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High Cycle Fatigue Analysis of a Lower Hopper Knuckle Connection of a Large Bulk Carrier under Dynamic Loading

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***Abstract*—**The fatigue of ship structural details is of major concern in the maritime industry as it can generate fracture issues that may compromise structural integrity. In the present study, a fatigue analysis of the lower hopper knuckle connection of a bulk carrier was conducted using the Finite Element Method by means of ABAQUS/CAE software. The fatigue life was calculated using Miner's Rule and the long-term distribution of stress range by the use of the two-parameter Weibull distribution. The cumulative damage ratio was estimated using the fatigue damage resulting from the stress range occurring at each load condition. For this purpose, a cargo hold model was first generated, which extends over the length of two holds (the mid-hold and half of each of the adjacent holds) and transversely over the full breadth of the hull girder. Following that, a submodel of the area of interest was extracted in order to calculate the hot spot stress of the connection and to estimate the fatigue life of the structural detail. Two hot spot locations were identified; one at the top layer of the inner bottom plate and one at the top layer of the hopper plate. The IACS Common Structural Rules (CSR) require that specific dynamic load cases for each loading condition are assessed. Following this, the dynamic load case that causes the highest stress range at each loading condition should be used in the fatigue analysis for the calculation of the cumulative fatigue damage ratio. Each load case has a different effect on ship hull response. Of main concern, when assessing the fatigue strength of the lower hopper knuckle connection, was the determination of the maximum, i.e. the critical value of the stress range, which acts in a direction normal to the weld toe line. This acts in the transverse direction, that is, perpendicularly to the ship's centerline axis. The load cases were explored both theoretically and numerically in order to establish the one that causes the highest damage to the location examined. The most severe one was identified to be the load case induced by beam sea condition where the encountered wave comes from the starboard. At the level of the cargo hold model, the model was assumed to be simply supported at its ends. A coarse mesh was generated in order to represent the overall stiffness of the structure. The elements employed were quadrilateral shell elements, each having four integration points. A linear elastic analysis was performed because linear elastic material behavior can be presumed, since only localized yielding is allowed by most design codes. At the submodel level, the displacements of the analysis of the cargo hold model to the outer region nodes of the submodel acted as boundary conditions and applied loading for the submodel. In order to calculate the hot spot stress at the hot spot locations, a very fine mesh zone was generated and used. The fatigue life of the detail was found to be 16.4 years which is lower than the design fatigue life of the structure (25 years), making this location vulnerable to fatigue fracture issues. Moreover, the loading conditions that induce the most damage to the location were found to be the various ballasting conditions.

***Keywords*—**Lower hopper knuckle, high cycle fatigue, finite element method, dynamic load cases.

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I. Introduction

SHIP structures encounter a number of different waves during service life that will cause several types of dynamic loads to the ship hull. One such type could be slam induced loads in seaway that will cause inertial pressure of the cargo/ballast to the ship hull. Such dynamic loads will result in different kind of cyclic loads to the global and local structure of the ship giving rise to fatigue phenomena that compromise the structure's integrity.

Fatigue failure occurs when a specimen will break in two parts although other situations might be defined such as the appearance of a crack having a specific size. Therefore, fatigue life would be the number of cycles to failure of the specimen. If failure occurs in less than 10^4 cycles, this is called low cycle fatigue, whilst for higher endurance it is called high cycle fatigue [1]. At high cycle fatigue analysis, linear elastic material behavior is assumed. Therefore, fatigue damage is estimated based on the Palmgren-Miner rule where fatigue damage for a given stress level can be considered to accumulate linearly with the number of stress cycles [2]. In this case, results of fatigue tests are presented as S-N curves. These are plots on a logarithmic scale of varying stress range (S) versus the number of cycles to failure (N). The S-N data used for fatigue damage calculations are developed under load resulting from principal stress acting normal to the weld toe line. However, in real structures, the principal stress direction may vary and may not be normal to the weld. The larger the angle of the principal stress is to the weld, the more conservative the fatigue damage calculations are, based on the assumption that the stress acts normal to the weld line. The IACS Rules assume that stress acting at 45 degrees to weld toe normal line is equivalent to a stress acting normal to weld line [3].

Traditionally, fatigue analysis was conducted by the use of nominal stress along with a catalogue of classified details where each type of detail was assigned to a particular S-N curve. This method was proposed by the International Institute of Welding (IIW) until 2009 where it was updated by including the structural hot spot and effective notch stress approaches [4]. The nominal stress approach does not consider the varying dimensions of the structural detail, which is a downside of the method [5].

