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Comparative fatigue assessment of ship structural details by a spectral direct calculation procedure

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Abstract

To provide plan approval surveyors, and ship designers with a design tool to assess the fatigue performance of ship structural details, Lloyd's Register (LR) has developed a multi-level fatigue design assessment. The ongoing research project has been supported by extensive analytical and numerical studies, experimental fatigue testing of large ship structural models, and in-service fatigue monitoring. In this paper, a brief overview of the components of the multi-level fatigue design assessment system is given, followed by a more detailed description of the procedural aspects of the level 3 spectral fatigue direct calculation.

1. Introduction

During the ship lifetime, two interrelated cumulative damage processes, namely fatigue and corrosion, progressively reduce the capability of the structural details. Over the last decade, the occurrence of fatigue failures on the highly optimised higher tensile steel structure of the second generation single skin VLCC, and the implementation of strict environmental regulations have lead the shipping industry to reconsider its attitude towards the fatigue of structural details. Since fatigue cracks can be possible points of initiation for the failure of the cargo containment barrier, and significant structural failures, resulting in costly repairs, and loss in revenue due to ship downtime, it has become essential to give more detailed considerations to fatigue performance.

To assist both Lloyd's Register's network of worldwide plan approval offices and the ship designers, a multi-level fatigue design procedure has been developed. In this paper, a brief overview of the components of the multi-level fatigue design assessment system is given, followed by a more detailed

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description of the Level 3 spectral fatigue direct calculation procedure.

2. Structural Detail Fatigue Performance Strategy

To *attain* and *maintain* a satisfactory fatigue performance during the ship lifetime, the following issues have been addressed:

- The conceptual design of ship structural details;
- The assessment of the fatigue performance by a direct calculation method;
- The construction tolerances and workmanship;
- The monitoring of the critical structural detail during the ship lifetime.

To perform these tasks, **L**loyd's **R**egister's (LR) has developed the *Ship**Right** **FDA*** **F**atigue **D**esign **A**ssessment procedure (Level 1,2 and 3)[3,4,5]. It is supported by the *Ship**Right** **CM*** **C**onstruction **M**onitoring procedure, and the *Ship**Right** **HCM*** **H**ull **C**ondition **M**onitoring procedure.

3. FDA Level 1 : Structural Detail Design Guide

The purpose of the Structural Detail Design Guide is to promote good detail design at an early stage of the design process. Compiled from the world-wide detail design and the in-service expertise of plan approval, newbuilding and field surveyors, it is based on a vast experience based knowledge database considering design and analysis, construction tolerances and fabrication issues, and in-service performance. In addition, extensive analytical and **F**inite **E**lement (**FE**) analyses have been performed to confirm and optimise the fatigue performance of the recommended structural details. The Detail Design Guide provides a qualitative assessment of the fatigue performance. The procedural steps can be summarised as follows:

- Identify the critical areas with respect to fatigue demand and construction;
- Identify the critical locations within the critical areas;
- Compare the detail design with the Level 1 recommended detail design Standard, and identify the degree of detail design improvement required;

The Detail Design Guide addresses the critical areas of double hull tankers, and bulk carriers. It is updated at regular intervals to reflect trends in service experience, design and construction practice and to incorporate results from the ongoing FDA and FE studies, and the LR fatigue testing programme. The effect of corrosion on structural detail design has also been reviewed in a separate paper where guidelines for a design strategy were proposed, Violette [1].

4. FDA Level 2 & 3 : Spectral Fatigue Direct Calculation

4.1 General Considerations

By virtue of the complex loading patterns generated by the wave environment, the prediction of the long term stress spectrum remains a difficult problem. The long term stress spectrum mathematical model must give due attention to the 10^5-10^8 stress cycles spectrum region where most of the fatigue damage is accumulated i.e. the low to medium seastates. To determine the long term stress spectrum, the maximum lifetime load approach, or the spectral approach

can be used.
The maximum lifetime load approach is well suited for design purposes, as the maximum lifetime loads can be determined using the strength assessment parametric expressions. Upon maximisation of a dominant load component, a load effect combination model and a Weibull shape factor can be selected to completely define the spectrum. However, this procedure is subject to a number of modelling simplifications. To achieve reliable fatigue life estimates, calibration using service experience data is essential. Reduced confidence levels may be introduced when the structural configuration or the loading patterns depart from the service experience base used for the calibration.
To enhance the level of confidence in the determination of the long term stress spectrum, the FDA procedure uses a first principles approach based on the spectral method of analysis. Two levels of analysis based on the same theory have been developed. Level 2 is for design purposes and uses parametric and analytical mathematical models, whilst Level 3 is based on first principles numerical mathematical models. The procedural steps of Level 2 and Level 3 are illustrated in Figure 1.

