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Fatigue strength of knuckle joints - a key parameter in ship design

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Abstract

Structural integrity of knuckle joints in inner hull for double hull oil tankers or membrane LNG carriers or at connection between the inner bottom and hopper tank sloping plating for bulk carriers is of primary importance taking into account the consequences on ship safety of initiation and propagation of cracks in those areas.

After showing typical cracks observed on knuckle joints of bulk carriers, the paper presents results of comparative fatigue analyses performed for various structural arrangements of welded knuckle joints. The calculations carried out according to the fatigue procedure developed by Bureau Veritas are based upon the use of S-N curves and Miner cumulative damage rule.

In 1995, Bureau Veritas launched a new Classification System (VeriSTAR) integrating design analysis and ship management. In particular, taking into account that fatigue and corrosion are the main causes of damage observed on ship structures, VeriSTAR includes a special module enabling to assess, on a systematic basis, the fatigue strength of main structural details. Determination of the fatigue life may be carried out according to two different approaches using either the results of fine mesh FEM local analyses or stress concentration factors pre-determined for typical structural details and stored in the system database. In its second part, the paper gives an overall view of the method considered in VeriSTAR to assess the fatigue strength of structural details.

In conclusion, recommendations are given to improve the fatigue life of knuckle joints taking into account experience gathered on ships in service and results of fatigue analyses performed by Bureau Veritas.

1 Introduction

1.1 There are many factors which affect the structural behaviour of welded or radiused knuckle joints; they are :

- overall scantlings of the structure and spacing of primary members,
- local design of the connection and, in particular, distance or eccentricity of the knuckle joint to the closest primary girder as shown in Figure 1,
- corner scallops for ease of fabrication,

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- workmanship,
- sea conditions,
- environmental conditions,
- operational conditions,

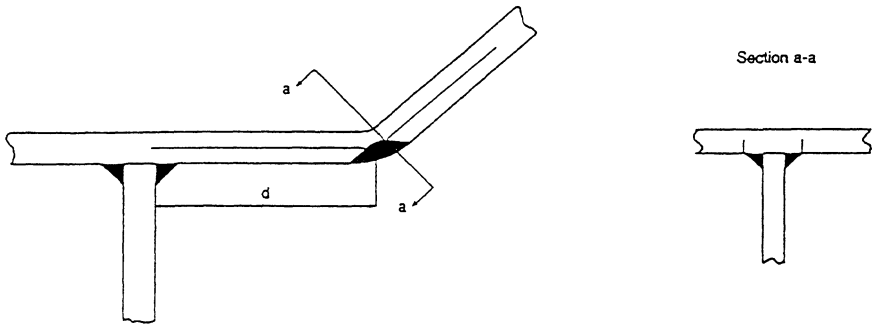

Figure 1

1.2 The local stress variations in the vicinity of knuckle joints due to fluctuating loads may lead to the initiation and propagation of fatigue fractures. Figure 2 shows typical examples of cracks occurring at connection of inner bottom plating to hopper plating of bulk carriers [1] which may be initiated :

- along the knuckle weld for welded knuckle joints,
- at the toe of fillet welds connecting transverse webs to the inner hull plating and propagating through the inner hull plating (see section a-a in Figure 1),
- at the edge of scallops and extending along the fillet weld without damage to the inner hull plating, as shown in Figure 3,
- at the edge of scallops and extending in the web plate.

