ship. When it comes to the internal corrosion caused by the seawater to the ship itself, NOBS is “essentially safe” and this is also once and forever.

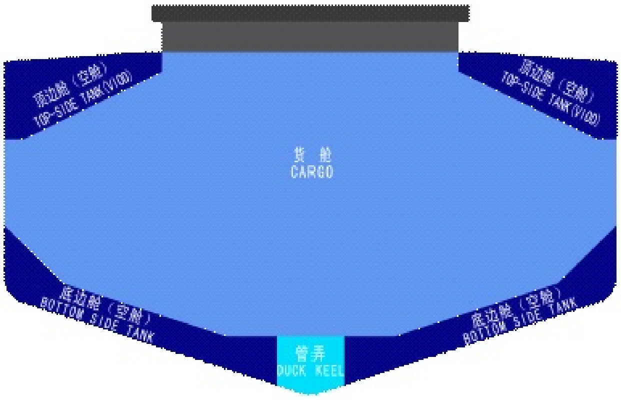

Figure 7.9

NOBS bulk cargo tank configuration [15]

The proposed configuration evolves the below mentioned advantages:

- The traditional U shaped flat bottom plating is modified to V shaped.
- Adopting V shape bottom plating, no need to equip the vessel with water ballast tank exists any more, and with reasonable arrangement of fuel, fresh water tank and optimization handling of molded lines. The lack of ballast water eliminates the costs of fitting ballast water processing units, minimizes the possibility of any water pollution on

the environment and the existence of residuals of ballast water and its influence on ship operating.

- Adopting V shape bottom plating for cargo oil tank, makes it much easier for stripping and furthermore for loading and unloading of bulk cargo (eg.grain,coal)
- The V shape bottom plating can reduce the wetted surface, therefore the resistance is reduced accordingly, thus an energy saving decision.

7.8 Ecoship 2020

Oshima shipbuilding[16], in collaboration with DNV[17], has proposed the major areas of innovation for the bulk carrier of the future, in order to achieve a flexible, cost effective, energy efficient and environmental friendly structure. Under this prism, an open hatch bulk carrier named Eco-ship 2020 was implemented, at first for specific pulp trade routes, with many port calls and trying to eliminate ballast voyages.

Among the main features of the concept, the ones affecting the structural design are:

- The ship is fully LNG fuelled, since LNG is the only fuel onboard.
- A minimum ballast design concept is adopted.
- The ship features composite hatch covers.
- Wide air lubricated twin skeg hull.

Since no oil is carried onboard, there is no risk of accidental oil pollution. The position selected for the two out of four LNG insulated pressurized tanks is beneath the aftmost hold (figure 7.10), leading to structural modifications in the vicinity of the double bottom at that place. The minimum ballast design concept is well depicted in the modification of the hold area, where the ballast hold is eliminated (figure 7.11), leading to 60% less ballast needs and the increment of cargo hold unloading efficiency. The hatch covers are manufactured with composite materials, thus a significant weight reduction (more than half of steel hatch cover weight) is achieved, less maintenance is required and they can easily be handed by deck cranes (no hydraulic systems are required). The implementation of an air lubrication system on the flat of bottom results to 3-7% energy saving for a Post Panamax type bulk carrier. This is achieved by the reduced fuel consumption due to the low total resistance of the hull in calm waters and on waves.

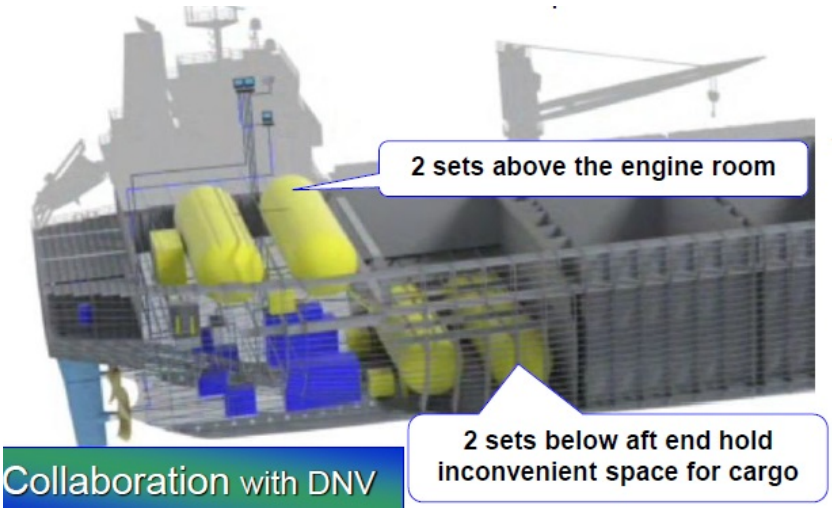

Figure 7.10

Eco-ship 2020 LNG fuel tanks positioning [17]

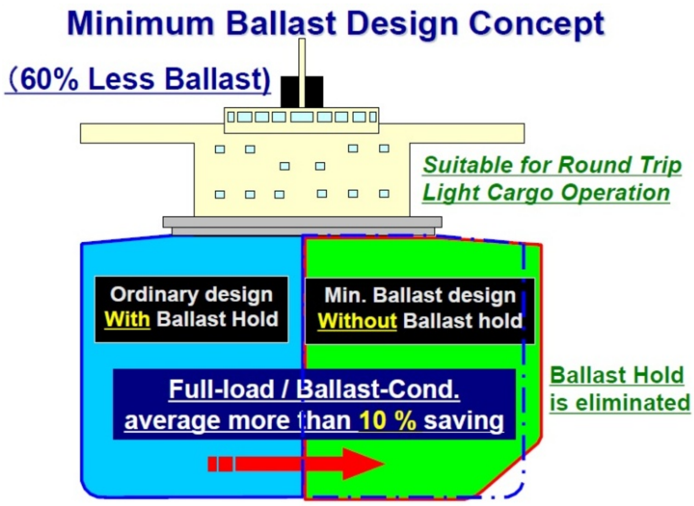

Figure 7.11

Midship profile modification for ECO-ship 2020 [16]

7.9 MALSADMMAX bulk carrier

Considering the abovementioned air lubrication system, Mitsubishi heavy industries has also implemented[18] the Mitsubishi Air Lubrication System (MALS) for the hull of three grain bulk carriers (figure 7.13) built for Archer Daniels Midland Company (ADM) of the United States. The system[19] reduces frictional resistance between the vessel hull and seawater by using air bubbles introduced at the vessel bottom by providing a large amount of air flow with high pressure (figure 7.12).

