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Safety Assessment of Double Skin Hull Structure against Ultimate Bending and Fatigue Strength

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이중선각구조 선박의 최종굽힘강도와 피로강도에 대한 안전성 평가

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Abstract

In this paper presented is the reliability analysis of a double skinned hull stucture against the ultimate bending moment and fatigue strength under longitudinal bending. The ultimate bending strength is obtained through the beam-column approach in which the load-end shortening curves(stress-strain curves) of stiffened plates under uni-axial compression are derived using the concept of plastic hinge collapse. The fatigue damage only is considered as fatigue failure for which the Miner's damage rule is employed.

Assessed are fatigue reliability for the possible joint types found at deck structure. Also included is the reliability analysis of a series system of which elements are ultimate and fatigue failure.

요 약

본 논문에서는 종굽힘하에서 이중선각구조선박의 최종굽힘강도와 피로강도에 대한 신뢰성평가를 다루었다. 최종굽힘강도는 beam-column approach의 개념을 이용하여 구하였고, 보강판의 응력-변형도 곡선들은 소성힌지의 개념을 이용하여 유도하였다. 피로강도는 피로손상에 대한 것만을 고려하였고, 이를 위해 Miner의 손상식을 이용하였다. 갑판에서 가능한 여러 연결부 형태에 대해 그 피뢰신뢰성을 추정하였고 또한 굽힘에 의한 파괴와 피로에 의한 파괴를 동시에 고려하는 일종의 Series System에 대한 신뢰성을 평가를 하였다.

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1. Introduction

Double skinned hull structure is a relatively new concept in the shipbuilding industry. Its merit has been emphasised at many places(for example see References 1, 2 and so on). The concept can be categorised into two, say single skin deck type and doubel skin deck type[3]. The present authors have already presented a work of estimation of ultimate bending strengh using the beam-column approach and the safety assessment based on reliability analysis under longitudinal bending of a such kind of double skin hull ship[3]. This paper deals with evaluation the ultimate longitudinal bending strength, and reliability analysis against bending and fatigue strength of Suezmax-size crude oil tanker of the single skin deck type double hull ship designed by Hyundai. She is outlined in Fig. 1[4]. The ultimate bending strength of the hull girder is evaluated using the beam-column approach as in the previous paper[3]. The sectional properties are listed in Table 1.

Table 1 Sectional properties of the present double skin hull structure model

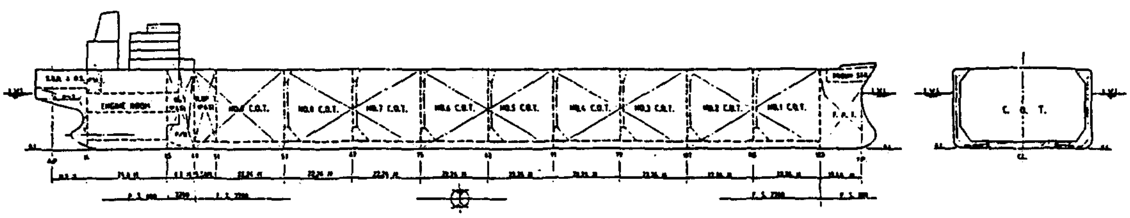
(unit : m, MN-m)

total sectional area	2.76
elastic neutral axis from bottom	10.52
plastic neutral axis from bottom	10.92
moment of inertia about elastic neutral axis (I_{NA})	557.2
plastic bending moment (M_P)	17074.4

In the previous paper by the present authors, the load-end shortening curves of stiffened platse were derived through the non-linear structural analysis. In this paper a very simplified method is propesed to reduce the computational efforts in deriving the load-end shortening curves. The method is based on the plastic hinge mechanism and it will be seen that the method can tremendously cut down the computational time and is also very efficient in estimating the ultimate bending strength.

In this paper reliability analysis against ultimate bending and fatigue strengths are presented and also included is the reliability analysis when considering both strengths.