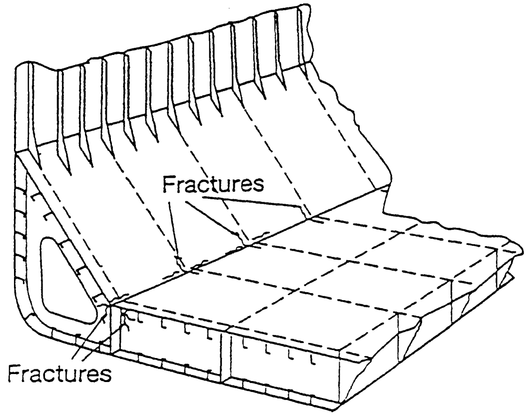

Figure 2

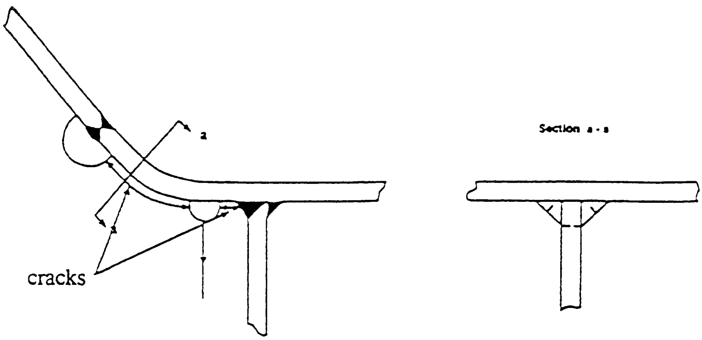

Figure 3

2. Comparative fatigue strength of welded knuckle joints

2.1 General

2.1.1 In order to determine the best structural arrangement for this particular structural connection, Bureau Veritas decided to carry out comparative calculations of the fatigue behaviour for typical welded knuckle joints of membrane LNG carriers :

- without eccentricity, and
- with eccentricity varying from 40 to 120 mm.

Where the eccentricity is nil, two different designs were examined, as shown in Figure 4 :

- detail No 1 with scarfing of the inner hull plating within the hopper tank structure,
- detail No 2 without scarfing.

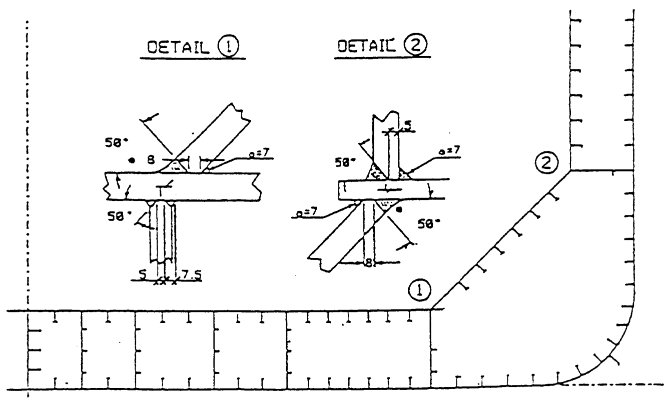

Figure 4

2.1.2 Calculations were carried out according to the procedure described in the Bureau Veritas Guidance Note NI 393 [2].

As for other limit states, e.g., yielding, buckling or ultimate strength, assessment of the fatigue reliability of structural details necessitates to determine :

- the demand or loads and stresses to which the ship's structure may be subjected during her life,
- the capacity of the structure which is generally represented by S-N curves giving the relationship between the stress range and the number of cycles to failure,

and to select a strength criterion above which the structure is considered as failed.

Therefore, the procedure described in NI 393 and summarized in Figure 5 is divided in three main steps :

- determination of the long term distribution of stresses,
- determination of the fatigue capacity,
- application of the Miner cumulative damage principle.

Procedure for Assessment of the Fatigue Strength

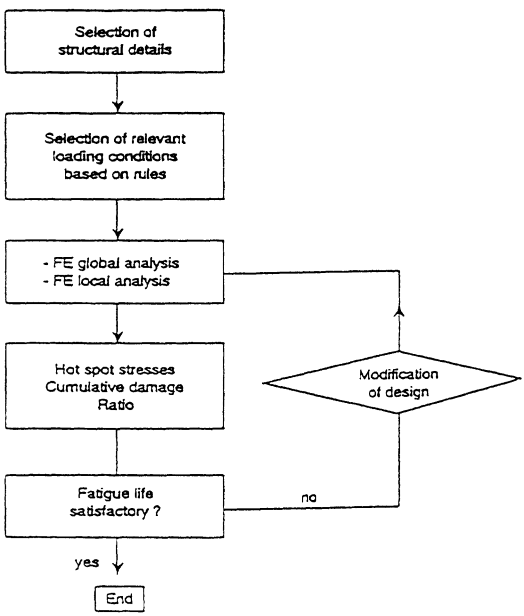

Figure 5

2.2 Loading conditions

2.2.1 The construction of the long term distribution of stresses necessitates generally to carry out a structural "spectral" analysis. Such a time consuming process cannot be used as a standard procedure and Bureau Veritas decided to develop a simplified procedure based on the assumption that the probability density function of the long term distribution of stresses may be represented by a two-parameter Weibull distribution, which enables to use the rule loads to determine the long term distribution of stresses.

