Thesis for the Degree of Master of Engineering

Fatigue Analysis and Reliability Assessment for a 17K Oil Tanker

by

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Fatigue Analysis and Reliability Assessment for a 17K Oil Tanker

17K Oil Tanker 에 대한 피로해석 및 신뢰성 평가

Advisor: Prof. Dong-Myung Bae

by

Tuo-Han Chen

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Abstract

Fatigue cracks and damages have been an important issue for ship and offshore structure in a long times. Initially the obvious remedy was to improve detail design. However, in the last decades, with the introduction of higher tensile steels in hull structures and dimension of ships become larger than before, the greater attention should be paid to fatigue problem. Most of researches focus on how to access the fatigue strength of ships more reasonably. Also, the major classification societies have already released their fatigue assessment notes. But due to the complexity of factor influencing fatigue performances，such as, wave load and pressure from cargo，the combination of different stress components，stress on concentration of local structure details，means stress and the corrosive environments etc，there are different specifications with varying classification societies，so it will lead to the different of results from different fatigue assessment methods. This paper founds on the DNV classification notes “fatigue assessment of ship structures” that expatiates on process of fatigue assessment and simplified method. Finally, a fatigue analysis was taken by use data of a real ship and assesses the reliability of the result.

The main aspects of this paper are as following:

1. Expatiation of elements of fatigue analysis and general method for fatigue strength analysis of ship structure----Focuses on base theory, analysis process and characteristic of S-N curves.
2. Simplified method of fatigue assessment----Expatiates the base theory and analysis process, and then, estimates the design life of the considered point by using simplified analysis provided by DNV fatigue assessment rules.
3. Reliability assessment---- Expatiates the base theory and analysis process, and assesses the reliability of result of fatigue analysis.

Key words: fatigue; reliability; S-N curve; DNV fatigue assessment rules

I. Introduction

1. Background

When one or some position of structure subjected to cyclic loading, some gradual, local, permanent change will be happen. And finally, the structure would create crack even failure if the cycle was enough. For this kind of failure, we call it fatigue damages. For a ship or an offshore structure whose period of service are 20 years that would be met 10^8 cycles and even more. It will make a fatigue damage problem. Since steel ships have been used, there were so many ship accidents have been reported for fatigue crack happen and extend in structure. According to the result of research, it indicates that the main cause of those accidents is the fatigue failure of structure.

In order to avoid fatigue of ship structure from lack of enough fatigue strength, many countries brought forward different method for calculation of general marine structure in design stage. The main classifications (ABS、BV、DNV、GL、KR、LR、NK、RINA and RS etc) have their own require of fatigue strength for structure in their classification.

For general, there are two method used for fatigue assessment:

(1) S-N curves. Find out S-N curves from experiment which suit for many kind of structure, and calculate fatigue life of structure with Palmgren-Miner cumulation formula.[1-2]

(2) Fracture mechanics. For microcosmic, there are some extent crack in both structure and assembly. When the tension stress was acted on it, stress field would be present nearby crack tip, and this stress field could be described by "stress

intensity factor"K. Brittle fracture will be happen when K reach the critical value K_c。If the expression of stress intensity factor and fracture toughness of material can be known, the maximum allowable stress and critical crack size of structure which make failure can be calculated. [3]

And reliability assessment usually taken in design stage [4][5][6]

In most of paper, simplified method based on S-N curves method was used for research or assessment [1]. And in shipyard, most of times, FE model method based on S-N curves method was used for high accuracy require of result. But, all of position should be calculated for check the most damage local detail with simplified method, and, fine mesh modeling should be used in FE model method. Both of them cost much times.

In this thesis, we suppose a new method which mixed simplified method and FE model method is suit for fatigue assessment, and stress range calculated form forepart of this thesis would be substitute design stress range in reliability assessment.

2. Objective of Study

For approximate assessment, both simplified method and FE model method are taken much time. However, it is not difficult to calculate in simplified and also not difficult to find the position with maximum stress in FE model method. So in approximate process, if we find out the position with maximum stress with FE model method, and then, take fatigue analysis by simplified method for the position, we can find out the damage life of the most damage position more easily. And use the maximum stress range to substitute design stress range, the result of reliability would be calculated also. There would be reduce more time for approximate the fatigue life and reliability of ship structure.

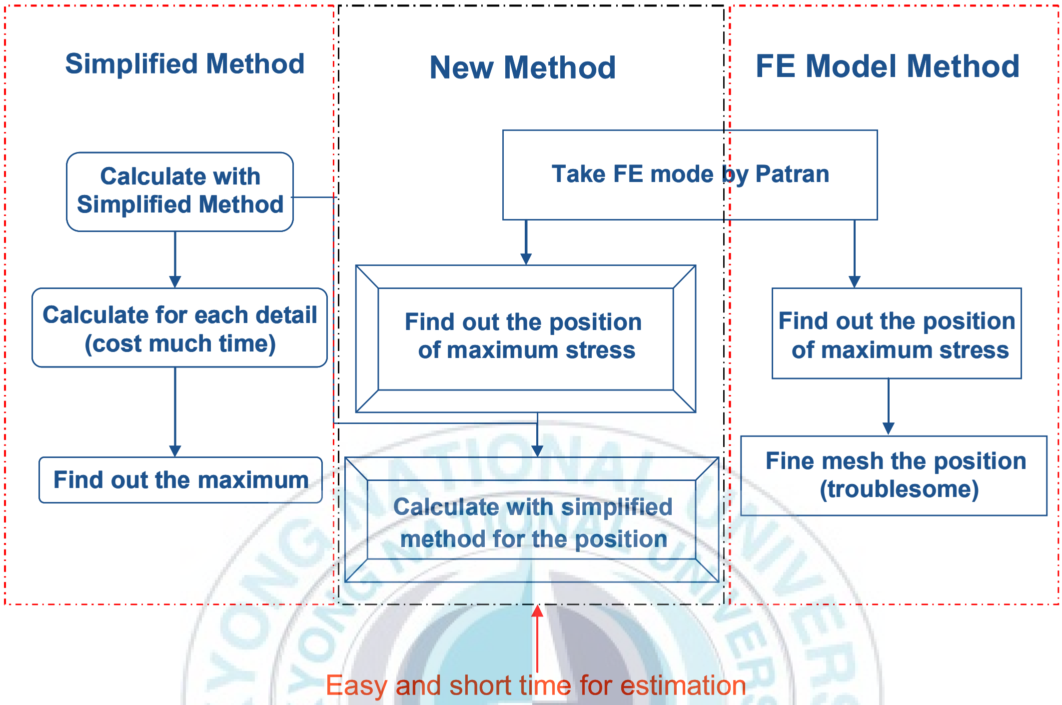


Fig. 1. 1 Relationship of different solution methods

3. Outline of Thesis

This research is comprised of 5 chapters. Except for the current introductory chapter, the rest of the chapters are summarized as follows:

Chapter Ⅱ contains overall contents of theory of fatigue assessment (S-N curves). The first half introduces general information on fatigue assessment, definition of fatigue and its history. The second half contains the detailed process of simplified method.

Chapter Ⅲ expatiate reliability of ship structure for the uncertainty factor, and introduce the reliability.

Chapter Ⅳ applies the new method and FE model method on a 17k oil tanker.

And analysis the result after compared.

The final discusses will be obtained in Chapter Ⅴ.

II. Fatigue Assessment

1. Methods for Fatigue Analysis

Fatigue design may be carried out by methods based on fatigues test(S-N data) and estimation of cumulative damage (Palmgrens-Miner rule). The long term stress range distribution is a fundamental requirement for fatigue analysis. In here, we outline two methods for stress range calculation:

(1) A postulated form of the long-term stress range distribution with a stress range based on dynamic loading as specified in the rules.

(2) Spectral method for the estimation of long-term stress range.

In the first method a Weibull distribution is assumed for the long term stress ranges, leading to a simple formula for calculation of fatigue damage. The load effects can be derived directly from the ship rules. The nominal stresses have to be multiplied by relevant stress concentration factors for calculation of local notch stresses before entering the S-N curve.

The second method implies that the long-term stress range distribution is calculated from a given (or assumed) wave climate. This can be combined with different levels of refinement of structural analysis.

Thus a fatigue analysis can be performed based on simplified analytical expressions for fatigue live or on a more refined analysis where the loading and the load effects are calculated by numerical analysis. The fatigue analysis may also be performed based on a combination of simplified and refined techniques as Fig. 2.1.

2. Fatigue Accumulation and S-N Curves

The fatigues life under varying loading is calculated based on the S-N fatigue approach under the assumption of linear cumulative damage (Palmgrens-Miner rule). The total damage that the structure is experiencing may be expressed as the accumulated damage from each load cycle at different stress levels, independent of sequence in which the stress cycles occur.

The design life assumed in the fatigue assessment of ships is normally not to be less than 20 years. The accumulated fatigue damage is not to exceed a usage factor of 1.0. The acceptance criteria is related to design S-N curves based on mean- minus- two- standard- deviations curves for relevant experimental data.

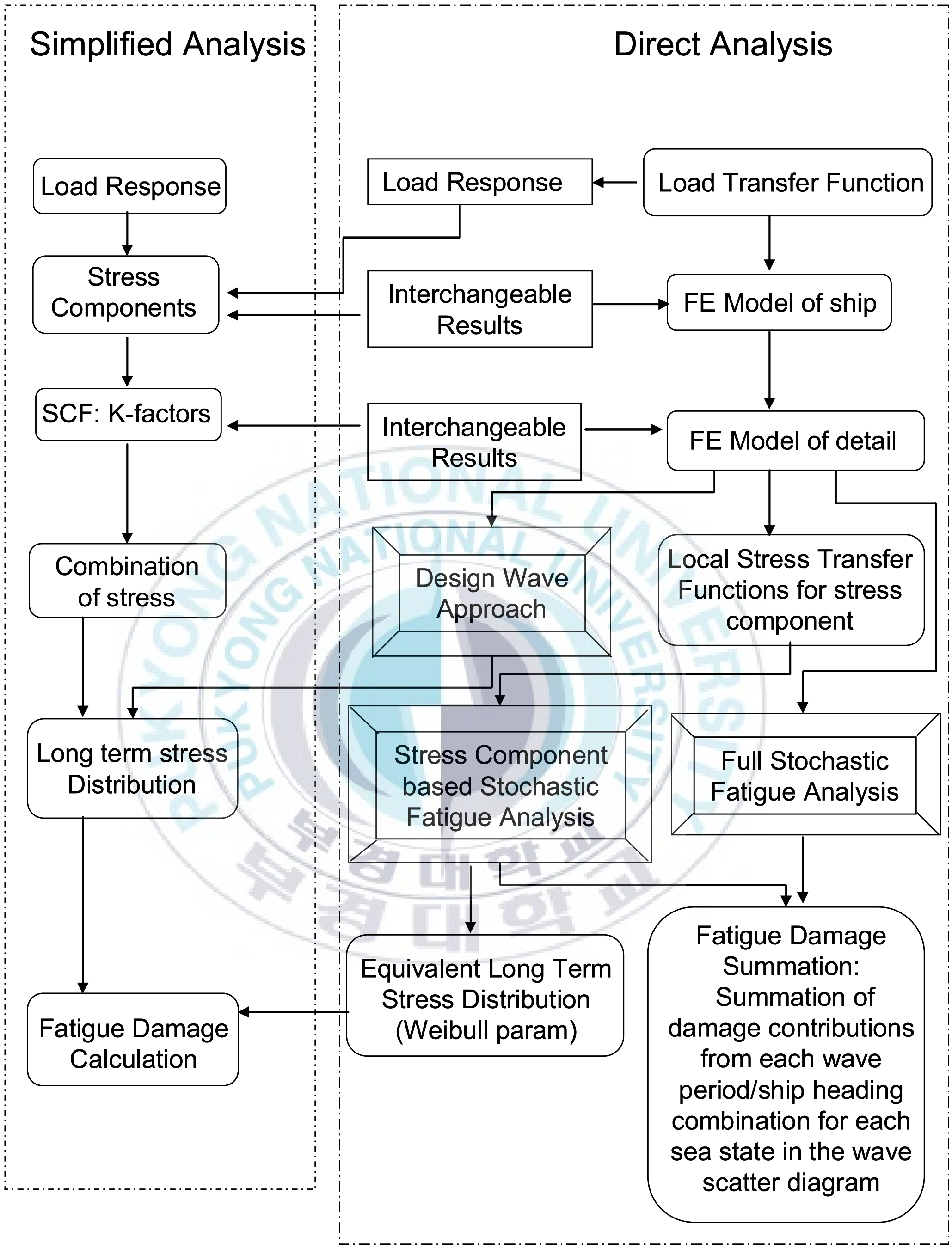


Fig. 2. 1 Flow diagram over possible fatigue analysis procedures

2.1 Cumulative Damage for Simplified Method

The fatigue life may be calculated based on the S-N fatigue approach under the assumption of linear cumulative damage (Palmgrens-Miner rule).