2. Simplified Method for Deriving the Load-End Shortening Curves of Stiffened Plates.



Principle Particulars

Length O.A.	274.0 m	**Draught, moulded**	16.1 m
Length B.P.	264.0 m	**Draught, scantling**	17.5 m
Breath, moulded	43.9 m	**Dead weight**	153000 dwt
Depth, moulded	24.4 m		

Fig. 1 Profile and midship section of the present double skin hull model (after reference[4])

The basic idea of the present method for deriving the load-end shortening curves of stiffened plates under axial compression is that the relation between stress and strain in the post-ultimate range is assumed to follow the curve derived by using the concept of plasitc hinge mechanism. For illustration let assume that hinge will be generated at mid-span of column as in Fig. 1. From the concept of plastic hinge collapse mechanism the relation between axial load, P and end-shortening, δ can be given as[5] :

$$\frac{P}{P_Y} = \sqrt{[\frac{P_Y}{2M_P}]^2 l\delta+1}-\frac{P_Y^{*}}{2M_P}\sqrt{l\delta} \qquad (1)$$

where P is the applied axial load, and $P_Y=\sigma_Y A$ is the crush load with σ_Y and A are yield stress and cross-sectional area, respectively. $M_P=Z_P\ \sigma_Y$ is plastic bending moment with Z_P plastic section modulus. l is span length of column. Let the normalised stress be σ' and that is easily given as :

$$\sigma'=P/P_Y \qquad (2)$$

The normalised strain ε' is :

$$\varepsilon'=\delta/l \qquad (3)$$

With Eqs.(2) and (3) it is easy to derive the normalised stress-strain relation, say the load-end shortening curve.

$$\sigma'\approx = \sqrt{[\frac{\sqrt{2}\,P_Y l}{2M_P}]^2\varepsilon'+1}-[\frac{\sqrt{2}\,P_Y l}{2M_P}]\sqrt{\varepsilon'} \qquad (4)$$

Let

$$\frac{\sqrt{2}\,P_Y l}{2M_P}=\frac{A l}{2\sqrt{2}\,Z_P}=\eta$$

be η which is hereinn termed as 'sectional coefficient' and then the equation of load-end shortening curve is given as :

$$\sigma' = \sqrt{\eta^2\varepsilon'+1}-\eta\sqrt{\varepsilon'} \qquad (5)$$

Typical example of Eq.(5) for the stiffened plate on deck of the present double skin hull ship(Fig. 1) are illustrated in Fig. 3, of which geometric properties are :

sectional area	: $A = 0.02\ m^2$
plastic sectional modulus	: $Z_P = 0.7\times10^{-3}\ m^3$
span length	: l = 5.56m
yield stress	: $\sigma_Y = 355 MN/m^2$

The whole load-end shortening curve is postulated to consist of 3 parts as shown in Fig. 4 :

(1) Upward hill(Line OA) : when the axial strain is less then ε_u which is the strain corresponding to ultimate stress σ_u, the linear relation between stress and strain is assumed to work.

(2) Plateau(Line AB) : after the ultimate state, the stress is assumed to keep constant as σ_u till the strain has the value corresponding to elastic elastic bucking.

