A Multihull Boat’s Fatigue Analysis at Early Design Phase

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Abstract: Fatigue analysis and life cycle prediction of a naval craft or a commercial ship are one of the most critical phases of structural integrity calculations. Many structural failures have occurred due to insufficient structural design, production methods, and inappropriate operational environmental conditions. As a result of the deterioration of the structural integrity, the occurrence of both vital and economic losses becomes inevitable. This paper emphasises the importance of fatigue and life cycle analyses at the early design phase since the fatigue phenomena are generally considered in the late design stages. In this study, fatigue calculations and life predictions were carried out by the Palmgren–Miner method. The wave rosette, also called long-term wave directionality, was used as the wave spectrum approach. Based on wave loads and accelerations, an FE analysis was carried out using the Maestro Marine Altair Partner Alliance (APA) software. The undamaged life prediction of the vessel was calculated based on different sea states and vessel speeds. A specific scenario, which was prepared based on the International Code of Safety for High-Speed Craft, was also analysed to compare the fatigue life of the vessel with regards to safe operating conditions for passengers and crew. In this study, the effect of sea states, environmental conditions, and materials used on the fatigue behaviour of the vessel are discussed and analysed. As a result of this study, the shortest and the longest undamaged life spans based on the loading scenarios are put forth clearly. Analyses and calculations prove that loading scenario 5 is the most effective scenario in terms of the undamaged life span of the boat. On the contrary, loading scenario 1 gives the shortest service life. The main reason behind this phenomenon is the relation between the fatigue endurance limit and load cycles.

Keywords: fatigue; Palmgren–Miner’s rule; AA5059; Aegean Sea; wave rosette spectrum approach

1. Introduction

Fatigue life prediction is a highly prominent phenomenon in today’s naval vessel design as in other structural engineering disciplines. The fatigue phenomenon is critical since several catastrophic fatigue-induced fractures have been observed even on ships less than five years old [1]. Therefore, this phenomenon should be carefully considered even in the early stage of the structural design phase. At the beginning of the structural design phase, scantling of the structural members is carried out based on global and local loads such as design bending moments and shear forces. The maximum allowable stress criteria have already been defined in class rules for different materials. A structural engineer uses the given limit values to decide on the size of the structural members under conventional loads. In contrast to the globally allowable stress approach, fatigue endurance limits for structural materials such as aluminium alloys are significantly reduced by incremental load cycles. Hence it is crucial to assess the fatigue endurance limits by S–N curves and operational conditions at the beginning of the project.

In this study, the AA5059 aluminium alloy was used as the structural material to emphasise this phenomenon. AA5059 has a significantly higher fatigue endurance limit compared to the AA5083 aluminium alloy, widely used in the shipbuilding industry. The AA5059 alloy is also known as the Alustar and has also been certified by the classification
societies such as LR, BV, and DNV-GL [2]. AA5059 also represents much better ballistic and corrosion resistance compared to AA5083. Only a limited fatigue analysis has been carried out in the literature for the AA5059 aluminium alloy. Most of the research is based on the weld joints of AA5083 and its H111 and H321 tempers. Grujicic et al. studied the statistical analysis of the high-cycle behaviour of friction stir welding for the AA5083 H321 temper [3]. Torzewski et al. carried out experimental low-cycle fatigue behaviour of friction stir welded butt joints and compared the experimental results with the standard data regarding the mechanical properties of the AA5083 H111 aluminium alloy [4]. Jesus J.S. studied the fatigue strength improvement of GMAW T welds in AA5083 by friction stir processing [5]. Unlike the references cited here, this study was conducted to assess and predict the undamaged life cycle of a semi-swath-type naval craft built with the AA5059 alloy at the initial design phase.

Previously, Peng et al. carried out a global finite element analysis using coarse mesh to investigate a trimaran-type vessel’s hot spot zones. This study also presented the spectral and simplified fatigue analysis methodology of a trimaran for a specific region [6]. Goncalves et al. investigated the difference between the numerical analysis and experimental studies for the fast monohull patrol boat built with the AA5083 H111 aluminium alloy for a specific bulkhead [7]. Marin et al. investigated the structural fatigue behaviour of a high-speed boat under a vibratory cyclical propeller load [8]. Several studies regarding fatigue life prediction are available in the literature, including these references. None of these studies were performed using the AA5059 aluminium alloy as a construction material.

