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Stress calculations for longitudinal hull members fatigue assessment

SUMMARY

Simple formulae for calculating stress ranges in longitudinal ship hull members, exceeded with an assumed probability, are proposed. The stress ranges can be applied to fatigue life assessment. The assessment is based on approximation of long term stress range distribution by the Weibull's distribution and S-N curves application. The total stress range is calculated in the form of sum of global and local components, multiplied by partial stress range coefficients. The global stresses are caused by general bending in vertical and horizontal planes. The local stresses result from bending of stiffeners or plating.

Approximate values of the partial stress coefficients were found on the basis of long term stress distribution direct calculations, performed for 6 ships. Approximate values of dynamic stress cycle numbers that occur during ship lifetime and coefficients defining the Weibull's distributions were also found.

INTRODUCTION

Hulls of large tankers, bulk carriers and containerships are often built with a wide use of high tensile steels. However, fatigue strength of welded structural details, where stress concentrations occur, is almost independent of the steel tensile strength. It means that the structures have to be evaluated with thorough consideration of the fatigue strength. For example, the shape and the possible minimum dimensions of large containership hatch corners are usually determined by the fatigue strength requirements.

Fatigue cracks of individual elements of ship structures usually do not directly lead to catastrophic events. However, the costs of multifold repairing of the cracked structure can be significant. The cracks occur mostly at the following ship structural details:

- the connections of tanker side longitudinals with transverse bulkheads and webs
- the connections of lower hopper tanks with side framing and those of lower stools with the double bottom, on bulk carriers
- the connections of bottom longitudinals with floors, in bulk carriers with alternate heavy cargo loading (low cycle fatigue hazard)
- hatch corners, longitudinal coamings and hatch covers of containerships.

The paper presents some results of the research project on the fatigue strength requirements to be applied in the Rules of the Polish Register of Shipping (PRS) [1].

SIMPLIFIED METHOD OF FATIGUE LIFE ASSESSMENT

The method will be explained by using as an example the fatigue life calculation of side longitudinal connections with transverse web frames (Fig.1).

The calculations can be performed by applying either the so - called nominal stress, or hot – spot stress or peak stress approach [2].

The nominal stress approach, which is the simplest of the three methods, will be applied here.

The dynamic component of normal nominal stress, σ, at the point P (Fig.1) can be calculated as follows:

$$\sigma = \sigma_v + \sigma_H + \sigma_p + \sigma_i \tag{1}$$

where:

- σ_v - the stress due to hull general bending in the vertical plane
- σ_H - the stress due to hull general bending in the horizontal plane
- σ_p - the stress due to local bending of the longitudinals under the external dynamic pressure p
- σ_i - the stress due to local bending of the longitudinals under the dynamic pressure p_i caused by the liquid cargo inside a tank

In the simplified fatigue strength calculations it is assumed that the long - term distribution of extreme values of σ, called further the dynamic amplitudes, is approximated by the two parameter Weibull distribution [3]. It means that the probability $Q(\sigma)$ of the exceedance of the stress amplitude level σ is determined by the formula:

$$Q(\sigma) = \exp\left[-\left(\frac{\sigma}{a}\right)^{\xi}\right] \tag{2}$$

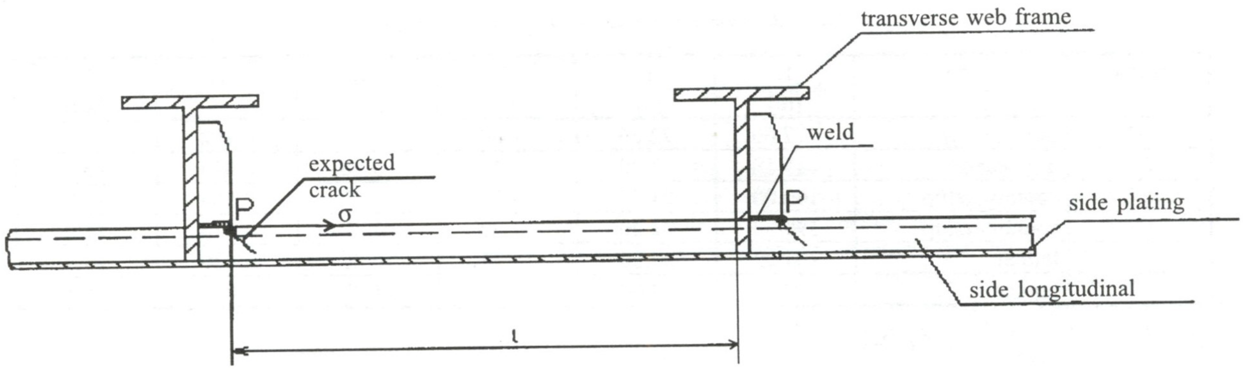

Fig. 1. Side longitudinal connection with transverse web frame

The value a can be determined if the value ξ (a, ξ, - parameters of Weibull's distribution) and a dynamic amplitude $\sigma = \sigma_R$, exceded with the probability $1/N_R$, are known (N_R - a natural number, e.g. 10^4).

Knowing $Q(\sigma)$, fatigue life of the welded connection shown in Fig. 1 can be assessed easily. To do this, the Miner - Palmgren rule can be applied [3] and a proper S–N curve should be chosen [2], [4].

The S-N curves are given in the form of the following simple equation:

$$N = \frac{K}{\Delta\sigma^m} \tag{3}$$

where:

- $\Delta\sigma$ - the stress range ($\Delta\sigma = 2\sigma$, except for the side longitudinals near the waterline level where $\Delta\sigma \approx \sigma$)
- K, m - S-N curve parameters

Applying the determined parameters and the Miner – Palmgren rule, the fatigue cumulative damage D can be expressed in the following form:

$$D = \frac{N_L}{K} \frac{(\Delta\sigma_R)^m}{(\ln N_R)^{m/\xi}} \Gamma\left(1 + \frac{m}{\xi}\right) \tag{4}$$

where:

- N_L - the number of stress cycles during the assumed period of ship life (usually of 20 years)
- $\Delta\sigma_R$ - the value of the stress range exceeded with the probability $1/N_R$
- Γ - the gamma function

The period of time L (in years) after which the fatigue cracks can occur is simply related to D as follows:

$$L = \frac{Y_L}{D} \tag{5}$$

where:

- Y_L - the number of years for which N_L was calculated (e.g. $Y_L = 20$)

The stress range $\Delta\sigma = \Delta\sigma_R$, exceeded with the assumed probability level $1/N_R$ has to be known to calculate the fatigue life. $\Delta\sigma$ is usually equal to 2σ; however σ is composed of a few components, see eq. (1).

The problem is that the dynamic amplitudes of the stresses σ_v, σ_H, σ_p and σ_i, determined by the rules of classification societies [1], are the maximum values, and they can not be added directly to calculate the amplitude σ (their extreme values do not occur at the same moment of time).

Polish Register of Shipping is currently preparing the fatigue strength requirements to be introduced into the Rules, by applying the concept of the so called partial stress range coefficients. According to the concept, the stresses in the longitudinal hull members are divided into:

- the global components, σ_v, σ_H and torsion stresses σ_T in the case of hulls with large deck openings, and
- the local components, σ_p, σ_i .

The dynamic amplitude of the global stress is first calculated. For the case shown in Fig.1, the global stress amplitude, σ_g, is calculated as follows:

$$\sigma_g = Max(\sigma_v, \sigma_H) + K_{VH}\, Min(\sigma_v, \sigma_H) \tag{6}$$

where:

- K_{VH} - the partial stress coefficient for combining the global stress components

In the next step, the dynamic amplitude of the local strees σ_l in the case of tankers or other ships carrying liquid cargo or ballast water, is calculated as follows:

$$\sigma_l = Max(\sigma_p, \sigma_i) + K_{pi}\, Min(\sigma_p, \sigma_i) \tag{7}$$

where:

- K_{pi} - the partial stress coefficient for combining the local stress components