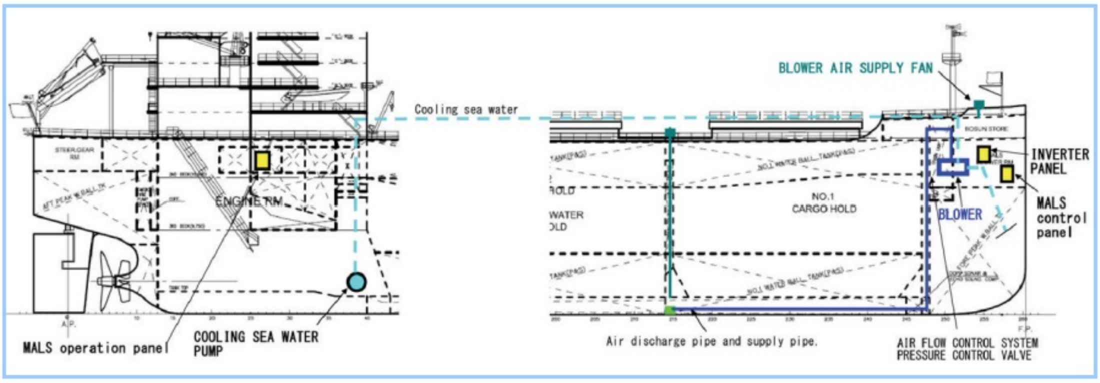

Figure 7.12

Outline arrangement of MALS [19]

Figure 7.13

MALSADMMAX air lubrication system [18]

7.10 Ecore ore carrier

Det Norske Veritas presented a futuristic alternative design considering the carriage of ore minerals by sea. Ecore[20] is a 250.000 DWT ore carrier destined for trade routes between Australia and China. Featuring relative overall low steel weight, the V-shaped hull form (figure 7.14) reflects the reduced need for ballast. Equipped with only one centre cargo hold, the cargo is evenly distributed from a single loading point. Finally, LNG tanks positioned in protected location inside the wing tanks (figure 7.15) offer no loss of cargo space and an alternative use of the wing tanks.

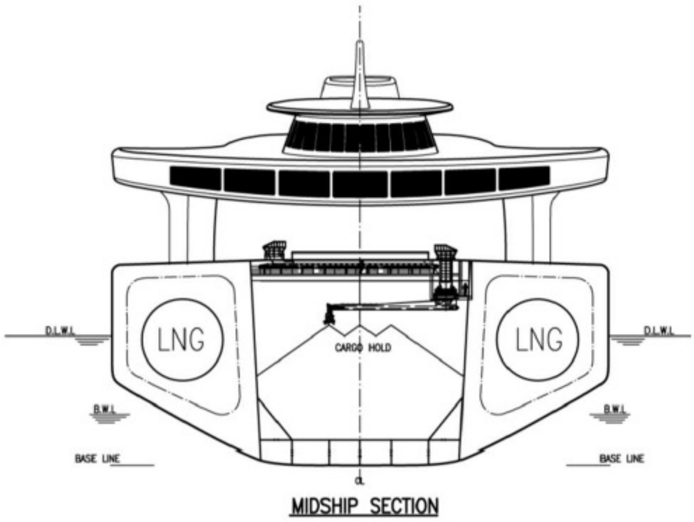

Figure 7.14

Ecore ore carrier midship section [20]

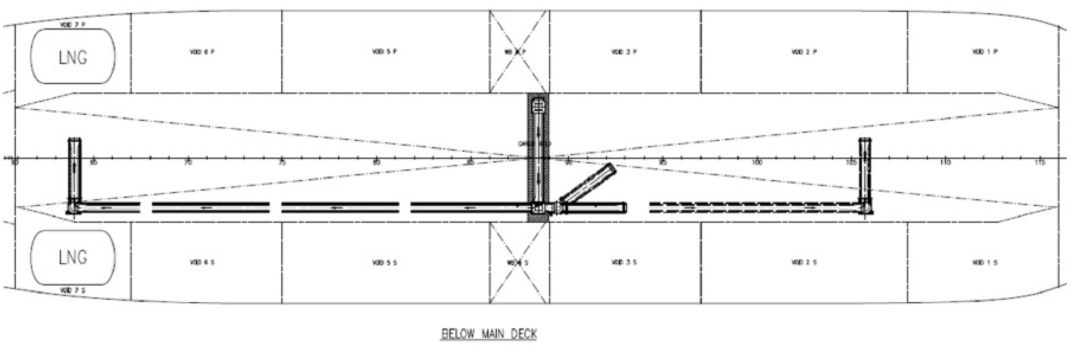

Figure 7.15

Ecore ore carrier LNG tanks relative positions [20]

7.11 Variable Buoyancy Ship

The Variable Buoyancy Ship[21] concept adopts solutions for the elimination of transporting ballast water around the globe. The development instead of adding ballast weight in ballast conditions uses reduced buoyancy to get the ship down to safe operating drafts. This is achieved by arranging the ship to have structural trunks of sufficient volume that extend most of the length of the ship below the ballast waterline and then opening these trunks to the sea in the no-cargo condition (figure 7.16). When the ship is at speed, the natural pressure difference between the bow and the stern of the ship induces a slow flow through these open trunks resulting in their always being filled with local seawater that is exchanged about once per hour. The trunks are connected to an inlet plenum at the bow and an outlet plenum at the stern. They are equipped with motor operated butterfly isolation valves at the bulkheads at the ends of the cargo region. When the vessel is ready to reload cargo these valves are closed and the ducts are pumped dry using conventional ballast pumps. This design essentially eliminates the transport of ballast water.