(3) Downward hill(Curve BC) : after point B the

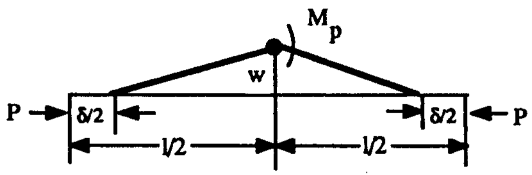


Fig. 2 Concept of plastic hinge mechanism

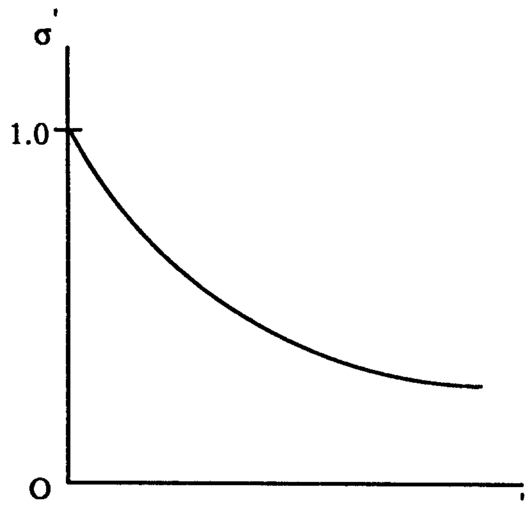


Fig. 3 Stress-strain curve vs plastic hinge mechanism

stress-strain relation is assumed to follow the curve obtained by the plastic hinge mechanism as described above. In this a part of the curve of which stress is less than the ultimate stress will be parallely moved by a distance of BB'.

Doing this we can get the load-end shortening curve designated as "OABC". This approach can allow for much reduction of computational time in getting the load-end shortening curves of stiffened plates compared with the non-linear strucural analysis. Fig. 5 shows the comparison between two method for the example stiffened plates of which geometric properties are given as in the last paragraph. Any method can be used to predict the ultimate stress[6]. It is, however, desirable to employ the method of which mean of modelling parameter(ration of actual value to predicted value) is close to unity. The method proposed by Faulkner[7]. is used in predicting the ultimate stress when the simplified method is applied. We can see the much difference in the post-ultimate range, i. e., when ε' is greater than ε_{cr} in Fig. 4.

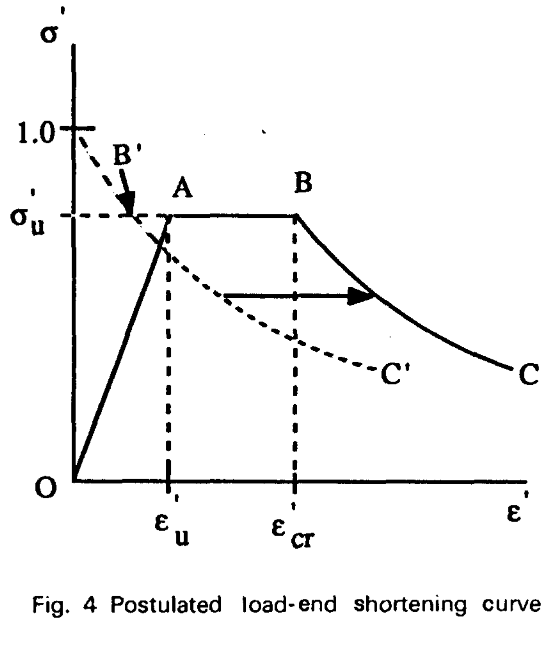


Fig. 4 Postulated load-end shortening curve

3. Beam-Column Approach for Ultimate Bending Moment

Once the load-end shortening curves of stiffened plates are at hand, we can get the ultimate bending moment by applying the beam-column approach.

When a hull girder is subjected to bending about its neutral axis of the section(see Fig. 6) the strain increment, $\Delta\varepsilon$ at any point(y, z) in the section is :

$$\Delta\varepsilon = z\ \Delta\phi y \tag{6}$$

where $\Delta\phi y$ is the increment of curvatures due to bending about the instantaneous neutral axis which changes progressively as a result of local buckling and yielding parts of the cross-section. The stress increment, $\Delta\sigma$, at the point in the section is simply given by :

$$\Delta\sigma = zE^*\ \Delta\phi y \tag{7}$$

where E^* is the effective tangential modulus at(y, z) in the section, which is the slope of stress-strain curve of stiffened panel and can be easily obtained from the curve at a given strain point. Increments