Fatigue life calculations and FEA (Finite Element Analysis) were carried out by the Maestro Marine FEA software due to its quick adaptation and capability of accurate quasi-static analysis for different sea conditions. MAESTRO is a finite element tool providing powerful structural design and analysis capabilities for naval architects [9]. Maestro Marine accomplishes this objective through a single Windows-based graphical user interface that completely encompasses the structural modelling (preprocessing), the ship-based loading, the finite element analysis, the limit state evaluation, and the postprocessing [9]. As the shell element, MAESTRO utilises the CQuadR element with a four-noded flat shell element with each node having 6 degrees of freedom. This shell element is also used in the NASTRAN The MacNeal-Schwendler Corporation (MSC) software developed for NASA, but in MAESTRO the task of calculating the orthotropic properties is automated [9]. In the following chapters, vertical accelerations for different sea states were calculated. The calculated vertical acceleration values were applied to the Maestro Marine FE model, and the related stress values were calculated accordingly. The loads were considered based on the vertical accelerations at the centre of gravity of the vessel. The model was also balanced in the seawater environment considering tank loading and weight distribution.

The wave rosette (also called long-term wave directionality), which describes the probability of each heading angle (the primary wave direction) at a site, was used as the spectrum approach. The directional convention should be followed using the rosette (e.g., for NOAA (National Oceanic and Atmospheric Administration) wave data, index 1 represents the wave coming from true north, and as the index increases, the wave direction changes clockwise). Directionality has significant effects on the structural response. In this approach, it is recommended that a realistic wave rosette be used in the fatigue analysis. If the wave rosette is unavailable, it is reasonable to assume an equal probability of all heading angles in open ocean conditions [10]. In light of this information, wave profiles and sea states, which cause maximum stress values and higher loading cycles, were chosen to simulate the worst-case scenarios.

The wave characteristics of the Aegean Sea were used to obtain the number of cycles that the vessel will encounter during its lifetime. Wavelengths, wave speeds, and load cycles were calculated in Section 3. This study was specifically carried out to predict the undamaged service life of the semi-swath-type vessel with an overall length of 55 m and 3 m draught at the preliminary design phase for the environmental conditions in the
Aegean Sea. This study can also be adapted easily if the sea state characteristics of different environmental conditions are known.

As a general approach, the fatigue analysis and life prediction are always considered later on in design stages. In contrast to this approach, this paper suggests and underlines that fatigue life prediction should be considered at the early design phase along with the global structural analysis. As a result of an early-stage fatigue analysis, it would be possible to determine critical locations where the fatigue cracks are expected to happen. In later stages, a detailed analysis must be carried out by implementing fine mesh to the hot spot areas based on the finalised scantling and detailed design of the structural members under the defined design loads. This study aims to predict the undamaged life span of a multi-hull naval craft at an early stage of design by using the Palmgren–Miner method.

Even fatigue life estimation is carried out based on the class requirements, a generally calculated assessment is inaccurate due to uncertainties such as loads, experimental S–N curves, operational conditions, and production processes [11]. The production process is another dominant factor for fatigue cracks, which is not addressed in this paper [12]. This study also aims to provide information on how long the service life of a naval craft would be without fatigue damage in different sea conditions and speed ranges, including operational requirements in terms of the International Code of Safety for High-Speed Craft (2000) [13].

This study differs from other previous studies on this subject regarding the material properties used during the analysis. Additionally, this study also shows originality in terms of comparing the fatigue life calculations according to the most challenging environmental conditions that a boat can encounter at the maximum vessel speed and the maximum environmental conditions and speeds that humans can withstand.

2. Palmgren–Miner Method and S–N Graphics

As mentioned above, Palmgren–Miner’s method was used for the fatigue analysis in this study. Palmgren–Miner’s method was chosen as the fatigue lifetime assessment method since the vessel is exposed to stress with a variable number of cycles and amplitudes during its lifetime (Figure 1).

![Figure 1. Typical block loading sequence [14].](image)

Palmgren–Miner’s method also provides exact and time-saving solutions. To obtain a cumulative damage assessment for steel specimens using this method, all the cycles above the fatigue endurance limit should be summed as shown in Figure 2 [15].

The standard approach is to use the Palmgren–Miner linear damage summation model to evaluate the cumulative damage (D) over the period selected, which is typically taken as the design lifetime of the ship [16]. This model assumes that the cumulative fatigue damage (D) is the sum of the fatigue damage inflicted by each stress cycle, independent of the sequence in which the stress cycles occur.

$$D = \sum_{i=1}^{N_i} \frac{1}{N_i}$$

(1)
If fatigue damage exceeds unity, it can be evaluated as the generation of a crack through the thickness of a material. In practice, such a crack may or may not be of a sufficient size to cause degradation of function (e.g., water tightness) or of an unacceptable risk of fracture, i.e., a crack that needs to be repaired. If the damage sum over the N stress cycles is less than unity, the ratio of the projected fatigue life to the time for the N cycles is the D–N total. Fatigue damage assessments can also be used to rank the structural members in the ship according to projected life and develop inspection strategies that focus attention on the highest risk areas. S–N curves are used to derive the damage at each stress cycle or stress range level. These curves are derived using the experimental data from fatigue tests conducted in air or a simulated seawater environment, usually on relatively small specimens simulating simple welded details of various configurations [17].