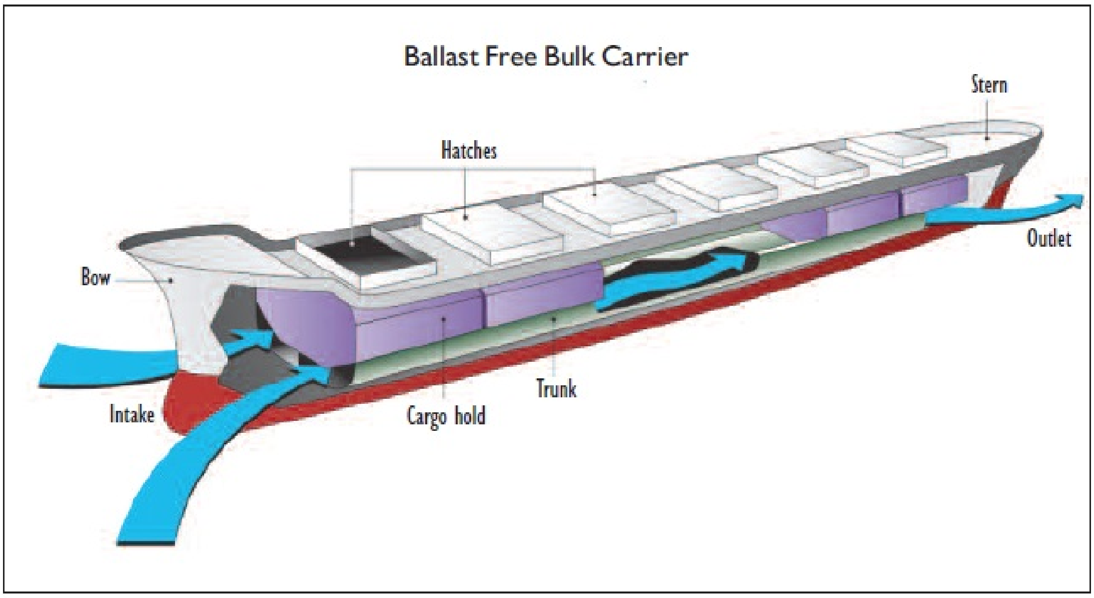

Figure 7.16

Variable Buoyancy Handy-Sized Bulk Carrier [21]

A midship section[22] of a Variable Buoyancy (or Ballast-Free) bulk carrier is shown at the right in figure 7.17, in comparison with a conventional bulk carrier. To provide full storm ballast volume below the ballast draft the inner bottom is raised from 1.6 m to 2.4 m. To maintain full grain capacity with the higher inner bottom, the hull depth is increased from 15 m to 16 m. There are three longitudinal trunks per side of the ship: two in the double bottom and one consisting of the hopper side region. The deeper double bottom will facilitate trunk cleaning with its full head room. To further aid the cleaning of the trunks, most of the floor plating is cutaway at the bottom plating so that the trunks could be more easily hosed clean below each cargo hold. The resulting hull steel weight increases from 5,553 t to 5,767 t (+3.85%) when designed to the ABS pre Common Structural Rules (CSR) Bulk Carrier rules (ABS 2002).

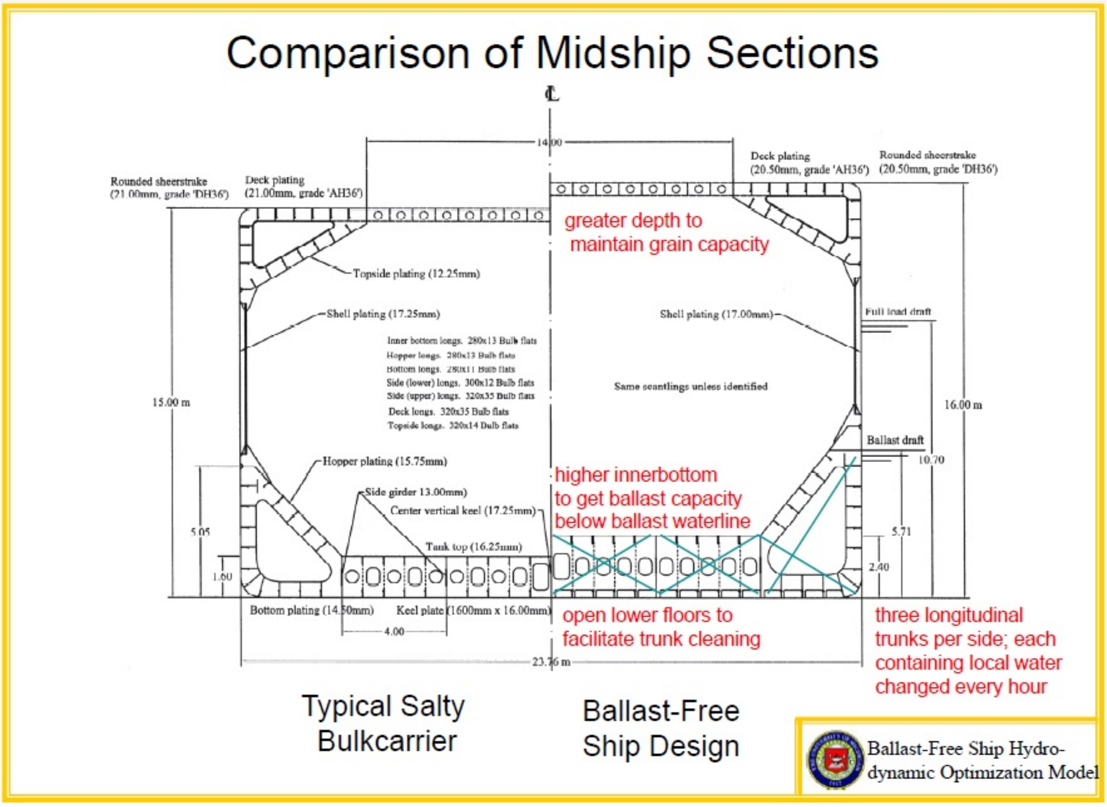

Figure 7.17

Midship Sections of Conventional and Variable Buoyancy Bulk Carriers [22]

[1] IACS Common Structural rules for Bulk Carriers and Oil Tankers (01 Jan 2014) pp 76
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8

8.1 Conclusions

8.1.1 Chapter 1

In the first chapter of this thesis an effort is being made to familiarize the reader on the concept of the bulk carrier design. At the beginning the term bulk carrier is defined as adopted by the IACS resolutions. Bulk carrier means a ship which is constructed generally with single deck, topside tanks and hopper side tanks in cargo spaces, and is intended primarily to carry dry cargo in bulk.

