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Fatigue Strength Assessment of a Longitudinal Hatch Coaming in a 3800 TEU Containership by ABS Dynamic Approach

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Abstract

Fatigue strength assessment procedures have been implemented in the ship design rules by many classification societies. However, a large variation in the details of the different approaches exists in practically all aspects including load history assessment, stress evaluation and fatigue strength assessment. In order to assess the influences of these variations on the prediction of fatigue lives, a comparative study is organized by the ISSC Committee III.2 Fatigue and Fracture. A pad detail on the top of longitudinal hatch coaming of a Panamax container vessel is selected for fatigue calculation. The work described in this paper is one set of results of this comparative study in which the ABS dynamic approach is applied. Through this analysis the following conclusions can be drawn. (1) With the original ABS approach, the fatigue life of this pad detail is very low, only 2.398 years. (2) The treatment of the stillwater bending moment in the ABS approach might be a source of conservatism. If the influence of stillwater bending moment is ignored, then the fatigue life for this pad detail is 7.036 years. (3) The difference between the nominal stress approach and the hot spot stress approach for this pad detail is about 26%.

Keywords : Fatigue strength assessment, Containership, Longitudinal hatch coaming, Stress concentration factor, Nominal stress approach, Hot spot stress approach

1 Introduction

Fatigue may be defined as a process of cycle by cycle accumulation of damage in a material undergoing fluctuating stresses and strains[Almar-Naess, 1985]. A significant feature of fatigue is that the load is not large enough to cause immediate failure. Instead, failure occurs after a certain number of load fluctuations have been experienced, i.e. after the accumulated damage has reached a critical level.

Fatigue cracking in ships has been a serious problem for many years[Munse et al., 1983]. For example, fatigue cracking was observed to occur repeatedly at the forward hatch corners of the SL-7 containerships during the early part of their service[Chen et al., 1986]. However, the problem of fatigue has not received adequate consideration by the ship designers for a long time[Faulkner, 1991]. One reason for this is that fatigue cracks do not generally constitute a safety problem and their main consequence is the continual repair required and the associated increase in maintenance costs.

Recently there is increasing attention paid to formal fatigue strength assessment by classification societies. The reasons are[Moan et al., 1997]: (1) the higher utilization of structures due to optimization by refined analyses; (2) the intensified application of higher-tensile steel as well as simplified structural details and fabrication techniques; (3) the increasing number of aging structures, with lacking maintenance in several cases; (4) the increased public sensitivity with respect to protection of human beings and the environment. Now most of the main classification societies have included in their rules the explicit design criteria related to ship fatigue strength[ABS, 1996][BV, 1994][DNV, 1995][GL, 1997][KR, 1995][LR, 1996][NK, 1996][RINA, 1995]. Most of these procedures are aimed at a simplified fatigue strength assessment. A typical procedure can be described as follows[Moan et al., 1997]:

—- Rule-based approach is used with Weibull two-parameter distribution. The spectral approach is recommended for more accurate fatigue assessment although guidance is not always given.

—- Simplified procedures are given for calculation of stresses due to hull girder global and local deformations. Partial FE-models are applied. Correlation factors are defined for calculating combined stresses.

—- The design life is taken as 20 years and safety factors are not used. Fatigue model is based on nominal or hot spot stress approach. Fatigue failure is defined by cumulative damage reaching one when calculated using the Palmgren-Miner linear summation and lower bound S-N curves.

A large variation in the details of the different approaches exists in practically all aspects including load history assessment, stress evaluation and fatigue strength assessment. In order to assess the influences of these variations on the prediction of fatigue lives, a comparative study is organized by the ISSC Committee III.2 Fatigue and Fracture. A pad detail on the top of longitudinal hatch coaming of a Panamax container vessel is selected for fatigue calculation. The work described in this paper is one set of results of this comparative study in which the ABS dynamic approach[ABS, 1996] is applied. As far as the fatigue strength assessment of container ships is concerned, many other relevant references are available[Xue et al., 1994][Cui, 1996][Chen et al., 1997][Baczynski et al., 1997][Cui & He, 1998].

2 Fatigue Strength Assessment Method and Determination of Permissible Stress Range

In ABS dynamic approach[ABS, 1996], the calculation of the fatigue damage of ship structures subjected to variable amplitude loading is made by using S-N curves together with Palmgren-Miner damage accumulation law. The key assumptions employed are listed below:

(1) A linear cumulative damage model (i.e. Palmgren-Miner rule), Eq.(1), has been used in connection with the S-N curve, Eq.(2).