In a fatigue analysis, other parameters may also be required to account for fabrication processes, including the effects of mean stress levels, residual stresses, and postweld treatments such as grinding and peening. If the fatigue calculations do not consider these parameters, the naval engineer ensures that these are incorporated into the fabrication processes [17].

If we rewrite and interpret the equation that is given in (1), we obtain the equations that are presented in Equation (2) and (3).

\[
\sum \frac{n_i}{N_i} = 1.0 \tag{2}
\]

\[
\frac{n_1}{N} + \frac{n_2}{N} + \frac{n_3}{N} + \ldots \leq 1.0 \tag{3}
\]

where \(n_1, n_2, n_3, \ldots\) are the numbers of cycles in each block, which are illustrated in Figure 1. Since the obtained maximum stress values in the loading scenarios, which are going to be defined in the following sections, are over or almost near the fatigue endurance limit (failure at stress level) of the aluminium alloy, the \(n_i\) number of cycles at the fatigue endurance stress level are included in the calculations [18].

For constant amplitude loadings, the allowable stress range is never less than the fatigue endurance limit [19]. For example, the yield stresses of the most widely used aluminium alloys in shipbuilding, AA5083 and AA5059, are 125 MPa and 155 MPa [20], respectively. From Figures 2 and 3, the endurance limit of the AA5059 alloy is approximately 115 MPa, and the endurance limit of the AA5083 alloy is less than 100 MPa. It can obviously be seen that there significant differences were observed between yield stresses and fatigue endurance limits of the aforementioned aluminium alloys. Based on Figures 2 and 3, the aluminium alloy AA5059 was chosen as the vessel building material. Even if the fatigue endurance limit was shown approximately as 117 ksi (117 MPa) at the \(10^8\) cycles in Figure 2, since Figure 3 is restricted with \(10^7\) cycles, which does not represent the fatigue endurance limit at \(10^8\) cycles, the fatigue endurance limit was assumed as 110 MPa for \(10^8\) cycles.

As shown in Figure 2, steel has an endurance limit plateau that provides reliability to this material leading to failures at the higher cycles. Provided a certain stress value is not exceeded, the fatigue endurance limit is independent of the number of cycles for the steel material. However, aluminium alloys are dependent on the number of cycles. As the number of cycles increases, the fatigue endurance limits of aluminium and composite material significantly decrease, endangering the vessel’s structural integrity and human life.
3. Loading Scenarios for Fatigue Calculations

At the first stage of the fatigue life cycle calculations, the definition of the loading scenarios emerges. Parametric work and scenarios should be created carefully considering how much time is spent in severe sea conditions. In this study, five different scenarios were created to simulate easy, moderately complex, and severe operating conditions. As mentioned before, a load case was created to assess the fatigue life behaviour of the vessel based on human safety. One of the most critical parameters in fatigue calculation is the wave encounter frequency of the vessel, which is dominated and calculated by the modal wave periods as given in Table 1.

As a general approach for naval ships, it is assumed that the vessel mentioned above is operated 2000 h annually. These operation hours change according to vessel capacities of the navy, but for this study, 2000 annual operation hours were taken as an assumption. As a result of this assumption, the calculated hours for each loading scenario and sea state are presented in Table 2.
Table 1. Characteristic wave heights and modal wave periods used for the study [23].

| Sea State | Characteristic Wave Height (m) |                  |                  |                  |
|-----------|-------------------------------|------------------|------------------|------------------|
|           |                               | Black Sea        | Mediterranean Sea | Aegean Sea       |
| 0–1       | 0.05                          | 3.53             | 4.42             | 3.63             |
| 2         | 0.3                           | 4.14             | 5                | 4.01             |
| 3         | 0.88                          | 5.41             | 6.25             | 4.86             |
| 4         | 1.88                          | 7.28             | 8.15             | 6.25             |
| 5         | 3.25                          | 9.1              | 10.16            | 7.96             |
| 6         | 5                             | 10.19            | 11.74            | 9.81             |

Table 2. Possible loading scenarios and operation conditions.