2.2.2 As it may be observed from examination of the loading manual for various types of ships, many different loading conditions occur during the ship's life. However, many ships are navigating most of their life with standard loading conditions, mainly full load and ballast. These two loading conditions have been selected in our analysis and considered as equi-probable.

2.2.3 According to Bureau Veritas Rules, for each of these two loading conditions, two basic cases which combine the various dynamic effects of the environment on the hull structure, are considered to determine the design loads; they are :

- head condition, and
- beam sea condition.

Finally, 28 different loading cases have been examined for calculation of the stress ranges.

2.3 Long term distribution of stresses

2.3.1 Since our procedure is based on the *notch stress* approach, ($\sigma_\ell = K_w \, \sigma_{\text{hot spot}}$), determination of stresses is generally carried out in two steps :

a) coarse mesh 3D FEM stress analysis to obtain the nodal displacements which will be used as boundary conditions in the fine mesh analyses,
b) fine mesh stress analyses for determination of the *hot spot stresses*.

2.3.2 Fine mesh analysis of the knuckle joints
The calculated fatigue life is very sensitive to the size of elements considered to determine the *hot spot stresses*. Our procedure follows the general modelling principles recommended by the International Institute of Welding (IIW) [3]. Therefore, the 3D FEM model representing the hopper tank structure was made of three levels of refinement and Figure 6 shows a typical fine mesh model of the lower knuckle.

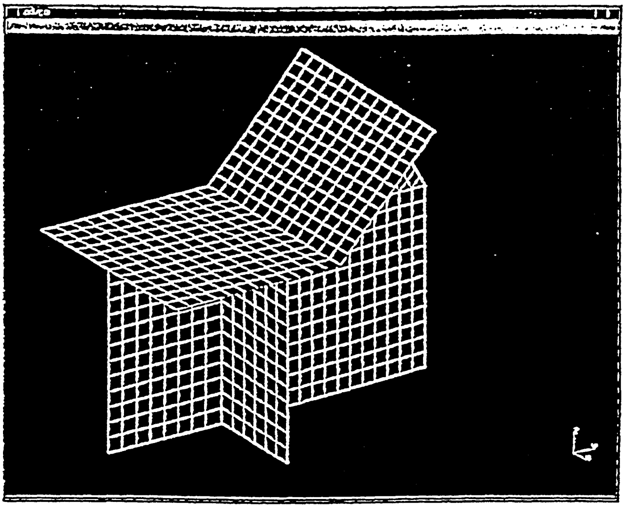

Figure 6

2.3.3 Hot spot stresses

For each relevant loading condition, i.e., full load and ballast, the *hot spot stresses* considered for calculation of the fatigue life were obtained by linear interpolation of principal stresses at the centroid of inner hull elements closest to the knuckle, forming an angle of less than 45 degrees with the normal to the longitudinal weld joint. The following two tables give, versus eccentricity, the maximum hot spot principal stress ranges, in MPa, for lower and upper knuckles.

Table 1 - Lower knuckle - Hot spot stresses

Eccentricity	d = 0	d = 40 mm	d = 80 mm	d = 120 mm
Full load	144	400	544	587
Ballast	124	376	513	551

Table 2 - Upper knuckle - Hot spot stresses

Eccentricity	d = 0	d = 40 mm	d = 80 mm	d = 120 mm
Full load	279	323	449	503
Ballast	233	358	408	543

2.4 Comparative fatigue life of the knuckle joints

2.4.1 The Miner cumulative damage rule is used for determination of the fatigue life of knuckle joints. Since the long term distribution of stresses is assumed to follow a Weibull distribution, the cumulative damage ratio may be expressed in a closed form equation :

$$D = \frac{N_L}{K} \frac{S_R^m}{(\ln N_R)^{m/\xi}} \mu\, \Gamma\left(1+\frac{m}{\xi}\right) \qquad (1)$$

where S_R is the *local stress* range, i.e., taking into account stress concentrations due to the effects of the structural geometry as well as the presence of welds :

$$S_R = K_w\, S_{hot\ spot} \qquad (2)$$

(K_w is the stress concentration factor due to weld geometry taken as 1,4 in our study).

2.4.2 Selection of the S-N curve

Since *local stresses* take into account stress concentrations due to the effects of structural geometry (K_g) and welds (K_w), the corresponding design S-N curve is to be such that $K = K_g\, K_w = 1$. The S-N curve considered to perform this comparative analysis is derived from data provided by the U.K Health and Safety Executive (HSE) for non tubular joints [4] and corresponds to the as-rolled condition with no flame-cut edges, i.e., modified B curve with a slope of 3.