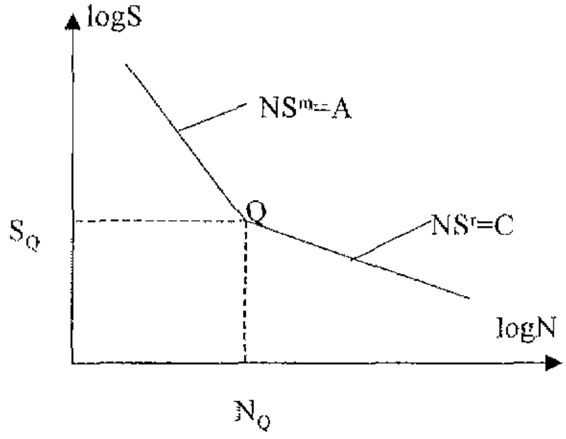

Figure 1. Two segments S-N curve.

$$D = N_T \int_0^{\infty} \frac{f(S)}{N(S)} dS \tag{1}$$

$$N(S) = \begin{cases} AS^{-m} & S > S_Q \\ CS^{-r} & S \leq S_Q \end{cases} \tag{2}$$

where

N_T - the total number of cycles in the service life
S - stress range, MPa
N - number of cycles at failure
S_Q - stress range at the knee point of $N_Q = 10^7$ in the S-N curve, see Figure 1
A, C - fatigue coefficients
m, r - fatigue exponents
$f(s)$ - probability density function for the long-term stress range distribution, Eq.(3).

(2) Cyclic stresses due to the wave-induced loads have been used and the effect of mean stress has been ignored.

(3) The target design life of the vessel is taken to be 20 years, i.e. $N_T = 5 \times 10^7$.

(4) The long-term stress ranges on a detail can be characterized by using a modified Weibull probability distribution parameter (k), Eq.(3).

$$f(S) = (\frac{k}{w})(\frac{s}{w})^{k-1} \exp\left[(-\frac{s}{w})\right]^k \tag{3}$$

where

k - the shape parameter
w - the scale parameter

(5) Structural details are classified according to UK DEn S-N curves[Dept. of Energy, UK, 1990].

(6) Simple nominal stress (e.g. determined by P/A and M/SM) is the basis of fatigue assessment rather than more localized peak stress in way of weld for details subject to a simple load pattern (one major member is loaded). Where the loading or geometry is too complex for a simple classification, the stress concentration factor is used.

By defining a "design"stress range, S_0, as

$$P(S > S_0) = 1/N_T \tag{4}$$

then the scale parameter w can be written as

$$w = S_0[\ln(N_T)]^{-1/k} \tag{5}$$

By substituting Eq.(2) and Eq.(3) into Eq.(1) and also replacing w by Eq.(5), the following equation can be obtained[Cui, 1995]:

$$D = \frac{N_T}{A}\frac{S_0^m}{(\ln N_T)^{m/k}}\Gamma(1 + m/k, Z_Q) + \frac{N_T}{C}\frac{S_0^r}{(\ln N_T)^{r/k}}\Gamma_0(1 + r/k, Z_Q) \tag{6}$$

where

$$Z_Q = \left(\frac{S_Q}{S_0}\right)^k \ln N_T \tag{7}$$

$$\Gamma(x, z) = \int_z^{\infty} u^{x-1}e^{-u}du \tag{8}$$

$$\Gamma_0(x, z) = \int_0^{z} u^{x-1}e^{-u}du \tag{9}$$

If we assume that the cumulative damage at failure for a particular structural detail is Δ and if N_T, k, Δ, A, C, m, r are known, then by equating $D = \Delta$, we can obtain a value of the "design" stress range. This design stress range is called the permissible stress range. The permissible stress range is a function of:

(1) the service life, usually 20 years;
(2) the modified Weibull shape parameter k of the long-term stress range distribution;
(3) the cumulative damage index at failure, usually $\Delta = 1$;
(4) the S-N curve for a particular structural detail (A, C, m, r).

The above procedure for predicting the permissible stress range is coded with Fortran 90 in a program named FSAP (Fatigue Strength Assessment Program). For the DEn S-N curves[Dept. of Energy, UK, 1990] used in the ABS approach, the relevant parameters are given in Table 1 and the predicted permissible stress range when $\Delta = 1$ and $N_T = 5 \times 10^7$ are given in Table 2. The values in Table 2 are the same as that given in the ABS approach[ABS, 1996] which validate the program FSAP.

Table 1. Fatigue strength parameters of DEn Basic S-N curves.

	class			
parameter	B	C	D	E
A	1.013×10^{15}	4.227×10^{13}	1.519×10^{12}	1.035×10^{12}
C	1.013×10^{17}	2.926×10^{16}	4.239×10^{15}	2.300×10^{15}
m	4.0	3.5	3.0	3.0
r	5.0	5.0	5.0	5.0
N_Q	10^7	10^7	10^7	10^7
S_Q(MPa)	100.32	78.19	53.36	46.95
S_R(MPa)*	150.02	123.84	91.24	80.29
	class			
parameter	F	F_2	G	W
A	6.315×10^{11}	4.307×10^{11}	2.477×10^{11}	1.574×10^{11}
C	9.975×10^{14}	5.278×10^{14}	2.138×10^{14}	1.016×10^{14}
m	3.0	3.0	3.0	3.0
r	5.0	5.0	5.0	5.0
N_Q	10^7	10^7	10^7	10^7
S_Q(MPa)	39.82	34.30	29.15	25.06
S_R(MPa)*	68.09	59.94	49.85	42.85

*S_R - reference fatigue strength at $N = 2 \times 10^6$ stress cycles.

