IACS NEW COMMON RULES FOR TANKERS: 
IMPACT ON STRUCTURAL DESIGN AND FATIGUE STRENGTH EVALUATION

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ABSTRACT
The Common Structural Rules (CSR) for Double Hull Oil Tankers establish new criteria that will result in more robust, safer ships. Scantling requirements have been increased to such an extent that it is not probable that an existing ship designed to comply with the previous criteria could meet the new standards and it is highly unlikely that a modification of the structure of an existing (ABS) approved vessel to bring it into compliance with the new Rules could be considered economically feasible.
The purpose of this study is to present the new Rules and evaluate their impact on the structural design. With this object in view a test case has been developed, focused on the midship cargo tank region of an existing vessel.
In this framework the fatigue analysis of critical details has been tackled. Initially, the theoretical basis on which the CSR formulations are founded has been investigated. In particular, a critical analysis was conducted for the fatigue strength design methods the new Rules require for this type of verification.

1. INTRODUCTION
With the emergence of advanced structural and hydrodynamic computational methods, designers have been able to meet classification Rule requirements while optimizing scantlings based on accepted alternative direct calculation methods. Many ship-owners expressed concern that this optimization process, based on these alternative methods, was leading to ships that, while fully conforming to class Rules, were less robust than in the past.
The new Common Structural Rules issued by The International Association of Classification Societies (IACS) have used these same computational methods to establish new criteria, applied in a consistent manner, that not only will result in more robust, safer ships but also eliminate the possibility of shipbuilders to use scantlings and steel weight as a competitive element when selecting a Classification Society to approve a new design.
The mandatory set of requirements for strength assessment involves a greater amount of calculations which lead to more robust ships: impact not only on ship structure strength, but also on steel weight is not negligible.

2. IACS NEW COMMON STRUCTURAL RULES
In the past, Classification Societies had to face approaches that they were competing for minimised scantling dimensions and steel weight to the detriment of ships’ safety. As a consequence, Classification Societies started to develop rules for the structural design of bulk carriers and double hull tankers several years ago which after adoption by all member Societies of the IACS – the principal forum for cooperation between Classification Societies – will be mandatory.
Informal discussion has been taking place between American Bureau of Shipping (ABS), Det Norske Veritas (DNV) and Lloyd’s Register (LR) for over twenty years. The central theme was that Classification Societies should not compete on the minimum safety standards.
A number of joint task groups were established to work on the development of common basic design criteria, which included consideration of loads and structural analysis. The progress was limited and it became clear that to achieve the goal of eliminating competition between the Classification Societies on structural standards, it was necessary to develop and implement a single set of common Rules for the design and construction of the hull structure.

Early in 2002 ABS, DNV and LR agreed to initiate a major project—the Joint Tanker Project, JTP—to jointly develop a single set of Classification Rules for the hull structure of oil tankers.

At this point of time there was discussion within the maritime community on the need for increased robustness in ship design and construction, with the aim of reducing the problems for ship owners during the service life. It was, therefore, decided that in developing a new common set of Rules for oil tankers there would be a deliberate effort to enhance robustness by design.

Since that decision, the three Classification Societies have been working to combine their experience in order to produce a common set of Rules for oil tankers. As a novelty during the development of the Rules, industry was given the opportunity to comment on the meanwhile published draft Rules. Extensive discussions with shipbuilding and shipping industries about the consequences and the background of the Rules took place and had influenced on the final draft of the Rules [1].

On 1 January 2006 the Rules took effect as the applicable structural Rules for each of the members of IACS. The new standards apply to all tankers of 150 metres in length and above for which the contract for construction is signed between the prospective owner and the shipbuilder on or after 1 April 2006.

It is worth to point out that the new Rules have been developed not to eliminate all competition between Classification Societies but to refocus those competitive efforts in the areas of service (as a matter of fact, there is not a common software to be used in applying the Rules) and responsiveness.

2.1 PHILOSOPHY ADOPTED BY THE PROJECT

Current Classification Society Rules have evolved over many years and have been mainly developed on an empirical basis. As a consequence, the basis of the Rules is not always transparent to the user. There have been many calls from the maritime industry for Classification Societies to adopt an approach which would lead to the development of Rules that are more easily understood and based on clearly identifiable scientific principles—even if the service history and statistical records have demonstrated that ships constructed to the existing Rules are of a satisfactory standard.

To meet the expectations of the maritime industry and to make use of best standards practice, it was decided by the project to develop a new set of Rules that would provide, through transparency, a better understanding of the design principles supporting the Rules.

The project team has also discussed the meaning of the call from industry for greater robustness in ship design and concluded that the concerns principally refer to the issues of safety and longevity. Safety is associated with classification strength requirements, whilst longevity refers to the period over which a ship keeps the necessary minimum strength without significant repair or renewal of steel.

