특집 : 구조물의 파괴 및 피로강도 해석 기법

Fatigue Assessment of Welded Ship Structures

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1. Introduction

Modern ocean-going vessels are highly dynamically loaded all-welded structures. Traditionally, this fact has been taken into account in the design of ship structures by keeping the overall stress level within permissible limits and by limiting notch effects in the design of local structures. With an improving knowledge of loads and the rapid development of calculation means more detailed calculation-based methods are increasingly introduced.

A brief review on fatigue assessment methods used in the design of ship structures has been given during the 1993 IIW International Conference in Glasgow. The rapid development during recent years underlines the importance of the subject and justifies its extended discussion in this paper.

2. Loads

Ships are floating bodies, which can move in six degrees of freedom. Moreover they are flexible and elastic vibrations may contribute to overall and local stress levels.

Time-dependent loads are already caused by varying loading conditions in still water(fig. 1). Dynamic random-distributed pressures in seaway lead to overall longitudinal bending and combined transverse bending/torsion of the hull as well as to dynamic loading of local structures, e. g. of side frames. Non-linear load effects must be taken into account in certain cases due to temporary immersion

Another type of local dynamic loading are impact loads, especially on the foreship bottom.

Finally, propulsion-induced vibrations must be mentioned as a possible source for dynamic loading of ship structures. Unbalanced masses as well as the firing process in reciprocating engines generate cyclic loads. The other importhant source is the propeller operating in a nonuniform wake field. Vibration-exciting forces generated at the propeller blades can be transmitted to the hull via shafting and bearings. More important is the generation of pressure pulses on the hull surface in way of the propel-

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ler. The amplitude of these pressure pulses and of the resulting vibration-exciting forces are very much dependent on extent and intensity of propeller blade cavitation. In areas remote from the exciting sources the vibration will affect crew and passengers on board rather than the structure. However, in certain areas severe structural damage may be caused by propulsion excited vibrations. Examples of such areas are rudders and tanks in the aftbody and engine room. Resonances play an important role in this connection. A further feature is a very high number of cycles up to about 100 billion during service life.

Summarizing it can be stated that ship structures are subjected to various dynamic loads, which may in individual areas be superimposed in a complicated way.

3. Response

While conventional design rules are mostly based on a direct estimate of dimensions as plate thicknesses or section moduli, fatigue life assessment methods require a stress analysis in each case. The calculation model depends on the assessment method applied.

Beam models of the hull or of local grillage systems, individual stiffeners or plates(fig. 2) can be used to calculate nominal stresses.

Finite element models of the whole hull (fig. 3) are more and more used[1]. In these membrane elements are used to model the hull structure, while local stiffeners are modelled by means of beam elements. Interacting load components can be applied simultaneously. Present developments are aiming at a direct derivation