Table 2. Permissible stress ranges for each class of the joints(MPa).

	class							
k	B	C	D	E	F	F_2	G	W
0.7	905.02*	776.76	586.73	516.81	438.15	385.70	320.95	276.05
0.8	744.53	626.83	463.54	408.40	346.20	304.76	253.65	218.18
0.9	629.95	522.82	380.58	335.40	284.29	250.26	208.32	179.26
1.0	545.28	447.61	321.95	283.79	240.52	211.74	176.28	151.67

* The permissible stress range cannot be taken greater than two times the specified minimum tensile strength of the material.

If we know that the "design" stress range is S_0 for the service life of 20 years which is denoted as N_{T0}, then the predicted fatigue life N_T can be obtained from the following equations:

$$\Delta = \frac{N_T}{A} W^m \Gamma(1 + m/k) + \frac{N_T}{C} W^r \Gamma_0(1 + r/k, Z_Q) - \frac{N_T}{A} W^m \Gamma_0(1 + m/k, Z_Q) \quad (10)$$

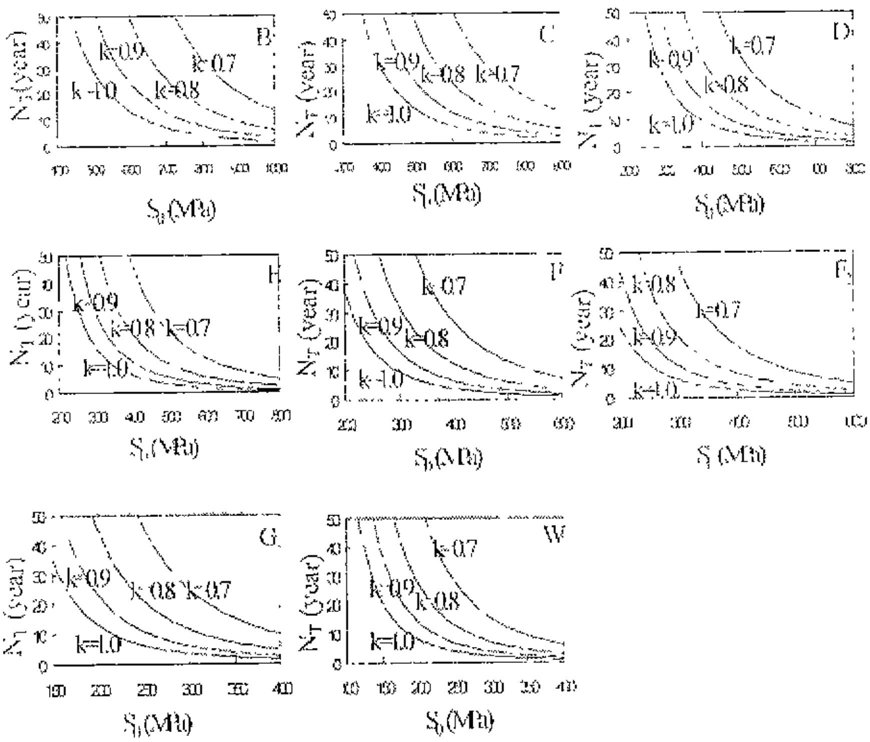

Figure 2. Relations between N_T and S_0.

where

$$W = \frac{S_0}{(\ln N_{T0})^{1/k}} \tag{11}$$

$$Z_Q = \left(\frac{S_Q}{W}\right)^k \tag{12}$$

If $\Delta = 1$, $N_{T0} = 5.0 \times 10^7$, (A, C, m, r, S_Q) are known for a given joint, then for a given value of k, N_T is a function of S_0. The relations between N_T and S_0 are shown in Figure 2 for DEn basic S-N curves.

3 Description of the Detail for Comparative Study

In order to make an interesting comparative study, a pad detail on the top of longitudinal hatch coaming of a Panamax container vessel is selected for fatigue calculation. The pad detail is shown in Figure 3 and it is assumed that the pad detail is located at midship, midway between the fore and aft ends of transverse coaming. The effects of torsional loading and warping stresses can then be disregarded. It should also be pointed out that the size of welding is specified by the throat thickness, not the leg length.

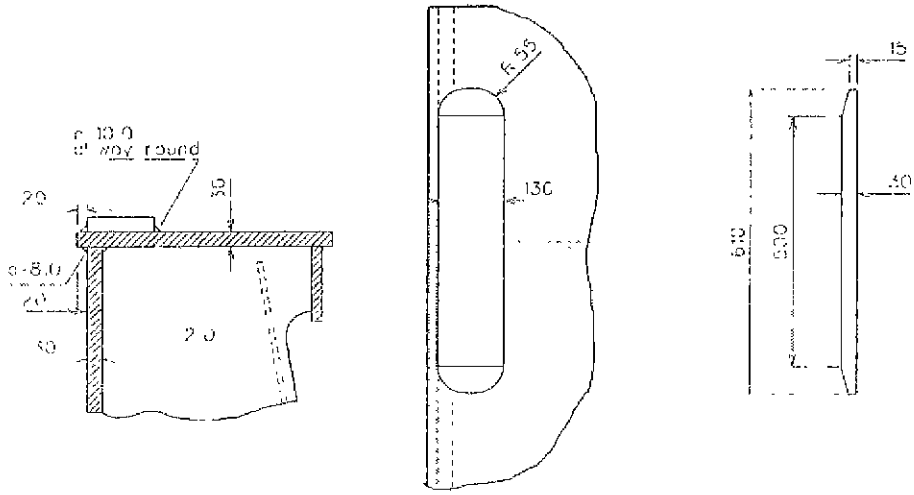

Figure 3. Pad detail on top of longitudinal hatch coaming.