Durability through life is shown in Fig. 1, which represents the overall level of safety reducing with time as the effects of wastage and fatigue, the principal time dependent factors, take place. Longevity (and life) is then extended by repair and renewal as necessary at periodical survey intervals.

![Fig. 1: Typical life pattern.](image)

The aim of the project has been to develop, within the new Rules, requirements that will satisfy safety issues and improve the longevity of ships by enhancing the requirements for fatigue life and wastage.

The new Rules will result in more robust ships that should be safer and more environmentally friendly. When properly operated and maintained, they should offer reduced through-life repair and maintenance costs and increase the margin of safety.

2.2 IMPACT ON THE STRUCTURAL DESIGN

The new Rules are not based on the existing Rules of any one society but reflect the combined experience of all the IACS members. The IACS Common Structural Rules have been developed within the IMO Goal Based Standards suggestions. For instance, the boundary conditions changed from the 20 year North Atlantic fatigue life used by ABS SafeHull to a 25 year North Atlantic fatigue life is based upon the indication from the IMO.

The IMO Goal Based Standards (GBS) initiative is aimed at introducing construction standards for new commercial ships. There is no intention that IMO would take over the detailed work of the Classification Societies, but rather that IMO would state what has to be achieved, leaving Classification Societies, ship designers and naval architects, marine engineers and ship builders the freedom to decide on how best to meet the required standards.

The result is a standard that exceeds the existing individual requirements of any one IACS member. Scantling requirements of the principal structural members have been increased to such an extent that it is not probable that an existing ship designed to SafeHull criteria could meet the new standards.
Indeed it is highly unlikely that a modification of the structure of an existing ABS Safehull approved vessel to bring it into compliance with the new Rules could be considered economically feasible. It would require replacement of the principal structural members to meet the new scantling requirements. Localized strengthening or additional details would not be sufficient.

In conclusion, the ships designed to the new Rules will be more robust than those built to existing classification requirements. Whether this enhanced robustness will be achieved through a more fatigue-resistant design, through an increase in the amount of steel or a combination of these will depend on the preference of the designer. It is obviously to be expected that some increase in steel will result.

3. CSR FOR TANKERS

3.1 KEY TECHNICAL ASPECTS

The Rules cover typical double hull tankers of greater than or equal to 150 m in length and with typical arrangements and proportions. The arrangements covered by the Rules (see Fig. 2) assume, in particular, that the structural arrangements include a narrow double side structure and double bottom structure with breadth/depth in accordance with statutory requirements, one or two longitudinal bulkheads of plane, corrugated or double skin construction (the number and location of bulkheads are arranged to comply with the statutory requirements).

The major portion of the Rules covers the prescriptive requirements, and include requirements for global longitudinal strength as well as local requirements for the hull envelope, transverse and longitudinal bulkheads and other primary supporting members. The prescriptive requirements incorporate all applicable IACS Unified Requirements, although these were redrafted to reflect the net thickness approach.

The net thickness approach adopted for the new Rules provides a direct link between the thickness that is used for strength calculations during the design stage and the minimum steel thickness accepted during the operational life of the ship. Since the IACS Unified Requirements are based on gross scantlings some changes are inevitable.

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The net thickness philosophy distinguishes between local corrosion (pitting, grooving and edge corrosion) and global corrosion (i.e.: uniform thickness reduction over an extensive area). As the hull girder cross section does not corrode uniformly, the philosophy adopted by the project takes this into account by using different thickness deductions for calculation of the global hull girder section properties and for local elements.

A distinctive requirement of the Rule is the mandatory assessment of the hull structure within the cargo region using finite element analysis (see Fig. 3). The objective of the FE structural assessment is to verify that the stress level, deflection and buckling capability of the main supporting hull structures are within the acceptable limits under the applied static and quasi-dynamic loads.

![Fig. 2: Typical arrangements of Double Hull Oil Tankers.](image)

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Acceptance criteria for working stresses in yield and buckling assessments are consistent with the structural model adopted in the calculation (i.e., making difference between FE calculations and analytical procedures).

In order to make a significant improvement in the transparency and internal consistency of the new Rules the definition of loads are included in a general section of the Rules. The other areas within the Rules and associated procedures refer to this section and define how the loads are applied in the particular application. By adopting this approach, the user of the new rules should be in no doubt about the origin of the loads or what the loads represent.

The dynamic loads cover normal service loads at $10^{-8}$ probability level. Rule design wave loads are based on the existing IACS Unified Requirements where possible or derived using the wave statistics of the North Atlantic sea area specified in IACS Recommendation 34.

In determining the dynamic load cases for both the prescriptive Rule requirements and for FE analysis the conditions for obtaining the most onerous structural response are established based on maximising primary load components for a 25 year return period. The simultaneously occurring secondary loads are accounted for by using load combination factors. The loads are applied to a number of fully loaded and partially loaded conditions in addition to the ballast condition.