In the third step, the global stress amplitude σ_g and the local stress amplitude σ_l are combined together to obtain the amplitude σ of the total stress as follows:

$$\sigma = Max(\sigma_g, \sigma_l) + K_{gl}\, Min(\sigma_g, \sigma_l) \tag{8}$$

where:

- K_{gl} - the partial stress coefficient for combining the global and local stress components

The values of K, ξ and N_L, to be applied in PRS Rules, are given below. The proposal for the K coefficients, ξ and N_L has been made on the basis of the calculations of long - term stress distributions, and of the so - called calibration studies where fatigue lives were calculated for hull members which cracked after known periods of time.

The long term response prediction for a ship or its structure is generally obtained by combining response in various sea states, taking into account sea state occurrence probability. There are different schemes of obtaining the long-term response predictions; and the results obtained from applying these schemes differ sometimes significantly. In this paper the approach presented in [5] has been applied.

RESULTS OF CALCULATIONS

The calculations of the long term stress distributions, (according to [5] ,[6]) were performed for six ships. The basic parameters of the ships are given in Tab. 1.

Tab.1. Basic parameters of the ships used in the calculations

Name	Type	L [m]	B [m]	T [m]	T_B [m]	C_B –	V [kn]
B1	bulk carrier	177.60	22.86	9.80	6.28	0.792	15.4
B2	bulk carrier	215.19	32.20	12.40	5.16	0.824	15.4
C1	containership	162.00	24.23	8.20	5.00	0.560	21.0
C2	containership	190.00	32.20	12.00	7.30	0.616	21.0
T1	tanker	284.62	48.06	15.33	10.80	0.820	16.3
T2	tanker	319.00	58.00	19.85	14.00	0.810	15.5

L - length, B - breadth, T - design draught, T_B - ballast draught, C_B - block coefficient, V - design speed