The container vessel has the following principal particulars:

Length between perpendiculars, L_{pp}	242.000 m
Rule length, $L = 0.97 \times L_{wl}$	234.740 m
Breadth moulded, B	32.250 m
Depth moulded, D	21.300 m
Draught moulded, T	14.000 m
Block coefficient, C_b	0.670
Maximum service speed, V_d	24.000 Kn
Frame spacing	3.144 m
Max still water bending moment	2060.100 $\mathrm{MN} \cdot \mathrm{m}$
Capacity	3800 TEU

The longitudinal elements in midship section is given in Figure 4. The present vessel has high tensile steel, HT36, in the upper part, noted AH and EH, and the rest of the steel is mild steel, noted A.

4 Determination of Fatigue Loads and Design Stress Ranges

For this pad detail, only the global hull girder bending moments are relevant. Furthermore, as mentioned before, the effects of torsional loading and warping stresses can also be neglected due to its midway location between the fore and aft ends of transverse coaming. According to the ABS approach[ABS, 1996], twelve combined load cases given in Table 3 are used to assess the fatigue strength of structural joints in a containership.

For each loading case, the nominal stress at the pad location can be calculated by the following

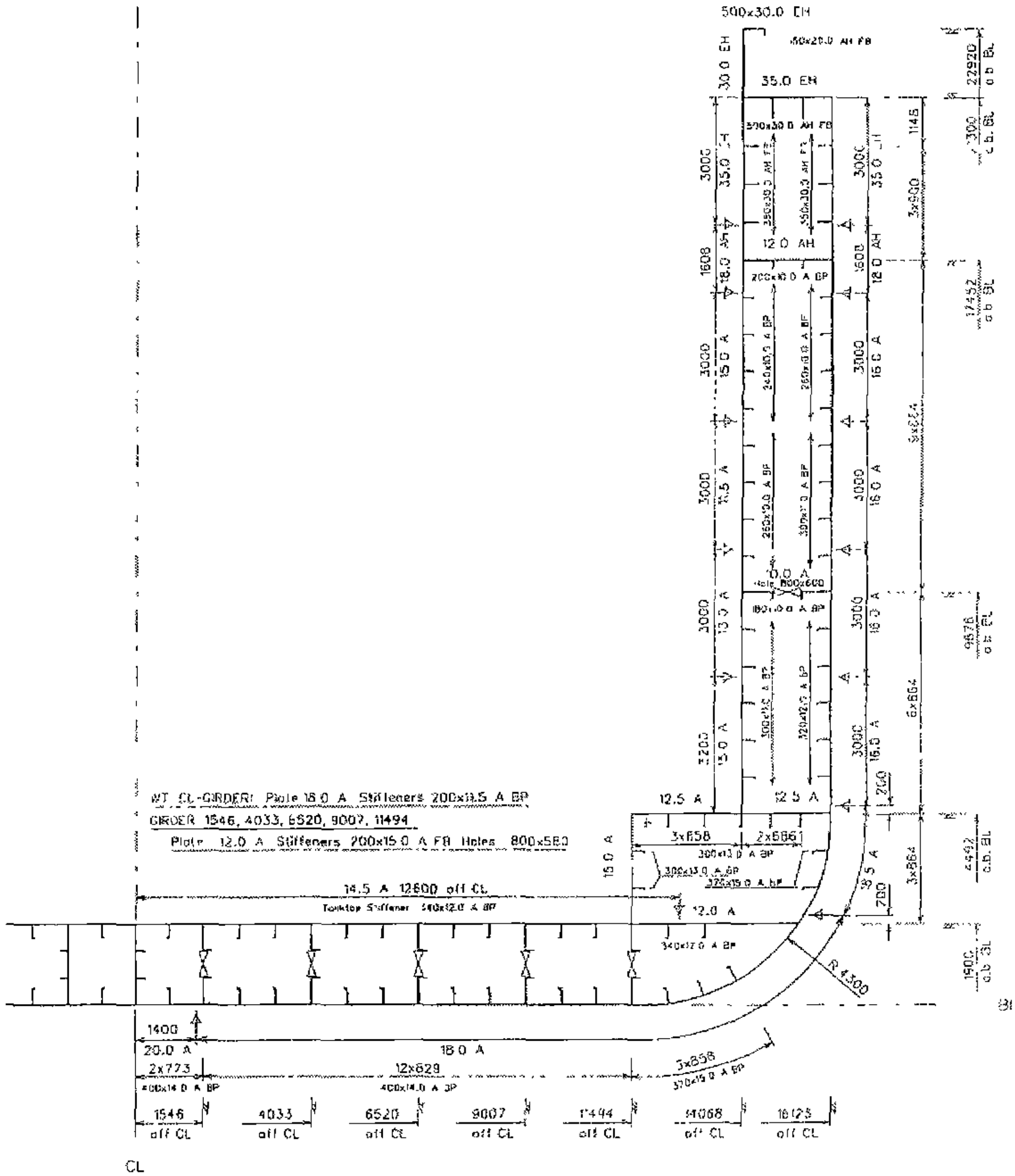

Figure 4. Midship section for a 3800 TEU Panamax container vessel.