When studying fatigue failure by crack initiation to the weld toe, the structural hot spot stress approach advances from the previous method as the stress established takes into account the dimensions of the detail. The stress calculated at the expected fatigue crack location is called structural hot spot stress. This stress includes the stress components of membrane and shell bending stress but not the non-linear stress peak caused by the notch at the weld toe. The notch effect is included in the hot spot S-N curves which are set experimentally. Fig. 1 portrays the stress components that the method calculates; the membrane stress (σ_{mem}), the shell bending stress (σ_{ben}) and the non-linear stress peak that is introduced by the S-N curves (σ_{nlp}) [5], [6]. In order for this method to be accurate, the fine meshing rules of the hot spot areas should be strictly followed. Moreover, to implement the structural hot spot stress method, the designer has to verify that the weld will not fail from the root or inner defects. For the case of weld root failure, the effective notch stress approach can be implemented, which also has limitations in its scope of application. Apart from these limitations, the approach is well established in tubular structures, shipbuilding and other areas [1], [5], [7].

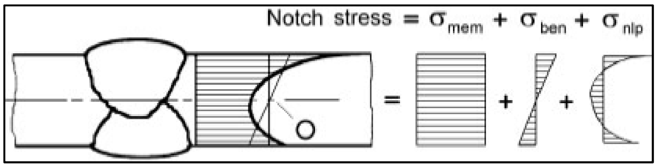

Fig. 1 Stress components at the hot spot

In the present study, dynamic loading was induced in the fatigue analysis by the implementation of the Equivalent Design Wave (EDW) concept. The basis of the concept concerns the creation of load cases at each loading condition, that are examined in the analysis. Each load case consists of combinations of a dominant load component (a global motion of the hull) and other significant load components (secondary loads) to accentuate the action of the dominant load component. One example of a load case scenario could be a vertical acceleration or vertical bending moment in the hull girder to act as dominant load component and secondary loads such as slamming and whipping whose outcome would be to accentuate the load effect of the vertical acceleration to the hull. The load combination factors that are used for the applied loads are calculated from transfer functions and phase angles between the dominant and secondary load responses for each instantaneous load case. The applicability of the equivalent design wave concept for the estimation of the maximum stresses has been demonstrated for any probability level at tankers and bulk carriers [8]-[10].

According to the CSR rules since the case study ship has length over 200 m, four loading conditions should be studied at a fatigue strength analysis. For each of the loading conditions (Homogenous-Alternate-Heavy Ballast and Normal Ballast) all four dynamic load cases should be examined. These are a load case (H) occurring due to head sea, a load case due to follow sea (F), a load case due to beam sea where there is maximum roll effect (R), and a load case due to beam sea where there is maximum external pressure (P) [11].

In the present study, a fatigue analysis is conducted to the lower hopper knuckle connection of a bulk carrier using the structural hot spot stress approach and the Palmgren-Miner's Rule. Literature and IACS Rules present this structural detail to have low fatigue life, therefore it is essential to conduct a fatigue strength analysis, at the design stage, in order to establish if its fatigue life is in compliance with the design fatigue life of the ship structure which is at 25 years [11], [12]. The fatigue damage that occurs in each of the loading conditions -Homogenous, Alternate, Normal Ballast and Heavy Ballast- is accumulated linearly in order to calculate the fatigue life of the detail. The direction of stress considered during the calculations is the one which acts perpendicularly to the ship's centerline axis.

For the simulations, the commercial finite element software ABAQUS/CAE is used. In the following sections, the material properties and the set-up of the finite element models are introduced in order to calculate the fatigue life of the case study detail.

II. Materials and Methods

A. Material Properties

Hull material is steel of normal yield stress at 235 MPa (for grade A) and high tensile steel of yield stress at 315 MPa (for grades of AH32 and DH32) and 355 MPa (for grades of AH36, DH36, and EH36). In the location of the lower hopper knuckle connection the steel used is high tensile steel grade AH36 with yield stress at 355 MPa.

The origin of coordinates is assumed to be in the stern-most plane. X-axis is directed to the ship bow, Y-axis is to the port side and Z-axis vertical up.

B. Finite Element Models

1. Cargo Hold Model

The principal dimensions and key characteristics of the case study-ship are shown in Table I.