Subsequently, the classification of bulk carriers in groups is outlined, since this procedure produces many categories that may overlap each other. The main feature that helps us categorize the bulk carries is their deadweight capacity. On the other hand, the vessels can be grouped according to the commodity they are built to carry or furthermore by the means that are used to load or discharge their cargo.

According to the deadweight capacity, bulk carriers can be classified in the following categories. Various other minor categories are also listed, usually named after the seaway or the port that imposes the restrictions to some of the principal dimensions of the vessel.

- Mini bulkers
- Handy
- Handymax
- Supramax
- Panamax
- Capsize
- Very Large Bulk Carriers (VLBC)

According to the commodity carrier onboard, bulk carriers can be categorized as follows:

- Ore carrier

- Combination carrier (combo)
- Belt self unloader
- Bulk cement carrier
- Bulk In-Bags Out (BIBO)
- Woodchip carrier
- Open hatch bulk carrier (conbulker)

Finally, bulk carriers of length 150m or more are assigned the following service features by IACS:

- BC-A
- BC-B
- BC-C

Afterwards, the typical structural configuration of a bulk carrier hold is presented. Figure 8.1 presents[1] the nomenclature used for the structural components of a cargo hold. Generally, the plating compromising structural items such as the side shell, bottom shell, strength deck, transverse bulkheads, inner bottom and topside and hopper tank sloping plating provides local boundaries of the structure and carries static and dynamic pressure loads exerted by the cargo, ballast, bunkers and the sea. This plating is supported by secondary stiffening members such as frames or longitudinals. These secondary members transfer the loads to primary structural members such as the double bottom floors and girders or the transverse web frames in topside and hopper tanks.

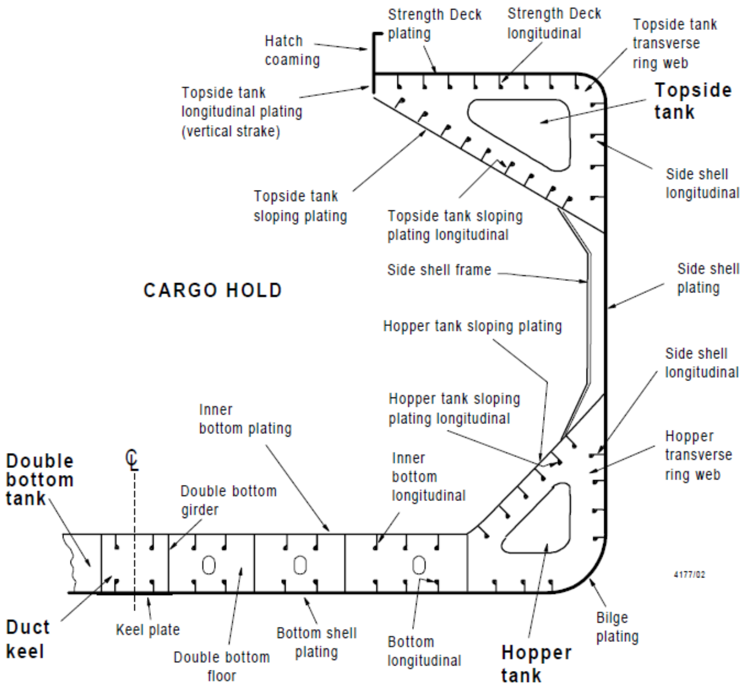

Figure 8.1

Nomenclature for typical transverse section in way of a cargo hold [1]

At the final part of the first chapter, the bulk carrier fleet is recorded, so that the reader understands the volume of the bulk carrier fleet, its age and the perspectives of the shipbuilding in this sector. In 2011, the overall bulk carrier fleet tonnage was estimated at 532 millions of DWT, with a remarkable 17% annual change (in comparison with 2010). At that time, the majority of the fleet DWT was on order status, with vessels up to 5 years of age being the second larger group. The size group with the larger number of vessels in service was 35.000-59.999 DWT, whereas China was the top bulk carrier building country. The average age of broken up bulk carriers in 2010 was 30.9 years.

8.1.2 Chapter 2

The second chapter of this thesis outlines the design principles followed in the structural design of a bulk carrier, starting from the structural design process, that is part of the design spiral. At this stage, a stepwise process determines the structural arrangements of the ship. Then the derivation of the hull scantlings is being made, followed by the assessment of the hull girder strength. Finally, the detail design of the components ends this part of the spiral.

The primary objective of the structural design of every ship is the development of a structure that will be able to withstand all the forces acting on it, thus to avoid any structural failure on the vessel. The most important of these forces are the bending moments and shear forces that result from the waves encountered at sea and the loading applied by the cargo carried. As the structure must continue to meet these forces throughout the ship's life, the scantlings must include allowances for the corrosion and wear which can be expected.

After presenting the three loading patterns usually encountered on bulk carrier operations, namely the homogeneous, alternate hold, and block loading, their effect of bending moments and shear forces is depicted, in comparison with the available allowable limits. It is evident that the two latter conditions push the vessel to the limits considering shear forces, whereas for the bending moment distribution the situation leading to maximum values is the alternate hold pattern.