Table 3. Combined load cases for fatigue strength assessment of container carriers.

	L.C.1	L.C.2	L.C.3	L.C.4	L.C.5	L.C.6
Vertical B.M.* K_c	Sag (-) 1.0	Hog (+) 1.0	Sag (-) 0.7	Hog (+) 0.7	Sag (-) 0.3	Hog (+) 0.3
Horizontal B.M. K_c	0.0	0.0	0.0	0.0	Stbd tens(-) 0.3	Port tens(+) 0.3
	L.C.7	**L.C.8**	**L.C.9**	**L.C.10**	**L.C.F1**	**L.C.F2**
Vertical B.M.* K_c	Sag (-) 0.4	Hog (+) 0.4	Sag (-) 0.4	Hog (+) 0.4	Sag (-) 0.4	Hog (+) 0.4
Horizontal B.M. K_c	Stbd tens(-) 0.5	Port tens(+) 0.5	Port tens(+) 0.7	Stbd tens(-) 0.7	Stbd tens(-) 1.0	Port tens(+) 1.0

*The following stillwater bending moment (SWBM) is to be used for structural analysis.
L.C.1,3,5,7,9,F1: Maximum sagging SWBM   L.C.2,4,6,8,10,F2: Maximum hogging SWBM

Table 4. Midship sectional properties of the 3800 TEU containership.

Property	Net section	Original section	
		present	Kierkegaard
I_y (m^4)	246.01	262.45	258.740
I_z (m^4)	613.92	662.29	659.765
Z_c (m)	8.570	8.411	8.496
W_C (m^3) (Coaming)	17.056**	18.195**	18.114*
W_D (m^3) (Deck)	19.325	20.362	20.208
W_B (m^3) (Bottom)	28.706	31.203	30.453

*The coaming height is defined according to the DNV rule, $z = 22.781$ m.
** The actual coaming height $z = 22.920$ m is used.

formula:

$$\sigma_n = \frac{K_c(VBM)(M_{sw} + M_{wv})Z_B}{I_y} + \frac{K_c(HBM)M_H Y_B}{I_z} \tag{13}$$

where

$K_c(VBM)$ and $K_c(HBM)$ are the correlation factors given in Table 3.

M_{sw}, M_{wv} and M_H are the stillwater bending moment, wave-induced vertical bending moment and wave-induced horizontal bending moment respectively and can be calculated by the following equations[ABS, 1996]:

$$M_{sw} = \begin{cases} -k_s C_1 L^2 B(C_b + 0.7) \times 10^{-6} & \text{MN} \cdot \text{m} \\ k_H C_1 L^2 B(8.167 - C_b) \times 10^{-6} & \text{MN} \cdot \text{m} \end{cases} \tag{14}$$

$$M_{wv} = \begin{cases} -k_w k_1 C_1 L^2 B(C_b + 0.7) \times 10^{-6} & \mathrm{MN} \cdot \mathrm{m} \\ k_w k_2 C_1 L^2 B C_b \times 10^{-6} & \mathrm{MN} \cdot \mathrm{m} \end{cases} \tag{15}$$

$$M_H = \pm k_w k_3 C_1 L^2 D(C_b + 0.7) \times 10^{-6} \quad \mathrm{MN} \cdot \mathrm{m} \tag{16}$$

where

$$\begin{aligned} & k_s = 13, \quad k_H = 15, \quad k_w = k_0^{1/2} \\ & k_0 = 1.09 + 0.029V - 0.47C_b \\ & k_1 = 110, \quad k_2 = 190, \quad k_3 = 104.2 \\ & C_1 = 10.75 - [(300 - L)/100]^{1.5} \\ & V = 0.75V_d \end{aligned} \tag{17}$$

I_y and I_z are the moments of inertia about the y and z axes respectively,
Y_B is the horizontal distance from the pad detail to the centerline,
Z_B is the vertical distance from the pad detail to the horizontal neutral axis.

From Table 3 it can be easily found that the maximum design stress range can either be the pair of L.C.1 and L.C.2 or the pair of L.C.F1 and L.C.F2. Therefore, only these four load cases need to be considered for the present pad detail.