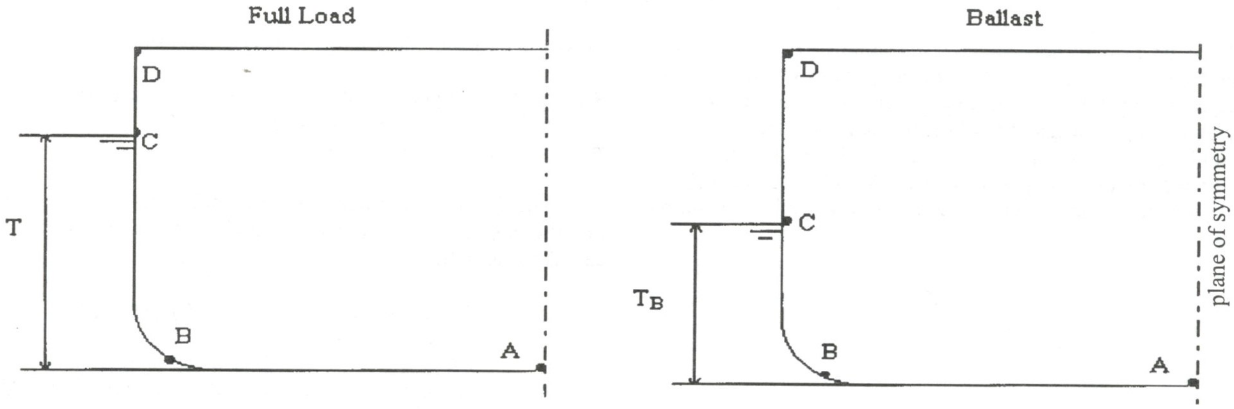

Fig. 2. Points selected for calculation of σ

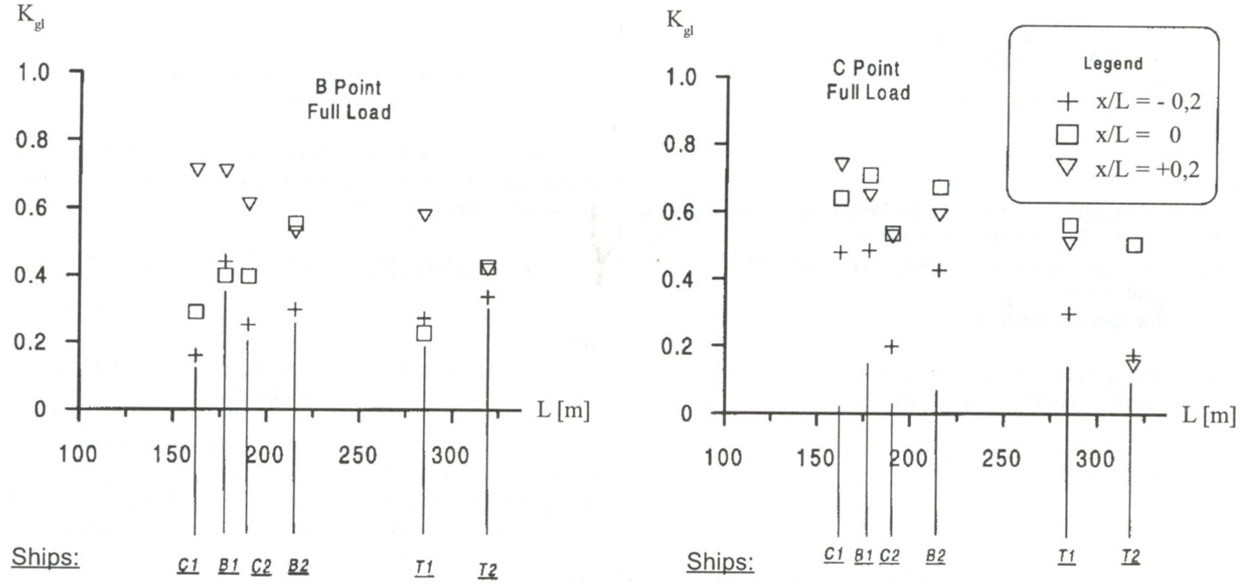

Fig. 3. Values of the K_{gl} coefficient, for $\sigma_i = 0$

The stresses σ were calculated at the points *P* (see Fig. 1) of the longitudinals placed at the points A, B, C, D of the transverse cross–sections shown in Fig. 2, the coordinates of which are equal to: $x = -0.2\,L$, $x = 0$ and $x = 0.2\,L$. *X* axis is directed along the ship with $x = 0$ at the midship cross–section. At the points D only the components σ_v and σ_H were considered.

The examplary *K* coefficients values calculated from (6), (7) and (8) for values of dynamic amplitudes σ, exceeded with probability 10^{-4}, are shown in Fig. 3.

SIMPLE FORMULAE FOR FATIGUE STRENGTH ASSESSMENT

The results of the analysis of the *K* values, determined for six ships of the most common types with the length varying from 160 m to 320 m (see Tab. 1), were used to evaluate the K_{gl} approximate values as follows:

$$\begin{aligned} &K_{gl} \approx 0.8 \;-\; \text{at the bottom, at plane of symmetry (point A in Fig. 2)} \\ &K_{gl} \approx 0.7 \;-\; \text{at the bilge (point B in Fig. 2)} \\ &K_{gl} \approx 0.6 \;-\; \text{at the waterline.} \end{aligned} \tag{9}$$

Linear interpolation can be applied between the values.

They are valid for $\sigma_i = 0$ and $\sigma_i \neq 0$, and approximately equal to the K_{gl} maximum values calculated directly.

The following values of the coefficients K_{VH} and K_{pi}, are proposed, irrespective of ship type and position of the longitudinal member within the hull:

$$K_{VH} = 0.3 \qquad K_{pi} = 0.4 \tag{10}$$

The number of cycles N_L for the six considered ships, calculated with the use of the total stress σ given by (1), was also analysed. It can be observed that the longer the ship the smaller value of N_L, and this is not clearly dependent on the ship type.