Experimental S-N curves are generally mean S-N curves determined in air environment and have to be corrected to take into account the following :

- probability of failure taken generally as 2,5 per cent,
- influence of static and residual stresses,
- effect of compressive stresses,
- workmanship,
- influence of the material,
- influence of the environment,
- Haibach effect (change in slope for $N = 5\ 10^6$ cycles).

2.4.3 Comparative fatigue life of the knuckle joints

Tables 3 and 4 give the comparative fatigue lives, in years, for the lower and upper knuckle joints versus the eccentricity. It is worth reminding that calculations were carried out without corner scallops and with inner bottom plating prolonged by horizontal brackets within the hopper tank structure.

Table 3 - Lower knuckle - Comparative fatigue life

Eccentricity	d = 0	d = 40	d = 80	d = 120
Damage ratio	0,22	2,43	5,08	6,10
Fatigue life	91	8,2	3,9	3,3

Table 4 - Upper knuckle - Comparative fatigue life

Eccentricity	d = 0	d = 40	d = 80	d = 120
Damage ratio	0,97	1,80	3,06	4,95
Fatigue life	20,6	11,1	6,5	4,0

2.4.4 As indicated in 2.3.3, calculations were carried out considering the maximum principal stresses in the inner hull plating forming an angle of less than 45 degrees with the normal to the longitudinal weld joint. For determination of the actual fatigue life of this type of structural detail, *hot spot stresses* should be calculated at all possible crack locations, in particular in transverse webs at the edge of scallops where fitted.

3. VeriSTAR

3.1 In 1995, Bureau Veritas presented a new approach to classification integrating design analysis and ship management [5]. The VeriSTAR System comprises two main functions :

- review of the structural design, and
- continuous hull monitoring from results of hull surveys and thickness measurements.

In particular, VeriSTAR enables to assess the fatigue strength of typical structural details by calculating the *hot spot stresses* according to two different methods :

- determination of *nominal stresses* and selection of the appropriate geometric stress concentration factor for the type of connection considered,
- use of fine mesh models.

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3.2 Nominal stress approach

Nominal stress is a general stress in a structural component calculated by beam theory or using a coarse mesh FEM stress analysis. This approach is particularly well fitted for standard structural details such as connections of longitudinal stiffeners to webs of transverse primary members [6], as shown in Figure 7. In that case, the *hot spot stress* may be given by :

$$\sigma_g = K_{ga}\,\sigma_{na} + K_{gb}\,\sigma_{nb} \tag{3}$$

$$\sigma_{nb} = \sigma_b + \frac{6\,E\,I_v\,\delta}{w\,\ell^2} \tag{4}$$

$$\sigma_b = \frac{p_{rs}\,\ell^2}{12\,w}\left\{1+\frac{t_f\,(a^2-b^2)}{2\,w_B}\,\frac{1}{1-\dfrac{b}{b_f}\left(1+\dfrac{w_B}{w_A}\right)}\right\} \tag{5}$$

The fatigue life of longitudinal connections to transverse webs is calculated on a systematic basis using geometric stress concentration factors K_{ga} and K_{gb} obtained from results of systematic FEM stress analyses carried out for various structural configurations.