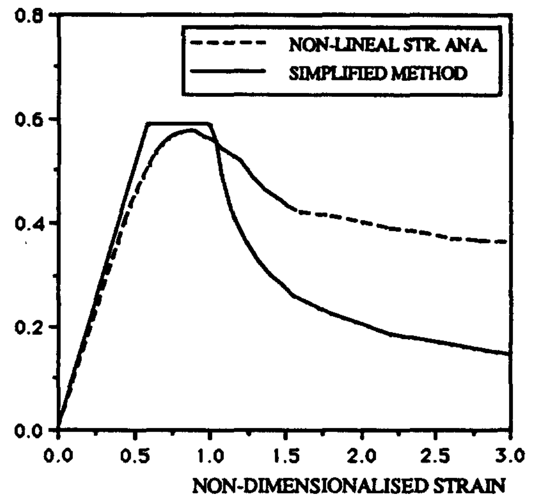


Fig. 5 Example of load-end shortening curve of stiffened plate on deck

of bending momeents about neutral axis, ΔM_Y is obtained from :

$$\Delta M_Y = \int_A z\Delta\sigma \, dA \qquad (8)$$

By incrementing the curvature $\Delta\phi y$ we can obtain the bending moment-curvature relation curve.

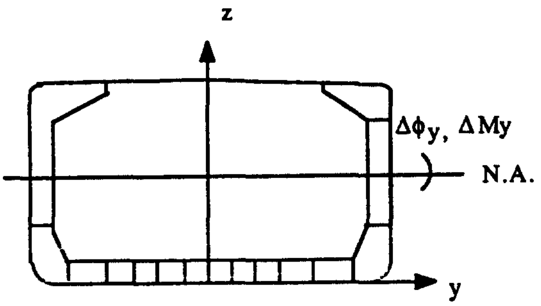


Fig. 6 Hull girder under bending

Fig. 7 shown the non-dimensional relations between bending moment and curvature when using the load-end shortening curves of stiffened plates derived by the non-linear structural analysis(dotted curve) and by applying the simplified method mentioned in Section 2 (solid curve). There is not much difference till ultimate state but we can see that

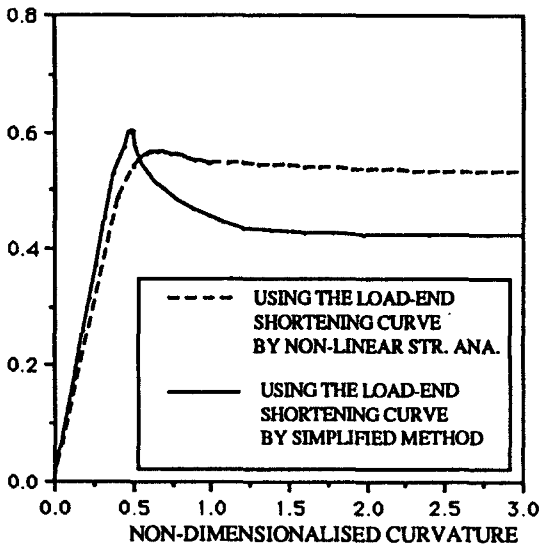


Fig. 7 Bending moment-curvature curve under sagging

there is much difference in the post-ultimate range. This is undoubtedly due to the effect of unloaded part of the curve in Fig. 4. In spite of this using the load-end shortening curves of stiffened plates derived by the simplified method described in the previous section can reasonably predict the ultimate bending moment(at least for the present ship model). The ultimate bending moment read from Fig. 8 should be increased by a factor of

$$(1-\alpha_Y+\alpha_Y\ \alpha_c\ \phi_c)\ \alpha_Y/\phi_c - 1 \qquad (9)$$

to account fot the systematic errors in yield strss and compression strength of stiffened plate as done in Reference 3.