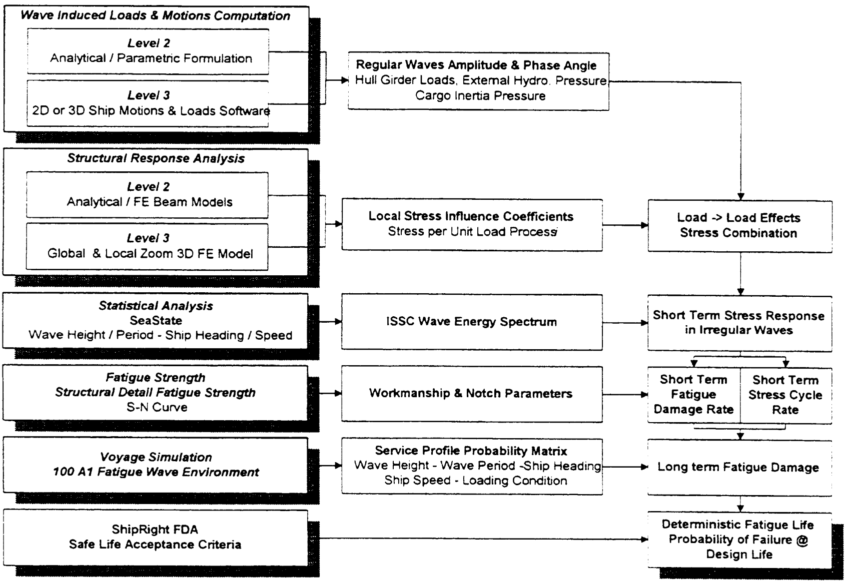

Figure 1 : ShipRight FDA Level 2 & 3 Procedure Flow Chart [3,4,5]

4.2 Computation of Wave Induced Loads & Motions

Using a 2D or 3D ship motions and loads computational method, the regular wave amplitude and phase angle of the following wave induced loads are computed for a range of wave parameters and loading conditions, see Table 1.

- External hydrodynamic wave pressure;
- Hull girder vertical & horizontal wave bending moment;

- Water ballast/cargo inertia pressure.

Parameter		*Range*	*Increment*
Ship to wave angle	χ	0 - 360 °	20°
Wave frequency	ω	0.2 - 1.2 rad/s	0.04 rad/s
Ship speed	V	0 - Vs	25% Vs
Tanker	*Bulk Carrier*		
• Normal ballast	• Normal & heavy ballast	• Coal homogeneous	
• Fully Loaded	• Iron ore alternate	• Grain for Panamax size	

Table 1 : Regular Wave Computation Array

4.3 Structural Response - Finite Element Analysis

In order to obtain the hot spot structural response at the critical locations of the structural detail, a top down approach using a global 3D and local 3D FE model is applied. Typical global and local FE models for a capesize bulk carrier are illustrated in Figure 2. In way of the critical locations, the local model FE mesh size is based on a *txt* element size, where t is the thickness of the plate within which the crack is likely to initiate. For each structural detail considered, the critical locations can be identified easily using the Structural Detail Design Guide.

In order to consider each regular wave conditions, a discrete unit load approach is used to accomplish the analysis with a reasonable number of loadcases. Typically, the total number of loadcases is in the region of 600 to 1100 depending on the size of the global model, the external hydrodynamic pressure patches resolution, and the number of loading conditions. The discrete unit load approach also permits a rigorous treatment of non linear / harmonic wave induced load features such as waterline splash zone and solid cargo inertia pressure. The loadcases required to be computed are summarised as follows :

- Unit hull girder vertical and horizontal bending moment applied at the ends of the global model separately.
- Individual unit pressure patch load applied over a discrete area of the bottom and side shell to represent the external hydrodynamic pressure.
- The cargo and water ballast inertia loads are represented by separate loadcases describing the load pattern due to the action of the individual acceleration vectors in the X, Y and Z directions. Loadcases for the linearised variation of the gravity forces due to angular motion (pitch & roll) are also considered.