Subsequently, the net scantling approach is presented. That part describes a process for the determination of the minimum hull scantlings that should be maintained throughout the ship's life to satisfy structural strength requirements. It clearly separates the net thickness from the thickness added for corrosion that is likely to occur during the ship in operation phase. The main concept of the procedure is the application of a general, average global hull girder and primary support member wastage (wastage allowance) such that the overall strength of these large structural members is maintained. The wastage allowance is the value of thickness diminution due to corrosion expected during the service life of the ship obtained by statistical analysis based on the thickness measurement data of ships and the steel renewal criteria which ensure that the net thickness is kept throughout the service life of the ship. Additionally, the value of the corrosion addition is obtained from wastage allowance by adding to the thickness diminution predicted till the next thickness measurement. The above mentioned principle is illustrated[2] in figure 8.2

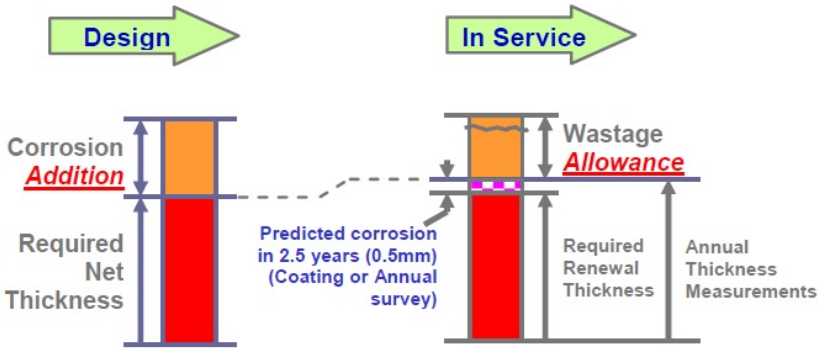

Figure 8.2

Net thickness principle [2]

Furthermore, the principal structural components of areas of major significance in bulk carrier structural design are outlined. Double bottom structure is longitudinally framed, with the height of double bottom having a significant influence on the overall hold design. Additionally, side structure on single skin bulk carriers is transversely framed, a feature that enables many problems during the loading and unloading operations, as discussed in following chapters. Finally, the bulkhead structure is presented, featuring the advantages of corrugation on strength issues. They may contain any flooding in the event of a compartment on one side of the bulkhead being bilged, whereas they serve as a hull strength member not only carrying some of the ship's vertical loading but also resisting any tendency for transverse deformation of the ship.

8.1.3 Chapter 3

The third chapter of this thesis outlines the determination of loads affecting the hull structure. The first loads assessed are the ones that are present in the still water conditions, meaning in conditions where the ship floats in calm water. The main components of this category are still water bending moments and shear forces. Those static loads should be supreimposed to the wave induced loads, in order to assess the total forces that result in negligible dynamic stress amplification of the structure. IACS formulas for the calculation of those loads are presented, whereas typical distributions of allowable and attained forces in specific cases are illustrated.

In order to generate the dynamic load cases for structural assessment (strength and fatigue), a variety of Equivalent Design Waves (EDW) is used. The term EDW refers to regular waves that generate response values equivalent to the long term response values of the load components considered being predominant to the structural members. Following the determination of the EDW's, for each situation, the ship motions responses are described in tables by the society and the global loads corresponding to each dynamic load case to be considered are mentioned for the strength assessment. Finally, the reference value of the global loads and the inertia load components (hull girder loads, longitudinal/transverse/vertical accelerations) is to be multiplied by a relevant Load Combination Factor (LCF), in order to achieve the desirable assessment.

Zhu et al[3] evaluated the design loads on primary structural members of bulk carriers and came up with the sea states having the maximum effect on structural strength. The dominant sea states were the:

- Vertical bending moment at head sea (L-180)
- Vertical bending moment at following sea (L-0)
- Roll (R)
- Hydrodynamic pressure at waterline (P)

The external loads that act as local transverse loads for the hull plating and the supporting structure consist of two components, one static and one dynamic. The static pressure is the hydrostatic pressure P_S that is related to the vertical distance between the free surface and the load point. Considering hydrodynamic loads, for each load case described by the classification society, the position of the waterline in comparison with the still wave situation is different, thus the distribution of the pressure is significantly modified. Thus, formulas providing the value of P_W are listed in the rules, whereas the use of various coefficients and other parameters is required, such as girth distribution coefficients, coefficients for non-linear effects and ballast water exchange scenarios, in order to clearly represent the load values.

Having determined the external pressures acting on the hull, the identification of the internal pressures that affect the structure is the next complex step. The major load component of this category corresponds to the forces due to the bulk cargo and the liquids loaded onboard, considered for static and dynamic scenarios. Special consideration is given on the cargo profile when loaded on hold, because this determines the forces acting internally on hull. Especially for heavy cargoes that partially fill the cargo hold, the effective upper surface of cargo is defined, in order to assess the internal pressures.

Finally, the hold mass curves are presented. They denote the maximum allowable and the minimum required mass of cargo in each cargo hold as a function of draught in seagoing condition as well as during loading and unloading in harbor. Additionally, they may provide the maximum allowable and minimum required mass of cargo and double bottom contents of any two adjacent holds as a function of mean draught in way of these holds.

8.1.4 Chapter 4

In the fourth chapter of this thesis, the calculations for the structural components in the midship section area of a bulk carrier are presented. This procedure is part of the preliminary design of a bulk carrier as conducted by students in the ship design laboratory of the National Technical University of Athens, as part of the preliminary design lesson of the department of Naval Architecture and Marine Engineering. The following procedure is a free translation from Greek of the tenth part of the work done by Antonis Dellis[4], whose kind permission was requested to reproduce the material in the present thesis, and was granted.

8.1.5 Chapter 5

The fifth chapter of this thesis copes with the overall design of the hold area in a bulk carrier, seen from an operational point of view. From the number of holds being required considering the size of the vessel and the density of cargo to be carried, to the definition of hold length and the transverse bulkheads to be fitted. Additionally, the purpose of topside tanks and hopper tanks presence is discussed, whereas the effects that ballast water management has on strength of the structure is also mentioned. Furthermore, the double bottom arrangement is presented, focusing on the effects of double bottom height on structural behavior of the ship.

When determining the general arrangement, the first approach is based on limited information that might include:

- Required volume of cargo spaces, based on type and amount of cargo.
- Method of stowing cargo and cargo handling system.
- Required volume of tankage, mainly fuel and ballast for a specific range.
- Required standard of subdivision and limitation of main transverse bulkhead spacing.