The fatigue strength assessment of ABS approach is based on a "net" ship approach. In performing structural analyses and fatigue strength assessments, the nominal design corrosion values as given in ABS approach[ABS, 1996] are to be deducted. These minimum corrosion values reflect an average overall corrosion wastage for 20 years in service assuming good maintenance schedules and an effective monitoring system for the coating protection in ballast tanks, and are not to be interpreted as renewal standards for local structures. The beneficial effects of these design margins on reduction of stresses and increase of the effective hull-girder section modulus will be accounted for by the adjustment factor $C_f = 0.95$ to reflect a mean wasted condition. Therefore, in order to calculate the nominal stress, the net sectional properties need to be calculated. Kierkegaard[Kierkegaard, 1999] has calculated the original sectional properties. In order to compare the difference between our program and his program for calculating the sectional properties, both original and net sectional properties have been calculated and the results are given in Table 4. It can be seen that the calculated sectional properties by our program are in good agreement with that calculated by Kierkegaard[Kierkegaard, 1999] .

5 Determination of the Stress Concentration Factor

For the present pad detail, both the nominal stress approach and the hot spot stress approach can be used. In the nominal stress approach, the cover plate can be regarded as the doubler on face plate, therefore G should be chosen as the corresponding S-N curve. This indicates a nominal SCF for this pad detail is 1.61. In the hot spot stress approach, E S-N curve is used with the corresponding SCF. The SCF is calculated using the finite element method.

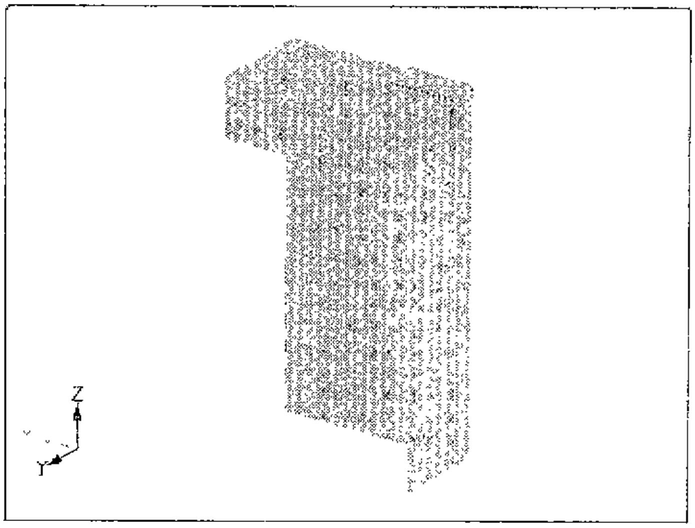

Figure 5. Finite element model of the longitudinal hatch coaming.

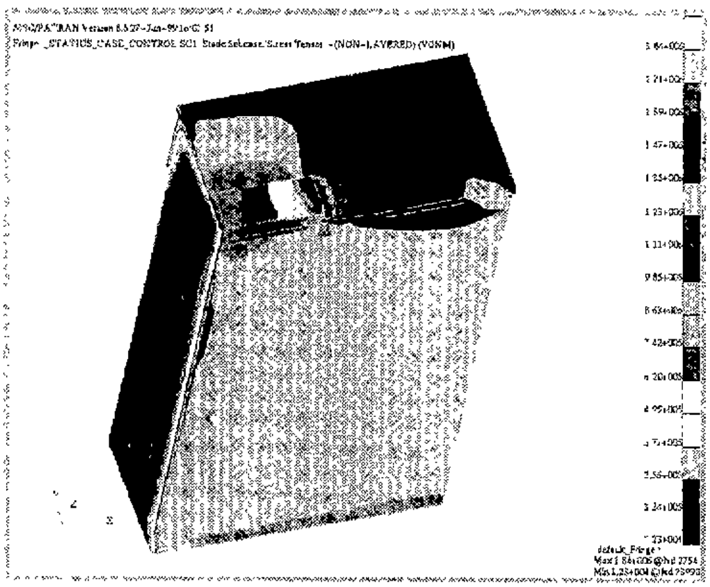

Figure 6. Stress distribution in the longitudinal hatch coaming.

In order to calculate the stress concentration factor caused by the welding of the cover plate on top of the hatch coaming, a part of length of 1810 mm in which the cover plate is located in the middle is taken out for the finite element analysis. Because of the symmetry, only half is modeled. The symmetry plane is in the middle of the web frame. The coordinate system is as follows. The origin is located at the intersection point among face plate, web and web frame. x is along the longitudinal direction, y is along the width direction and z is along the height direction. For simplicity, the cover plate is assumed to be a rectangular plate which means that

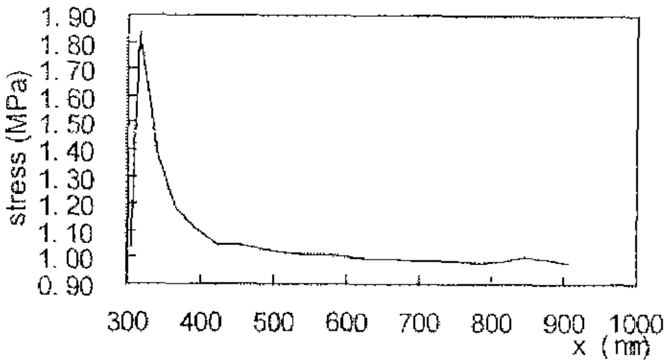

Figure 7. Stress distribution in front of the maximum stress.