In (9) $\alpha = 1+\xi = 1+$(systematic error) and ϕ_c is the compressive strength parameter of a stiffened plate defined as $\phi_c = \sigma_u/\sigma_Y$ in which σ_u is the ultimate compressive stress and σ_Y is the average yield stress of the combined section of plate and stiffener. The minimum value of ϕ_c of stiffened plates on deck or bottom will be inserted in (9). The systmatic errors, ξ_Y and ξ_c can take account of the objective uncertainaties in predicting yield stress and in using idealised design formulae to evaluate σ_u. They are given as :

(1) systematic error in yield strength :

$$\xi_Y = 0.1 \qquad (10.a)$$

(2) systematic error in compression strength :

$$\xi_c = (1-\phi_c)\ (2+\xi_Y)\ \xi_{co} \text{ with } \xi_{co} = 0.15 \qquad (10.b)$$

The results are summarised in Table 2 in which the ultimate bending moments are normalised by

Table 2 Comparison of ultimate bending moments(normalised by the plastic bending moment : M_u/M_P)

loading condition	method of deriving load-end shortening curve	
	non-linear str. ana.	by simplified method
sagging	0.67	0.70
hogging	1.03	0.98

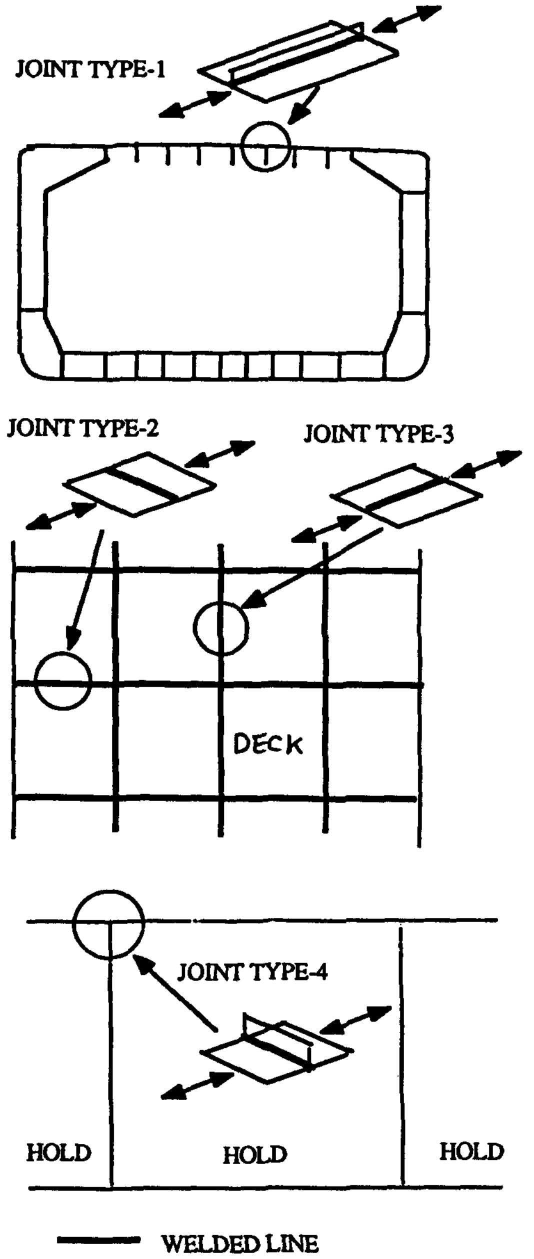


Fig. 8 Joint types found on deck structure

the plastic bending moment of cross-section, M_P in Table 1. It can be seen that the use of load-end shortening curves derived by the simplified method can give close ultimate bending moments using the curves by non-linear structural analysis.

4. Reliability against Ultimate Bending Strength

For the reliability analysis against the ultimate bending strength the safety margin is expressed in terms of bending moments as :

$$M^U = X_{M_R} X_{M_C} M_u - (X_{M_{SW}} M_{SW} + X_{M_W} M_W) \qquad (11)$$

where M_u is the ultimate bending moments M_W and M_W are still water and wave bending moments, respectively. X_{M_R}, X_{M_W} and X_{M_W} are the strength modelling parameter in predicting M_u, and load modelling parameter for still water and wave bending moments, respectively. X_{M_C} is the strength modelling parameter in predicting the ultimate compressive strength. Thses modelling parameters are introduced to account for the uncertainties in predicting the ultimate strength and loading, and they are characterised by its mean bias and COV. The super script "U" is referred to the safety margin for the ultimate bending strength.