A typical matrix of loadcases for a three hold model of a capesize bulk carrier is summarised in Table 2. Figures 3 & 4 illustrate the typical discretisation of the hull envelope into hydrodynamic patches.

In addition to the discrete loadcases, reference loadcases may also be computed to provide a test case to check the convergence of the unit load approach. Since the reference loadcase is based on the application of adequately combined maximum lifetime loads, it may also be used to perform a maximum lifetime load approach fatigue analysis as an additional check.

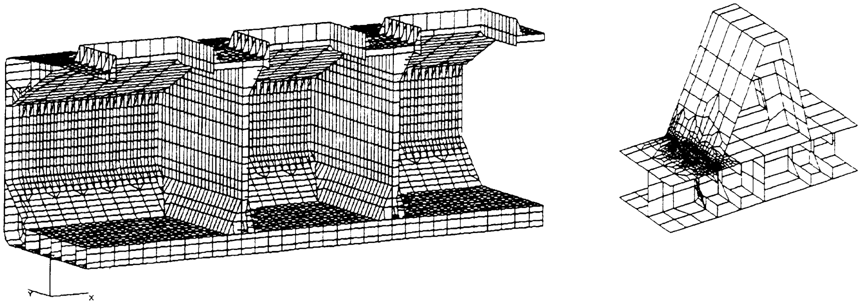

Figure 2 : Bulk Carrier Typical 3D Global & Local 3D FE Model [7]

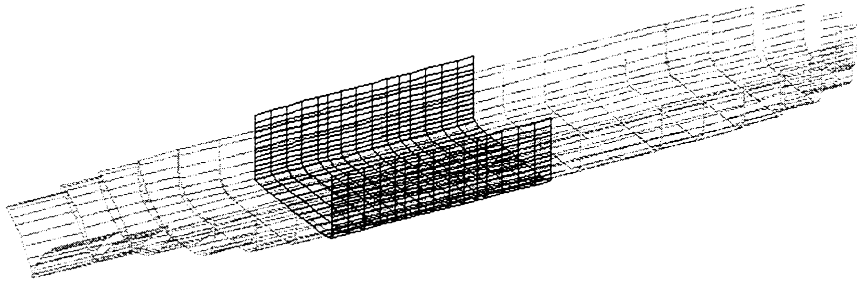

Figure 3 : Hydrodynamic & Finite Element Pressure Patch Meshes [7]

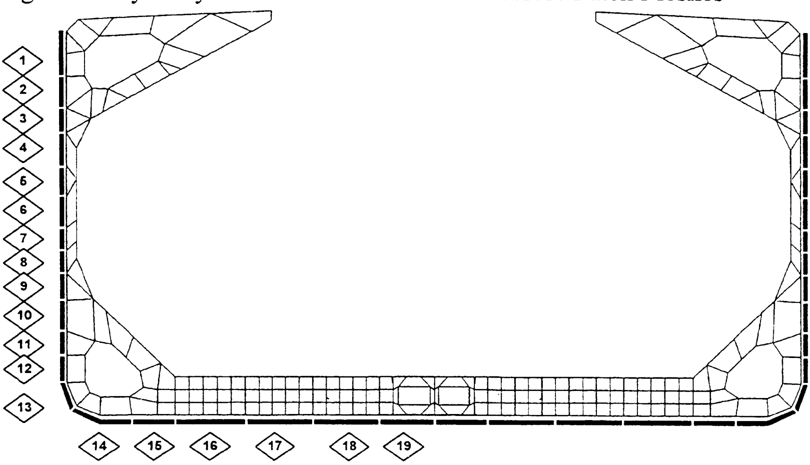

Figure 4 : Typical Transverse Pressure Patch Distribution [7]