For fatigue assessment the loads are used to calculate the expected stress range history based on a suitable distribution function. The reference load value used is at the $10^{-4}$ probability level of the Weibull long-term probability distribution [1].
3.2 THE STRUCTURAL DESIGN PROCESS

The basic principle in structural design is to apply the defined design loads, identify possible failure modes and employ appropriate capacity models to determine the required structural scantlings. The approaches applied for the determination of the structural response to the applied design load combinations are Beam Theory, used for prescriptive requirements, and the FE Analysis, applied on a coarse mesh for cargo hold model (yield and buckling assessment), on fine mesh for local models (yield assessment), and on very fine mesh for fatigue assessment.

The strength and acceptable safety of the hull and the structural elements is verified through the application of Rule requirements listed below, and they are applied into two subsequent phases:

- prescriptive scantling requirements,
- design verification requirements based on load-capacity methods.

The following diagram in Fig. 4 represents a general overview of the structural design process.

In the first phase minimum requirements are verified, which are in the form of minimum thickness, independent of the yield stress, based on service experience and expressed as a linear function of the ship length, and minimum stiffness and proportions, based on prescriptive buckling requirements. These are to be applied irrespective of all other requirements, hence thickness below the minimum is not allowed.

Also load-capacity based requirements are assessed through beam theory approach; they are based on the Working Stress Design (WSD) method, in which the stress in a local structure calculated in a critical point is to be below a fixed allowable stress.

In the second phase strength assessments using the Finite Element Analysis are performed. A linear elastic three dimensional finite element analysis of the cargo region (a FE model length of three tanks is required) is carried out to assess and verify the structural response of the proposed hull girder and primary support members and assist in specifying the scantlings requirements for the primary support members (PMS). The purpose of the finite element analysis is to verify that the stresses and buckling capability of the PMS are within acceptable limits for the applied design loads.

In the same phase fatigue assessments are required to verify that the fatigue life of critical structural details is adequate. The fatigue capacity assessment models are in either a prescriptive format or require the use of FEA methods. Also hull girder ultimate strength calculations are required.

The prescriptive minimum requirements define the minimum acceptable scantlings. These may not be reduced by any form of alternative calculations such as load-capacity prescriptive requirements or strength analysis such as FEM. The philosophy is that a coarse approach should be more conservative than a detailed approach.

3.3 ASSUMPTIONS FOR THE FATIGUE CHECK

In recent years Classification Societies, have introduced fatigue assessment requirements in order to improve the fatigue life of critical details by providing attention to construction details at the design stage. Additionally, improvements in the fatigue life have been gained by limiting the use of higher tensile steel. The new Rules include a mandatory set of requirements for fatigue assessment of all construction details currently covered within the classification requirements of ABS, DNV and Lloyd's Register.

The fatigue assessment procedure within the Rules is based on the following principal assumptions:

- S-N curve approach for the fatigue strength capacity,
- the use of cyclic stresses derived from the application of the different kinds of specified loads (the effect of mean stress is included),
- a design fatigue life of the vessel equal to 25 years,
- a long-term stress ranges load history of each structural detail represented by a two-parameter Weibull probability distribution,
- long-term environmental data relevant to the North-Atlantic Ocean (based on the IACS Wave Data),
- a linear cumulative damage model, based on the Palmgren-Miner’s summation method.

The procedure is specifically developed to evaluate the design fatigue life of tanker structural details that are known from experience to be potentially vulnerable to fatigue damage.

Fatigue strength is to be assessed to evaluate the design fatigue life of welded connections. This is done for:

- standard details, basically all longitudinal stiffener end connections in the cargo tanks and associated ballast spaces, using a nominal stress approach,
- other critical details (mandatory for the hopper knuckle connection) using the FE-based hot spot stress approach.

A structural detail classification based on the construction detail, joint geometry and consideration of the applied loading, is given in an appendix within the new Rules. Where the loading or geometry is too complex for a simple classification, a finite element analysis of the detail is to be carried out to determine the hot-spot stress at that detail.

Guidance on the finite element analysis required to determine the hot-spot stress at the weld is included in the new Rules.
4. THE FATIGUE DAMAGE

Fatigue damage can be defined as the process of progressive localized permanent change occurring in a material subjected to conditions which produce fluctuating stresses and strains at some point or points and which may culminate in cracks or complete fracture after a sufficient number of fluctuations [2].

Fatigue cracks and fatigue damages have been known to ship designers for several decades. With the introduction of higher tensile steels (HT steels) in hull structures, at first in deck and bottom to increase hull girder strength, and later on in local structures, the fatigue problem became more imminent.