Data for reliability study are listed in Table 3. Still water bending moment is obtained through the longitudinal strength analysis of a ship and values of severe conditions in sagging and hogging. DnV rule is used to calcuate wave bending moment. The advanced first-order reliability method (AFORM) is used in the present reliability analysis. Results are summarised in Table 4 and it can be seen that the present ship model has sufficient safety against longitudinal bending.

5. Reliability against Fatigue Strengrh

Fatigue failure is one of the most important failure modes in a structure mainly subjected to variable loads. As well known, there is much more uncertainties associated with fatigue strength than the ultimate strength and then, reliability analysis must be a suitable tool to rationally assess the structural

Table 3 Data for reliability analysis against ultimate bending moment

Variable	mean	COV	dist. type
X_{M_R}	1.0	0.10	log-normal
X_{M_C}	1.0	0.078	log-normal
M_u	0.70 in sagging	–	
	0.98 in hogging	–	
$X_{M_{SW}}$	1.0	0.10	log-normal
X_{SW}	0.17 in sagging	0.10	log-normal
	0.23 in hogging	–	
X_{M_W}	0.1	0.13	normal
M_W	0.18 in sagging	–	
	0.17 in hogging	–	

Table 4 Reliability indices(β^U) and failure probability (P_f^U) against ultimate bending moment

	β^U	P_f^U
sagging	4.66	0.155×10^{-5}
hogging	6.11	0.503×10^{-9}

safety. In this paper failure due to fatigue damage is concerned. The characteristic S-N curve is given by :

$$NS^m = A \tag{12}$$

where m is the fatigue strength exponent, A is the fatigue strength coefficient, S is stress range and N is cycles to failure at the stress range S. Assuming that fatigue strength is defined by Eq.(12) and that Miner's rule works, fatigue damage ratio, D can be written as :

$$D = M_T B^m E[S^m]/A \tag{13}$$

where N_T is the total unmber of cycles applied and E[·] is expected value. B is introduced as a bias factior on stress range and reflect the uncertainty in estimating the stress range, S. The Weibull distribution is commonly employed for the long-term distribution of S. Its cummulative distribution function is given as :

$$F_S(s) = 1 - \exp\,[-(s/\delta)^{\xi}],\ s>0 \tag{14}$$

in which δ and ξ are Weibull scale and shape parameters. Let S_c be the design stress range, say the characteristic stress range, and then the probability that Sc is exceeded by s on an average of once every N_T times is :

$$P(s>S_c) = 1/N_T \tag{15}$$

Let N_T be the total number of cycles in the service life. It can be easily shown that the scale parameter, δ is expressed in terms of Sc, ξ and N_T as :

$$\xi = S_c[ln\ N_T]^{-1/\xi} \tag{16}$$

With Eq.(14) for ξ, Eq.(12) is rewritten as :

$$F_S(s) = 1 - \exp[-(ln\ N_T)\ (s/S_c)^{\xi}],\ \text{for}\ s>0 \tag{17}$$

Using the cumulative distribution function for S given as Eq.(17), the expected value $E[S^m]$ in Eq.(13). is written as :

$$E[S^m] = \int_0 S^m\ f_S(s)\ ds$$
$$= S_c^m (ln\ N_T)^{-m/\xi}\,\Gamma(1+m/\xi) \tag{18}$$

where $f_S(s)$ is the probability density function for S and Γ(·) is the Gamma function. Substituting Eq.(18) for $E[S^m]$ in Eq.(13), we can have the fatigue damage ratio as :

$$D=\frac{N_T}{A}B^m\ S_c^m[(ln\ N_T)^{-m/\xi}\ \Gamma(1+m/\xi)] \quad (19)$$

Let Δ be the fatigue strength (damage index at failure) and then failure occurs when Δ be the fatigue strength (damage index index at failure) and then failurs occurs when Δ<D. Hence the safety margin is written as :

$$M^F=\Delta-D$$

$$\Delta-\frac{N^T}{A}\ B^m\ S_c^m[(ln\ N^T)^{-m/\xi}\ \Gamma(1+m/\xi)] \quad (20)$$

The superscript "F" denotes that the safety margin is associated with the fatigue failure.