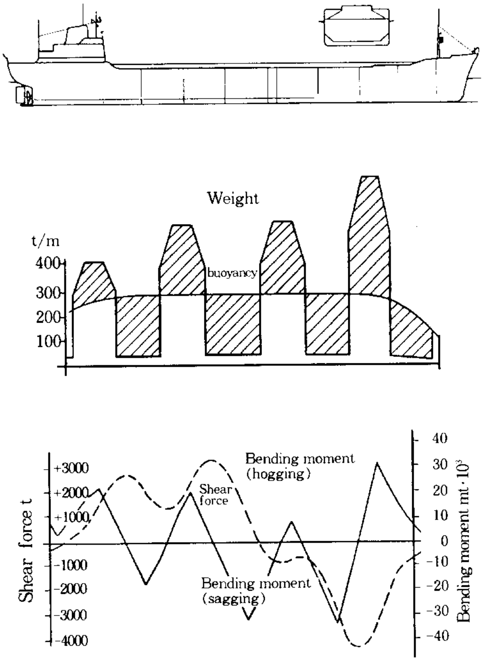


Fig. 1 Still water hull load of a bulkcarrier with ore cargo

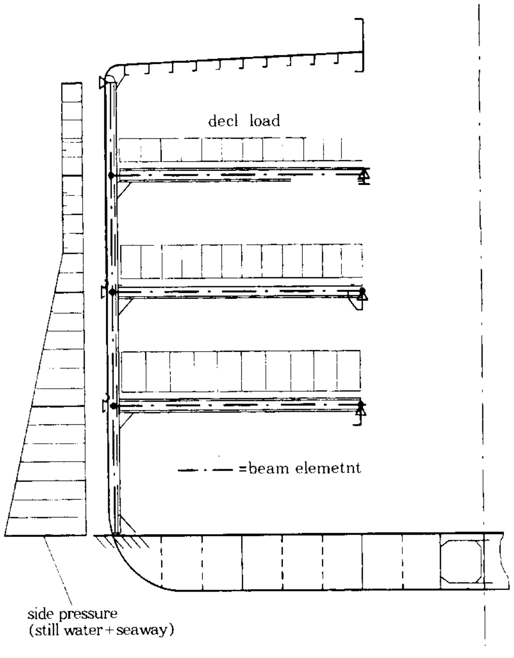


Fig. 2 Transverse structure of a reefer vessel

of the finite element model from CAD models. The results from the global analysis are considered as nominal stresses.

Global finite element models are further useful to calculate the hull and superstructure response to vibration exciting forces in terms of resonance frequencies and modes or non-resonant forced vibrations.

Local models are used to calculate hot spot or notch stresses(fig. 4)[1]. Stresses at free edges of plates can often be obtained by means of membrane elements. In the calculation of notch stresses at welds plate elements or, as shown in fig. 4, volumetric elements are used. Models with a linear stress distribution over the plate thickness are required to calculate hot spot stresses.

In the stress analysis, linear-elastic material behaviour is usually assumed. Cyclic stress-strain curves together with approximative relations as Neuber's rule can help to estimate local elasto-plastic stress and strain[2].

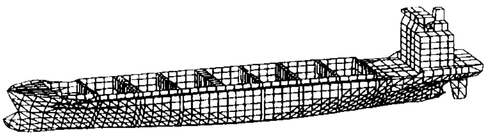
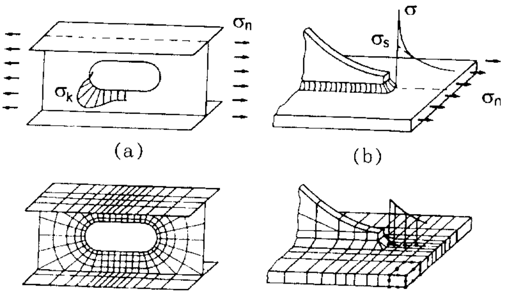


Fig. 3 Global FE-model of a container vessel



(a) Cutout with themal cut edge
(b) Welded stiffener end

Fig. 4 FE-models of structural details

4. Fatigue-life prediction methods

The mostly used method is the nominal stress concept[3] which also forms the basis of most fatigue codes for ship structures. The concept is based on a grid of SN-curves together with notch classes corresponding to stress ranges at a given number of cycles(usually 2 million) and survival probability. Design details are related to notch classes by means of test results. It must, however, be kept in mind that deviations from standard in design or loading condition may lead to large errors in the fatigue strength assessment.

Variable amplitude loading is usually taken into account by means of a Miner damage summation. The two-slope S-N-curve first proposed by Haibach is used to take the decreasing fatigue limit during the damage process into account.

Attempts have been made during recent years to apply the hot spot stress concept originally developed for tubular nodes to welded ship structures[4]. Problems have been found in the assessment of individual details as shown in fig. 5. Such problems relate to details of the finite element model and to the extrapolation of stresses to the hot spot.

More refined local concepts are not enclosed in present codes. However concepts as the elastic notch stress concept proposed by Radaj for welds[5] and the elasto-plastic stress/strain concept for free plate edges[6] are applied with benefit in individual cases.

The fracture-mechanics based stable crack propagation approach is applied in certain cases. In a recent interesting study remaining fatigue life and required inspection intervals of pre-cracked structures containing structural notches depending on notch stress gradients are discussed[7].

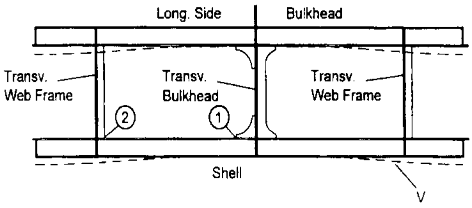


Fig. 5 Longitudinal side frame in a VLCC

5. Fatigue design rules of Ship Classifiction Societies

Rules and procedures for the fatigue assessment of ship structures have been developed by Ship Classification Societies and included in their design rules. It is not surprising that such procedures basically contain the same steps. However, differences exist in detail between these approaches making their application even more difficult for designers not so experienced in this kind of assessment. A harmonization of the fatigue design rules of individual Classification Societies within the IACS(International Association of Classification Societies) activities would therefore be very welcome from the designer's point of view.