When the long-term stress range distribution is defined applying Weibull distributions for the different load condition, and a one-slope S-N curves is used, the fatigue damage is given by,

$$D = \sum_{i=1}^{k} D_i = \frac{v_0 T_d}{\bar{a}} \sum_{n=1}^{N_{load}} p_n q_n^m \Gamma(1 + \frac{m}{h_n}) \leq \eta \tag{2.1}$$

where N_{load} is total number load conditions considered; p_n is fraction of design life in load condition n, $\sum p_n \leq 1$, but normally not less than 0.85; T_d is design life of ship in seconds (20years=6.3*10^8 secs.); h_n is Weibull stress range sharp distribution parameter for load condition n; q_n is Weibull stress range scale distribution parameter for load condition n; v_0 is long-term average response zero-crossing frequency; $\Gamma(1 + \frac{m}{h_n})$ is gamma function.

The Weibull scale parameter is defined from the stress range level, $\Delta\sigma_0$,as,

$$q_n = \frac{\Delta\sigma_0}{(\ln n_0)^{1/h_n}} \tag{2.2}$$

where n_0 is the number of cycles over the time period for which the stress range level $\Delta\sigma_0$ is defined.

$$v_0 = \frac{1}{4 \cdot \log_{10}(L)} \tag{2.3}$$

where L is the ship Rule length in meters.

2.2 S-N Curves

The S-N curves recommended for the fatigue assessment are obtained from experience and fatigue tests. The S-N curves are to be applied together with the notch stress, the local stress at the weld toe due to structural discontinuities (hot-spot stress) and the weld geometry. Different S-N curves are defined for welded joints and base material in air/cathodic protected environments and for corrosive environments. The design S-N curves are based on the mean-minus-two-standard-deviation curves for relevant experimental data, and are thus associated with a 97.6% probability of survival. The basic design S-N curve is given,

$$N = \frac{\bar{a}}{\Delta\sigma^{m}} \quad \text{or} \quad \log N = \log \bar{a} - m \log \Delta\sigma \tag{2.4}$$

$$\log N = \log \bar{a} - \frac{m}{4} \log\left(\frac{t}{25}\right) \log \Delta\sigma \quad \text{for thickness larger than 25mm} \tag{2.5}$$

With S-N curve parameters given in Table 2.1

where N is predicted number of cycles to failure for the stress range $\Delta\sigma$; and m is negative inverse slope of S-N curve; $\log \bar{a}$ is intercept of logN-axis by S-N curve;

$$\log \bar{a} = \log a - 2s \tag{2.6}$$

where a is constant relating to mean S-N curve; s is standard deviation of $\log N$ equal to 0.20.

Table 2. 1 Parameters of S-N curve

	Type	Environment	N≤10^7		N>10^7	
Two-slope S-N curves						
			$\log \bar{a}$	m	$\log \bar{a}$	m
Ⅰ	Welded joint	Air-cathodic prt.	12.65	3.0	16.42	5.0
Ⅱ	Welded joint	Corrosive	12.38	3.0	12.38	3.0
Ⅲ	Base Material	Air-cathodic prt.	12.89	3.0	16.81	5.0
Ⅳ	Base Material	Corrosive	12.62	3.0	12.62	3.0
One-slope S-N curves						
Ⅰb	Welded joint		12.76	3.0	12.76	3.0
Ⅱb	Base Material		13.00	3.0	13.00	3.0

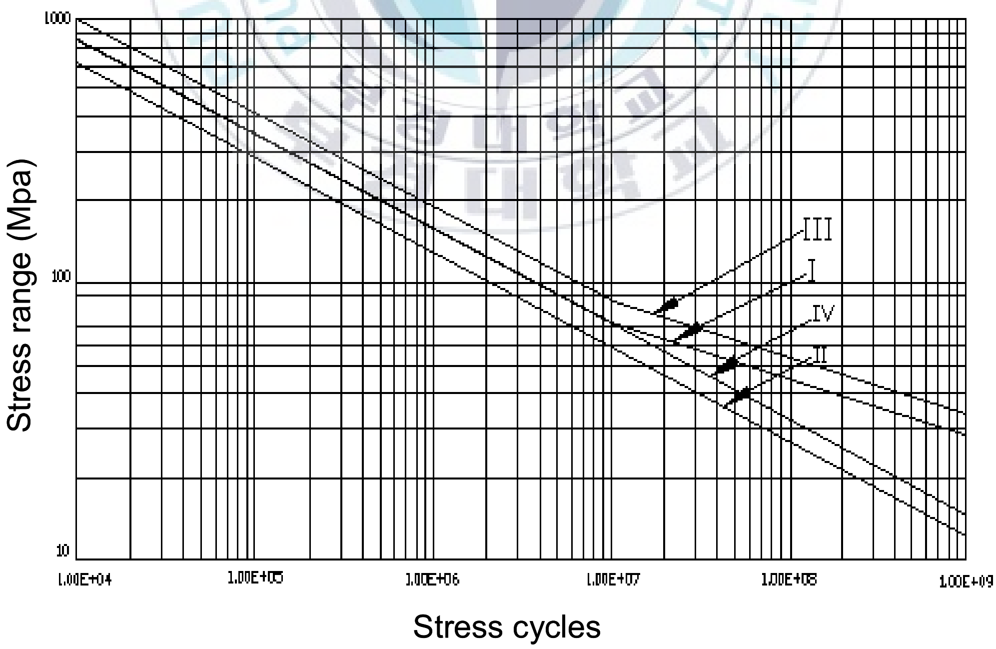


Fig. 2. 2 S-N Curves for two slope

3. Simplified Stress Analysis

The simplified approach for estimating the long term stress range distribution is based on the assumption of Weibull distributed stress ranges. In this approach, simplified formulas for estimating the individual stress response components, the combination of global and local stress response components and the modelling of the shape of the stress response distribution etc were defined.

The wave induced dynamic loading is estimated from empirical rule formulations or dynamic pressure load analyses. The corresponding stress response is derived applying empirical and analytical expressions, or for more accurate estimation, the use of frame analyses or finite element analyses.

Almost all of the classification societies have them own simplified method, for general, the approach will be taken as:

(1) estimating of fatigue load

(2) estimating of individual stress response components

(3) combination of global and local stress response components

(4) fix on stress concentration factor

(5) calculation of accumulated fatigue damage

3.1 Long Term Distribution of Stresses

The long term distribution of stress ranges can be shown to be well described through a Weibull distribution, having cumulative probability,

$$Q(\Delta\sigma) = \exp[-(\frac{\Delta\sigma}{q})^h] \quad (2.7)$$

where Q is probability of exceedance of the stress range; h is Weibull shape parameter; q is Weibull scale parameter, defined as,

$$q = \frac{\Delta\sigma_0}{(\ln n_0)^{1/h}} \tag{2.8}$$

The shape parameter depends on the prismatic parameters of the ship, the location of the considered detail and the sailing route over the design life. In lieu of more accurate calculations, the shape parameter may be taken as,

$h = h_0$ For deck longitudinals

$h = h_0 + h_a(D - z)/(D - T_{act})$ For ship side above the waterline $T_{act} < z < D$

$h = h_0 + h_a$ h=h0+ha For ship side at the waterline $T_{act} = z$

$h = h_0 + h_{az}/T_{act} - 0.005(T_{act} - z)$ For ship side below the waterline $T_{act} > z$

$h = h_0 - 0.005T_{act}$ For bottom longitudinals

$h = h_0 + h_a$ For longitudinal and transverse bulkheads

where $h_0 = 2.21 - 0.54\log_{10}{}^{(L)}$ is the basic shape parameter; h_a is the additional factor depending on the motion response period, equal to 0.05 in general and 0.0 for plating subjected to roll motions for vessels with roll period over 14 seconds. D is the moulded depth of the ship, T_{act} is the actual draught and z is the location height above the keel. And for hopper knuckle connections, the Weibull shape parameter for ship side at the waterline may be used.

3.2 Definition of Stress Components

Dynamic stress variations are referred to as either stress range ($\Delta\sigma$) or stress amplitude (σ). For linear responses, the following relation applies $\Delta\sigma = 2\sigma$.

In the fatigue analysis, for the global dynamic stress components (primary stresses) should be considered. Like as,

Wave induced vertical hull girder bending stress σ_v ; Wave induced horizontal hull girder bending stress σ_{hg} .

And the local dynamic stress amplitudes which should considered also. Like as:

Total local stress amplitude due to dynamic external pressure loads σ_e ; Total local stress amplitude due to dynamic internal pressure loads or forces σ_i

While the local stress components are defined as:

σ_2 secondary stress amplitude resulting from bending of girder system;

σ_{2A} stress amplitude produced by bending of stiffeners between girder supports;

σ_3 tertiary stress amplitude produced by bending of un-stiffened plate element between longitudinals and transverse frames.

3.3 Combination of Stresses

For each loading condition, the local dynamic stress components due to internal and external pressure loads are to be combined with the global stress components induced by hull girder wave bending. The stress components to be combined are the notch stresses.

If a combined long term stress response analysis is not carried out, the combined stress range response from the combined global and local stress may be taken as,

$$\Delta\sigma = f_m \Delta\sigma \tag{2.9}$$

$$\Delta\sigma = f_e \max\begin{cases}\Delta\sigma_g + b\cdot\Delta\sigma_l \\ a\cdot\Delta\sigma_g + \Delta\sigma_l\end{cases} \tag{2.10}$$

where f_e is reduction factor on derived combined stress range accounting for the long-term sailing routes of ship considering the average wave climate the vessel will be exposed to during the lifetime. For world wide trade, the reduction factor may be taken as 0.8. For shuttle tankers and vessels that frequently operates in the North Atlantic or in other harsh environments, $f_e = 1.0$ should be used. f_m is the reduction factor on derived combined stress range accounting for the effect of mean stresses. a,b equal to 0.6 are load combination factors, accounting for the correlation between the wave induced local and global stress ranges. σ_l is combined local stress range due to lateral pressure loads. σ_g is combined global stress range.

In general, the combined global stress range may be taken as,

$$\Delta\sigma_g = \sqrt{\Delta\sigma_v{}^2 + \Delta\sigma_{hg} + 2\rho_{vh}\Delta\sigma_v\Delta\sigma_{hg}} \qquad (2.11)$$

where the long term correlation of ρ_{vh} is the average correlation between vertical and horizontal wave induced bending stress which defined as 0.10.

The combined local stress range, σ_l , due to external and internal pressure loads may be taken as,

$$\Delta\sigma_l = \sqrt{\sigma_e{}^2 + \sigma_i{}^2 + 2\rho_p\sigma_e\sigma_i} \qquad (2.12)$$

ρ_p is average correlation between sea pressure loads and internal pressure loads be taken as,

$$\rho_p = \frac{1}{2} - \frac{z}{10 \cdot T_{act}} + \frac{|x|}{4 \cdot L} + \frac{|y|}{4 \cdot B} - \frac{|x| \cdot z}{5 \cdot L \cdot T_{act}} \qquad z \le \text{Tact} \qquad (2.13)$$

$$\rho_p = \frac{1}{2} - \frac{1}{10} + \frac{|x|}{4 \cdot L} + \frac{|y|}{4 \cdot B} - \frac{|x|}{5 \cdot L} \qquad z > \text{Tact} \qquad (2.14)$$

3.4 Calculation of Stress Components

The global and local stress components are derived from the wave induced bending moments and the external and internal wave induced pressure loads.

The stress contributions are estimated applying simple analytical and empirical expressions, accounting for the effective span of longitudinal/stiffeners, the effective breadth of plate flanges and the relative deflection between transverse bulkheads and adjacent web frames.

The wave included vertical hull girder stress taken as,

$$\sigma_v = 0.5K[M_{wo,h} - M_{wo,s}]10^{-3}|z - n_0| / I_N \tag{2.15}$$

and $M_{wo,s(h)}$ is vertical wave sagging (hogging) bending moment amplitude. In this equation $|z - n_0|$ is vertical distance in m from the horizontal neutral axis of hull cross section to considered member. I_N is moment of inertia of hull cross-section in m^4 about transverse axis. K is stress concentration factor for considered detail and loading.

The wave included horizontal hull girder stress taken as,

$$\sigma_h = KM_H 10^{-3}|y| / I_C \tag{2.16}$$

where M_H is horizontal wave bending moment amplitude. y is distance in m from vertical neutral axis of hull cross section to member considered. I_C is the hull section moment of inertia about the vertical neutral axis.