The following approximation has been proposed for calculations of the fatigue damage D (assuming that the typical ship operates for 20 years and spends about 80% of time at sea):

$$\begin{aligned} N_L &= 5\cdot 10^7 \quad \text{for} \quad L = 150\ \text{m} \\ N_L &= 4\cdot 10^7 \quad \text{for} \quad L = 350\ \text{m} \end{aligned} \tag{11}$$

Linear interpolation between the values given by (11) has been proposed.

The fatigue life of the connections shown in Fig. 1 was calculated for different values of ξ both on the basis of the direct calculations of σ and approximate distribution given by (2). The values ξ that minimize the difference between the values of D, obtained by applying the two methods, have been considered as giving the best approximation in the form of eq. (2). It has been observed that the values of ξ depend on the type and length of ship.

They are greater for the containerships with L<220 m than for bulk carriers and tankers.

It means that the values of ξ are influenced by the ship speed and block coefficient.

The following values of ξ have been proposed to be applied to the fatigue strength assessments:

- for bulk carriers and tankers:

$$\begin{aligned} \xi &= 1.2 \quad \text{for} \quad L = 150\ \text{m} \\ \xi &= 0.95 \quad \text{for} \quad L = 350\ \text{m} \end{aligned} \tag{12}$$

- for containerships:

$$\begin{aligned} \xi &= 1.25 \quad \text{for} \quad L = 150\ \text{m} \\ \xi &= 1.05 \quad \text{for} \quad L = 250\ \text{m} \end{aligned}$$

Linear interpolation between the values can be applied.

The values of M_V, M_H, p and a_v, necessary for calculating the dynamic amplitudes of σ_v, σ_H, σ_p and σ_i can be determined from the Rules [1].

(M_V, M_H – wave bending moments in vertical and horizontal plane, respectively , p – external dynamic pressure, a_v - vertical acceleration of the ship).

The formulae (9) to (12) can be used as the Rules criteria for determining the fatigue life of longitudinal hull members. They were used to check the fatigue life of the side longitudinal connections with transverse frames (see Fig. 1) of a 258 000 dwt tanker. Fatigue cracks were discovered on the ship at some connections after 3 years of operation only. The ship was built of high tensile steel with the yield point of 360 MPa.

The nominal stress approach in the form of eq. (1) and the F2 S–N curve, recommended in [4] for this type of connections, was applied.

The calculated fatigue life was 3.2 years, which well complies with the fatigue life observed on the ship.

Further calibration studies are recommended to check thoroughly the accuracy of the proposed formulae for K – coefficients, N_L and ξ.

CONCLUSIONS

The aim of the research project was to work out the Rules requirements by elaborating the method that makes effective fatigue strength assessments of longitudinal hull members possible.

The proposed formulae provide the approximate values of the results of very complex long – term prediction of the stresses in the members and therefore they are rather rough. In some cases too conservative results of the fatigue strength may be obtained.

The Rules recommend, in some cases, that direct calculations of the fatigue strength, based on long – term prediction of the dynamic stresses, should be performed. Such a method is strongly recommended e.g. for checking the fatigue strength of hatch corners in open type ships (container ships, multi–purpose vessels).

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Current *reports*

TECHNICAL UNIVERSITY OF SZCZECIN
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Modular design system of shipboard food store rooms

The Refrigeration Department of the Institute proposed, in contrast to the traditional design of one common store space, to combine the shipboard refrigerated food store from several separate refrigerated rooms connected by a common installation with the microprocessor control of individual refrigerating units of the rooms. The modular design system is based on the idea of assembling just on shipboard several, fabricated in advance, complete refrigerated space modules (each of them equipped with an individual refrigerating unit comprising an evaporator, compressor and condenser).

Bogusław Zakrzewski, D.Sc., the author of the proposal, underlines the following advantages of such design:

- lower energy consumption due to possible load distribution of individual rooms
- smaller volume of the entire refrigerated space
- smaller loss of the refrigerant out of the installation
- easiness of repair (without any need of stopping work of the entire refrigerated store)
- higher operation reliability of the food store
- more freedom in arranging refrigerated rooms within ship's space.