Four types of joint are considered found on deck structure as shown in Fig. 8. The design stress range, S_C can be easily calculated using the bending moment data in Tables 1 and 3. Data for fatigue damage analysis are listed in Table 5[8]. As can be seen in Eq.(20) the random variables for farigue reliability analysis are Δ, B and A. Their uncertainties usually varies within a wide range. COV of

Table 5 Data for fatigue damage analysis

joint type	S-N curve*	m	$\log_{10}{}^A$
Joint type-1&-2	CURVE-D	3.0	12.18
Joint type-3	CURVE-C	3.5	13.63
Joint type-4	CURVE-F	3.0	11.80

o Design stress range $S_C=144.5\ N/mm^2$
o Weibull shape parameter $\xi=1.0$
* see for example Refernce 8

the bias factor B can be calculated from that of wave bending moments is sagging and hoggin.

Since COV of wave bending moments in both condition is 13% each as Table 3 the COV of B (this is the COV of design stress range) is 18.4%. For conservative COV of B is given as 20%.

Data for fatigue reliability are shown in Table 6 and results are summarised in Table 7. Referring to that the allowable reliability index of marine structures ranges 2.0 to 3.0 [9], it can be seen that Joint type. 1, 2 and 3 seems to have acceptable reliability to fatigue damage. Joint type 4 has low reliability index and so the junction of deck plate and bulkhead is liable to fail due to fatigue.

Table 6 Data for fatigue damage analysis

varible	mean	COV	dist. type
Δ	1.0	0.3	log-normal
B	1.0	0.2	log-normal
A	see Table 5	0.4	log-normal

Table 7 Results of fatigue reliability analysis (β^F & P_f^F)

	β^F	P_f^F
Joint type-1 & -2	2.08	0.187×10^{-1}
Joint type-3	3.84	0.603×10^{-4}
Joint type-4	0.94	0.174

A case study of fatigue reliability analysis has bees carried out for Joint type 4 with varying the service life, N_T to look into the change of fatigue reliability to operating year. Fig. 9 shows the result. As mentioned before the allowable fatigue reliability index($\beta_0{}^F$) of marine structures lies between

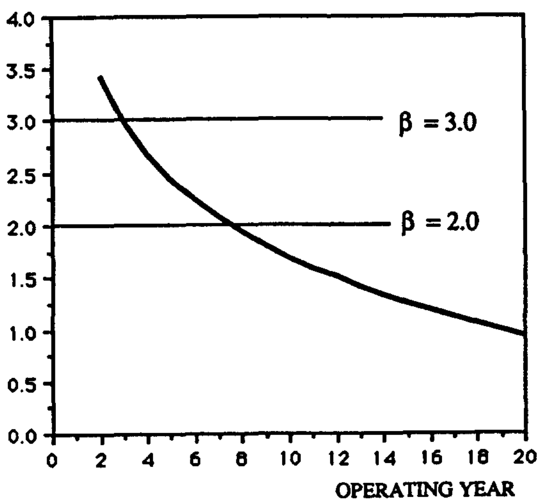


Fig. 9 Change of fatigue reliability on operation year

2.0 and 3.0. Selecting β_0^F usually depends on the importance of a structure part and the structure itself. With refer to these values, in the case that β_0^F is selected as 3.0 and 2.0, the junction of deck plate and bulkhead should likely be checked at least in 3 and 7 years of operation, respectively.