Local secondary bending stresses(σ_2) are the results of bending due to lateral pressure of stiffened single skin or double hull cross-stiffened panels between transverse bulkheads. The preferred way of determining secondary stresses is by means of FE model analysis or alternatively by 3(2)-dimensional frame analysis models. When such analyses are not available, secondary stresses may be estimated by the expressions given in Appendix B of DNV Classification Notes No.30.7.

Longitudinal local tertiary plate bending stress amplitude in the weld at the plate/ transverse frame/ bulkhead intersection is midway between the longitudinals taken as,

$$\sigma_3 = 0.343p(s/t_n)^2 K \tag{2.17}$$

Where p is later pressure (p_e for dynamic sea pressure when p_i for internal dynamic pressure). s is stiffener spacing, t_n is "net" plat thickness. Similarly, the transverse stress amplitude at stiffener mid-length is,

$$\sigma_{3T} = 0.50 p(s / t_n)^2 K \tag{2.18}$$

3.5 Calculation of Loads

The linear dynamic load components for which the individual stress contributions are estimated and calculated from empirical rule expressions as defined in the DNV Rules.3 The load components considered are the global wave induced bending moments the external sea pressure acting on the hull and the internal inertia pressure acting on the tank boundaries. The rule expressions are adjusted for an excess probability of 10^{-4} per wave cycle. The fatigue damage should in general be calculated for all representative load conditions combined with the expected operation time within each of the considered conditions.

The dynamic internal inertia pressure loads should be calculated based on the combined acceleration in longitudinal, transverse and vertical direction. As an approximation, however, the inertia pressure can be estimated as the maximum inertia induced pressure in the longitudinal, transverse or vertical direction.

The moments, at 10^{-4} probability level of exceedance, may be taken as,

$$M_{wo,s} = -0.11 f_r K_{WM} C_W L^2 B(C_B + 0.7) \quad \text{(KNm)} \tag{2.19}$$

$$M_{wo,h} = 0.19 f_r K_{WM} C_W L^2 B \quad \text{(KNm)} \tag{2.20}$$

Where C_W is wave coefficient; K_{WM} is moment distribution factor and it equal to 1.0 between 0.40L and 0.65L from A.P., for ships with low/moderate speed, and it equal to 0.0 at A.P. and F.P. (Linear interpolation between these values); f_r is factor to transform the load from 10^{-8} to 10^{-4} probability level; h_0 is long-term Weibull shape parameter; L is rule length of ship (m); B is the greatest moulded breath of ship in meters measured at the summer waterline; C_B is block coefficient.

$$C_W = \begin{cases} 0.0792L & L < 100\text{m} \\ 10.75 - [(300-L)/100]^{3/2} & 100\text{m} < L < 300\text{m} \\ 10.75 & 300\text{m} < L < 350\text{m} \\ 10.75 - [(L-350)/150]^{3/2} & 350\text{m} < L \end{cases} \quad (\text{KNm}) \qquad (2.21.a)$$

$$f_r = 0.5^{1/h_0} \qquad h_0 = 2.21 - 0.54\log^{(L)} \qquad (2.21.b)$$

The horizontal wave bending moments amplitude at 10^{-4} probability level be taken as,

$$M_H = 0.22 f_r L^{9/4}(T_{act} + 0.30B)C_B(1-\cos(2\pi x/L)) \quad (\text{KNm}) \qquad (2.22)$$

where x is distance in m from A.P. to section considered.

The dynamic external pressure amplitude (half pressure range), P_e , related to the draught of the load condition considered, may be taken as,

$$P_e = r_p P_d \quad (\text{KN/m}^2) \qquad (2.23)$$

$$P_d = \max\begin{cases} P_{dp} = P_l + 135\dfrac{|\bar{y}|}{B+75} - 1.2(T_{act} - Z_w) \\ Pdr = 10[|\bar{y}|\dfrac{\phi}{2} + C_B\dfrac{|\bar{y}| + K_f}{16}(0.7 + 2\dfrac{Z_w}{T_{act}})] \end{cases} (\text{KN/m}^2) \qquad (2.24)$$

The dynamic pressure from liquid cargo or ballast water should be calculated basted on the combined accelerations. The gravity components due to the motions of the vessel should be included. The dynamic internal pressure amplitude, P_i in KN/m^2, may be taken as,

$$P_i = f_a \max\begin{cases} P_1 = \rho a_v h_s \\ P_2 = \rho a_t |y_s| (\text{KN/m}^2) \\ P_3 = \rho a_t |x_s| \end{cases} \qquad (2.25)$$

III. Relatively Assessment

As noted earlier, the principal objective of this investigation is the development of a ship structure fatigue design criteria. Although a variety of factors have been considered in developing the design criteria, only the three most important factors have been included: the mean fatigue resistance of the local fatigue details; a "Reliability Factor" that is a function of the slope of the S-N curve, level of reliability, and a coefficient of variation; a "Random Load Factor" that is a function of the expected loading history and slope of the S-N curve.

1. S-N Relationships

The mean fatigue resistance of the local fatigue details (as shown in Fig.3.1), and the basic information used for design are presented in Table 3.1. The equation can be taken as,

$$\bar{n} = \frac{C}{S^m} \quad \text{or} \quad \log \bar{n} = \log C - m \log S \tag{3.1}$$

where $\bar{n}$ is mean fatigue life; C is constant (C is intercept of logN-axis by S-N curve) same to $\bar{a}$; S is stress range same to $\Delta\sigma$; m is negative inverse slope of S-N curve.

The values in the table are based on laboratory test data and presented in terms of stress range; Because of the relatively small differences in fatigue strength for details of various steels, and the magnitude of scatter generally obtained in fatigue data, it is considered appropriate to neglect any material factor in fatigue assessment of most details. Similarly, because of the complexities caused by a consideration of mean stress and the relatively small magnitude of the mean stress effect, it is recommended that this factor also be neglected in design; and thermal effects, residual stresses, shifting of ballast, distribution and magnitude of cargo, consumption of fuel, etc., all affect the mean stress, sometimes increasing it and in other instances decreasing it. Consequently, there will be a tendency for the mean stress effects to balance one another and thus justify neglecting the mean stress effects in design.

Clearly, the use in design of the mean fatigue stress range is desirable and makes possible the development of a simple fatigue design criteria and assessment for ship structures.