As an example the fatigue design rules of Germanischer Lloyd[8] are briefly highlighted. They have been chosen because the author of this paper is especially familiar with these rules.

The load spectrum is derived from design loads(maximum expected load during assumed service life) and standards for shape of the spectrum and number of cycles. In case of superimposed global and local load effects stress combinations are defined, in which factors shall take account of the random nature of superposition.

A nominal stress assessment procedure is given, which is based on the IIW fatigue design rules and Eurocode 3. A catalogue of standard notch cases is presented, which is related to detail categories. This detail category corresponds to the stress range for 2 million cycles and a survival probability of 97.5%. Correction factors are applied to take account of material, stress ratio, weld shape, stress spectrum and risk assessment.

With regard to base material a bonus is given for higher strength steels in case of thermal-cut free plate edges. No influence is assumed for welded connections.

A favourable influence of mean compression stresses is assumed for free plate edges and for welds subjected to variable stress range spectra. This assumption deviates from those made in other codes. It is intended to reflect the experience from the survey of ships in service.

The effect of post-weld treatment, e. g. by grinding, can be taken into account by a weld shape factor.

The most important influence is that of spectrum shape and number of cycles. The pertinent factor is obtained from a Miner-summation and a two-slope S-N-curve.

Finally, a reduction factor to the permissible stress range has to be applied in cases where local damage can lead to immediate failure of larger components or where high crack propagation rates must be expected due to low gradients in the notch stress field, e. g. in hatch corners with large corners plate radii.

If an dffective corrosion protection cannot be assumed, a modified S-N-curve corresponding to a fatigue life reduction by a factor of 2 should be used.

It should further be noted that usual angular distortion and misalignment effects are assumed to be covered by the detail categories chosen. Further investigations are considered desirable with respect to the combined influences of plate thckness and geometric imperfections.

A hot spot design curve is given, which is derived from the notch stress concept and which corresponds to the detail category 100. Further rules for the stress analysis and extrapolation to the hot spot exist. Comparison with standard notch cases based on nominal stress has led to the introduction of a second design curve for load-carrying transverse fillet welds, which corresponds to detail category 90.

6. Examples

The application of the fatigue assessment methods discussed is shown in three examples[9].

Example 1 *Longitudinal side frame in a large crude oil tanker(VLCC).*

Experience has shown that longitudinal side frames in large VLCC's are critical with regard to fatigue. A fatigue assessment is therefore generally required.

The frame considered here(fig. 6) is located in the midship region of a 320 m long twin hull tanker 2.55 m below the design load line. In the critical loading condition to be considered the cargo tanks are fully loaded while the ballast tanks in the twin hull are empty.

The following stress components have to be calculated and superimposed

—hull girder bending stress

—local bending stresses caused by side pressure(fig. 7)

—local bending stresses caused by relative deformation v between transverse girder and bulkhead(fig. 6)

The relative deformation v is obtained from a global finite element model of the cargo tank area.

Superposition of the three stress components is carried out taking a factor of 0.75 for the hull girder wave bending stress into account. The signs of the stress components have to be carefully observed. From maximum and minimum total stress the stress range and mean stress are obtained. At the bulkhead(point 1 in fig. 6) local bending stresses add unfavourably,

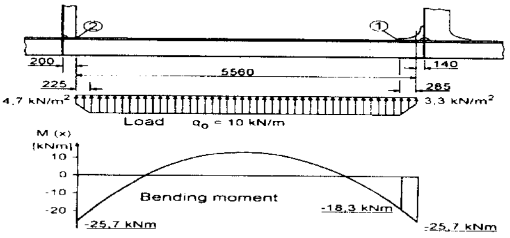


Fig. 6 Local bending of longitudinal side frame

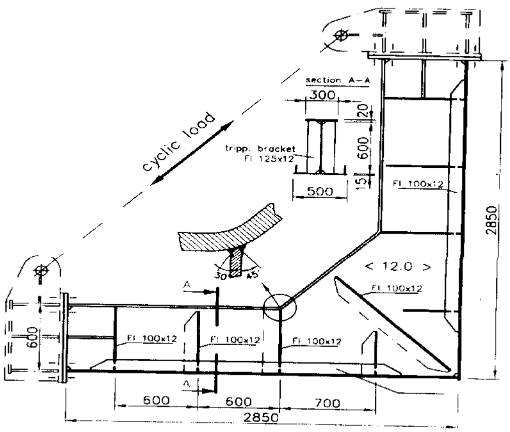


Fig. 7 Frame corner model

while at point 2 they partly cancel each other out.