| Loading Scenarios | LS1-Operation Condition (LS1) | LS2-Operation Condition (LS2) | LS3-Operation Condition (LS3) |
|-------------------|-------------------------------|-------------------------------|-------------------------------|
|                   | %                             | Equivalent Hours %           | Equivalent Hours %           |
| Sea State 1       | 25                            | 500                           | 10                            | 200                           | 5                             | 100                           |
| Sea State 2       | 25                            | 500                           | 10                            | 200                           | 5                             | 100                           |
| Sea State 3       | 30                            | 600                           | 20                            | 400                           | 15                            | 300                           |
| Sea State 4       | 10                            | 200                           | 25                            | 500                           | 20                            | 400                           |
| Sea State 5       | 5                             | 100                           | 20                            | 400                           | 25                            | 500                           |
| Sea State 6       | 5                             | 100                           | 15                            | 300                           | 30                            | 600                           |

As mentioned above, the wave spectrum rosette was used to obtain the worst wave attacking angles for the critical sea states. In each sea state, the wave attacking angles were defined as 0, 45, 60, and 90 degrees, and the highest stress values were reflected in the design to simulate the worst loading case.

3.1. LS1-Operation Condition Calculations

In the LS1 operation condition, the vessel was assumed to be operated primarily on sea state two and sea state three, as can be followed in Table 2. Under these circumstances, it is expected that the life span should be longer than the other (higher sea state) loading conditions since the maximum stresses are calculated lower than the other operating conditions. However, in reality, due to the fact that the aluminium alloys’ fatigue endurance limits reduces by the cycles significantly the expected undamaged lifetime of the vessel will not be as long as predicted as calculated in the following sections.

The following equation is used to calculate the wavelength, assuming the water is deeper than half of the wavelength so that the sea can be evaluated as deep water [24]. With this assumption, the following wavelength calculation equation (dispersion relationship) was used to calculate the cycles that the vessel can encounter [25].

\[
\lambda = \frac{gT^2}{2\pi} = 1.56T^2
\]  

(4)

Considering the modal wave periods in the Aegean Sea for different sea states, which are given in Table 1, the following wavelengths which were presented in Table 3 have been calculated.

Wave encounter frequencies(rad/s) should be obtained to calculate the cycles for each sea state separately. Assuming that the vessel is in high sea states with a velocity of 20 kts, then the waves seem to meet the vessel faster than the actual frequency of the waves. The
observed and obtained new frequency is called an encounter frequency, \( \omega \). If the waves are incident on the ship at some angle, \( \mu \), then the component of the ship’s speed in the direction of wave propagation is \( V_a = V \cdot \cos \mu \) \[^{[25]}\]. The wave crest moves at phase speed, \( C_p = \omega / k \), and the relative speed between the ship and the wave is calculated with:

\[
V_R = V_a + C_p = V \cdot \cos \mu + \omega / k \tag{5}
\]

In this equation:

\[
k = \frac{2\pi}{\lambda} = \text{wave number (rad/m)} \tag{6}
\]

\[
\omega = \frac{2\pi}{T} = \text{wave frequency (rad/s)} \tag{7}
\]

\[
\vartheta = \lambda f = \lambda T \tag{8}
\]

Table 3. Sea state–wave length relations for the Aegean Sea.

| Sea State | 0–1 | 2  | 3  | 4  | 5  | 6  |
|-----------|-----|----|----|----|----|----|
| \( \lambda \) (m) | 20.6 | 25.1 | 36.9 | 61 | 98.9 | 150.3 |

Based on the equations given above, the relative speed between the vessel and waves was calculated for each wave attacking angle as given in Table 4.

Table 4. Relative speed calculation considering wave attacking angle.

| Wave Attacking Angle Vs. Relative Speed |
|-----------------|-----------------|-----------------|-----------------|-----------------|-----------------|-----------------|
| \( V \) (m/s) | \( T \) (s) | \( \omega \) (rad/s) | \( \omega \) (Hz) | \( k \) (rad/m) | \( V_\theta \) (0 Degrees) | \( V_\theta \) (30 Degrees) | \( V_\theta \) (45 Degrees) | \( V_\theta \) (60 Degrees) | \( V_\theta \) (90 Degrees) |
| 10.28 | 3.63 | 1.73 | 0.275 | 0.31 | 15.95 | 14.57 | 12.94 | 10.81 | 5.67 |
| 10.28 | 4.01 | 1.57 | 0.249 | 0.25 | 16.54 | 15.17 | 13.53 | 11.40 | 6.26 |
| 10.28 | 4.86 | 1.29 | 0.205 | 0.17 | 17.87 | 16.49 | 14.86 | 12.73 | 7.59 |
| 10.28 | 6.25 | 1.00 | 0.159 | 0.10 | 20.04 | 18.67 | 17.03 | 14.90 | 9.76 |
| 10.28 | 7.96 | 0.79 | 0.126 | 0.06 | 22.71 | 21.34 | 19.70 | 17.57 | 12.43 |
| 10.28 | 9.81 | 0.64 | 0.102 | 0.04 | 25.60 | 24.23 | 22.59 | 20.46 | 15.32 |

As presented in Table 5, the angle of encounter at which the maximum cycle occurred results from the waves that come from 0 degrees. For this reason, in order to simulate the worst cycle loading condition, the number of cycles generated by the waves from zero degrees were taken into account in the calculations.