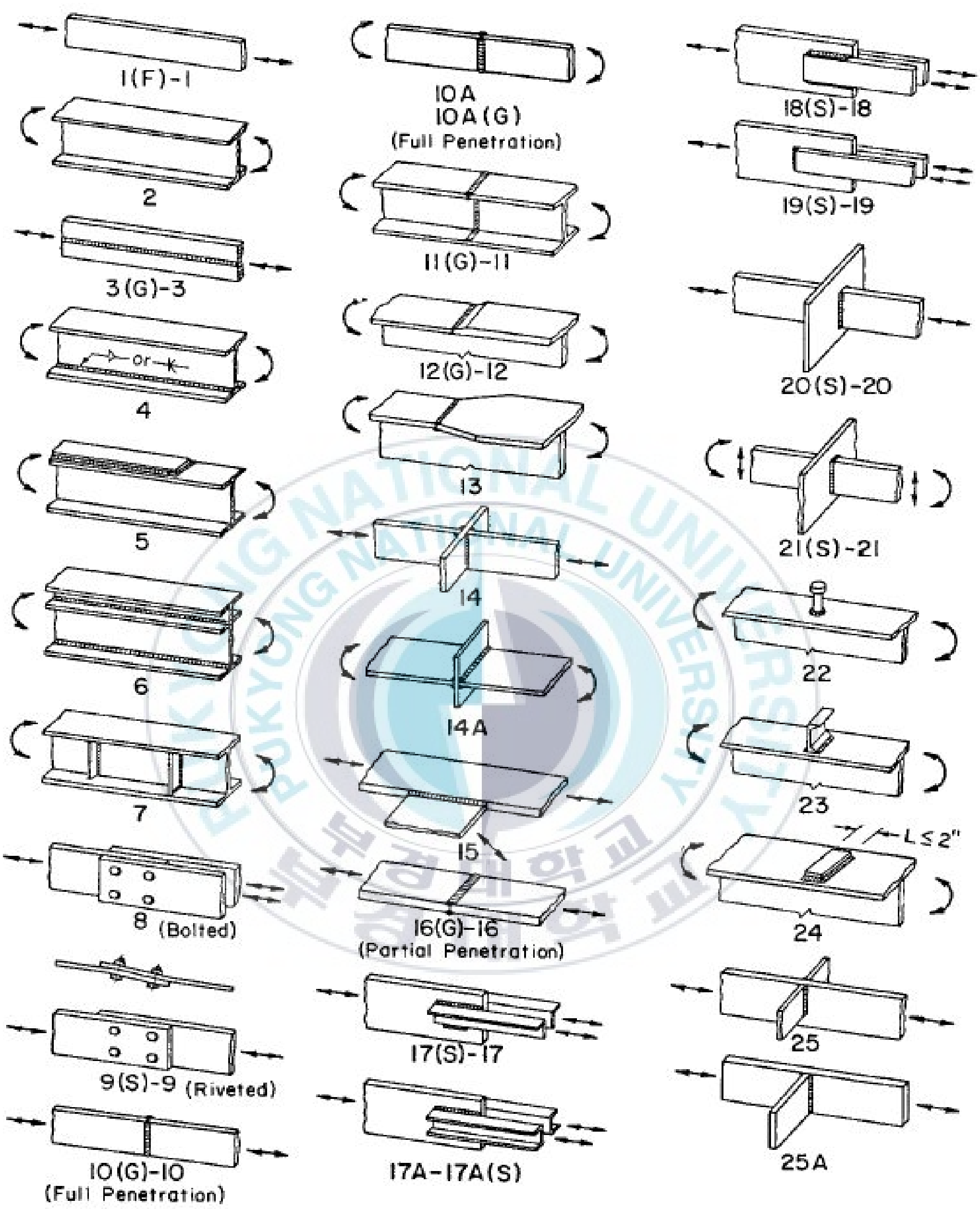


Fig. 3. 1 Details of local

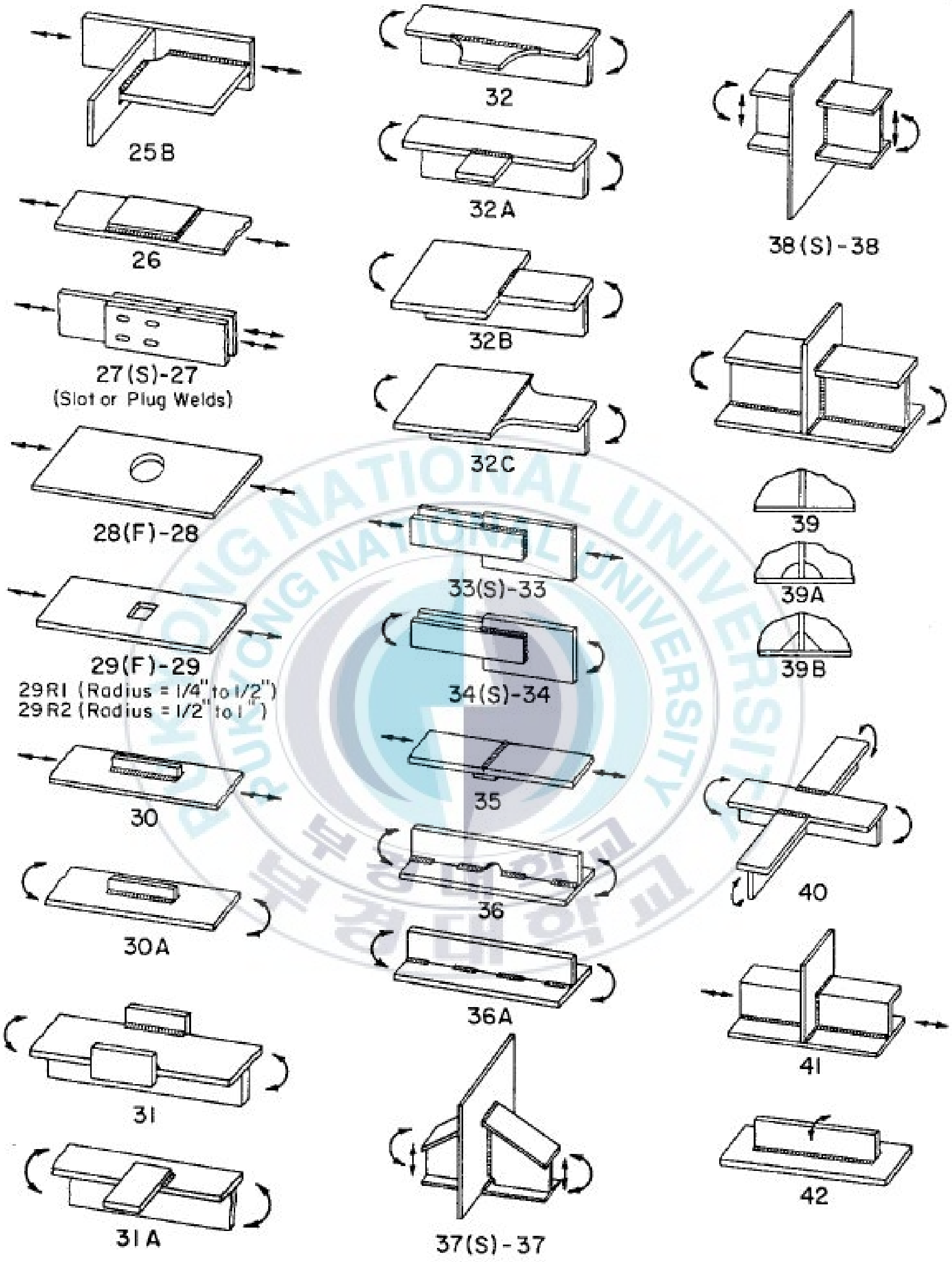
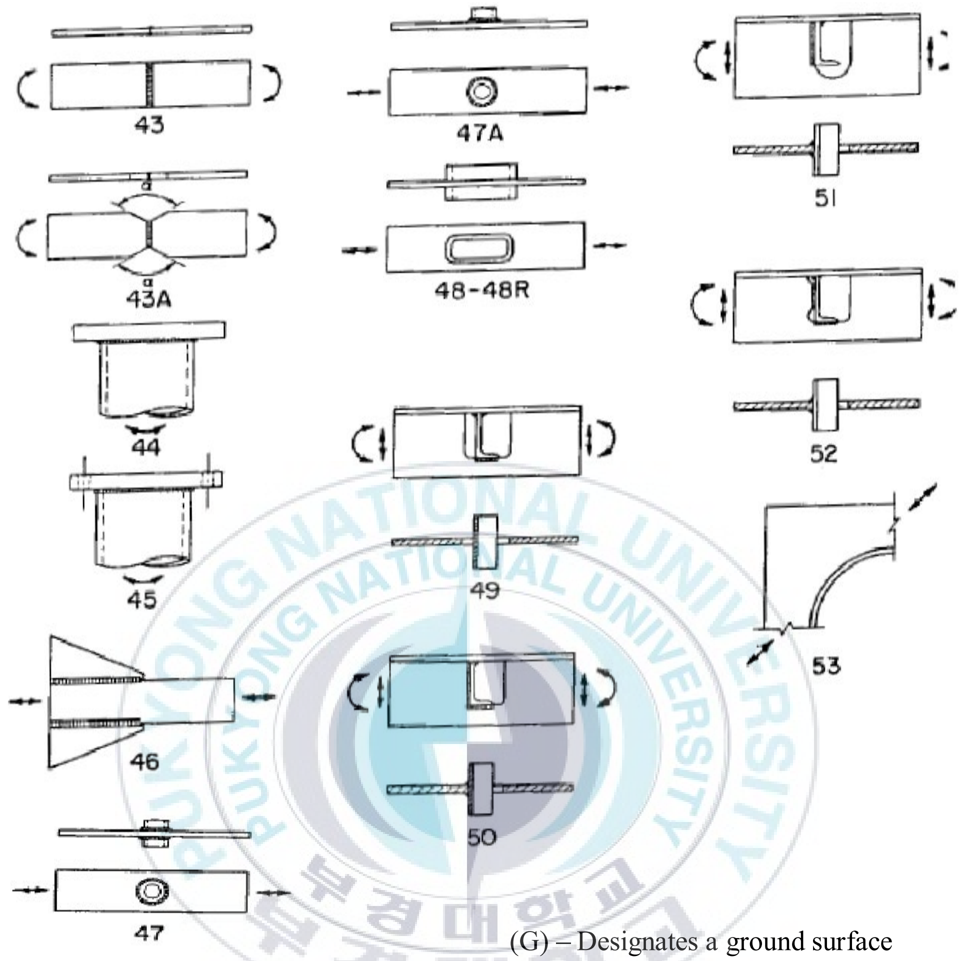


Fig. 3.1 Details of local (Cont.)



(G) – Designates a ground surface
(S) – Designates shear on weld or fastener
(F) – Designates flame cut edges for
comparison with machined edges

Fig. 3.1 Details of local (Cont.)

Table 3. 1 Mean fatigue strength for range of fatigue of details

Detail No. (See Fig.3.1)	S-N curve slope, m	Stress range, ksi, for n cycles			
		$n=10^5$	$n=10^6$	$n=10^7$	$n=10^8$
1 (all steels)	5.729	69.4	46.5	31.1	20.8
1M	12.229	46.2	38.3	31.7	26.3
1H	15.449	56.3	48.5	41.8	36.0
1Q	5.199	80.6	51.8	33.2	21.3
1(F)	4.805	67.1	41.5	25.7	15.9
2	6.048	61.5	42.0	28.7	19.6
3	5.946	44.6	30.3	20.5	13.9
3(G)	6.370	44.9	31.3	21.8	15.2
4	5.663	42.5	28.3	18.8	12.5
5	3.278	26.3	13.0	6.4	3.2
6	5.663	42.5	28.3	18.8	12.5
7(B)	3.771	44.8	24.3	13.2	7.2
7(P)	4.172	35.5	20.4	11.8	6.8
8	6.549	55.8	39.2	27.6	19.4
9	9.643	32.6	25.7	20.2	15.92
10M	7.589	34.1	25.2	18.6	13.7
10H	12.795	43.2	36.1	30.1	25.2
10Q	5.124	48.9	31.2	19.9	12.7
10G	7.130	47.1	34.1	24.7	17.9
10A	5.468	47.1	30.9	20.3	13.3
10A(G)	--	--	--	--	--
11	5.765	33.2	22.3	14.9	10.0
12	4.398	33.2	19.6	11.6	6.9
12(G)	5.663	40.8	27.2	18.09	12.05
13	4.229	48.3	28.0	16.3	9.44
14	7.439	40.6	29.8	21.8	16.03
14A	--	--	--	--	--
15	4.200	24.4	14.1	8.2	4.7
16*	4.631	32.8	19.9	12.1	7.37
16(G)*	6.960	32.8	23.6	16.9	12.2

Table 3.1 Mean fatigue strength for range of fatigue of details (Cont.)

Detail No. (See Fig.3.1)	S-N curve slope, m	Stress range, ksi, for n cycles			
		$n=10^5$	$n=10^6$	$n=10^7$	$n=10^8$
17	3.736	27.8	15.0	8.1	4.4
17(S)	7.782	28.2	21.0	15.6	11.6
17A	3.456	30.4	15.6	8.0	4.1
17A(S)	7.782	28.2	21.0	15.6	11.6
18	4.027	20.3	11.5	6.5	3.6
18(S)	9.233	25.7	20.0	15.6	12.2
19	7.472	23.1	17.0	12.5	9.2
19(S)	7.520	27.5	20.3	14.9	11
20	4.619	26.5	16.1	9.8	5.9
20(S)	6.759	27.5	19.6	13.9	9.9
21(1/4” weld)	14.245	33.5	28.5	24.2	20.6
21(3/8” weld)	15.494	21	18.1	15.6	13.4
21(S)	7.358	42.4	31.0	22.7	16.6
22	3.147	39.8	19.2	9.2	4.4
23	3.187	35.4	17.2	8.3	4.1
24	3.187	35.4	17.2	8.3	4.1
25	7.090	33.2	24.0	17.4	12.5
25A	8.518	49.9	38.1	29.1	22.5
25B	6.966	28.6	20.6	14.8	10.6
26	3.348	34.0	17.1	8.6	4.3
27	3.146	25.0	12.0	5.8	2.8
27(S)	5.277	21.8	14.1	9.1	5.9
28	7.746	40.1	29.8	22.1	16.4
28(F)	--	29.3	--	--	--
30	3.159	34.7	16.7	8.1	3.9
30A	3.368	45.6	23.0	11.6	5.8
31	4.348	20.16	11.87	6.99	4.12
31A	3.453	30.6	15.7	8.1	4.1
32A	4.200	24.4	14.1	8.2	4.7
32B	3.533	21.5	11.21	5.84	3.04

Table 3.1 Mean fatigue strength for range of fatigue of details (Cont.)

Detail No. (See Fig.3.1)	S-N curve slope, m	Stress range, ksi, for n cycles			
		$n=10^5$	$n=10^6$	$n=10^7$	$n=10^8$
33	3.660	21.3	11.4	6.1	3.2
33(S)	10.368	25.5	20.5	16.4	13.1
35	3.808	32.4	17.7	9.7	5.3
36	6.966	28.6	20.6	14.8	10.6
36A	5.163	33.6	21.5	13.8	8.8
38	3.462	31.1	16.0	8.2	4.2
38(S)	10.225	16.3	13.0	10.4	8.3
40	3.533	21.5	11.21	5.84	3.04
42	7.358	42.4	31.0	22.7	16.6
46	4.348	20.16	11.87	6.99	4.12
51(V)	3.813	35.9	19.6	10.8	5.87
52(V)	4.042	34.9	19.8	11.2	6.32

2. Uncertainty- Coefficient of Variation

The reliability model for fatigue design is a function of the “total” uncertainty in fatigue life. For establishing this total uncertainty, all sources of uncertainty should be taken into account: the scatter in the fatigue data; the uncertainty in the fatigue model, the uncertainty in the fatigue damage model (Miner’s linear damage rule); the uncertainty in the stress-range distribution and error in stress analysis; the effects of the quality of fabrication and workmanship; and the uncertainty produced by any other design and fabrication factors.

The measure of total uncertainty in fatigue life Ω_n is given as

$$\Omega_n^2 = \Omega_f^2 + m^2\Omega_s^2 + \Omega_c^2 \tag{3.2}$$

Ω_n is the total uncertainty in fatigue life.

Ω_f is the uncertainty in fatigue data life; $\Omega_f^2 = \delta_f^2 + \Delta_f^2$ in which δ_f is the coefficient of variation in the fatigue life data about the S-N regression line; and the Δ_f is the error in the fatigue model.

Ω_c is the uncertainty in the mean intercept of the S-N regression line and includes in particular the effects of workmanship and fabrication.

Ω_s is measure of total uncertainty in mean stress range, including the effects of impact and error of stress analysis and stress determination.

Table 3. 2 Summary of uncertainties in fatigue parameters

Detail No. (See Fig.3.1)	m	$\log_{10}^{c}$	δ_f	Δ_f	Ω_c	Ω_c	Ω_n
1 (all steels)	5.729	15.55	0.75	0.15	0.40	0.10	1.04
1M	12.229	25.36	0.71	0.15	0.40	0.10	1.48
1H	15.449	32.04	0.91	0.15	0.40	0.10	1.84
1Q	5.199	14.91	0.68	0.15	0.40	0.10	0.96
1(F)	4.805	13.78	0.60	0.15	0.40	0.10	0.88
2	6.048	15.82	0.64	0.15	0.40	0.10	0.98
3	5.946	14.80	0.63	0.15	0.40	0.10	0.96
3(G)	6.370	15.52	0.74	0.15	0.40	0.10	1.07
4	5.663	14.22	0.61	0.15	0.40	0.10	0.93
5	3.278	9.65	0.48	0.15	0.40	0.10	0.72
6	5.663	14.22	0.61	0.15	0.40	0.10	0.93
7(B)	3.771	11.23	0.53	0.15	0.40	0.10	0.78
7(P)	4.172	11.46	0.51	0.15	0.40	0.10	0.78
8	6.549	16.44	0.81	0.15	0.40	0.10	1.13
9	9.643	19.59	0.90	0.15	0.40	0.10	1.39
10M	7.589	16.63	0.88	0.15	0.40	0.10	1.24
10H	12.795	25.92	0.96	0.15	0.40	0.10	1.66

Table 3.2 Summary of uncertainties in fatigue parameters (Cont.)