TABLE I
Key Dimensions of the Case Study Vessel

Type of ship	Bulk Carrier BC-A
Hull type	Single skin
Length overall, LOA	292.00 m
Length between perpendiculars, LBP	283.00 m
Breadth molded, B	45.00 m
Depth molded, D	24.80 m
Scantling draught, T	18.20 m
Design draught, T	16.50 m
Maximum service speed	14.9 knots

Fig. 2 illustrates the mid-ship drawing (taken from the ship's technical drawings) and the detail of the lower hopper knuckle connection in 3D sketch, designed in ABAQUS/CAE where a detailed fatigue analysis will be conducted.

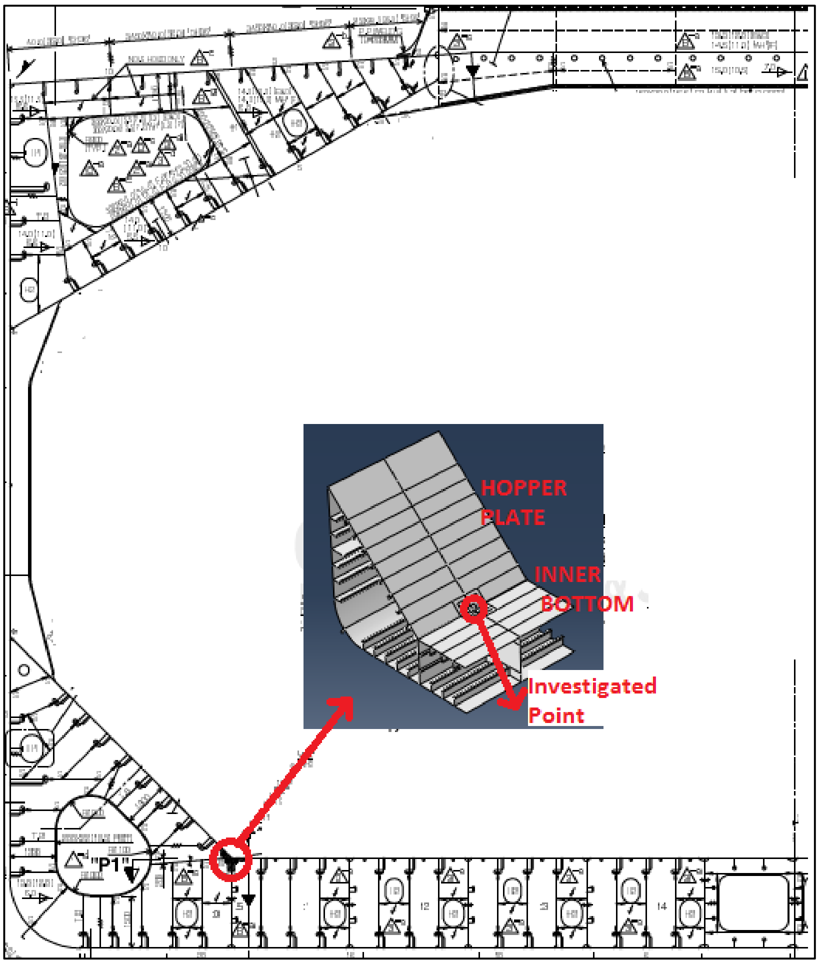

Fig. 2 Mid-ship section of the ship at the starboard side and location of the sub-model

A global cargo hold model comprised by two holds is normally employed for fatigue assessment. This is important as at this analysis the displacements and rotations of the external nodes will be provided that will act as boundary conditions for the local model. The overall aim is to obtain the hot spot stress range from specific areas in the structure that are prone to fatigue cracks; however, in the present study, the lower hopper knuckle connection is targeted. The hull structure of the case study vessel is modeled transversely on its full breadth and longitudinally by two hold lengths (½ + 1 + ½), between Fr 145 and Fr 200 with frame spacing at 930 mm. The extent of the model is set in that way so that symmetrical boundary conditions could be applied. Reduced plate scantlings (gross scantlings minus corrosion addition) were used for the model. The cargo hold model is comprised by all main longitudinal and transverse structural elements; shell, deck, double bottom, girders, transverse web frames, hatch coaming, stringers, all plates, and longitudinal stiffeners. Large openings were also modeled.