During recent years a growing number of fatigue crack incidents in local tank structures in HT steels have demonstrated that a more direct control of fatigue is needed [3].

The better static properties of high-tensile steels are not coupled with comparable better strength when cyclic loads are applied; moreover, reduced scantlings lead to increased stresses and as a consequence greater stress ranges, which are obviously more demanding in relation to the cumulative fatigue damage.

The main cause of cyclic loads on ship structures is the action of the sea. Waves give rise to varying hydrodynamic pressures acting directly on the hull as well as cause ship motions that in turn create inertial loads [4].

Calculated fatigue lives, calibrated with the relevant fatigue damage data, may give the basis for a sound structural design (steel selection, scantlings and local details). Furthermore, they can form the basis for efficient inspection programs during fabrication and throughout the life of the structure.

4.1 BASIC CONCEPTS

The process starts with a microscopic crack, called the initiation site, which then widens with each subsequent loading. The number of cycles required for failure depends on material and failure is essentially probabilistic as consequence of the nature of loads. Damage is cumulative thus materials do not recover when rested.

The magnitude of stresses inducing fatigue damage includes stress concentrations caused by part geometry, quality of the surface – surface roughness, scratches, etc. cause stress concentrations or provide crack nucleation sites which can lower fatigue life depending on how the stress is applied. A reduction in fatigue life is also ascribed to weld shape and to possible weld defects. Moreover, environmental conditions can cause erosion and corrosion which both affect fatigue life.

In order to estimate the fatigue damage it is required to establish a reliable load history for the structure.

This generally implies the definition of a service profile for the ship, so that the environmental loads may be determined making use of appropriate wave statistics. It stands to reason that the procedure for the direct calculation of the demand in terms of stress range history, besides being quite complex is also extremely time consuming, since large numbers of sea states and ship conditions need to be considered.

In order to reduce the computation work, it is appropriate to create an analytical model applicable to structural details that are similar in shape and location. Such a model should adequately represent the severity of the expected load history. From several measurements it has been found that the long-term distribution of the stress ranges can be satisfactorily represented by the two-parameter Weibull probability function.

The evaluation of the response of the structure – in terms of stress ranges – in the various critical spots is generally accomplished through a multi-level FEM. In the finer mesh model the stress analysis is focused on critical details, and more specifically on hot-spots as weld fillets, cut-outs, scallops or other similar discontinuities, where fatigue cracks usually initiate, growing normal to the direction of the largest principal stress [5,6].

The comparison between demand and capability can be performed making use of proper S-N curves and resorting to Palmgren-Miner’s linear cumulative damage model. In this framework the fatigue resistance of the detail under consideration, when exposed to a given series of loads, is checked through the evaluation of its fatigue life, i.e. through the prediction of the time at the end of which a possible fatigue crack reaches the critical size that triggers a failure process.

For the fatigue analysis different reference stresses can be considered (see Fig. 5):  
- the real stress at the hot spot, defined as notch stress \( \sigma_n \),
- the conventional stress defined as hot-spot stress \( \sigma_k \) – a stress obtained by a linear extrapolation of the geometrical (or structural) stresses to the hot spot location; is obtained by FE analyses using very fine mesh models.
- the nominal stress \( \sigma_N \) calculated at the hot spot location, generally by analytical formula based on the beam theory.

\[
\begin{align*}
\sigma_n &= \text{notch stress} \\
\sigma_k &= \text{hot spot stress} \\
\sigma_N &= \text{nominal stress}
\end{align*}
\]

Fig. 5: Stress definitions at the hot spot [5].

The notch stress is affected by the weld shape and by the gross geometry of the detail. In the region close to the hot-spot, where only the influence of the structural discontinuity is perceived, the stresses are called geometrical or structural stresses in order to emphasize the fact that their values depend mainly on the gross geometry of the structure.
The hot-spot stress is obtained through a linear extrapolation based on the structural stress values calculated on the plate at 0.5 \( t \) and 1.5 \( t \) (\( t \) is the thickness of the plate) from the hot spot (see Fig. 5). In the standard practice, structural stresses based on FE calculations are collected at the centroid of the closest four shell elements located in front of the hot spot. Then, the two stress values on the above mentioned location can be simply obtained by resorting to the Lagrange method for the polynomial interpolation. The ratio between the notch stress and the hot-spot stress is defined as stress concentration factor due to the weld geometry \( K_w \):

\[
K_w = \frac{\sigma_n}{\sigma_s} \tag{1}
\]

The stresses in the area relatively far from the hot-spot, where all the local effects are negligible, correspond to the nominal stresses. The ratio between the hot-spot stress and the nominal stress at the hot spot is defined as stress concentration factor due to the detail gross geometry \( K_g \):

\[
K_g = \frac{\sigma_s}{\sigma_k} \tag{2}
\]

### 4.2 THE FATIGUE DAMAGE CALCULATION

In 1945, M. A. Miner popularized a rule that had first been proposed by A. Palmgren in 1924. The rule is based on the assumption that damage accumulated after \( n \) cycles of a certain applied dynamic load yielding to a stress range \( S \) is quantified by the ratio \( n / N \), being \( N \) the number of cycles to failure of the constant stress range \( S \). Therefore, the rule states that where there are \( k \) different stress magnitudes in a spectrum, each contributing \( n_i \) cycles, the fatigue damage is calculated as the sum of \( n_i / N_i \) ratios each corresponding to the \( i \)-th stress range (being \( i \) an integer between 1 and \( k \)). Failure occurs when this ratio is equal or greater than one.