Detail No. (See Fig.3.1)	m	$\log_{10}^{c}$	δ_f	Δ_f	Ω_c	Ω_c	Ω_n
10Q	5.124	13.65	0.76	0.15	0.40	0.10	1.01
10G	7.130	16.93	0.94	0.15	0.40	0.10	1.25
10A	5.468	14.14	0.79	0.15	0.40	0.10	1.05
10A(G)	--	--	--	0.15	0.40	0.10	--
11	5.765	13.77	0.68	0.15	0.40	0.10	0.99
12	4.398	11.69	0.43	0.15	0.40	0.10	0.75
12(G)	5.663	14.12	0.60	0.15	0.40	0.10	0.93
13	4.229	12.12	0.45	0.15	0.40	0.10	0.75
14	7.439	16.96	0.91	0.15	0.40	0.10	1.25
14A	--	--	--	0.15	0.40	0.10	--
15	4.200	10.83	0.43	0.15	0.40	0.10	0.74
16*	4.631	12.02	0.58	0.15	0.40	0.10	0.85
16(G)*	6.960	15.55	0.95	0.15	0.40	0.10	1.25
17	3.736	10.39	0.34	0.15	0.40	0.10	0.66
17(S)	7.782	16.28	0.65	0.15	0.40	0.10	1.10
17A	3.456	10.14	0.39	0.15	0.40	0.10	0.67
17A(S)	7.782	16.28	0.65	0.15	0.40	0.10	1.10
18	4.027	10.26	0.65	0.15	0.40	0.10	0.88
18(S)	9.233	18.02	0.75	0.15	0.40	0.10	1.26
19	7.472	15.19	0.93	0.15	0.40	0.10	1.27
19(S)	7.520	15.83	0.93	0.15	0.40	0.10	1.27
20	4.619	11.57	0.66	0.15	0.40	0.10	0.92
20(S)	6.759	14.73	0.93	0.15	0.40	0.10	1.22
21(1/4" weld)	14.245	26.72	--	0.15	0.40	0.10	--
21(3/8" weld)	15.494	25.49	--	0.15	0.40	0.10	--
21(S)	7.358	16.98	0.83	0.15	0.40	0.10	1.19
22	3.147	10.04	0.32	0.15	0.40	0.10	0.62
23	3.187	9.94	0.13	0.15	0.40	0.10	0.55
24	3.187	9.94	0.13	0.15	0.40	0.10	0.55
25	7.090	15.79	0.78	0.15	0.40	0.10	1.14

Table 3.2 Summary of uncertainties in fatigue parameters (Cont.)

Detail No. (See Fig.3.1)	m	$\log_{10}^{c}$	δ_f	Δ_f	Ω_c	Ω_c	Ω_n
25A	8.518	19.47	0.91	0.15	0.40	0.10	1.32
25B	6.966	15.15	0.63	0.15	0.40	0.10	1.03
26	3.348	10.13	0.61	0.15	0.40	0.10	0.82
27	3.146	9.40	0.58	0.15	0.40	0.10	0.78
27(S)	5.277	12.06	0.54	0.15	0.40	0.10	0.87
28	7.746	17.41	0.81	0.15	0.40	0.10	1.20
28(F)	--	--	--	0.15	0.40	0.10	--
30	3.159	9.87	0.31	0.15	0.40	0.10	0.62
30A	3.368	10.58	0.10	0.15	0.40	0.10	0.55
31	4.348	10.67	--	0.15	0.40	0.10	--
31A	3.453	10.13	0.44	0.15	0.40	0.10	0.71
32A	4.200	10.83	0.43	0.15	0.40	0.10	0.74
32B	3.533	9.71	--	0.15	0.40	0.10	--
33	3.660	9.86	0.50	0.15	0.40	0.10	0.75
33(S)	10.368	19.59	0.81	0.15	0.40	0.10	1.38
35	3.808	10.75	0.28	0.15	0.40	0.10	0.64
36	6.966	15.15	0.63	0.15	0.40	0.10	1.03
36A	5.163	12.88	0.46	0.15	0.40	0.10	0.81
38	3.462	10.17	0.36	0.15	0.40	0.10	0.66
38(S)	10.225	17.39	0.88	0.15	0.40	0.10	1.42
40	3.533	9.71	--	0.15	0.40	0.10	--
42	7.358	16.98	0.83	0.15	0.40	0.10	1.19
46	4.348	10.67	--	0.15	0.40	0.10	--
51(V)	3.813	10.93	0.07	0.15	0.40	0.10	0.58
52(V)	4.042	11.24	0.19	0.15	0.40	0.10	0.62

3. Reliability Factor

The fatigue life of a structural detail is a random variable, and it is assumed that the distribution of life can be represented by the Weibull distribution. This distribution is often used in fatigue for a variety of reasons.

If $L(n)$ is the probability of survival through a given number of loading cycles, then,

$$L(n) = L(n-1)[1 - h(n)] \tag{3.3}$$

where $h(n)$ is the hazard function.

Assuming the Weibull Distribution for fatigue life, the hazard function is given by,

$$h(n) = \frac{k}{w - \varepsilon}\left(\frac{n-\varepsilon}{w-\varepsilon}\right)^{k-1}; k \geq 1.0 \tag{3.4}$$

where ε is the minimum life; w is the characteristic life; k is the Weibull sharp parameter.

The parameters w, k and ε can be related to $\bar{n}$ and σ_n, the mean fatigue life and its standard deviation, as follows,

$$\bar{n} - \varepsilon = (w - \varepsilon)\Gamma(1 + \frac{1}{k}) \tag{3.5}$$

$$\sigma_n = (w - \varepsilon)[\Gamma(1 + \frac{2}{k}) - \Gamma^2(1 + \frac{1}{k})]^{1/2} \tag{3.6}$$

where Γ is the gamma function.

And if the minimum life ε is assumed to be equal to zero or very small, the ratio of the standard deviation to the mean life is,

$$\frac{\sigma_n}{\bar{n} - \varepsilon} = \frac{[\Gamma(1 + \frac{2}{k}) - \Gamma^2(1 + \frac{1}{k})]^{1/2}}{\Gamma(1 + \frac{1}{k})} \quad \text{or} \quad \frac{\sigma_n}{\bar{n}} \cong \Omega_n \tag{3.7}$$

The shape parameter k can be approximated by ,

$$k \cong {\Omega_n}^{-1.08} \tag{3.8}$$

This equation is shown in Fig.3.2. The use of this approximation greatly simplifies the development of a simple reliability relationship.

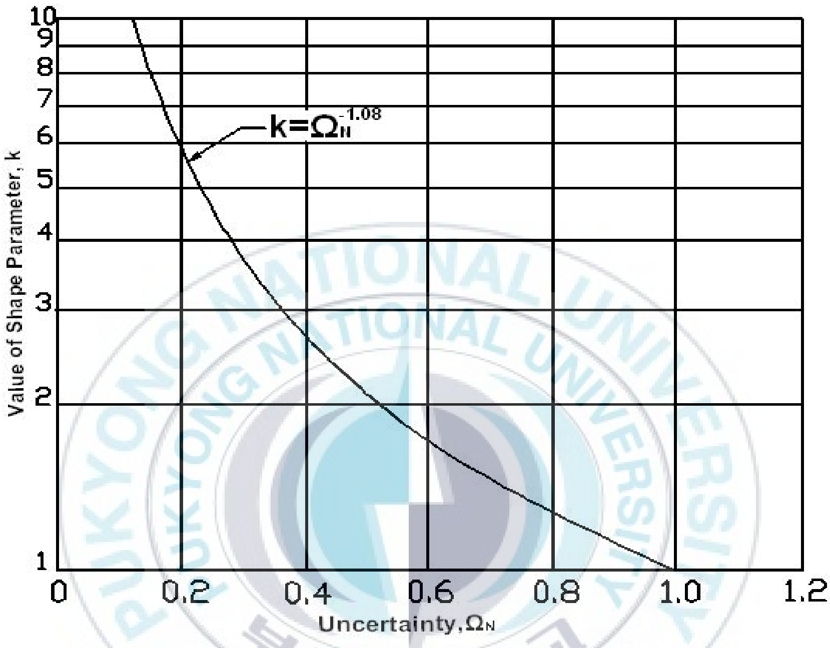


Fig. 3. 2 Relationship between k and Ω_n

Using the Weibull distribution and the hazard function of Eq. (3.4), the reliability function can be expressed as,

$$L(n) = \begin{cases} exp[-(\frac{n-\varepsilon}{w-\varepsilon})^k]; n \geq \varepsilon \\ 1.0; n < \varepsilon \end{cases} \tag{3.9}$$

Introducing Eqs.3.5 and 3.9, this can be written as,

$$L(n) = \exp\{-[\frac{n}{\bar{n}}\Gamma(1+\Omega_n^{1.08})]^{\Omega_n^{-1.08}} \tag{3.10}$$

Then, for a specified probability of survival, $L(n)$,

$$\frac{n-\varepsilon}{w-\varepsilon}=[-\ln L(n)]^{1/k} \cong [P_f(n)]^{1/k} \tag{3.11}$$

and for a mean life $\bar{n}$,

$$\frac{\bar{n}-\varepsilon}{n-\varepsilon} \cong \frac{\Gamma(1+\frac{1}{k})}{[-\ln L(n)]^{1/k}} \cong \frac{\Gamma(1+\frac{1}{k})}{[P_f(n)]^{1/k}} \tag{3.12}$$

This ratio has been defined as the fatigue life factor, γ_L ,

$$\gamma_L = \frac{\Gamma(1+\Omega_n^{1.08})}{[P_f(n)]^{\Omega_n^{1.08}}} \quad \text{and} \quad \bar{n} = n\gamma_L \tag{3.13}$$

where $\bar{n}$ is the required mean life that would be necessary to insure a useful life *n* with a reliability of $L(n)$ or probability of failure of $P_f(n)$. The relationships between the fatigue risk factor and the uncertainty in fatigue life for various probabilities of failure are shown in Fig.3.3

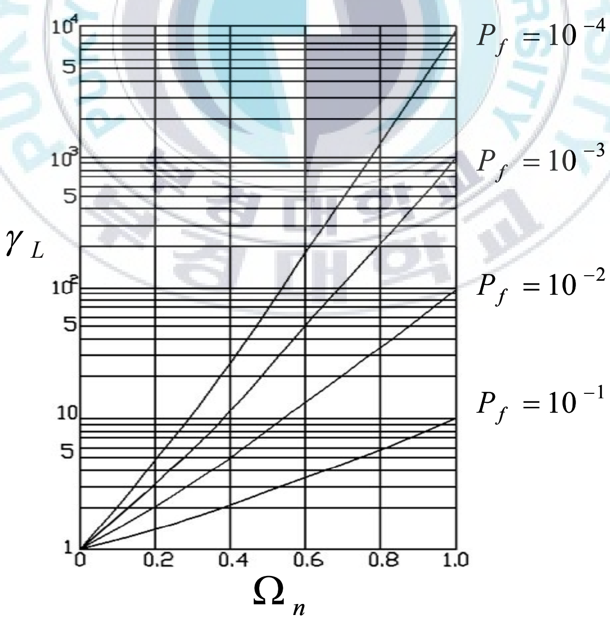


Fig. 3. 3 Relationship of γ_L and the uncertainty for various $P_f(n)$

Under a constant stress range the required design stress is then given by,

$$S_D = (\frac{C}{n})^{1/m} \quad \text{or} \quad S_D = (\frac{C}{n\gamma_L})^{1/m} = (\frac{C}{n})^{1/m}(\frac{1}{\gamma_L})^{1/m} \tag{3.14}$$

Designating the last term of Eq. (3.14) as the reliability factor, R_F ,

$$R_F = (\frac{1}{\gamma_L})^{1/m} = \left\{\frac{[P_f(n)^{\Omega_n^{1.08}}]}{\Gamma(1+\Omega_n^{1.08})}\right\}^{1/m} \tag{3.15}$$

Then, the allowable design stress would be,

$$S_D = (\frac{C}{n})^{1/m} \cdot R_F = S \cdot R_F \tag{3.16}$$

where $1/m$ corresponds to the slope of the S-N curve for the member in question, C is the intercept of the S-N curve, and S is the stress range corresponding to the desired useful life n. A summary of computed values of reliability factors for three levels of reliability (0.90, 0.95, 0.99) and five coefficients of uncertainty (0.40, 0.60, 0.80, 1.0, 1.2) are given in Fig. 3.4& Fig. 3.5 & Fig. 3.6. Summary of Reliability Factor R_F as shown in Table 3.3.