The mentioned members are modeled by 4-noded, quadrilateral shell elements. In the model, between the girders, there are eight elements whilst between each frame two elements. The mesh of the plates should at least represent the actual plate panel so that stresses can be read out directly. The geometry of the cargo hold model and the mesh generated are depicted in Fig. 3.

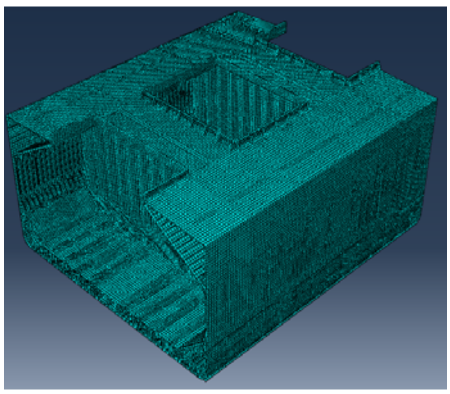

Fig. 3 Cargo hold model-view of the mesh

2. Submodel

The minimum extent of the submodel should be such that the boundaries of the submodel correspond to the locations of adjacent supporting members. Here, the boundaries of the submodel correspond to the location of a girder (16150 mm off CL) in the transverse direction and of two web hopper plates in the longitudinal direction.

In order to calculate the hot spot stress at the hot spot location, a very fine mesh zone was generated. According to [11], the very fine mesh zone should be at least within a quarter of frame spacing in all directions from the hot spot position. The element size in very fine mesh areas is to be approximately equal to the representative net thickness in the assessed areas, and the aspect ratio of elements is to be close to 1 [11].

Here, the very fine mesh zone will extend for 232.5 mm in all directions from the connection area since the frame spacing is at 930 mm. The element size in the very fine mesh zone will be 16 mm x16 mm. After the zone, the mesh size will gradually change from very fine to fine through transitional areas. These are depicted in Fig. 4.

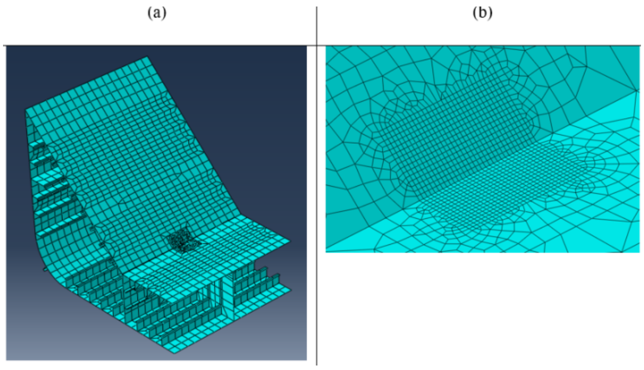

Fig. 4 (a) Submodel view of the mesh, (b) Submodel view of the very fine mesh zone and the transitional zone

C. Loads and Boundary Conditions

1. Loads and Boundary Conditions on Cargo Hold Model

As discussed at the introduction, dynamic loading is induced by the implementation of the equivalent design wave concept. Among all four dynamic load cases that should be studied for each loading condition, the predominant load case for each loading condition would be the one where maximum stress range accrues and this specific load case contributes to the fatigue life of the detail. However, since it is sought to maximize the transverse-to the ship's coordinate system-stress range, higher fatigue damage to the case study detail will occur for the load case with the highest stress range at the transverse direction. Table II presents the effect of its load case on the ship hull. Both R and P load cases (Beam Sea) demonstrate that they will affect the transverse stress range but it is unclear on which degree. For that reason, in all four loading conditions, these load cases will be studied in order to establish which load case (R or P) will be used for each condition. For both of these cases, inertial pressure due to dry bulk cargo for homogenous and alternate loading conditions, inertial pressure due to liquid for normal and heavy ballast conditions, and dynamic external pressure for all loading conditions are applied [11].