In general, the density of the anticipated cargo controls the location of the inner bottom. For dense cargoes, it is advised that the hold should be narrow at the top, in order to prevent problems of cargo shifting. Furthermore, to prevent violent motions that would result from excessive metacentric height, it is desirable that the centre of gravity of the cargo should be relatively high. Those considerations lead to a configuration with high inner bottom and large wing tanks.

Contrary, low density bulk carrier needs much more volume to carry the cargo, that results in a lower inner bottom. This leads to the configuration that includes the high slopping inner bottom at the bilge and the topside tanks. Variations may enable the omission of topside tanks or the presence of inner side shell that makes easier the cleanup of cargo and provides extra space for water ballast.

IACS suggests that the *bulkheads* in the cargo hold region are to be spaced at uniform intervals as far as practicable. This, apart from the standard structural blocks to be considered on ship construction, enables a constant cargo hold length that leads to standard hatch cover sizes. Contrary, Taggart[5] mentions that for shallow draft bulk carriers that carry heavy cargoes, the arrangement enables alternatively long and short holds in order to achieve an acceptable metacentric height. This distribution creates very high vertical shear forces near the bulkheads, that may lead to the need for increases in the shell plate thickness.

According to Paik et al[6], transverse bulkheads in dry cargo holds are usually designed to withstand three load components:

- Lateral pressure due to dry cargo and/or flooding water
- Carry-over bending moment, resulting from overall double bottom bending, which is important in alternate hold loading situations
- In-plane axial force due to the net double bottom pressure.

The two first components are mainly related to cargo mass, whereas the latter is a function of cargo mass and draft. Frystock et al[7], note that the vertical bending moment acting on the transverse bulkhead is a function of torsional rigidity of the upper and lower stools, and the stiffness of the double bottom structures, with additional bending moment components transmitted from the double bottom to the bulkhead.

The bulk carrier configuration with inclined *upper and lower wing tanks*, according to Taggart[8] allows:

- A small area for clean up under the square of the hatch once most of the cargo has been discharged, as the remaining cargo slides down to canted sides. This also allows discharging gear to reach all areas, as the tank top breadth is roughly equal to the hatch opening breadth.
- Stowage free of shifting boards or other temporary devices to prevent the load from shifting to one side. Thus the upper wing tank configuration presents minimum free surface when the bulk cargo is stowed to the top of the hold. Furthermore, Isbester[9] mentions that the upper hopper thanks occupy space into which bulk cargo would never flow, a valuable feature for grain trades.

Additionally, the methods used in ballast water exchange (sequential, flow-through and dilution method) are described, whereas the potential hazards each one can have on structural components is generally outlined.

Double bottom structure results in a strong bottom that is well adapted to withstand the upward pressure of the sea as well as the longitudinal hull girder bending stresses, especially the compression resulting from hogging stresses. It provides tankage for liquids such as fuel oil, fresh water and ballast, thus using space that is unsuitable for other purposes. It results in a structure which can withstand a considerable amount of bottom damage caused by grounding without flooding of the holds or machinery spaces, provided the inner bottom remains intact. Additionally, a smooth inner hull free of stiffening structure is produced, which provides easier cleaning accessibility.

The effect of double bottom height on the structural behavior of bulk carriers was assessed by Contraros et al.[10] They found that the shear stress decreases considerably as the double bottom height increases. This is due to the additional shear area available by the corresponding increased height of girders and floors. They also mention that the dominant loading condition for the double bottom grillage to produce the maximum stress values, is the oblique sea conditions.

Considering the tank arrangements, various studies have been made to assess the influence of the arrangement and location of bunker tanks to the oil outflow from collision and grounding. IMO proposed a probabilistic based procedure for assessing oil outflow performance, by using the probability of zero outflow P_O, that represents the likelihood that no oil will be released into the environment, given a collision or grounding casualty which breaches the outer hull. Additionally, the *mean* outflow parameter O_M is the nondimensionalized mean or expected outflow.

Concluding, the findings of the evaluation conducted by Michel et al[11] related to outflow for bulk carriers arrangements are :

- Tanks located in the engine room, confined to a short length of the ship reduce the probability of penetration in collisions. Breaching the tanks in a grounding scenario is very unlikely, since they are located aft and above the inner bottom.
- Forward deep tanks are susceptible to damage from both collisions and groundings. When double hull protection is arranged outboard of the bunker tanks the mean outflow is significantly reduced.
- Double bottom configurations have the poorest outflow performance. Large center double bottom tanks have a high probability of damage and because of their size, may spill more oil than small wing tanks located at the bottom of the hold.

Hatch covers are formed by several steel panels which rest horizontally across the hatchway, sealing the hatch opening. Even though these panels should be stiff, in order to maintain weathertightness at sea, the steel structure of a hatch cover, as well as the bearing pad and sealing arrangements must adapt to the varying shape of the coaming top while the hull is working and flexing at sea. The optimal stiffness of the steel structure of a hatch cover panel is a compromise between the above issues.

The large size of the hatches reduces the torsional stiffness of the hull and causes twisting and diagonal changes in the hatchway, as well as warping of the deck plane in rough seas. The longitudinal bending of the hull or hogging/ sagging causes considerable changes in the hatch length. The third major type of flexible deformation is bending of the sides inwards and outwards. This not only occurs at sea but also in port when the draught changes due to variations in loading.

The final part of this chapter focuses on the *diversity of cargoes* that a bulk carrier is set to carry, and the various aspects of structural design that each of them affects. Ore cargoes loading rates could influence the strength of the bulk carrier, whereas liquefaction phenomena could become a cause of bulk carrier loss. Additionally, carriage of certain types of ore cargoes, under specific circumstances, could result in spontaneous combustion of the cargo. Coal cargoes, if mixed with water onboard, are notable for their corrosivity. The main problem associated with grain carriage is its tendency to shift when the ship rolls, leading to loss of stability. Moreover, steel cargoes may lead to tanktop area exceeding the maximum permissible loads assigned by the classification society. Finally, hazards associated with timber cargoes are identified and measures for safe carriage of such cargoes are listed.