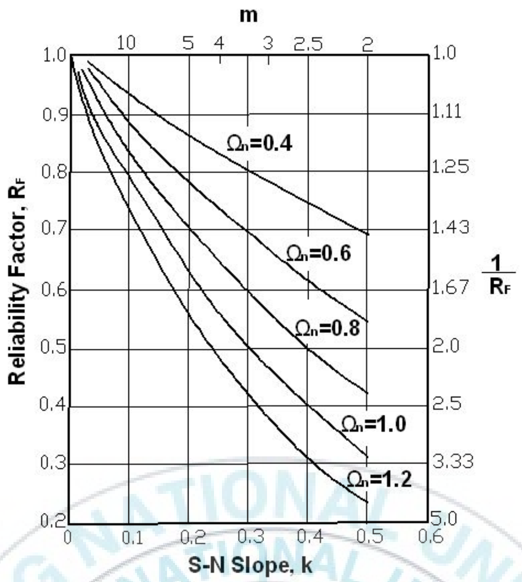


Fig. 3. 4 Reliability factor VS S-N slope 90% reliability

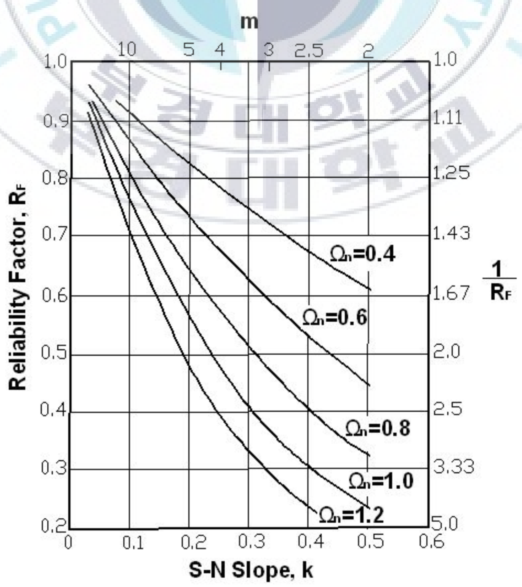


Fig. 3. 5 Reliability factor VS S-N slope 95% reliability

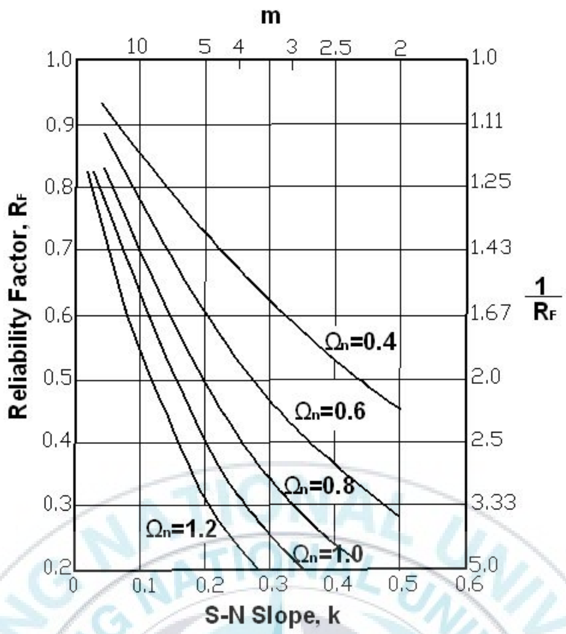


Fig. 3. 6 Reliability factor VS S-N slope 99% reliability

Table 3. 3 Summary of reliability factor R_F for local details

Detail No. (See Fig.3.1)	m	Ω_n	Reliability, $L(n)$		
			90%	95%	99%
1 (all steels)	5.729	1.04	0.655	0.578	0.431
1M	12.229	1.48	0.732	0.671(M)	0.549(M)
1H	15.449	1.84	0.719	0.660	0.540
1Q	5.199	0.96	0.657	0.578	0.430
1(F)	4.805	0.88	0.666	0.587	0.438
2	6.048	0.98	0.690	0.617	0.475
3	5.946	0.96	0.692	0.619	0.478
3(G)	6.370	1.07	0.674	0.600	0.457
4	5.663	0.93	0.690	0.616	0.474
5	3.278	0.72	0.629	0.542	0.384
6	5.663	0.93	0.690	0.616	0.474
7(B)	3.771	0.78	0.640	0.557	0.402

Table 3.3 Summary of reliability factor R_F for local details (Cont.)

Detail No. (See Fig.3.1)	m	Ω_n	Reliability, $L(n)$		
			90%	95%	99%
7(P)	4.172	0.78	0.668	0.589	0.438
8	6.549	1.13	0.663	0.587	0.444
9	9.643	1.39	0.694	0.626	0.494
10M	7.589	1.24	0.670	0.597	0.457
10H	12.795	1.66	0.707	0.644	0.518
10Q	5.124	1.01	0.634	0.553	0.403
10G	7.130	1.25	0.650	0.575	0.431
10A	5.468	1.05	0.639	0.559	0.410
10A(G)	--	--	--	--	--
11	5.765	0.99	0.674	0.599	0.454
12	4.398	0.75	0.695	0.619	0.474
12(G)	5.663	0.93	0.690	0.616	0.474
13	4.229	0.75	0.685	0.608	0.460
14	7.439	1.25	0.662	0.588	0.447
14A	--	--	--	--	--
15	4.200	0.74	0.688	0.610	0.463
16*	4.631	0.85	0.667	0.589	0.440
16(G)*	6.960	1.25	0.643	0.567	0.422
17	3.736	0.66	0.694	0.617	0.468
17(S)	7.782	1.10	0.725	0.657	0.523
17A	3.456	0.67	0.670	0.588	0.435
17A(S)	7.782	1.10	0.725	0.657	0.523
18	4.027	0.88	0.615	0.530	0.374
18(S)	9.233	1.26	0.715	0.649	0.519
19	7.472	1.27	0.658	0.583	0.441
19(S)	7.520	1.27	0.659	0.585	0.444
20	4.619	0.92	0.639	0.557	0.405
20(S)	6.759	1.22	0.644	0.567	0.422
21(1/4” weld)	14.245	--	--	--	--
21(3/8” weld)	15.494	--	--	--	--

Table 3.3 Summary of reliability factor R_F for local details (Cont.)

Detail No. (See Fig.3.1)	*m*	Ω_n	Reliability, $L(n)$		
			90%	95%	99%
21(S)	7.358	1.19	0.676	0.604	0.464
22	3.147	0.62	0.670	0.587	0.432
23	3.187	0.55	0.600	0.635	0.411
24	3.187	0.55	0.600	0.635	0.411
25	7.090	1.14	0.681	0.608	0.468
25A	8.518	1.32	0.679	0.609	0.472
25B	6.966	1.03	0.709	0.640	0.504
26	3.348	0.82	0.586(m)	0.496	0.336
27	3.146	0.78	0.586(m)	0.495(m)	0.335(m)
27(S)	5.277	0.87	0.694	0.620	0.477
28	7.746	1.20	0.687	0.616	0.478
28(F)	--	--	--	--	--
30	3.159	0.62	0.671	0.589	0.434
30A	3.368	0.55	0.724	0.650	0.506
31	4.348	--	--	--	--
31A	3.453	0.71	0.649	0.565	0.409
32A	4.200	0.74	0.688	0.610	0.463
32B	3.533	--	--	--	--
33	3.660	0.75	0.646	0.562	0.407
33(S)	10.368	1.38	0.714	0.650	0.522
35	3.808	0.64	0.709	0.633	0.488
36	6.966	1.03	0.709	0.640	0.504
36A	5.163	0.81	0.711	0.639	0.498
38	3.462	0.66	0.675	0.594	0.441
38(S)	10.225	1.42	0.702	0.635	0.505
40	3.533	--	--	--	--
42	7.358	1.19	0.676	0.604	0.464
46	4.348	--	--	--	--
51(V)	3.813	0.58	0.738(m)	0.667	0.528
52(V)	4.042	0.62	0.732	0.661	0.521

4. Variable Loading–Random Load Factor

In considering fatigue in terms of a constant amplitude stress-range, the mean fatigue life is given by the well know S-N relationship like Eq. (3.1). However, such a relationship cannot be applied directly to ship structures that are subjected to a variable or random loading. Other relationships must be developed to modify it.

A relationship between a variable amplitude stress range and the mean fatigue life, comparable to Eq. (3.1), has been presented utilizing the S-N relationship and the Palmgren-Miner linear damage rule.

From the Miner damage rule, we can obtain,

$$D = \sum_{i=1}^{k} \frac{n_i}{\bar{n}(S_i)} \tag{3.17}$$

A variable amplitude stress range can be considered as a random variable S with a probability density function $f_s(s)$. Then, for n cycles of the variable stress range, the number of cycles at stress range $S = s$ is $nf_s(s)ds$. Based on Eq. (3.17), the expected cumulative damage is,

$$E(D) = \int_0^\infty \frac{nf_s(s)ds}{\bar{n}(s)} \tag{3.18}$$

Where $\bar{n}(s)$ is mean fatigue life under constant amplitude stress range s. $f_s(s)$ is a probability density function representing the random cyclic stress range.

Introducing the basic relationship of Eq. (3.1) and equating the damage to 1.0, the damage relationship may be written,

$$E(D) = \frac{\bar{n}}{C} \int_0^\infty S^m f_S(s)ds = 1.0 \tag{3.19}$$

Rearranging terms yields,

$$\bar{n} = \frac{C}{\int_0^\infty s^m f_S(s)ds} = \frac{C}{E(S^m)} \tag{3.20}$$

where $E(S^m)$ is m^{th} moment of S, the randomly varying stress range (or the expected value of S^m).

Eq. (3.20) represents the relationship between an applied variable amplitude stress range and the mean fatigue life.

Very little fatigue testing has been done under variable loading conditions. In order to utilize the vast amount of constant amplitude fatigue data (i.e., S-N curve data) available, it is necessary to develop a relationship between the constant-cycle and variable-cycle cases. For a given detail, a random stress range S can be related to a constant-cycle stress range Sc with the same mean fatigue life by combining Eqns.3.1 and 3.20 to give the following,

$$E(S^m) = S_C^m \quad \text{or} \quad [E(S^m)]^{1/m} = S_C \tag{3.21}$$

Eq. (3.21) applies to any distribution of applied stress range S. A convenient design relationship can then be developed from Eq. (3.21) by introducing a random load factor, ξ , such that,

$$[E(S^m)]^{1/m} = \frac{S_0}{\xi} = S_C \quad \text{or} \quad S_0 = \xi \cdot S_C \tag{3.22}$$

where S_0 is the maximum stress range in a random loading that can be represented by a β Distribution (For the other distributions presented herein the value is the maximum stress range expected only once in 10^8 cycles of loading, $S_{10}{}^{-8}$); ξ is “ random load factor”.

By combining the above equations, the following general relationships are obtained,

$$E(S^m) = \int_0^\infty S^m f_S(s)ds = (\frac{S_0}{\xi})^m = (S_C)^m \tag{3.23}$$

$$\xi = \frac{S_0}{[\int_0^\infty S^m f_S(s)ds]^{1/m}} = \frac{S_0}{[E(S^m)]^{1/m}} \tag{3.24}$$

and $$S_C = \frac{S_0}{\xi} = [\int_0^\infty S^m f_S(s)ds]^{1/m} \tag{3.25}$$

Thus, the constant-cycle stress range representing the variable load distribution can be represented as a function of the constant amplitude stress range having the same mean fatigue life $\bar{n}$. Shown as Fig. 3.7,

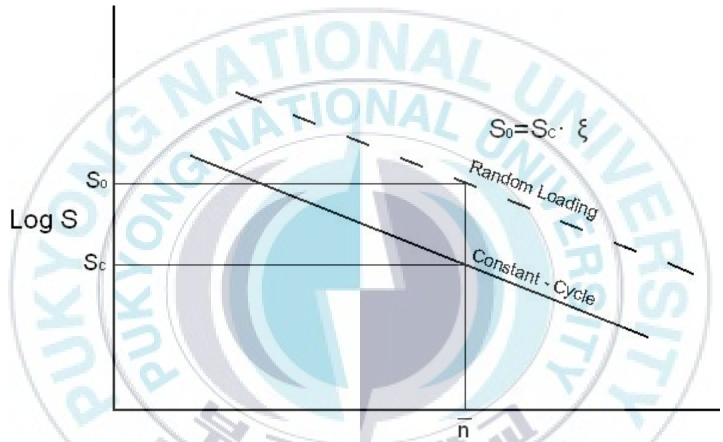


Fig. 3. 7 Max-stress range of random loading and constant-cycle

For the Weibull distribution function , the random load factor in terms of $S_{10^{-8}}$ is a function of the inverse slope m of the S-N curve for the detail to which the loading is to be applied and the Weibull shape parameter k,

$$\xi = (18.42)^{1/k}[\Gamma(1+\frac{m}{k})]^{-1/m} \tag{3.26}$$

A summary of the random load factors for various values of m and k is presentd in Fig.3.8.