the taper of the thickness and the round end were not accurately modeled. This is thought not to significantly affect the calculational results because the two factors have different effects and they might counteract with each other. The tapered thickness of the cover plate at the end will increase the stress concentration at the cover plate end while the round end will reduce the stress concentration. However, the weld beads of the triangular shape are included in the FE model. The finite element analysis is carried out using MSC/NASTRAN software. 20-node together with some 15-node solid elements are used. For all the nodes located in the connection between hatch coaming and deck, the translations along y and z directions and the rotations along x and z axes are constrained. For the nodes located in the connection between web frame and deck, all the six degrees of freedom are constrained. For all the nodes in the web frame surface, symmetrical conditions are applied. The bending stress distribution according to the beam theory is applied at one end in which the maximum nominal stress in the face plate of the hatch coaming is 1 MPa. The net thickness is used in which 1 mm corrosion margin is deduced for all the plates according to the ABS rule. The finite element model is shown in Figure 5.

The stress distribution is shown in Figure 6. It is found that the maximum stress is 1.84 MPa which occurs at the transverse end of the cover plate. The longitudinal stress distribution in front of the maximum stress point is shown in Figure 7. The hot spot stress then is calculated by a linear extrapolation of stresses at point $t/2$ and point $3t/2$ (where t is the face plate thickness) from the weld toe. The hot spot SCF is found to be 1.7355. From this calculation it can be seen that this SCF is 7% higher than the nominal SCF.

6 Fatigue Strength Assessment of the Pad Detail

6.1 Load Calculation

The variations of deadweight in containership is relatively small which is opposed to what happens with tankers where clearly different deadweights are experienced in the ballast and loaded conditions, therefore the stillwater bending moment is generally hogging for containerships[Guedes Soares et al., 1996]. For this containership, Kierkegaard[Kierkegaard, 1999] also only provided the maximum hogging stillwater bending moment. However, in ABS approach[ABS, 1996], both sagging and hogging stillwater bending moments are defined. From Eq.(14), these can be calculated as

$$C_1 = 10.75 - [(300 - 234.74)/100]^{1.5} = 10.223$$

$$M_{sw0}^{sag} = -13 \times 10.223 \times 234.74^2 \times 32.25 \times 1.37 \times 10^{-6} = -323.55 \text{ MN} \cdot \text{m}$$

$$M_{sw0}^{hog} = 15 \times 10.223 \times 234.74^2 \times 32.25 \times 7.497 \times 10^{-6} = 2042.966 \text{ MN} \cdot \text{m}$$

The actual hogging stillwater bending moment calculated by Kierkegaard[Kierkegaard, 1999] for this containership is $M_{sw}^{hog} = 2060.1$ MN · m, the actual sagging stillwater bending moment is calculated as follows:

$$M_{sw}^{sag} = \frac{M_{sw}^{hog}}{M_{sw0}^{hog}} M_{sw0}^{sag} = -326.26 \text{ MN} \cdot \text{m}$$

The actual hogging and sagging stillwater bending moments will be used in the fatigue strength assessment.

The wave-induced vertical bending moments are calculated as follows:

$$V = 0.75 \times 24 = 18 \text{ kn},\ k_0 = 1.09 + 0.029 \times 18 - 0.47 \times 0.67 = 1.2971,\ k_w = 1.1389$$

$$M_{wv}^{sag} = -1.1389 \times 110 \times 10.223 \times 234.74^2 \times 32.25 \times 1.37 \times 10^{-6} = -3118.04 \text{ MN} \cdot \text{m}$$

$$M_{wv}^{hog} = 1.1389 \times 190 \times 10.223 \times 234.74^2 \times 32.25 \times 0.67 \times 10^{-6} = 2633.88 \text{ MN} \cdot \text{m}$$

The wave-induced horizontal bending moments are:

$$M_H = \pm 1.1389 \times 104.2 \times 10.223 \times 234.74^2 \times 21.3 \times 1.37 \times 10^{-6} = \pm 1950.77 \text{ MN} \cdot \text{m}$$

6.2 Nominal Stress Calculation

For each loading case the nominal stress can be calculated by Eq.(13) The results are:

L.C.1: $\sigma_n = \frac{(M_{sw}^{sag} + M_{wv}^{sag}) Z_B}{I_y} = \frac{-3444.3 \times 14.35}{246.01} = -200.91$ MPa

L.C.2. $\sigma_n = \frac{(M_{sw}^{hog} + M_{wv}^{hog}) Z_B}{I_y} = \frac{4693.98 \times 14.35}{246.01} = 273.80$ MPa

L.C.F1: $\sigma_n = \frac{0.4 \times (-3444.3) \times 14.35}{246.01} + \frac{-1950.77 \times 14.068}{613.92} = -125.07$ MPa

L.C.F2: $\sigma_n = \frac{0.4 \times 4693.98 \times 14.35}{246.01} + \frac{1950.77 \times 14.068}{613.92} = 154.22$ MPa

Therefore the maximum nominal design stress range is given by the load pair of L.C.1 and L.C.2:

$$S_0^{nom} = 273.80 - (-200.91) = 474.71 \text{ MPa}$$

The maximum nominal mean stress is

$\sigma_{mean}^{nom} = \frac{273.80-200.91}{2} = 36.45$ MPa

6.3 Calculation of the Modified Weibull Shape Parameter

In ABS approach[ABS, 1996], the modified Weibull shape parameter k is defined as below:

$$k = m_s \gamma_0 \tag{18}$$

where

$m_s = 1.05$	for deck and bottom structures of vessels with a forebody parameter, $A_r d_k = 155$ m^2
$= 1.02$	for deck and bottom structures of vessels with a forebody parameter, $A_r d_k = 112$ m^2
$= 1.00$	for structures elsewhere, and all structures of vessels without bowflare slamming, $A_r d_k \leq 70$ m^2.