Table 5. Encounter cycle vs. sea state summary.

| Calculated Cycles On Each Sea State Considering Wave Encountering Angle |
|-----------------|-----------------|-----------------|-----------------|-----------------|-----------------|-----------------|
| Sea States | % | Hours | Seconds | Cycles For 0° Wave Enc. Angle | Cycles For 30° Wave Enc. Angle | Cycles For 45° Wave Enc. Angle | Cycles For 60° Wave Enc. Angle | Cycles For 90° Wave Enc. Angle |
| Sea State 1 | 25% | 500 | 1,800,000 | 1,394,833 | 1,274,395 | 1,131,532 | 945,350 | 495,868 |
| Sea State 2 | 25% | 500 | 1,800,000 | 1,185,539 | 1,086,845 | 969,776 | 817,208 | 448,878 |
| Sea State 3 | 30% | 600 | 2,160,000 | 1,046,262 | 965,634 | 869,994 | 745,353 | 444,444 |
| Sea State 4 | 10% | 200 | 720,000 | 236,499 | 220,248 | 200,971 | 175,849 | 115,200 |
| Sea State 5 | 5% | 100 | 360,000 | 82,617 | 77,607 | 71,665 | 63,921 | 45,226 |
| Sea State 6 | 5% | 100 | 360,000 | 61,315 | 58,017 | 54,105 | 49,006 | 36,697 |
| \( \Sigma \) | 4,007,064 | 3,682,745 | 3,298,043 | 2,796,689 | 1,586,313 |
The Palmgren–Miner rule predicts that failure occurs when damage equals “1“. In this equation, “N” represents the total number of cycles given in Figure 3, the AA5059 Wöhler diagramme. The number of cycles given in Table 5 at the “0”-degree encounter angle should be divided into the total cycle suggested by the class rules.

\[
\frac{1394833}{10^8} + \frac{1185539}{10^8} + \frac{1046262}{10^8} + \frac{236499}{10^8} + \frac{82617}{10^8} + \frac{61315}{10^8} = 0.08014 (9)
\]

\[
\text{Life time Without Any Fatigue Crack} = \frac{1}{0.08014} = 12.5\text{ years} \quad (10)
\]

As mentioned above, to stay on the conservative side, the fatigue endurance limit of the aluminium alloy was taken as 110 MPa for 10^8 cycles. In the FEA calculations, which were conducted by the Maestro Marine FEA, additional acceleration values should be calculated for different sea states and reflected in the FEA model to obtain the results from the quasi-static stress analysis. In this study, the 2012 Germanischer Lloyd rules were used to obtain the vertical acceleration at the vessel’s centre of gravity [20].

\[
a_{CG} = C_{HSC} \cdot C_{RW} \cdot \frac{V}{\sqrt{L}} \quad (11)
\]

Parameters in the Equation (11) are as given below in Tables 6 and 7.

Table 6. Service coefficients.

| Service Range Coefficients |
|---------------------------|
| \( C_{RW} \)             |
| 1.00 for unlimited service range |
| 0.90 for restricted service area RSA (200) |
| 0.75 for restricted service area RSA (50) |
| 0.66 for restricted service area RSA (20) |
| 0.60 for restricted service area RSA (SW) |

Table 7. Vessel operation type coefficients [20].

| Type of Service | Passenger, Ferry, or Cargo | Supply | Pilot | Rescue |
|----------------|---------------------------|--------|-------|--------|
| \( C_{HSC} \)   | 0.24                      | 0.36   | 0.5   | 0.6    |

During vertical acceleration calculations at the vessel’s centre of gravity, the service range coefficient was taken as “1” to simulate the unlimited service range condition.

The operation type coefficient of the vessel was assumed as 0.6 in case the vessel may be used for rescue operations.

The \( H_s \) values used in the FEA calculations are given in Table 8. These significant wave height values affect the vertical acceleration of the vessel significantly and, therefore, the maximum stress values obtained.