Form Eq. (3.16) and Eq. (3.26) we can know that the stress range of design for variable loading can be represented as,

$$S_D = S_N \cdot \xi \cdot R_F \tag{3.27}$$

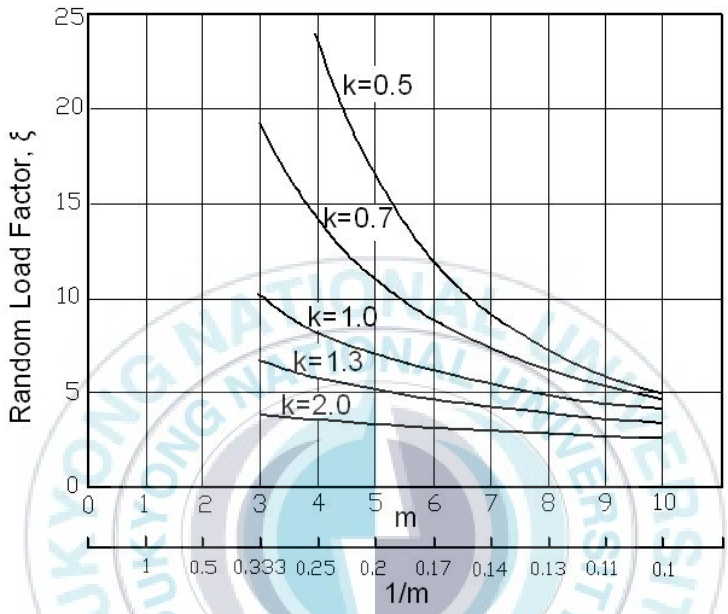


Fig. 3. 8 Variation of ξ with m for various Weibull shapes k

IV. Numerical Calculation for a 17K Oil Tanker

Numerical calculation for 17K Oil Tanker was presented in this chapter. Two kinds of fatigue assessment method would be used and the results would be compared with each other. The fatigue processes as following:

①. Take a modeling of tanker with MSC.PATRAN

②. Create load condition for Ballast and Full Load.

③. Find out the position with maximum stress.

④. Take fatigue analysis by new method for the position.

⑤. Fine mesh the local detail and take fatigue analysis

⑥. Compare results obtained from two methods.

⑦. Take reliability assessment, stress range data obtained from new method.

1. Basic Data and Principal Dimension of a 17K Oil Tanker

The principal dimension and part of basic data of 17,000 Ton Oil Tanker were presented in Table 4.1. And the mid-section was shown as Fig.4.1,

Table 4. 1 Principal dimension and basic data

Items	Symbol	Value
Length	L_{OA}	144.00 (m)
Length	L_{BP}	136.00 (m)
Breadth	B	22.60 (m)
Depth	D	12.50 (m)
Draft	T_d	9.10 (m)
Speed	V	14 (Kt)
Block Coefficient	C_B	0.795
I_{hor}	--	$8.94 x 10^{13}$ (mm^4)
I_{ver}	--	$1.89 x 10^{14}$ (mm^4)
Dis. form N.A. to	--	5471 (mm)

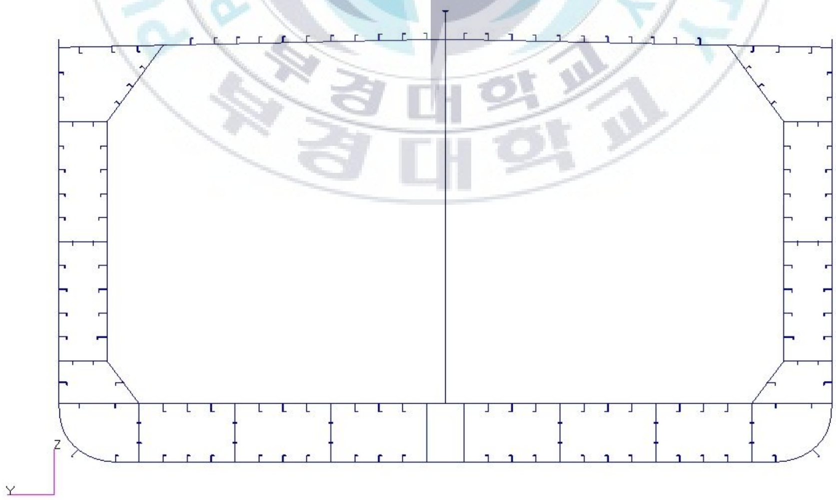


Fig. 4. 1 Mid-section of tanker

2. Modeling

The NO.5 tanker have been analyzed, and the position and model as shown in Fig.4.2. Because of the construction of tanker are symmetrical, half taker was modeling for analysis as shown in Fig.4.3 (Cargo hold model is normally to cover the considered tank/hold and addition one half tank/hold outsider each end of considered tank/hold, i.e. the model extent is 1/2+1+1/2 hold or tank).

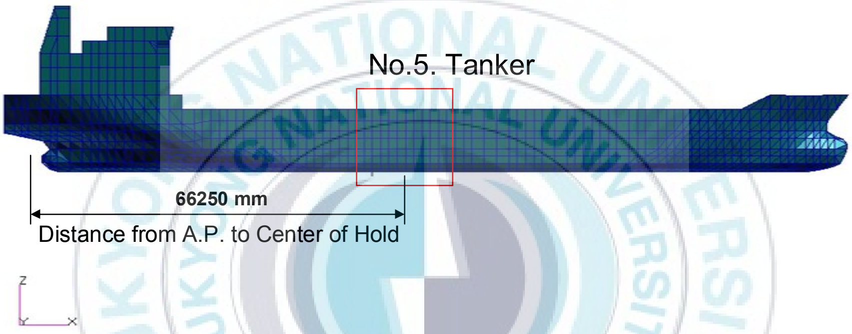


Fig. 4. 2 Position of considered tanker

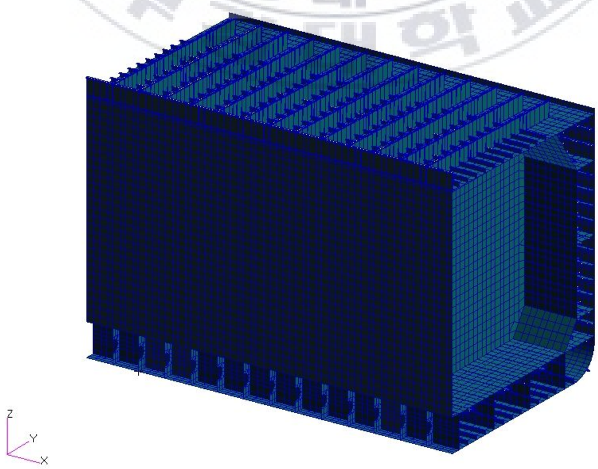


Fig. 4. 3 Finite element model of considered tanker

3. Load Condition

The internal pressure and external pressure for different condition as shown in Table 4.2 and Fig.4.4 & Fig. 4.5,

Table 4. 2 Load condition

Items	Value	
	Ballast	Full Load
ρ for internal	1.025 (t/ m^3)	0.90 (t/ m^3)
ρ for external	1.025 (t/ m^3)	1.025 (t/ m^3)
T_d	5.6 (m)	9.10 (m)
Part of time	0.4	0.45

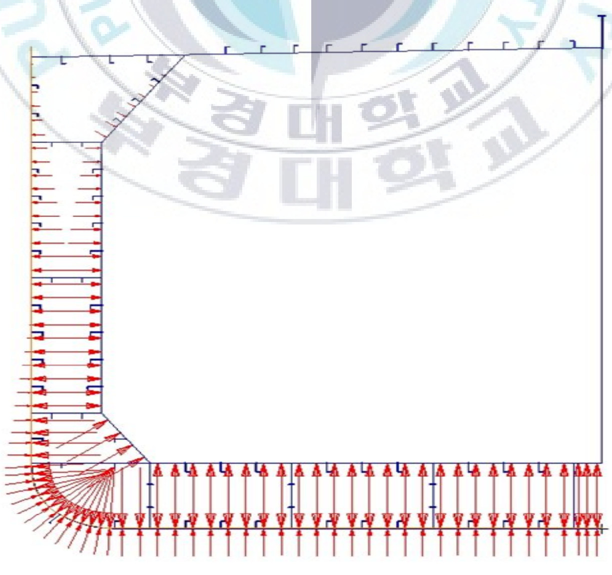


Fig. 4. 4 Load case for ballast condition

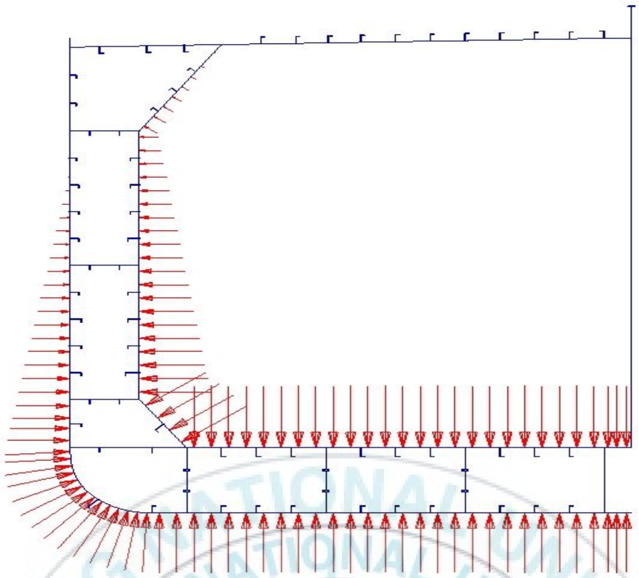


Fig. 4. 5 Load case for full load condition

4. Considered Position

After calculation, we can find out the maximum stress of hold, and take the position as assessment object. See Fig.4.6,

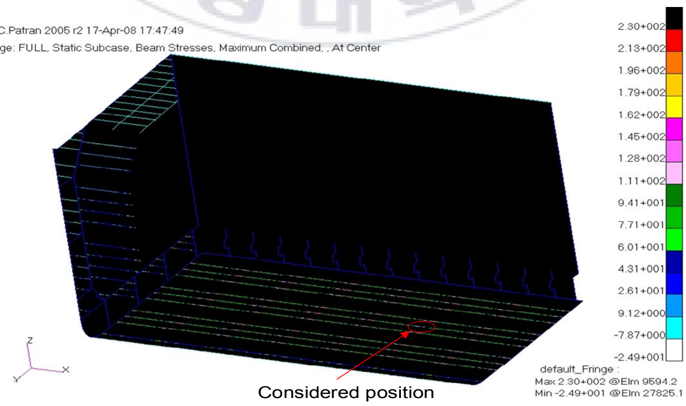


Fig. 4. 6 Considered position

The stresses are to be calculated at the considered point on the weld connection stiffener and longitudinal, as shown in Fig. 4.7 & Fig. 4.8. The dimensions of stiffener given in Table 4.3,

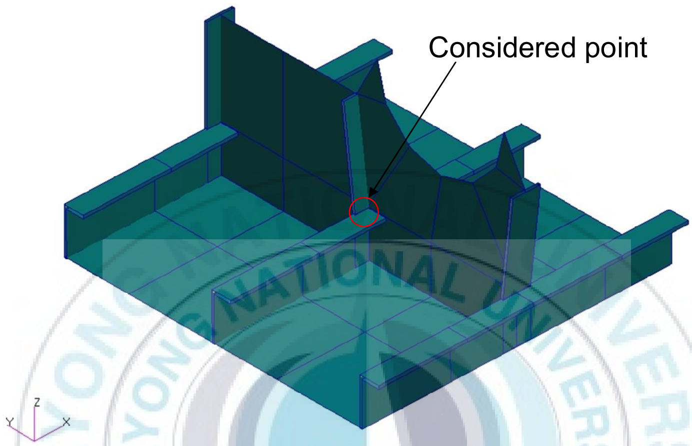


Fig. 4. 7 FE model of considered position

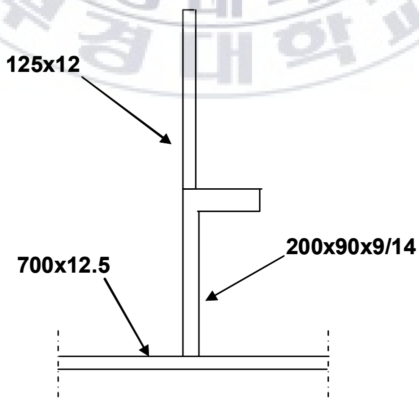


Fig. 4. 8 Details of considered position

Table 4. 3 Geometry of stiffener considered

Description	Symbol	Value
Stiffener sectional modulus at top of flange	Zs	5.83×10^5 (mm^3)
Distance above keel	z	176.3 (mm)
Effective span length	l	2675 (mm)
Web frame spacing	l_s	2800 (mm)
Stiffener spacing	s	700 (mm)
Thickness of plate	t_p	12.5 (mm)
Height of stiffener	h	200 (mm)
Thickness of web	t_w	9 (mm)
Width of flange	b_f	90 (mm)
Thickness of flange	t_f	14 (mm)
Distance from neutral axis to top flange	z_{01}	176.3 (mm)

5. Calculation Using New Method

The data of calculation and result obtain from new method as shown in Table.4.4 and 4.5,