Loadcase Description	*No.*
Global Hull Girder Loads	
Vertical Bending Moment Aft End Global Model	1
Vertical Bending Moment Forward End Global Model	1
Horizontal Bending Moment Aft End Global Model	1
Horizontal Bending Moment Forward End Global Model	1
External Hydrodynamic Pressure	
19 transverse x 15 longitudinal patches - symmetric CL Plane	285
19 transverse x 15 longitudinal patches - antisymmetric CL Plane	285
Cargo Inertia Loads	
$\ddot{X}$ Acceleration DB & Hopper *Ballast Tanks* - symmetric & antisymmetric	2
$\ddot{Y}$ Acceleration DB & Hopper *Ballast Tanks* - symmetric & antisymmetric	2
$\ddot{Z}$ Acceleration DB & Hopper *Ballast Tanks* - symmetric & antisymmetric	2
....	
$\ddot{X}$ Acceleration *Cargo Hold No 6 Water Ballast* - symmetric	1
$\ddot{Y}$ Acceleration *Cargo Hold No 6 Water Ballast* - antisymmetric	1
$\ddot{Z}$ Acceleration *Cargo Hold No 6 Water Ballast* - symmetric	1
ϕ Roll Positive & Negative *Cargo Hold No 6 Water Ballast*	2
θ Pitch Positive & Negative *Cargo Hold No 6 Water Ballast*	2
$\ddot{X}$ Acceleration *Cargo Hold No 5 Iron Ore* positive & negative	1
...... *Repeat for each holds and loading conditions*	
....	

Table 2 : Summary of Discrete Loadcases [7]

4.4 Computation of Short Term Fatigue Damage

For a given ship to wave angle, wave frequency, ship speed and loading condition, the total stress can be expressed as follows:

$$\sigma(t) = \sum_{i=1}^{n} C_i \cdot P_i(t) = \sum_{i=1}^{n} H_i(t) \tag{1}$$

For the given stress check point location, ship loading condition, ship speed, ship to wave angle, and seastate $(H_{1/3}, T_Z)$, the short term stress statistics are calculated. The spectral function $S_\sigma(\omega)$ is calculated directly from the wave spectral function $S_\omega(\omega)$ (ISSC spectrum), the transfer function $H_i(\omega)$ of the *ith* load process, and the complex conjugate of the $H_j^*(\omega)$ of the *jth* load process as follows :

$$S_\sigma(\omega) = S_\omega(\omega) \sum_i^n \sum_j^n C_i C_j H_i(\omega) H_j^*(\omega) \qquad m_i = \int_0^\infty \omega^i \cdot S_\sigma(\omega) d\omega \tag{2}$$

For the side shell, where the wave free surface creates a non-linear and non-harmonic behaviour of the pressure, a time domain simulation procedure is performed to calculate the short term stress statistics. The same procedure is applied for the non-harmonic behaviour of solid cargo inertia pressures. Using a closed form solution, the short term fatigue damage rate and associated stress cycle rate can be calculated. For a given stress check point location, ship loading condition, ship speed, ship to wave angle, and seastate $(H_{1/3}, T_Z)$, and assuming that the stress process is narrow banded, the accumulated fatigue damage is expressed as follows

$$D = \sum_{i=1}^{n} \frac{n(S_i)}{N(S_i)} = n_T \int_0^{\infty} \frac{P(S)}{N(S)} dS \qquad p(S) = \frac{S}{4\sigma^2} \cdot \exp\left(-\frac{S^2}{8\sigma^2}\right) \tag{5}$$

$$D == \left[n_T \frac{B^m}{K} \int_0^{\infty} S^m p(S) dS \right] \mu(.) = n_T \frac{B^m}{K} \mu(.)(2\sqrt{2})^m \sigma^m \Gamma\left(\frac{m}{2}+1\right) \tag{6}$$

Since the stress process is not a strictly narrow banded process, a rainflow correction factor $\lambda(m,\varepsilon)$ is introduced, Wirsching [8]. The expected number of stress cycles is obtained as follows :

$$n_T = T\upsilon_0^+ \quad and \ \upsilon_0^+ = \frac{1}{2\pi} \sqrt{\frac{m_2}{m_0}} \tag{7}$$

The deterministic fatigue damage accumulated in a given seastate $(H_{1/3}, T_Z)$ can be obtained from the following expressions:

$$D = \frac{TB^m \Omega}{K} \qquad \Omega = \lambda(\mathrm{m},\varepsilon)\mu(.)\upsilon_0^+ (2\sqrt{2})^m \sigma^m \Gamma\left(\frac{m}{2}+1\right) \tag{8}$$

For each seastate $(H_{1/3}, T_Z)$, the short-term fatigue damage, and stress cycle rate are computed in order to determine the total long term fatigue damage.