Table 8. Sea state–wave height relations summary [20].

| Significant Wave Height | Smooth Sea Service | Moderate Environment Service | Restricted Open Sea | Open Sea |
|-------------------------|--------------------|------------------------------|---------------------|----------|
| \( H_s \) Ranges        | \( H_s \leq 0.5 \) | \( 2.5 \leq H_s \leq 0.5 \) | \( 4 \leq H_s \geq 2.5 \) | \( H_s \geq 4 \) |

Table 9 presents the relation between significant wave heights and sea state codes. If we compare Tables 8 and 9, the open sea service condition refers to sea state six or over. The calculated vertical acceleration values for each sea state condition are summarised in
Table 10. These $a_{\text{CG}}$ values are directly utilised in the model as an input as can be seen in Figure 4.

Table 9. Sea state specifications [26].

| Beaufort Number | Wind Speed Range (knots) | NATO Sea State Code | Wave Heights ($H_s$) Range (m) |
|-----------------|--------------------------|---------------------|-------------------------------|
| 1               | 0–3                      | 1                   | 0–0.1                         |
| 2               | 4–6                      | 1                   | 0.1–0.5                       |
| 3               | 7–10                     | 2                   | 0.5–1.25                      |
| 4               | 11–16                    | 3                   | 1.25–2.5                      |
| 5               | 16–21                    | 4                   | 2.5–4                         |
| 6               | 22–27                    | 5                   | 4.0–6.0                       |
| 7               | 28–33                    |                      |                               |
| 8               | 34–40                    | 6                   | 6.0–9.0                       |
| 9               | 41–47                    |                      | 9.0–14.0                      |
| 10              | 48–55                    | 7                   | >14                           |
| 11              | 56–63                    | 8                   |                               |
| 12              | >63                      | >8                  |                               |

Table 10. Calculated $a_{\text{CG}}$ values based on sea states.

| CHSC | CRW | Significant Wave Height | V (knots) | $L_{\text{WL}}$ (m) | $a_{\text{CG}}$ (g) |
|------|-----|-------------------------|----------|--------------------|---------------------|
| 0.6  | 0.6 | Smooth Service Area (up to SS2) | 25       | 40                 | 1.42                |
| 0.6  | 0.66| Smooth Service Area (up to SS3) | 25       | 40                 | 1.54                |
| 0.6  | 0.75| Moderate Environment Service (up to SS4) | 25       | 40                 | 1.78                |
| 0.6  | 0.9 | Restricted Open Sea (up to SS5) | 25       | 40                 | 2.10                |
| 0.6  | 1   | Open Sea (up to SS9) | 25       | 40                 | 2.37                |

Figure 4. Stress values for LS1 loading case.

Based on the above calculated $a_{\text{CG}}$ values, maximum von Mises stress values were calculated as presented below in the Maestro Marine FEA software screenshots. As a general approach, von Mises and Tresca are popular approaches for stress analyses but not recommended for fatigue analyses. Von Mises and Tresca are nondirectional, whereas fatigue cracks are generally directional [27]. In the analysed FEA models and obtained
results, all the directional stresses were calculated lower than the von Mises stress values. Since this study was carried out to assess the undamaged life cycle of the vessel in the preliminary design phase and to simulate the worst loading conditions, maximum von Mises values were chosen to predict the entire life cycle of the vessel.

As can be seen from Figures 4–6, the maximum stress values were calculated by the waves coming from the head of the ship at “0” degrees. The other stress values caused at 30, 45, 60, and 90 degrees are not presented since they have lower stress values than the waves coming at “0” degrees. Normally for swath-type vessels, the stress values generated by the waves with a 45-degree attacking angle result in higher stress values than others. Whereas since the analysed vessel is semi-swath, which has a stronger outrigger–main body connection than the swath-type vessels, the maximum stress values were obtained for the forward side of the vessel. For the loading case (LS1), the maximum von Misses stress was calculated as 116 MPa with a 1.54 g additional acceleration, which was calculated and is presented in Table 10 as per the Germanisher Lloyd rules for sea state 3 (SS3).