For intermediate values of $A_r d_k$, m_s may be obtained by linear interpolation. For $A_r d_k > 155$ m^2, m_s is to be determined by direct calculations.

$$\gamma_0 = \begin{cases} 1.40 - 0.2\alpha L^{0.2} & \text{for } 130 < \text{L} < 305 \text{ m} \\ 1.54 - 0.254\alpha^{0.8} L^{0.2} & \text{for L} \geq 305 \text{ m} \end{cases} \tag{19}$$

where

$\alpha = 1.0$	for deck structures, including side shell and longitudinal bulkhead structures within $0.1D$ from the deck.
$= 0.93$	for bottom structures, including inner bottom, and side shell and longitudinal bulkhead structures within $0.1D$ from the bottom.
$= 0.86$	for side shell and longitudinal bulkhead structures within the region of $0.25D$ upward and $0.3D$ downward from the mid-depth.
$= 0.80$	for side frames, vertical stiffeners on longitudinal bulkhead and transverse bulkhead structures.

A_r, d_k are defined in the ABS approach. In this calculation, it is assumed that $m_s = 1.02$ and $\alpha = 1.0$, then $\gamma_0 = 1.40 - 0.2 \times 1.0 \times 255^{0.2} = 0.7942$, $k = 0.81$.

6.4 Fatigue Life Calculation

The fatigue life for this pad detail can be calculated using Eq.(10) in the following manners.

(1) Nominal Stress Approach Using G S-N curves

The actual design stress range $S_0 = 0.95 \times 471.71 = 450.97$ MPa, $N_{T0} = 5 \times 10^7$ for 20 years, $k = 0.81$, if G S-N curve is chosen, then $N_T = 0.754 \times 10^7$ which is equivalent to 3.017 years.

(2) Hot Spot Stress Approach Using E S-N curve

The maximum SCF calculated by F.E. for this pad detail is $K = 1.7355$ and the predicted fatigue life is $N_T = 0.599571 \times 10^7$ which is equivalent to 2.398 years. Therefore, the difference between the nominal stress approach and the hot spot stress approach is about 25.8%.

From the above calculations one conclusion can be drawn is that with ABS approach the fatigue life for this pad detail is very low. This may be true or might be too conservative. By examining the ABS approach in detail and also comparing this approach with others, e.g. DNV[DNV, 1995], one can find that the inclusion of the stillwater bending moments in the stress range calculation might be a conservative source. This is because the change of the stillwater bending moment has a totally different nature when comparing with the wave-induced loads. Also the number of cycles for the stillwater bending moment is much smaller than that of wave-induced bending moments. Therefore if we ignore the effect of stillwater bending moment, then the nominal design stress range becomes:

$$S_0 = 0.95 \times \frac{(2633.88+3118.04)\times 14.35}{246.01} = 318.74 \text{ MPa}$$

Using the same approach as before the fatigue life for this pad detail is recalculated as follows:

(1) Nominal Stress Approach Using G S-N curves

If G S-N curve is chosen, then $N_T = 2.22911 \times 10^7$ which is equivalent to 8.916 years.

(2) Hot Spot Stress Approach Using E S-N curve

$N_T = 1.75906 \times 10^7$ which is equivalent to 7.036 years.

7 Summary and Conclusions

Fatigue cracking in ships has been a problem for many years. However, only until recently, attention has been paid to this problem by researchers and classification societies. Now many classification societies have implemented fatigue strength assessment procedures in their ship design rules. However, a large variation in the details of the different approaches exists in practically all aspects including load history assessment, stress evaluation and fatigue strength assessment. In order to assess the influences of these variations on the prediction of fatigue lives, a comparative study is organized by the ISSC Committee III.2 Fatigue and Fracture. A pad detail on the top of longitudinal hatch coaming of a Panamax container vessel is selected for fatigue calculation. The work described in this paper is one set of results of this comparative study in which the ABS dynamic approach is applied. Through this analysis the following conclusions can be drawn.

(1) With the original ABS approach, the fatigue life of this pad detail is very low, only 2.398

years.

(2) Through some detailed analysis of the approach, it is thought that the treatment of the stillwater bending moment in the ABS approach might be a source of conservatism. If the influence of stillwater bending moment is ignored, then the fatigue life for this pad detail is 7.036 years. This result is recommended.

(3) For this pad detail the difference between the nominal stress approach and the hot spot stress approach is about 26%.

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