The assessment of the fatigue life of a structural detail is performed resorting to the linear cumulative damage model of the Palmgren-Miner’s rule expressed in the following form:

\[
DM = \sum_{i=1}^{k} \frac{n_i}{N_i} \tag{3}
\]

Though Miner’s rule is a useful approximation in many circumstances, it has a major limitation, because it fails to recognize the probabilistic nature of fatigue.

In order to evaluate the cumulative damage \( DM \), it is necessary to refer to an appropriate \( S-N \) curve for the number of cycles to failure \( N_i \), and to the load history for the corresponding demand in terms of \( n_i \).

As well known, the \( S-N \) curve is a graph of the magnitude of cyclical stresses \( S \) against the relevant cycles to failure \( N \). \( S-N \) curves are derived from tests on samples of the material (or structural detail) to be characterized where a regular sinusoidal stress is applied by a testing machine which also counts the number of cycles to failure. Thus, the capacity of welded steel joints with respect to fatigue strength is characterized by \( S-N \) curves which give the relationship between the stress ranges applied to a given detail and the number of constant amplitude load cycles to failure. \( S-N \) curves are represented by:

\[
\log(N) = \log(K) - m \log(S) \tag{4}
\]

where \( m \) and \( K \) are constants depending on material and weld type, environmental conditions (exposure to air or sea water), type of loading and geometrical configuration. \( K \) is implicitly related to a certain survival probability to cover the scatter in the experimental results.

The \( S-N \) curve to be used for the fatigue damage calculation has to be consistent with the reference stresses considered.

In general terms, if the fatigue capacity does not vary for a set of structural details made with a given material, then the \( S-N \) curve in terms of real stresses is the same for whole set and it may be referred to as the parent curve (and the relevant \( K \) parameter may be called \( K_P \)). Thus for each structural detail it is possible to define a \( S-N \) curve in terms of conventional stresses simply by scaling the parent curve. In other words, the \( K \) parameter for the considered structural detail comes out to be:

\[
\log(K) = \log(K_P) - m \log(K_S) \tag{5}
\]

where the scaling factor \( K_S \) is equal to \( K_w \) or to the product \( K_w K_g \), depending on the choice for the conventional stresses to be referred to, hot-spot stresses or nominal stresses respectively. Clearly, the \( S-N \) curve characterized by \( K = K_P \) is that drawn from testing a sample of material for which \( K_w K_g = 1 \).

On the other hand, to any of such \( S-N \) curves may be related all the structural details characterized by the same value of the scaling factor \( K_S \). Tables of these scaling factors related to “nominal stresses \( S-N \) curves” are provided for typical structural configurations and loading conditions, taking into account the detail geometry, the weld shape and the peculiarities of the manufacturing process.

If a two slope \( S-N \) curve in log-log scale is considered, having a change of slope from \( -1/m \) to \( -1/m' \) at \( N_C \) cycles (which corresponds to the stress range \( S_C \)), Eq. 4 is valid only for \( S \geq S_C \), and for \( S \leq S_C \) the \( S-N \) curve is:

\[
\log(N) = \log(N_C) + m' \log(S_C / S) \tag{6}
\]

where, as usual for ship structural details, \( N_C = 10^7 \) cycles.

As previously noted, to represent the load history \( p(S) \), it is possible to resort to a Weibull distribution once its two parameters \( \lambda \) and \( k \) are defined with suitable values. Therefore the long term stress range probability density function can be written as follows:

\[
p(S) = k \lambda S^{k-1} \exp \left[ - \left( \frac{S}{\lambda} \right)^k \right] \tag{7}
\]

The scale parameter \( \lambda \) is correlated to the severity of the load history, and its value can be determined by imposing a stress range \( S_0 \) to have a certain probability of exceedance \( Q(S_0) \). The expression for \( \lambda \) is:

\[
\lambda = \frac{1}{S_0} \left[ - \ln \left( Q(S_0) \right) \right]^{1/k} \tag{8}
\]
The shape parameter $k$ is correlated to the distribution of the occurrences $n/N$ of load cycles $S$. To explain the meaning of the shape parameter, let us consider the effect of the waves occurring with a relatively higher frequency (those that are comparatively medium or small in size) on a small vessel: they cause a great number of high stress range values. As a consequence, the load history area will have a barycentre shifted towards higher stress range values and this situation corresponds to a higher value of the shape parameter.