4.5 Voyage Simulation – 100 A1 Fatigue Wave Environment

Using a concept similar to the 100 A1 North Atlantic longitudinal strength standard, the 100 A1 Fatigue Wave Environment standard has been formulated. It is obtained by a voyage simulation procedure for a combination of trading routes. The trading routes applied are a function of the ship type and size, and they have been determined from statistical analysis of worldwide trading patterns. The Global Wave Statistics data, BMT [9], is used to determine a service profile matrix giving the probabilities of occurrence of the seastates $(H_{1/3}, T_Z)$, loading conditions, ship to wave angles and ship speeds.

4.6 Computation of Long Term Fatigue Damage

The total lifetime accumulated fatigue damage *Dt* for a service period T_S is computed as follows :

$$Dt = \frac{T_S B^m \Omega_t}{K} \qquad \Omega_t = \sum_{i,j,k,l} p_i p_j p_k p_l \Omega_{i,j,k,l} \tag{9}$$

where $\Omega_{i,j,k,l}$ is the stress level parameter for a given seastate i,j,k,l

4.7 Fatigue Acceptance Criteria

- Deterministic fatigue life with a 97.5% probability of survival S-N curve, and a fatigue damage factor of 1.0 for 20 years;
- Probability of failure and safety index for a given number of service years.

5. Critical Fracture Plane Stress Criteria for Fatigue Analysis

In welded details, the presence of welding residual stresses, initial material imperfections, and the mainly unidirectional stress field in way of the stress concentration areas promote the initiation of fatigue cracks in preferred directions, such as directions perpendicular to adjacent planes between welding and base materials. Since the normal stress on a given fracture plane is a major contributor to the crack initiation and propagation phase, and the shear stresses may be considered to influence mainly the direction of crack propagation, a fatigue failure criterion based on the normal stress on a preselected critical fracture plane has been formulated. When 2D finite elements are used, the crack plane is assumed perpendicular to the free surface of the plate. For a given stress check point location, a principal critical fracture plane is assigned as shown in Figure 5. When, the local stress field exhibit a bi-axial stress behaviour or the finite element does not have one of its local axis perpendicular to the principal crack plane, the maximum fatigue damage may occur on a plane at an angle from the principal critical fracture plane. Therefore, a range of critical crack plane is investigated by sweeping the range [-40, +40] degrees about the normal to the principal critical fracture plane in increment of 10 degrees. At a given stress check point finite element, the direct stress components S_{11}, S_{22} and the shear stress S_{12} are defined with respect to the local element coordinate system. For a given critical fracture plane, the resultant stress normal to the critical crack plane is defined at an angle θ about the local finite element x axis, and can be obtained from the direction cosines as follows:

$$S_\theta = S_{11}\cos^2\theta + S_{22}\sin^2\theta + 2S_{12}\cos\theta\sin\theta \qquad (10)$$

The LRSN reference S-N Curve is defined as the mean S-N Curve with a stress range of 75.62 N/mm^2 at 10^7 stress cycles. The reciprocal of the anti log of the standard deviation is taken as 0.60. The LRSN Curve is a two slopes S-N curve modified with an inverse slope varying from 3 to 5 at 10^7 stress cycle. The LRSN curve is applicable to fillet welds, and should be used in conjunction with finite element analysis using shell element of a mesh size equivalent to the plate thickness *t x t* in the locations where cracks are likely to initiate.

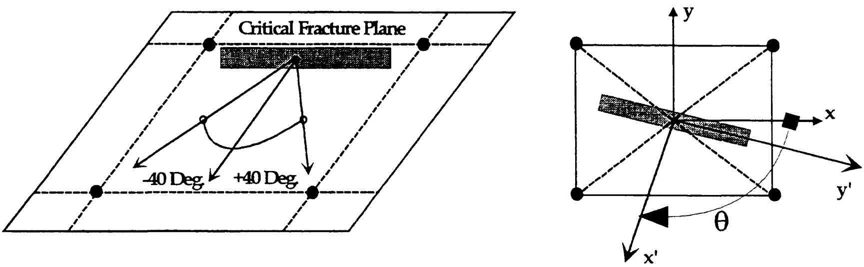

Figure 5 : Critical Fracture Planes & Normal Stress Vector

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6. Comparative FDA Level 3 Applications