TABLE II
IDENTIFICATION OF THE LOAD CASES THAT INDUCE TRANSVERSE STRESS AMPLITUDE [11]

Load Case	H1	H2	F1	F2	R1	R2	P1	P2
EDW	"H"		"F"		"R"		"P"	
Heading	Head		Follow		Beam		Beam	
Effect	Max. Bending Moment		Max. Bending Moment		Max. Roll		Max. Ext. Pressure	
	Sagging	Hogging	Sagging	Hogging	(+)	(-)	(+)	(-)

Inertial pressure due to dry bulk cargo for each load case for homogenous and alternate condition at load cases R, P is described by:

$$p_{CW} = \rho_C[0.25a_y(y - y_G) + K_C a_Z(h_C + h_{DB} - z)] \quad (1)$$

where ρ_c is the density of the dry bulk cargo, a_y is the transverse acceleration at the center of gravity of the hold, K_c is a coefficient taken equal to $K_c=\cos^2 a+(1-\sin\psi)\sin^2 a$, where a is the angle between the panel considered and the horizontal plane and ψ is the assumed angle of repose of the dry bulk cargo, az is the vertical acceleration at the center of gravity of the hold, h_c is the vertical distance from the inner bottom to the upper surface of the bulk cargo, h_{db} is the height of the double bottom at the centerline, z is the coordinates of the load point with respect to the reference co-ordinates.

Inertial pressure due to liquid for Normal and Heavy Ballast condition is described by:

$$\text{Load case R P:} p_{BW} = \rho_L[a_Z(z_B - z) + a_y(y - y_B)] \quad (2)$$

At (1) and (2), ρ_L is the density of the internal liquid, z_B is the z-co-ordinate of the tank top for completely filled spaces and the top of the hatch coaming for ballast holds, y_B is the y-co-ordinate, in m, of the tank top located at the most lee side when the weather side is downward, or of the most weather side when the weather side is upward.

Dynamic sea pressure is applied for each loading condition at the bottom and side shell below the waterline.

For Load case P it is described as:

$$P_P = 4.5 * fp * fnl * C\sqrt{\frac{L+\lambda-125}{L}}\left(\frac{2|z|}{Tlc} + \frac{3|2y|}{B}\right), \quad (3)$$

Table III explains the distribution of external dynamic pressure for load cases P1-P2.

TABLE III
DISTRIBUTION OF DYNAMIC EXTERNAL PRESSURE IN LOAD CASE P

Load case	Weather side	Lee side
P1	Pp1=Pp	Pp1=Pp/3
P2	Pp2=-Pp	Pp2=-Pp/3

For Load Case R:

$$P_{R1} = fnl\left(10ysin\theta + 0.88fp\,C\sqrt{\frac{L+\lambda-125}{L}}\left(\frac{|2y|}{B}+1\right)\right),$$
$$P_{R2} = -P_{R1} \tag{4}$$

At (3) and (4), factor f_p is the coefficient corresponding to the probability level taken equal to 0.5 for probability level 10^{-4}, f_{nl} is the coefficient considering non-linear effect, taken equal to 1.0 for the probability level of 10^{-4}, C is the wave coefficient, L is the rule length, T_{LC} is the draught at the considered cross section for the considered loading condition, λ is the wave length corresponding to the load case, and B is the molded breadth at the waterline for the considered cross-section.

Regarding the boundary conditions, both ends of the model are simply supported. The nodes of the longitudinal members of both end sections are rigidly linked to independent points that lay in the neutral axis of the section. Table IV shows the support condition of the independent points in the aft and fore end of the model.

TABLE IV
SUPPORT CONDITION OF THE INDEPENDENT POINT

Location of the independent point	Translational			Rotational		
	Dx	Dy	Dz	Rx	Ry	Rz
Independent point on aft end of the model	-	Fix	Fix	Fix	-	-
Independent point on fore end of the model	Fix	Fix	Fix	Fix	-	-

2. Loads and Boundary Conditions on Submodel

The sub-model runs as a separate analysis. The only link between the sub-model and the cargo hold model is the transfer of the variables to the relevant boundary nodes (the “driven nodes”) of the sub-model. More specifically, the displacements of the analysis of the cargo hold model to the region of the sub-model acted as boundary conditions and loading to the sub-model. Only these displacements were taken into consideration to the analysis.

III. RESULTS

A. Load Case Applied

By Figs. 5 and 6, it is identified that the most damaging load case is the “R” in loading conditions homogenous, alternate and normal ballast. Load case “P” was proven to be more damaging in heavy ballast condition. Hence, beam sea contributes to the fatigue damage in the transverse direction.