For ship structures the shape of the load history for a structural detail depends mainly on the use made of the detail: location on board, type of structure in which it is inserted and type of ship where it is applied. It has been found that the value of the shape parameter $k$ can be related to the ship dimensions (higher value of the shape parameter for the smaller vessels) and location of the detail under investigation. To take into account these two aspects, $k$ is usually defined as:

$$k = f(L) f(P)$$

where the function $f(L)$ defines the dependence on the ship length $L$ and $f(P)$ defines the dependence on the plating area to which detail is joined (e.g., bottom plating at centreline in way of the midship section).

To sum up the whole process, the availability of an analytical representation of the fatigue demand through the Weibull function $p(S)$ together with an analytical representation of the capability in terms of $S-N$ curve, make it possible to carry out an analytical evaluation of the fatigue damage. Assuming the long term distribution of stress ranges fits a two-parameter Weibull probability distribution, the cumulative fatigue damage $DM$ is:

$$DM = \frac{N_L}{K_S^m} \mu \Gamma \left[1 + \frac{m}{k}\right]$$

where $\mu = \mu(m, m', k, N_c, N_{cs}, S_c, S_{0}, Q(S_0))$ is a coefficient taking into account the change in slope of the $S-N$ curve ($\mu$ is equal to one for a single slope $S-N$ curve), $N_L$ is the number of cycles for the expected design life and $\Gamma[\cdot]$ is the complete Gamma function.

The cumulative fatigue damage ratio $DM$ may be converted to a calculated fatigue life, equal to the ratio between the design life and the damage ratio.

### 4.3 CSR FATIGUE STRENGTH ASSESSMENT

Fatigue analyses are to be carried out for representative loading conditions according to the intended ship’s operation. Two conditions are to be examined: full load at design draught and ballast at normal ballast draught. Therefore, the cumulative damage is calculated as the sum of the cumulative fatigue damages $DM_j$ corresponding to each loading condition:

$$DM = \sum_j \alpha_j DM_j$$

where $\alpha_j$ refers to the proportion of ship’s life spent in each loading condition. Adjustments to the $S-N$ curves can be made, to take into account the effect of mean stresses, plate thickness and weld improvements.

The fatigue capacity assessment models are in either a prescriptive format or require the use of more advanced calculations such as FE analyses. These methods account for the combined effects of global and local dynamic loads [7].

For each load history, the stress range $S_0$ used for the evaluation of the scale parameter $\lambda$ is that having a probability of exceedance $Q(S_0) = 10^{-4}$.

With regard to longitudinal structures, a fatigue strength assessment is to be carried out for the end connections of longitudinal stiffeners to any transverse structures and for scallops in way of block joints on the strength deck within the cargo tank region.

Each structural detail is idealised, classified and related to the proper $S-N$ curve, defined in terms of nominal stresses and characterized by a proper scaling factor $K_S$ (the fatigue analysis is carried out using a nominal stress approach). To determine the load history, and hence the cumulative damage, the suitable value for $S_0$ is a nominal stress obtained by empirical formulae.

With regard to transverse structures a fatigue strength assessment is to be carried out for the welded knuckles between inner bottom and hopper plate for at least one transverse frame close to amidships.

When necessary, the hot spot stress approach may be applied to demonstrate the adequacy of longitudinal stiffener end connections.

The $S-N$ curve has to be one that takes into account the weld effect only (i.e., $K_S = K_{SW}$), being the stress used the hot spot stress.

To determine hot spots stresses, local 3D plate elements very fine mesh FE models are to be used (see Fig. 6). The fatigue damage calculation is to be based on the hot spot stress range $S_0$ evaluated close to the potential crack location in a direction perpendicular to intersection of the inner bottom plate and hopper plate (i.e., perpendicular to the potential direction of the crack).

![Fig. 6: Typical local FE model of knuckle between inner bottom and hopper plate.](image)

It has to be noted that stress ranges on the hopper knuckle do not depend on hull girder bending but only on the distortion of the transverse web frame in which the hopper knuckle is under analysis, and therefore stress ranges depend on the loads deforming the web frame.
5. THE CASE STUDY

Objective of the study was that of establishing representative minimum scantlings required to meet the Common Structural Rules for Double Hull Oil Tankers for an existing standard design and of assessing the predicted change of steel weight resulting from the difference in the minimum CSR scantlings for the midship cargo region compared to as-built scantlings.

The test case study was carried out for the midship section region of an existing vessel, an ABS class 80,000 dwt oil tanker (see Fig. 7).