6. Reliability Analysis Considering both Ultimate and Fatigue Srtengths

In this section the safety level by considering both the ultimate and fatigue strength is briefly described. Let think the failure due to lack of ultimate strength and due to fatigue damage as failure elements[10]. These two failure elements make a series system. The probability of failure is evaluated from :

$$P_{f_S}=P[(M^U>0)\cup(M^F>0)] \qquad (21)$$

where M^U and M^F are the safety margin associated with ultimate and fatigue failure, respectively given as Eqs(11) and (20). Hereafter the superscript "U" and "F" denotes that they are associated with the ultimate and fatigue failure. The probabilities of ultimate and fatigue failure are obtained through the separate analysis :

$$P_f^U=P[(M^U<0)] \quad P_f^U=P[(M^F<0)]$$

The corresponding reliability indices are :

$$\beta^U=-\Phi^{-1}(P_f^U)$$
$$\beta^F=-\Phi^{-1}(P_f^F)$$

Where $\Phi(\cdot)$ is the cumulative distribution function of standard normal distribution. The failure probability of series system given as Eq.(21), P_{f_s} is then given by :

$$P_{f_s}=1-[\Phi(\beta^U)\ \Phi(\beta^F)+\int_0^\rho \phi_2(\beta^U,\ \beta^F : z)\ dz] \qquad (22)$$

where ρ is the correlation correlation coefficient between two safety margins, M^U and M^F, and $\phi_2(\cdot)$ is the binormal density function (e.g., see Reference 10) :

$$\phi_2(x,\ y : \rho^2)=\frac{1}{2\pi\sqrt{1-\rho^2}}\exp[-\frac{1}{2(1-\rho^2)}\cdot(x^2+y^2-2\rho xy)] \qquad (23)$$

Then the corresponding reliability index of the series system, β_S is obtainde from :

$$\beta_S=-\Phi^{-1}(P_{f_s})$$

Let the safety margins, M^U and M^F be rewritten in the linear form as :

$$M^U=\sum_{i=1}^{n}\alpha_i^U X_i^U+\beta^U$$
$$M^F=\sum_{i=1}^{n}\alpha_i^F X_i^F+\beta^F \qquad (24)$$

in which n is the number of random variables. X's are variables and α's are corresponding sensitivity factors. The correlation coefficient is defines as :

$$\rho=\{\alpha^U\}^T\{\alpha^F\} \qquad (25)$$

The sensitivity factors corresponding to the common variables between the safety margins M^U and M^F sensitivity factor are included in calcuation of ρ from Eq.(25). Although there may be several common variables between safety margins M^U and M^F, only the wave bending stress is selected in this study.

Eqs.(21) to (25) are applied to 4 joint types shown in Fig. 8 and data are listed in Table 8. It is seen that there is not much close correlation between ultimate and fatigue failure. Since the reliability index against fatigue strength is lower than the ultimate strength, it is natural P_{f_S} and β_S are close to those of fatigue strength.

7. Conclusions

Table 8 Data for evaluation of failure probability considering both ultimate and fatigue failure and results

joint	ultimate failure		fatigue failure		ρ	P_{fS}	β_S
	β^U	α	β^F	α			
joint type-1 & -2	4.66	−0.4,779	2.08	−0.7801	0.3728	0.187×10^{-1}	2.08
joint type-3	4.66	−0.4,779	3.84	−0.8247	0.3941	0.618×10^{-4}	3.84
joint type-4	4.66	−0.4,779	0.94	−0.7801	0.3728	0.174	0.94

This paper has concerned with the reliability analysis of double skinned hull ship against the ultimate strength and fatigue strength under longitudinal bending. Fatigue damage is considered. The ultimate bending moment of midship section has been obtained through the beam-column analysis in which a simplified method is adopted to derive the load-end shortening curves of stiffened plates. It is shown that using the simplified method gives reasonable ultimate bending strength. From the reliability study following are drawn :

(1) the present ship model has sufficient safety against longitudinal bending strength.

(2) some joint types found on deck structure seems to have reasonable safety against fatigue damage, but the joint of deck plate and transverse bulkhead is liable to fail in some years due to fatigue damage, say in 3 or 7 years if the allowable fatigue reliability index is selected as 3.0 or 2.0.

An extension of the present study to design will be presented at some place in the near future.

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