B. Fatigue Analysis of the Lower Hopper Knuckle Connection

1. Establishment of Equivalent Notch Stress Range

Further down, the process that was implemented to calculate the equivalent notch stress range as introduced by [11] is explained. As mentioned in §II.C.1, the predominant load case for each loading condition is the one where maximum stress range accrues. Fig. 7 depicts the two hot spot locations that are studied in the fatigue analysis. Therefore,

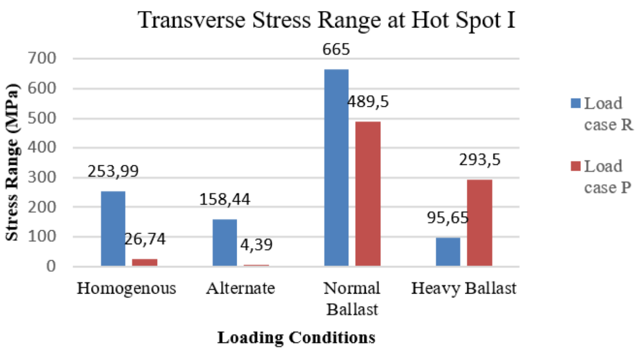

Fig. 5 Transverse stress range at Hot Spot I for load cases "R" and "P"

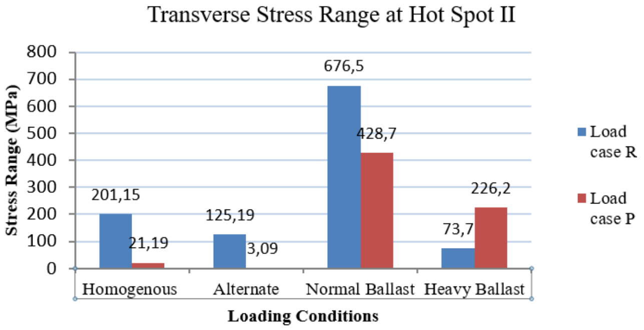

Fig. 6 Transverse stress range at Hot Spot II for load cases "R" and "P"

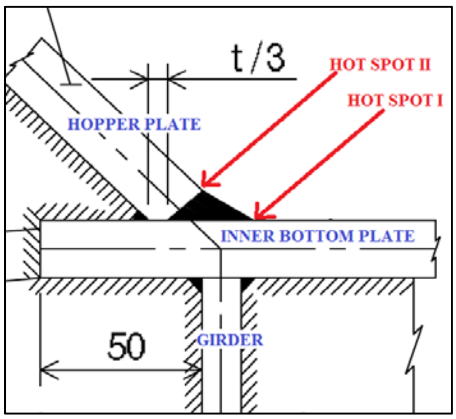

Fig. 7 The two hot spots that will be explored by the FE analysis

$$\Delta\sigma_w = \max(\Delta\sigma_{w,k}) \tag{5}$$

where k denotes the predominant load case. The equivalent hot spot stress range would then be

$$\Delta\sigma_{equiv} = f_{mean}\,\Delta\sigma_w \tag{6}$$

where f_{mean} is the correction factor accounting for mean stress

The equivalent notch stress range would then be:

$$\Delta\sigma_{eq} = K_f\,\Delta\sigma_{equiv} \tag{7}$$

where K_f is a fatigue notch factor.

The equivalent notch stress range is corrected by inserting factors regarding the corrosive environment (f_{coat}), the material ($f_{material}$), and the plate thickness effect (f_{thick}). Therefore, the equivalent notch stress range is:

$$\Delta\sigma_E = f_{coat}\, f_{material}\, f_{thick}\, \Delta\sigma_{eq} \tag{8}$$

This specific stress range is then used to the cumulative probability density function of the long-term distribution of combined notch stress range. It is taken as a two-parameter Weibull distribution [11].

$$F(x) = 1 - exp\left[-\left(\frac{x}{\Delta\sigma E}\right)^{\xi} (lnN_R)\right], \tag{9}$$

where factor ξ is the Weibull shape parameter taken equal to 1.0, and N_R is the number of cycles. Following that, the elementary fatigue damage for each loading condition can be calculated by:

$$D = \frac{aN_L}{K}\frac{\Delta\sigma_E{}^4}{(lnN_R)^{4/\xi}}\left[\Gamma\left(\frac{4}{\xi}+1,\nu\right)+\nu^{-3/\xi}\,\gamma\left(\frac{7}{\xi}+1,\nu\right)\right], \tag{10}$$

where: K= S-N curve parameter, equal to $1.014*10^{15}$, a= Coefficient depending on the loading condition, N_L= Total number of cycles for design ship's life, equal to $N_L = \frac{0.85T_L}{4logL}$, T_L=Design life in seconds representing 25 years of life, Γ= Type 2 incomplete gamma function, γ= Type 1 incomplete gamma function, and

$$\nu = \left(\frac{100.3}{\Delta\sigma E}\right)^{\xi} lnN_R,$$

The elementary fatigue damage ratio is calculated for all loading conditions and then it is linearly accumulated to identify the cumulative fatigue damage ratio for the detail.