Some changes were necessary to comply with the CSR requirements and the modifications have been kept to a minimum, in order to stick to the original structural design as much as possible.

The modifications were made to the structure in accordance with the widely known “trial and error” method, favoured by the availability of the software set up by ABS [8]. In some cases the search for the intended goal led to choices that a designer could not make as easily as have been made. Indeed, optimisation in the “real world” involves several economic and practical construction aspects that were not considered in this study.

An iteration process could not obviously be avoided: a local modification in the structure can bring in a change in the ship section modulus that in turn affects the stresses acting on the structure and therefore the section modulus that in turn affects the stresses acting on the stiffeners, and the latter allows to refer to a better twisted modifications can be made (see Fig. 8):

− increase of the stiffener section modulus could also be accomplished by raising the height of the neutral axis of 0.4 metres.

As far as the deck longitudinal stiffeners are concerned, several longitudinal stiffeners of the existing vessel do not pass the CSR checks in terms of web thickness and section modulus requirements. This specific lack is shown by the majority of the longitudinal stiffeners, most of which do not present a sufficient section modulus and, in general, the deficiency ranges from 15% to 45%. Only a small number of strakes do not prove to satisfy the scantling requirements and the necessary thickness increments range from 0.5 mm to 3.5 mm.

Once the minimum scantlings were obtained, a fatigue life evaluation was made. The application of the method described in chapter four to evaluate the fatigue life of longitudinal stiffeners end connections gave unsatisfactory results for several stiffeners located on deck and side shell.

With regards to the fatigue check of longitudinal stiffeners end connections two different situations have been faced, according to the type of the predominant cyclic load, that is, global loads or local loads. For a location where the stress ranges are mainly due to global loads (i.e., along the deck), local changes in the structure can not provide a sufficient reduction of stress ranges, whilst a change in the ship section modulus can obviously bring more benefit. On the other hand, local changes result more effective where local loads are predominant (i.e., on the ships side).

With regards to the deck longitudinal stiffeners, a solution to the problem could lie in increasing the ship section modulus at deck, in order to reduce the high stress (and therefore the stress ranges) acting on the stiffeners. This can be obtained by increasing the deck plating thickness, and/or by introducing deck girders. In other words this is obtained by raising the height of the neutral axis, bearing in mind that this could worsen the situation on the bottom structure, although for a tanker double bottom structure this is generally not a major problem.

It is noteworthy that the material employed in the deck structure is HT steel. This type of steel has a better behaviour compared to mild steel basically when the yield strength is concerned. When repetitive loadings are considered the two materials basically show the same fatigue strength and the reduction in steel requirements for the HT steel (in terms of steel thickness) based on yield-strength conditions, causes a raise in acting stresses that, although within the limits of yielding stress, are high enough to sensibly reduce the fatigue life.

As far as the deck longitudinal stiffeners are concerned, calculation gave satisfactory results in fatigue life by a raising in the height of the neutral axis of 0.4 metres.

When the side shell stiffeners are concerned, two intertwined modifications can be made (see Fig. 8):

− reduce the effective length of the stiffeners by choosing a proper size for the connection attachment lengths,
− choose a different structural detail for the stiffener end connections (use of “soft toe” connections).

The former solution reduces the actual stresses acting on the stiffener, and the latter allows to refer to a better class of S-N curve to calculate the fatigue damage. An increase of the stiffener section modulus could also be
considered, of course, but this would lead to major increases in the steel weight required by the structure. A reduction in web spacing would be an even more unfortunate drastic initiative, and clearly could not be considered a proper solution.

**Fig. 8:** A schematic representation of the stiffeners connections.

### 5.2 TSA STRUCTURAL EVALUATION

As already seen, Common Structural Rules require a strength assessment of the hull structure by means of FE analysis. The assessment procedure is referred to as Total Strength Assessment (*TSA*).

At present the FE global model, as far as model geometry and mesh are concerned, has to be produced by the user with the use of finite element modelling software (such as FEMAP) in accordance with the requirements of CSR, whereas constraints and loads on model, based on data both from CSR-*ISE* and from user input given at this stage, are automatically generated by CSR-*TSA*, once the unladen model has been imported.

The three cargo hold model used for the strength assessment was based on the scantlings obtained for the modified design. **Fig. 9** shows a slice of the coarse mesh FE model.

**Fig. 9:** FE model employed in the stress assessment.

Fine mesh local models and very fine mesh models are similarly handled: their mesh is created through the use of a modelling software and then imported into CSR-*TSA*. Once the finer mesh model has its location in the global model defined by the user, it basically inherits the relevant constraints and loads from the parent model.