![Figure 5. Additional acceleration applied due to the related sea state.](image)

![Figure 6. Applied wave specifications to FEA model.](image)

3.2. LS2-Operation Condition Calculations

In this operating condition, the vessel was assumed to be operated mostly in sea states three, four, and five. Hence, the wavelength was taken as 80 m, and the wave amplitude was taken as 2 m, resulting in a 4-metre wave height. The calculations and results were
derived similarly to the results for the LS1 loading condition. In this loading condition, the maximum stress values were calculated for the waves coming from the head of the ship with “0” degrees as well as LS1. The calculated cycle values are presented in Table 11.

\[
\frac{557933}{10^8} + \frac{474215}{10^8} + \frac{697508}{10^8} + \frac{591247}{10^8} + \frac{330466}{10^8} + \frac{183945}{10^8} = 0.05671 \quad (12)
\]

\[\text{Life time Without Any Fatigue Crack is;} \frac{1}{0.05671} = 17.6 \text{ years} \quad (13)\]

As can be summarised in the LS2 loading scenario, a lifetime without fatigue cracks is calculated as 17.6 years if the fatigue endurance limit is not exceeded. This result reveals that, especially for aluminium vessels, the most dominant parameter against fatigue cracks is the wave encounter frequency (number of cycles) since aluminium alloys have no horizontal fatigue endurance limit. As can be seen from the LS3 loading condition, which has a lower wave encounter frequency due to a longer wavelength, it offers the most extended lifetime without any fatigue cracks if the fatigue endurance limit is not exceeded. Obtained stress values, applied vertical accelerations and wave specifications have been presented respectively in Figures 7–9.

Table 11. Encounter cycle vs. sea state summary.

| Sea States   | %  | Hours | Seconds | Cycles For 0° Wave Enc. Angle | Cycles For 30° Wave Enc. Angle | Cycles For 45° Wave Enc. Angle | Cycles For 60° Wave Enc. Angle | Cycles For 90° Wave Enc. Angle |
|--------------|----|-------|---------|-----------------------------|-------------------------------|-------------------------------|-------------------------------|-------------------------------|
| Sea State 1  | 10%| 200   | 720,000 | 557,933                     | 509,758                       | 452,613                       | 378,140                       | 198,347                       |
| Sea State 2  | 10%| 200   | 720,000 | 474,215                     | 434,738                       | 387,910                       | 326,883                       | 179,551                       |
| Sea State 3  | 20%| 400   | 1,440,000 | 697,508                    | 643,756                       | 579,996                       | 496,902                       | 296,296                       |
| Sea State 4  | 25%| 500   | 1,800,000 | 591,247                    | 550,619                       | 502,428                       | 439,623                       | 288,000                       |
| Sea State 5  | 20%| 400   | 1,440,000 | 330,466                    | 310,429                       | 286,661                       | 255,685                       | 180,905                       |
| Sea State 6  | 15%| 300   | 1,080,000 | 183,945                    | 174,051                       | 162,314                       | 147,018                       | 110,092                       |
| \(\Sigma\)   |    |       | 2,835,315 | 2,623,350                  | 2,371,921                     | 2,044,253                     | 1,253,191                     |

Figure 7. Stress values for LS2 loading case.
3.3. LS3-Operation Condition Calculations

In this operation condition, the vessel was assumed to be operated mostly in sea states five, six, and higher. Hence, the wavelength was taken as 120 m, and the wave amplitude was taken as 3 metres. The calculations and results were derived similarly to the results for the LS1 loading condition. In this loading condition, the maximum stress values were calculated by the waves coming from the head of the ship with “0” degrees as well as LS1. The calculated cycle values are presented in Table 12.

\[
\frac{278967}{10^8} + \frac{237108}{10^8} + \frac{523131}{10^8} + \frac{472997}{10^8} + \frac{413083}{10^8} + \frac{367890}{10^8} = 0.023
\]  

(14)

Life time Without Any Fatigue Crack is; \( \frac{1}{0.023} = 43.47 \) years  

(15)

The longest lifetime without any fatigue crack was calculated as 43.5 years for the LS3 loading condition. However, having calculated the 221 MPa stress value, the vessel reached the highest von Mises stress in all the loading conditions. Obtained stress values, applied vertical accelerations and wave specifications have been presented respectively in Figures 10–12.
Table 12. Encounter cycle vs. sea state summary.

| Sea States    | %     | Hours | Second | Cycles For 0° Wave Enc. Angle | Cycles For 30° Wave Enc. Angle | Cycles For 45° Wave Enc. Angle | Cycles For 60° Wave Enc. Angle | Cycles For 90° Wave Enc. Angle |
|---------------|-------|-------|--------|------------------------------|------------------------------|------------------------------|------------------------------|------------------------------|
| Sea State 1   | 5%    | 100   | 360,000| 278,967                      | 254,879                      | 226,306                      | 189,070                      | 99,174                       |
| Sea State 2   | 5%    | 100   | 360,000| 237,108                      | 217,369                      | 193,955                      | 163,442                      | 89,776                       |
| Sea State 3   | 15%   | 300   | 1,080,000| 523,131                      | 482,817                      | 434,997                      | 372,677                      | 222,222                      |
| Sea State 4   | 20%   | 400   | 1,440,000| 472,997                      | 440,496                      | 401,942                      | 351,699                      | 230,400                      |
| Sea State 5   | 25%   | 500   | 1,800,000| 413,083                      | 388,036                      | 358,326                      | 319,607                      | 226,131                      |
| Sea State 6   | 30%   | 600   | 2,160,000| 367,890                      | 348,101                      | 324,628                      | 294,037                      | 220,183                      |
| Σ             |       |       |        | 2,293,175                    | 2,131,697                    | 1,940,154                    | 1,690,530                    | 1,087,885                    |

Figure 10. Stress values for LS3 loading case.