2. Establishment of Fatigue Life at Hot Spots I and II

The elementary fatigue damage that occurs for each loading condition is depicted in Table V. At the bottom row of Table V, the fatigue life for the detail is established at 13.53 years for Hot Spot I and at 14 years for Hot Spot II. The fatigue life that accrues is very low compared with the design fatigue life for the ship structure is set for 25 years.

TABLE V
ELEMENTARY FATIGUE DAMAGE FOR EACH LOADING CONDITION

	Hot Spot I	Hot Spot II
$D_{HOMOGENOUS}$	0.30	0.14
$D_{ALTERNATE}$	0.07	0.03
$D_{NORMAL\ BALLAST}$	2.62	2.70
$D_{HEAVY\ BALLAST}$	0.55	0.25
D_{FINAL}	3.54	3.12
$T_{(YEARS)}$	13.53	14.00

IV. CONCLUSION

In the present study, a fatigue strength analysis was conducted at the lower hopper knuckle connection of a bulk carrier. Dynamic loading was applied by implementing the equivalent design wave concept. First, the cargo hold model was generated and then the submodel, which included the area of interest, was extracted. The fatigue life in both hot spots was calculated. All steps were described in detail which will be useful for fatigue analyses of other parts of the ship.

At the application of dynamic loads in the cargo hold analysis, it was identified that beam sea condition induces fatigue damage in the case study detail. Also, it was noticed that normal ballast condition poses the higher stress range in the connection. This is important as this condition induces a loading scheme that promotes high stress concentration in the detail. Therefore, this condition can be used as the worst case scenario in optimization purposes when minimization of stress range is sought.

At the end of the fatigue analysis, the fatigue life of Hot Spot I and Hot Spot II was calculated. It was found to be at 13.53 years and 14 years, respectively. This is significantly lower than the design fatigue life of the ship which is at 25 years. This result agrees with literature and the rules where the lower hopper knuckle connection is presented as prone to fatigue fracture.

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REFERENCES

[1] Pook L., *Metal Fatigue; What it is, why it matters*, Springer, 2007, ch.1.
[2] DNV, "Fatigue Assessment of Ship Structures", *Det Norske Veritas*, 2014.
[3] IACS, Common Structural Rules, "Technical Background for CSR BC&OT Report List", *International Association of Classification Societies,* 2015.
[4] Fricke, "Guideline for the Fatigue Assessment by Notch Stress Analysis for Welded Structures", *Annual Assembly of International Institute of Welding*, Vol. IIW, 2010.
[5] Niemi E, Fricke W, Maddox SJ, *Designer's guide to the structural hot spot stress approach*, 1st ed., Woodhead Publishing, 2006, ch.1.
[6] Hobbacher A F, "New Developments at the Recent Update of the IIW Recommendations for Fatigue of Welded Joints and Components", *Steel Construction*, Vol.3, 2010, No 4, pp. 231-242.
[7] Fricke W, Kahl A, "Fatigue assessment of weld root failure of hollow section joints by structural and notch stress approaches", Conference: Tubular Structures XI, 2006, London.
[8] Kim B, Yung-Sup S, Wang X, "Combination of Long Term Dynamic Loads for Fatigue Assessment", *International Society of Offshore and Polar Engineers*, 2006.
[9] ABS, "Dynamic Load Approach and Direct Analysis for High Speed Craft", *American Bureau of Shipping*, 2003.
[10] ABS, "Dynamic Load Approach for Loading Production, Storage and Offloading (FPSO) Installations", *American Bureau of Shipping*, 2014.
[11] IACS Common Structural Rules, "Common Structural Rules for Bulk Carrier with length 90m and above", *Det Norske Veritas AS*, 2012.
[12] Liao P, Quemener Y, Lee C, Chen K, "Load Uncertainty Effects on the Fatigue Life Evaluation by the Common Structural Rules", *OMAE2015*, Proceedings of the ASME 2015 34TH International Conference on Ocean, Offshore and Arctic Engineering, May 31-June 5, 2015, St. John's Newfoundland, Canada.