The results of total strength assessment, in terms of yielding and buckling checks, presented the necessity for further modifications to some longitudinal structures. Increases made at this stage regarded both plates thickness and stiffeners section modulus in some areas of the global model. In some cases peak values above threshold values were found in details such as the lower bracket of vertical webs or on the horizontal stringers. In such cases the intervention on the structure is made locally. For example, a bracket can be built up from plates of different thickness and the thickest parts go to cover the areas under higher stresses.

The fatigue strength evaluation made on the connection between inner bottom and hopper plate (see **Fig. 10**) gave satisfactory results, since the fatigue life of the hot spot has been found to be much greater than 25 years (the safety factor was of about 3.7).

**Fig. 10:** Hopper knuckle connection fine mesh FE model.

### 5.3 STEEL WEIGHT INCREASE

An estimate of the minimum steel weight increase required in the midship cargo tank region is provided in **Tab. 1**. It has to be noted that this estimate does not include the modifications the structure requires to pass the fatigue check for longitudinal stiffener end connections. It is likely to expect the required modifications to result into a further increase in steel weight.

<table>
<thead>
<tr>
<th>Plates and stiffeners position</th>
<th>Influence on the steel weight increase (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>plates</td>
</tr>
<tr>
<td>Bottom &amp; bilge</td>
<td>6,6</td>
</tr>
<tr>
<td>Side</td>
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</tr>
<tr>
<td>Deck</td>
<td>6,6</td>
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<tr>
<td>Inner bottom</td>
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</tr>
<tr>
<td>Inner skin</td>
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<tr>
<td>Centreline BHD</td>
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<tr>
<td>NT bottom girders</td>
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<td>WT bottom girders</td>
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<tr>
<td>Side transverses</td>
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</tr>
</tbody>
</table>

**Tab. 1:** Estimate of the minimum steel weight increase.
The modifications made to the original design resulted in a steel weight increase of 3.3% (corresponding to an weight of 121 t) of the steel weight of as built longitudinal structures of the cargo tank region (stiffeners required an increase of 4.4% whereas plates increased their weight of 3.2%). The length of the midship cargo tank region includes 22 transverse web frames and 3 transverse bulkheads, for which it has been estimated a steel weight increase of 2.9% (corresponding to a weight of 33 t). This yielded a steel weight increase of 3.2% for the midship cargo region. This value is in accordance with what has been predicted by other studies.

In early 2005, the effect of applying the 2nd draft of the Common Structural Rules for Tankers to current tanker designs was assessed by the CSR authoring Societies. The assessment compared the new requirements with the as-built scantlings of some existing designs: the result obtained concerning the overall effect on the steel weight change in the midship cargo area is an increment ranging from 3% to 5% [9].

It has to be said that for any particular ship size a range of steel weight differences is possible since the estimates are highly dependent on the degree of structural optimisation and the original classification of the as built design.

6. CONCLUSIONS

The purpose of this study is to present the new Common Structural Rules (CSR) for Double Hull Oil Tankers and evaluate their impact on the structural design.

With this object in view a test case has been developed, focused on the midship cargo tank region of an existing vessel.

The original design was modified to conform it to minimum scantlings requirements of the new criteria, making use of a new ABS software based on CSR for double hull oil tankers.

In this framework the fatigue analysis of critical details has been tackled. Initially, the theoretical basis on which the CSR formulations are founded has been investigated. In particular, a critical analysis was conducted for the fatigue analysis methods (nominal stress approach and hot spot stress approach) the new Rules require for this type of verification.

Nominal stress approach was applied to study details of standard geometry. The application of the new Rules allowed to locate on the side shell and along the deck a certain amount of longitudinal end connections characterised by an unsatisfying fatigue life. Having identified the most demanding cyclic loads for each location, adequate interventions to improve fatigue strength of such details have been outlined, to provide useful guidelines for basic design. Improvements in the fatigue life can also be gained by limiting the use of higher tensile steel.

Hot Spot stress approach was applied to verify the lower hopper knuckle connection. Being this detail not standardised, for its verification it is necessary to resort to a laborious study based on FEA. With the introduction of the new Rules, such procedure is now mandatory, being this detail particularly exposed to possible fatigue fracture. The analysis carried out on the examined vessel did not point out a critical behaviour of the structure.

The structural adjustment of the examined vessel involved a steel weight increase. The evaluation of such increase, caused by the modifications brought to the structure, resulted to be 3.2%. This is overall in accordance with the prediction made in 2005 by the CSR authoring Societies. A steel weight increase is a natural consequence of the changes brought along by the new Rules, being its design conditions more severe and requiring a 25 year North Atlantic fatigue life (i.e. 5 years more than previous Rule requirements).

Scantling requirements of the principal structural members have been increased to such an extent that it is not probable that an existing ship designed to “old” criteria could meet the new standards.

The interventions required to bring the design into compliance with the new Rules resulted so invasive that such modification does not seem economically feasible.

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REFERENCES