8.1.6 Chapter 6

The sixth chapter of this study copes with the strength analysis of the hull structure. At first, the Finite Element Method analysis is presented, in order to assess the strength of longitudinal hull girder structural members, primary supporting structural members and bulkheads. Additionally, using this method, detailed stress levels in local structural details can be obtained, whereas the fatigue capacity of the structural details can be determined. The typical process of structural analysis using the finite element method is discussed, while the areas of concern in bulk carrier structures are displayed.

The finite element analysis[12] (figure 8.3) consists of three parts: (a) Cargo hold analysis to assess the strength of longitudinal hull girder structural members, primary supporting structural members and bulkheads. (b) Fine mesh analysis to assess detailed stress levels in local structural details, and (c) Very fine mesh analysis to assess the fatigue capacity of the structural details.

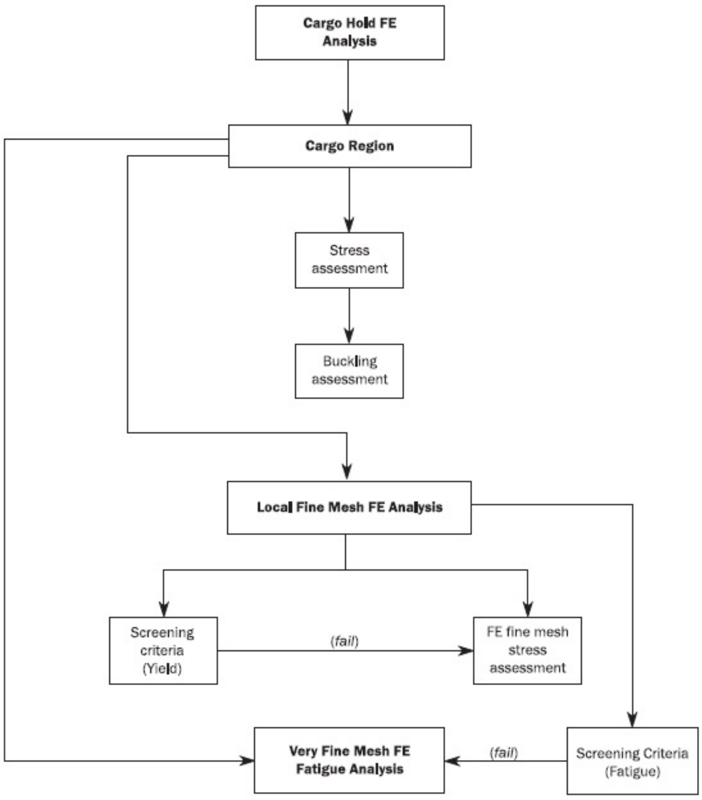

Figure 8.3

Flow diagram of finite element analysis [11]

Lehmann et al[13] mention that the scope of strength analysis in bulk carrier design should cover the following aspects:

- Global hull girder strength with particular view to bending and shear stresses in the hull girder.
- Strength of the double bottom grillage, particularly in case of heavy cargo and/or empty holds, considering supporting effects by the lower wing tanks and/or bulkhead stools.
- Strength of the bulkheads, taking into account interaction effects especially with the bulkhead stools and double bottom.
- Local strength of structural details considering stress concentrations and fatigue. Particular attention has to be paid to knuckles in the upper and lower wing tanks, connections between the stools and the bulkhead plating and/or inner bottom, end connections of side frames, hatch corners, terminations of coamings and transitions at the ends of the hold area.

Subsequently, the procedure for direct strength analysis is explained. The yielding strength check is discussed, while buckling and ultimate hull girder strength assessment is further analyzed. Prone to buckling areas of bulk carriers are presented, while the findings of an ultimate hull girder strength assessment are represented.

Finally, the assessment of the fatigue life of the various structural members subject to fatigue failure is presented. The types of stresses that are considered for this type of assessment are discussed, while the selection of the correct S-N curve is mentioned. Last but not least, an example of a fatigue performance analysis of bulk carriers side frame structure is noted, whereas the findings of this study are featured.

The fatigue assessment is required to verify that the fatigue life of critical structural details is adequate. A simplified fatigue requirement is applied to details such as end connections of longitudinal stiffeners using stress concentration factors (SCF) to account the actual detail geometry. Areas to be assessed for fatigue on bulk carriers are shown in figure. An illustration[14] of the abovementioned areas is shown in figure 8.4

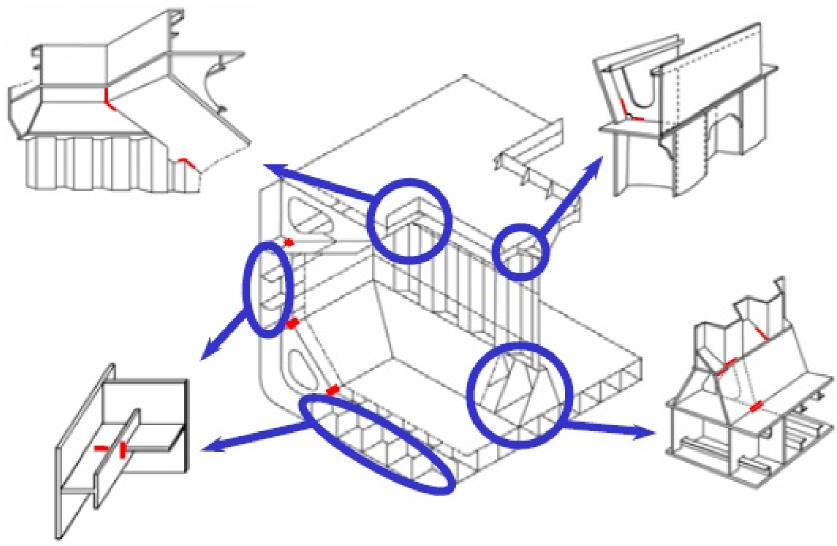

Figure 8.4

Bulk carrier details to be checked in fatigue [13]

8.1.7 Chapter 7

The seventh chapter of this thesis outlines the main alternative designs that have been implemented last decades in bulk carrier structural design, whereas the major areas of concern for the design of the future are also listed.