Table 4. 4 Ballast condition

Description	Symbol	Value
K-factor for axial stresses	K_{axial}	2
K-factor for local stiffener bending stresses	$K_{lateral}$	4.117
Stress from internal pressure loads	σ_i	109 (KN/m^2)
Stress from external pressure loads	σ_e	-115 (KN/m^2)
Combined local stress	$\Delta\sigma_l$	212 (KN/m^2)
Wave sagging moment amplitude	$M_{wo,s}$	-350483.7 (KNm)
Wave hagging moment amplitude	$M_{wo,h}$	322869.84 (KNm)
Horizontal wave bending moment amplitude	M_H	161008.88 (KNm)
Combined global stress	$\Delta\sigma_g$	82.67 (KN/m^2)
Combined global and local stress range	$\Delta\sigma_0$	178 (KN/m^2)
Part damage in Ballast	D	0.233

Table 4. 5 Full load condition

Description	Symbol	Value
K-factor for axial stresses	K_{axial}	2
K-factor for local stiffener bending stresses	$K_{lateral}$	4.117
Stress from internal pressure loads	σ_i	58.9 (KN/m^2)
Stress from external pressure loads	σ_e	-117 (KN/m^2)
Combined local stress	$\Delta\sigma_l$	195 (KN/m^2)
Wave sagging moment amplitude	$M_{wo,s}$	-350483.7 (KNm)
Wave hagging moment amplitude	$M_{wo,h}$	322869.84 (KNm)
Horizontal wave bending moment amplitude	M_H	206528.35 (KNm)
Combined global stress	$\Delta\sigma_g$	84.16 (KN/m^2)
Combined global and local stress range	$\Delta\sigma_0$	167 (KN/m^2)
Part damage in Full Load	D	0.212

When considering the corrosive environment the χ-factor selected as 1.3, the total damage from Eq. (2.1) and fatigue life should be,

$$D = (0.233 + 0.212) \cdot 1.3 = 0.578$$

$$T_{life} = 20/D = 20/0.578 = 34.6 \text{ years}$$

6. Calculation Using Fine Mesh Method

The different from the new method in here is that model should be fine mesh and the result of calculated stress could be used directly. The fine meshed model and result as shown in Fig.4.9&4.10,

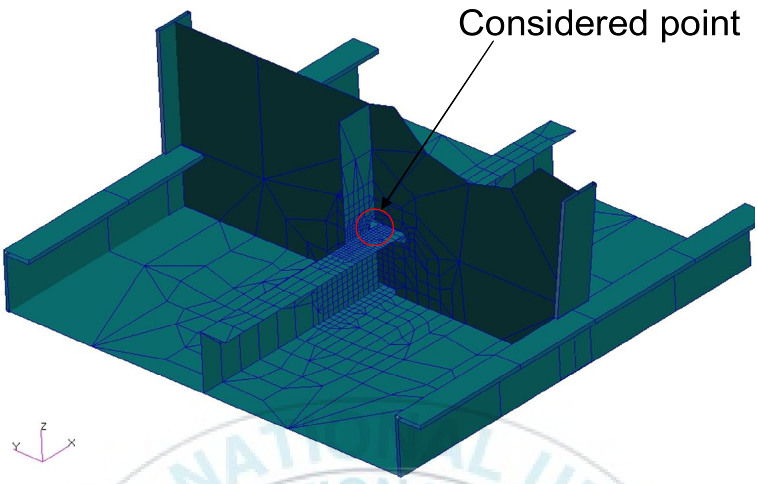


Fig. 4. 9 Fine mesh for details of considered position

Fig. 4. 10 Stress range of considered point

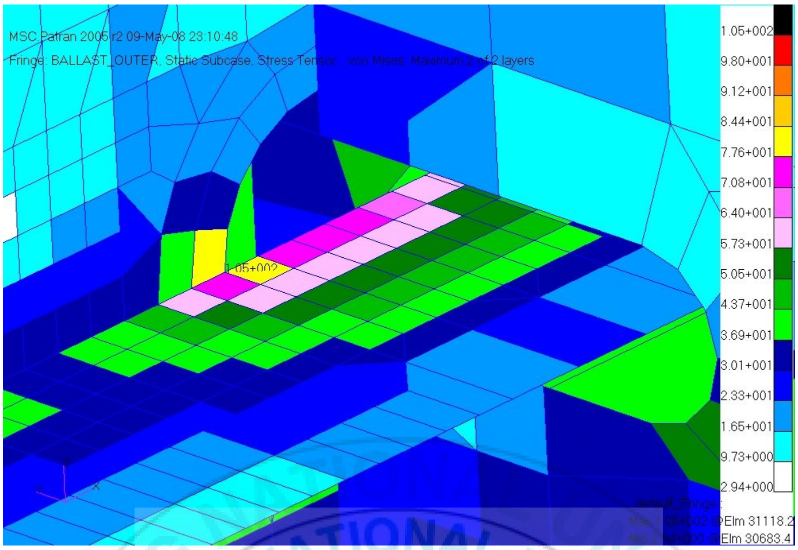


Fig. 4. 11 Zooming-01 for stress range of considered point

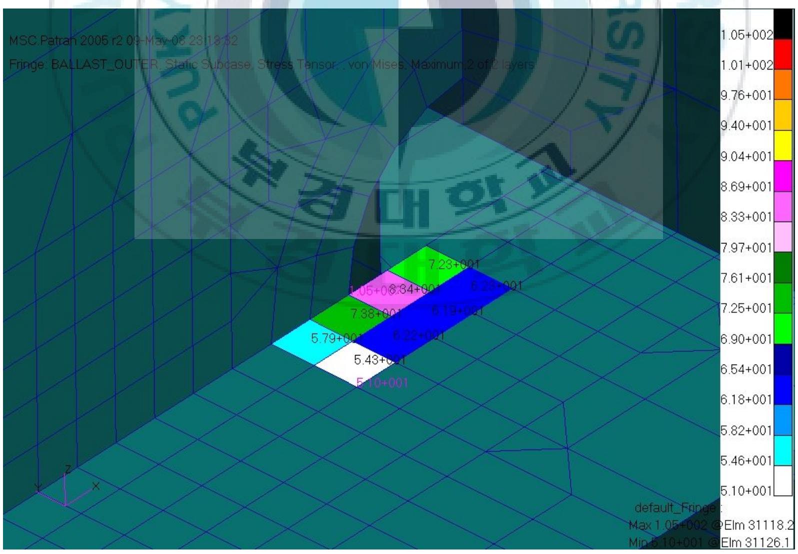


Fig. 4. 12 Zooming-02 for stress range of considered point

Table 4. 6 Result from fine mesh mothod

Description (Full Load)	Symbol	Value
Stress from internal pressure loads	σ_i	57.8 (KN/m^2)
Stress from external pressure loads	σ_e	130 (KN/m^2)
Combined local stress	$\Delta\sigma_l$	254.6 (KN/m^2)
Combined global stress	$\Delta\sigma_g$	0
Combined global and local stress range	$\Delta\sigma_0$	173.16 (KN/m^2)
K-factor	$k_w \cdot k_g$	1.5
Part damage in Full Load	D	0.236
Description (Ballast)	**Symbol**	**Value**
Stress from internal pressure loads	σ_i	124.2 (KN/m^2)
Stress from external pressure loads	σ_e	79.6(KN/m^2)
Combined local stress	$\Delta\sigma_l$	271.16 (KN/m^2)^2
Combined global stress	$\Delta\sigma_g$	0
Combined global and local stress range	$\Delta\sigma_0$	184.39 (KN/m^2)
K-factor	$k_w \cdot k_g$	1.5
Part damage in Ballast	D	0.257

When considering the corrosive environment the χ-factor selected as 1.3, the total damage from Eq. (2.1) and fatigue life should be,

$$D = (0.236 + 0.257) \cdot 1.3 = 0.641$$

$$T_{life} = 20/D = 20/0.641 = 31.2 \text{ years}$$

7. Reliability Assessment of a 17K Oil Tanker

For reliability assessment, according compare constructions, the fatigue detail of considered position can be considered like details of local No.30 as shown in Fig 4.11,

Fig. 4. 13 Construction comparing

Fatigue data from the constant amplitude stress range experiment, the real stress accord with Weibull distribution, so that random load factor ξ should be calculated. The design life is 20 years, the stress cycles are 10^8. From Eq. (3.26), we obtain that,

$$\xi = (18.42)^{1/k}[\Gamma(1+\frac{m}{k})]^{-1/m} = 9.92$$

where k=1.02 from Chapter 2.3.1. m=3.159 from Table 3.1. And when fatigue life n=10^8, the reliability factor R_F can obtained from Table3.2. The mean fatigue strength of fatigue detail as shown in Table 4.7.

From Eq. (3.27), $S_{RN} = S_N \cdot \xi \cdot R_F$, the stress range can be calculated and the relationship between stress range and reliability level as shown in Table 4.8,

Table 4. 7 Mean fatigue strength

Items	Value			
Cycles n	10^5	10^6	10^7	10^8
Mean stress range (Mpa) S	239.1	115.1	55.8	26.9
Random load factor ξ	9.92	9.92	9.92	9.92

Table 4. 8 Stress for different reliability level

Items	Value		
Reliability level $L(n)$	0.90	0.95	0.99
R_F	0.671	0.589	0.434
S_{RN}	179.06	157.17	115.81

For it, we can calculate the reliability level $L(n)$ by interpolate, and the stress range calculated from Chapter 4.5 (178 Mpa for Ballast Condition) was used,

$$L(n) = 0.90 + \frac{(0.95-0.90)\times(179.06-178)}{179.06-157.17} = 0.903$$

V. Conclusions

Most of ship structures are failure while the stress range not reaches the yield point. The reason for this situation is fatigue damage which can make structure failure at low stress with load cycles. So that fatigue assessment is necessary for ship structure. There are vary analysis method can be used, however, at most of times, the FE model was been used and the Finite element model is necessary.

For the FE Model, the size of mesh is pivotal for the accuracy of respond result. For the mode analysis, the big size can be taken while the small size should be used in structure analysis.

When take a FEM analysis, it is not different to fix the position of maximum stress with the finite element model. But the work of fine mesh is complicated and troublesome. From the result, we know that the different from two methods is not so big with same corrosive parameter. So that, if we want to approximate fatigue life for some section faster, we no need to check every detail, and no need to take a fine mesh model also. Use this kind of simplified method that the times of working should be short.

17K Oil Tanker 에 대한 피로해석 및 신뢰성 평가

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국 문 요 약

오랜 기간동안 피로로 인한 균열 및 파괴는 선박과 해양구조물에 대한 가장 중요한 문제중의 하나로 인식되고 있다. 최근 10 년동안, 고장력강(HTS)을 이용한 선박의 대형화로 인하여 피로 문제가 더욱 크게 대두되었다. 대부분 연구에서 선박의 피로 강도를 어떻게 합리적으로 평가하는지를 고민하고 있다. 이에 따라,세계 주요 선급에서도 적절한 피로 평가 지침을 발행하였다. 그러나 많은 복잡한 요소들이 피로 성능에 영향을 끼치고 있다. 예를 들어, 파랑 및 화물의 하중, 여러 가지 응력의 합성, 응력 집중, 평균응력과 부식 환경 등이 그러한 것들이다. 그리고 각 선급에 따라 기술지침서의 내용도 다르다. 본 논문에서는 DNV 선급에 따른 피로 평가 절차 및 간략화한 계산방법을 이용하였고 이를 실제 선박에 적용하여 이에 대한 신뢰성을 평가하였다.

References

[1] Det Norske Veritas, Classification Note No. 30.7, Fatigue Assessment of Ship Structures, Hovik, 2003.

[2] Det Norske Veritas, Classification Note No. 30.6, Structural Reliability Analysis of Marine Structures, Hovik, 1991.

[3] Knott, J. F, Fundamentals of fracture mechanics. Wiley, 1973

[4] Ang,A.H.-S., A Comprehensive Basis for Reliability Analysis and Design, Tokyo, 1974

[5] Munse W H, et al., Fatigue characterization of fabricated ship details for design, SSC-318, 1983

[6] Elsayed E.A., Reliability Engineering, Addison Wesley Longman, 1996

[7] ABS&DNV&LR, Common Structural Rules for Double Hull Oil Tankers, 2005

[8] Cramer E.H. Fatigue Assessment of Ship Structures, Marine Structures 8 (359-383), 1995

[9] Melchers,R.E, Structural Reliability Analysis and Prediction, John Wiley& Sons, 2000

[10] Jensen,J.J., Load and Global Response of Ships, Elsevier , 2001

[11] O. Ditlevsen, H.O. Madsen, Structural Reliability Methods, John Wiley, 1996.

[12] Bathe K.J.,Finite Element Procedure. Prentice-Hall,1996.

[13] R. W. Clough, J. Penzien, Dynamics of structures. McGraw-Hill,1993

[14] Bendat,J.S, Random Data Analysis and Measurement Procedures, John Wiley& Sons, 2000

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