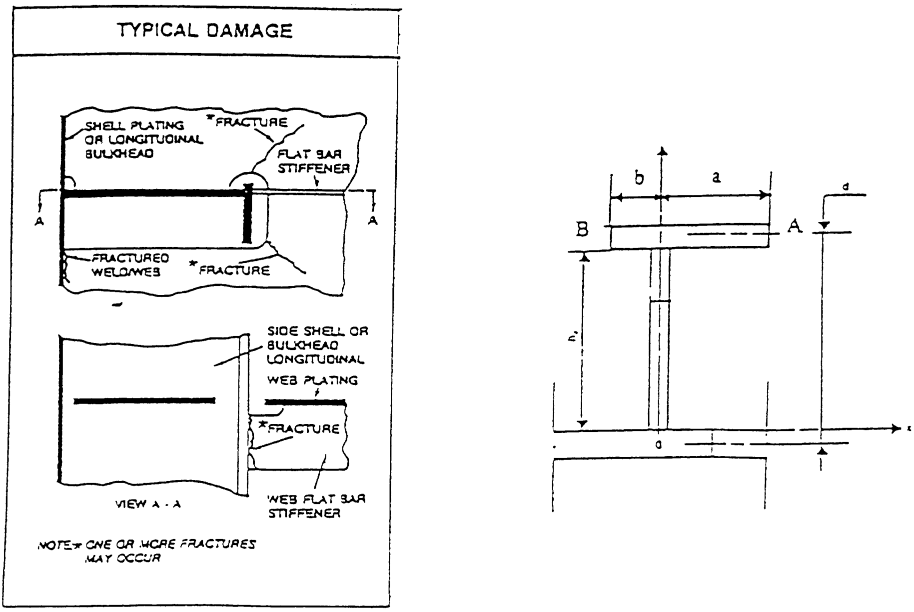
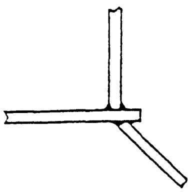
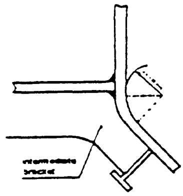
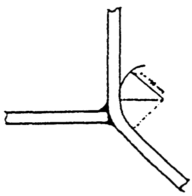
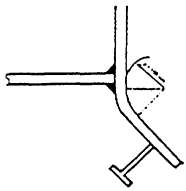
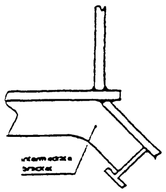
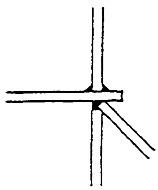
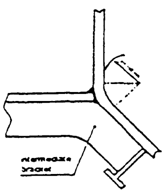


Figure 7

VeriSTAR enables to calculate the fatigue life of knuckle joints considering *nominal stresses* as obtained from coarse mesh stress analyses and longitudinal and transverse stress concentration factors (K_{gx} and K_{gy}) calculated for various configurations of knuckle joints as given in Table 5.

Table 5 - Knuckle joints - Geometric stress concentration factors

Configuration	K_g	Configuration	K_g
	$K_{gx} = 3,5$ $K_{gy} = 3,9$		$K_{gx} = 3,9$ $K_{gy} = 3,9$
	$K_{gx} = 3,7$ $K_{gy} = 3,4$		$K_{gx} = 8,8$ $K_{gy} = 4,5$
	$K_{gx} =3,2$ $K_{gy} = 3,6$		$K_{gx} = 1,8$ $K_{gy} = 2,4$
	$K_{gx} = 3,5$ $K_{gy} = 3,2$		

3.3 Fine mesh analyses

VeriSTAR includes also the possibility to calculate, for particular structural details, *hot spot stresses* from 2D or 3D FEM fine mesh models complying with the modelling principles recommended by IIW.

4. Conclusions

Though the fatigue lives given in Tables 3 and 4 consider only one particular mode of crack propagation, this analysis enables to compare the fatigue strength of various configurations of welded knuckle joints and to draw essential conclusions on their design :

- the fatigue life decreases with the eccentricity,
- the optimum fatigue life is obtained for no eccentricity with scarfing of the inner hull into the hopper tank structure,

Moreover, additional calculations have shown that intermediate brackets fitted at mid-span between transverse webs have a minor influence on the fatigue life while scallops increase the stress concentration factor by 40 per cent.

These conclusions which are also applicable to radiused knuckle joints, are introduced in our Rules which recommend, depending on the type of ship :

- to limit the eccentricity to 40 mm for bulk carriers and 10 mm for membrane LNG Carriers, including construction tolerances,
- to prolonge, for welded constructions, the inner bottom plating within the hopper tank structure,
- to increase locally, where necessary, the thickness of the inner hull plating depending on the eccentricity,
- to avoid scallops in transverse webs of topside and hopper tanks,
- to improve the weld geometry.

References

[1] Guidelines for Surveys, Assessment and Repair of Hull Structures, International Association of Classification Societies, 1994.
[2] Assessment of the Fatigue Strength of Welded Ship Structures, Bureau Veritas NI 393, September 1994.
[3] Recommendations on Fatigue of Welded Components, International Institute of Welding, April 1996.
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[6] Guidance Manual for the Inspection and Condition Assessment Tanker Structures; Tanker Structure Co-operative Forum, 1986.