At first, the implementation of double side skin configuration is discussed. The benefits arising from this introduction are listed and a comparison with the conventional single side design is made. Additionally, some alternative designs proposed for the side structure area are also discussed. Strength aspects such as collision resistance and the residual strength of the structure are mentioned, whereas the reliability levels of the proposed structure in comparison with the single side structure are also described.

Double sides on bulk carriers enhance protection of the primary structural members against cargo related corrosion and mechanical damage, as well as provide a barrier against extensive flooding due to low-impact side shell damage. The exposure of damage-prone transverse frames of conventional bulk carriers is eliminated on double side bulkers, whereas the creation of stiffer side structure eliminates the flexing or fatigue of conventional side frame structures.

The advantages and disadvantages of the double side skin (DSS) in comparison with the single side skin (SSS) bulk carriers have been summarized by ABS[15] in table 8.1, which examines various aspects of the design, such as corrosion, flooding, mechanical damage, maintenance and steel weight.

Table 8.1

Comparison of DSS bulk carriers to SSS bulkers [14]

	DSS Bulk Carriers	SSS Bulk Carriers
Perception	Pros: • Safer in structure • Flexible in operation	Pros: • Commercially competitive
	Cons: • Loss of grain capacity (for handymax vessels)	Cons: • Vulnerable to side structure failure • Effect of regulations yet to be evaluated
Corrosion	Pros: • High corrosion resistance, only when double hull is left void	Pros: • Easy blasting, re-coating and renewing of side structure if necessary
	Cons: • Extensive corrosion is envisioned if the hull space were used for ballast	Cons: • Hold frames are exposed to cargoes with high corrosion rates
Flooding resulting from damage to side structure	Pros: • Improved resistance against low energy collision resulting in holds flooding	Pros: • Hold structure and hull girder are strengthened against one hold flooding, and easily maintained
		Cons: • If side shell integrity were breached, one hold flooding may lead to a progressive flooding and loss of the ship
Mechanical Damage	Pros: • Hold side structure is protected from possible mechanical damage	Pros: • Hold frames are easily accessible for repairs
	Cons: • Repair work of DSS structure may require hot work in confined space – both outer/inner hull	Cons: • Hold frames are vulnerable to mechanical damage during unloading

Table 8.1 (continued)

Comparison of DSS bulk carriers to SSS bulkers [14]

	DSS Bulk Carriers	SSS Bulk Carriers
Inspection and Maintenance	Pros: • Access to DSS spaces will be facilitated using the hull structure – in the absence of ballast	Pros: • Hold structure and hull girder are strengthened against one hold flooding and easily maintained
	Cons: • Maintenance work could be more challenging due to DSS spaces being confined	Cons: • Special means of access is necessary (permanent means of access is not feasible)
Steel Weight	Pros: • Small difference as long as the strengthening for hold flooding is exempted in SOLAS XII	Pros: • Lighter than the same size for DSS BCs
	Cons: • Heavier than the same size of SSS BCs – such effect may become larger of smaller BC	

Subsequently, the general characteristics of a Newcastlemax ore carrier (202,500 DWT) are presented, mainly by listing the structural arrangement of such a vessel and its advantages compared to a conventional bulk carrier. Moreover, a hybrid configuration (Hycon) bulk carrier is presented, demonstrating double sides in the fore and aftmost holds, whereas the other holds remain single sided. Furthermore, the Optimum 2000 is listed, a bulk carrier providing each cargo hold with a longitudinal bulkhead. This leads to advanced strength and stiffness of the structure.

Alternative designs are then presented. The curved inner bottom bulk carrier aims to reduce local stresses in the hold area by modifying the flat inner bottom and hopper tanks with an upside down arch plate. Non ballast seawater bulk carrier (NOBS) is further discussed, a design aiming to reduce the ballast seawater used by implementing an alternate hull shape. The Ecoship 2020 is a design listing a number of proposed innovations that can lead to more flexible, cost effective, energy efficient and environmental friendly structure. Mitsubishi air lubrication system (MALS) design is then discussed, a system aiming to reduce frictional resistance of the hull. Ecore ore carrier, a 250,000 DWT ore carrier is featured, implementing the use of one centre cargo hold, and alternative use of the wing tank areas.

Finally, the variable buoyancy ship is introduced, a bulk carrier adopting solutions aiming to eliminate the transportation of ballast water around the globe. This is achieved by having trunks that extend most of the length of the ship below the waterline, which are open when ship is at speed, leading to ballast water exchange.

8.2 Further research

There are so many aspects on bulk carrier structural design thus any proposal at this point could only provide the general outline on research on areas of concern.

- In a similar way to bulk carrier design, structural design of other types of ships could be recorded, and a comparison including the different approaches or the similarities could finally be conducted. The supervisor has already assigned such thesis projects for tankers and LNG carriers, as far as this is to the author's knowledge.
- Evaluation of actual corrosion rates with direct measurements onboard in comparison with the ones proposed by the net scantling approach.
- Contribution of hopper tank sloping plate angle and or topside tank sloping plate angle on various load conditions, on the overall strength of the bulk carrier.
- Use of FEM for assessing specific structural members and proposed alternative designs (eg curved inner bottom or various side skin configurations)
- Assignment of bending moments according to the rules in comparison to the actual bending moments a bulk carrier is to encounter during its lifetime, thus differences arising from using wave data from other than North Atlantic ocean areas.
- Provided detailed bibliography and computational methods could be used, more specific items could be assessed. For example the existence of cranes on deck could be assessed, the structural issues governing their fitting, with respect to the underdeck structure that has to withstand the overall loads.
- Other areas of concern could be the use of composite materials on structural design, assessing the weight reduction and the structural characteristics of the proposed structures.
- Assessing the strength of various structural members (eg corrugated transverse bulkheads or hatch covers), on specific cargo loads (for example buckling strength of hatch covers in conjunction with the logs loaded on deck or the existence of sloshing loads in nickel ore bulk carriers)
- The additional measures on structural design for bulk carriers destined for polar navigation since this is an area expected to notably evolve in the years to follow (North Sea Route in the Arctic sea)

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