The fatigue performance of a VLCC double hull tanker radiused hopper knuckle connection has been investigated [6] on a comparative basis with a similar bulk carrier structural detail which had suffered fatigue damage. Since the trading routes and the loading patterns of both ship types are different, the use of a first principle approach is essential. The predicted fatigue life of the bulk carrier structural detail has been found to be 5.64 years in comparison with the actual service experience life of 7.10 years. A stress bias factor has been determined and was subsequently introduced in the calculation of the fatigue life for the VLCC double hull tanker structural detail. Using the maximum lifetime load approach procedure with individual spectrums for the loaded and ballast loading condition, the fatigue life of the bulk carrier structural detail was found to be 15.69 years assuming an F2 SN curve.
The fatigue performance of a capesize bulk carrier ballast hold inner bottom to lower bulkhead stool connection has been investigated [7]. The fatigue life was found to be 3.53 years in comparison with the service experience life of 1.10 years. In this analysis, the proportion of heavy ballast loading condition has been shown to have a significant effect on the fatigue life, and the default 100 A1 Fatigue Wave Environment value was used. Figure 6 illustrate the variation of the fatigue damage along the critical fracture planes of a finite element on the ballast hold inner bottom plating adjacent to the lower bulkhead sloping plate of a capesize bulk carrier.
Taking into consideration the scatter shown fatigue tests, and the uncertainties due to the ship operation, it is considered that the results correlate well with service experience.

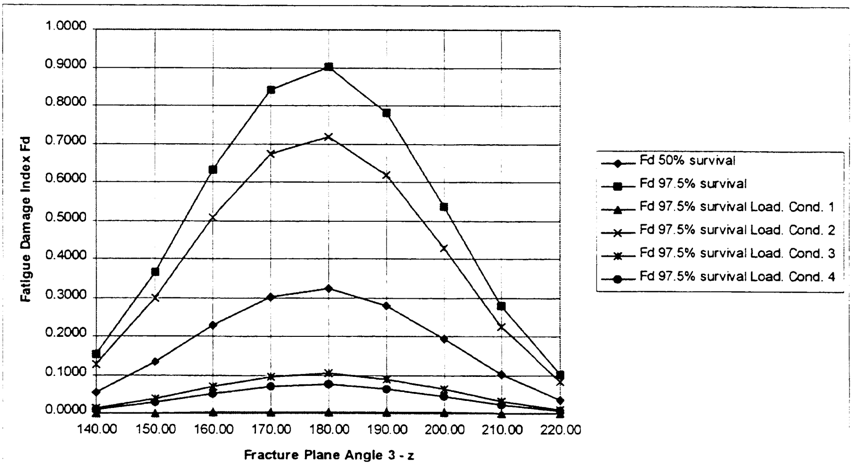

Figure 6 : Fatigue Damage versus Critical Fracture Plane Angle

7. Summary & Conclusions

The spectral FDA Level 3 procedure, and the critical fracture plane stress criteria have been reviewed in this paper. The procedure has shown to yield fatigue life estimates close to the fatigue life of structural details for which service experience data is available. Due to the cumulative nature of the fatigue process, it is considered that a spectral model is a better suited that a maximum lifetime model to yield consistent and stable results. To facilitate the FE loadcase generation, global-local boundary conditions transfer, and the FDA computations, an integrated software procedure is now being developed.

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9. Nomenclature

C_i	Structural influence coefficient of the ith load process $P_i(t)$
$P_i(t)$	Load process i
$H_i(t)$	Stress process i complex form
n	Total number of load processes
p(S)	Stress range probability function
σ	Standard deviation of the stress process
S_0	S-N curve stress range at 10^7 stress cycles
$N(S_i)$	Number of allowable stress cycles at stress range S_i
$n(S_i)$	Number of stress cycles with stress range S_i for the given seastate
n_T	Expected number of stress cycles in the given seastate
B	Modelling bias for the stress prediction model
$\mu(m,m_1,S_0,B)$	Correction factor for multi-linear S-N curve
m	Slope of selected S-N curve
K	Intercept of selected S-N curve
ε	Spectral bandwidth
T	Seastate duration
υ_0^+	Mean zero crossing frequency
$p_i = p(i \vert jkl)$	probability ith loading condition, jth ship to wave angle, kth ship speed, lth seastate $(H_{1/3},T_z)$