It can, therefore, be concluded that point 1 is critical with regard to fatigue cracking. By means of the nominal stress concept it is shown that the detail design applied is still permissible.

Example 2 *Cicular hole in a hatch coaming flange plate.*

With regard to hatch cover operation circular holes often become necessary in the flange plates of continuous hatch coamings(fig. 8). As these coamings are part of the upper hull girder flange, they are highly stressed. The fatigue assessment is based on the elasto-plastic stress/strain concept. It includes the following assumptions and calculations steps.

The nominal design stress is 208 N/㎟ for a yield stress 355 N/㎟ of the base material.

The nominal stress spectrum is Weibull-distributed with a shape factor 1.0 and a total number of cycles of 50 million. The stress spectrum is converted into an 8-step spectrum (fig. 9)

A notch factor of 3.0 for an infinite plate with a circular hole is assumed. The Max. linear-elastic stress then becomes 624 N/㎟. The design stress ratio is R=0(max. still water and wave bending stress equal).

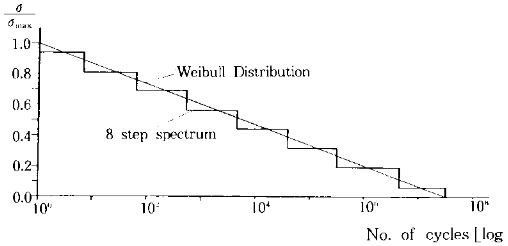


Fig. 8 Long-term hull wave bending spectrum

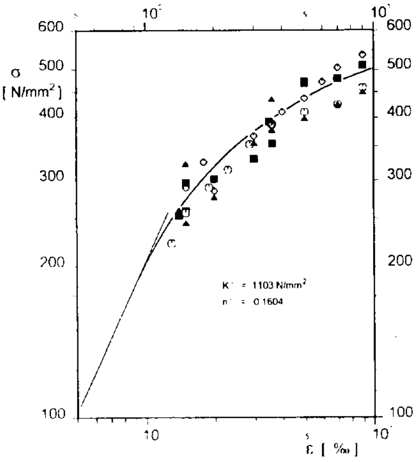


Fig. 9 Cycle stress-strain curve for flame-cut edge in DH 36 shipbuilding steel

It is further assumed that the vessel will operate half of its life with full and half with 50 % of permissible still water bending stress.

Local nonlinear strain amplitudes and maximum stresses for each step are calculated by means of a given cyclic stress-strain curve(fig. 10) and Neuber's rule.

The damage is assessed by menas of Smith, Watson and Toppers damage parameter PSWT and a Miner summation. The most damaging part of the load spectrum is found in the second lowest step(step no. 7). This underlines the importance of the PSWT-N-curve in the high cycle regime and of the damage accumulation rule applied. With the two-slope approach a damage sum of 0.6 is obtained. However, this figure refers to a survival probability of about 50%. This fact and uncertainties regarding fabrication quality and corresponding effective stress concentration factor lead to the conclusion that circular holes cannot be accepted in the upper hull girder flange built from higher strength steel.

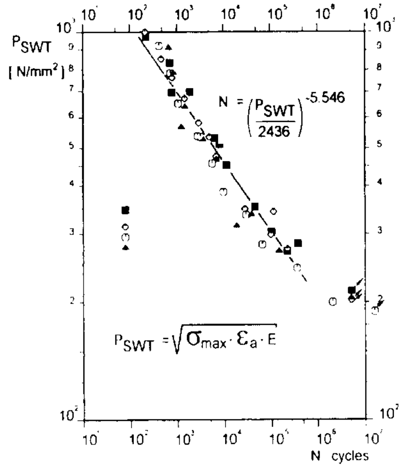


Fig. 10 Damage parameter curve for DH 36 shipbuilding steel with flame cut edge

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