Figure 11. Additional acceleration applied due to the related sea state.
3.4. LS4-Operation Condition Calculations

This operating condition aims to implement more realistic loading conditions than the previously created worst-case scenarios. In this scenario, the number of cycles the vessel may encounter during its lifetime was calculated considering different wave attack angles. The calculated cycles based on the different wave attacking angles are given in Table 13. Table 13 reveals that randomly chosen sea conditions yield fewer cycles than the LS1 loading scenario and higher cycles than the LS2 and LS3 loading scenarios.

Table 13. Random sea condition loading scenarios.

| Possible Scenarios | % Spent Time | Hours | Seconds | SS | Wave Attacking Angle | V | T | λ | w | k | VR | Transition Time | Cycles       |
|--------------------|--------------|-------|---------|----|----------------------|---|---|---|---|---|-----|-----------------|--------------|
| PS1                | 14           | 280   | 1,008,000 | 2  | 0                    | 10.3 | 3.63 | 20.58 | 1.73 | 0.31 | 15.95 | 1.29              | 781,107      |
| PS2                | 10           | 200   | 720,000  | 3  | 15                   | 10.3 | 4.86 | 36.90 | 1.29 | 0.17 | 17.52 | 2.11              | 341,919      |
| PS3                | 15           | 300   | 1,080,000 | 2  | 30                   | 10.3 | 4.86 | 36.90 | 1.29 | 0.17 | 17.52 | 1.65              | 652,780      |
| PS4                | 9            | 180   | 648,000  | 3  | 45                   | 10.3 | 4.86 | 36.90 | 1.29 | 0.17 | 14.86 | 2.48              | 260,998      |
| PS5                | 8            | 160   | 576,000  | 4  | 60                   | 10.3 | 6.25 | 61.02 | 1.00 | 0.10 | 14.90 | 4.09              | 140,679      |
| PS6                | 6            | 120   | 432,000  | 5  | 75                   | 10.3 | 7.96 | 98.98 | 0.79 | 0.06 | 15.09 | 6.56              | 65,884       |
| PS7                | 7            | 140   | 504,000  | 4  | 90                   | 10.3 | 6.25 | 61.02 | 1.00 | 0.10 | 9.76   | 6.25             | 80,640       |
| PS8                | 5            | 100   | 360,000  | 6  | 45                   | 10.3 | 9.81 | 150.33 | 0.64 | 0.04 | 22.59 | 6.65              | 54,105       |
| PS9                | 18           | 360   | 1,296,000| 3  | 15                   | 10.3 | 4.86 | 36.90 | 1.29 | 0.17 | 17.52 | 2.11              | 615,453      |
| PS10               | 8            | 160   | 576,000  | 4  | 20                   | 10.3 | 6.25 | 61.02 | 1.00 | 0.10 | 19.42 | 3.14              | 183,347      |
| Σ                  | 100          | 2000  | 7,200,000| 3  |                      |      |      |      |     |     |       |                  | 3,176,912    |

The scenarios which have been defined in Table 13 can also be followed in Figures 13 and 14. These 3D scatter graphs and matrix assembly equations were created by the Mathcad Prime 4.0 software.

\[
\frac{781107}{10^8} + \frac{341919}{10^8} + \frac{652780}{10^8} + \frac{260998}{10^8} + \frac{140679}{10^8} + \frac{65884}{10^8} + \frac{80640}{10^8} + \frac{54105}{10^8} + \frac{615453}{10^8} + \frac{183347}{10^8} = (16)
\]

\[
\text{Life time Without Any Fatigue Crack is; } = \frac{1}{0.0318} = 31.44 \text{ years} \quad (17)
\]
3.5. LS5-Operation Condition Calculations

Unlike the first four operational conditions, the LS5 operation condition was created to assess the fatigue life of the vessel if the vessel operates under the International Code of Safety for High-Speed Craft. So far, fatigue life predictions, encounter frequencies, and cycles have been calculated based on the vessel’s maximum speed and different sea states. Nevertheless, as given in Figure 15, a 1.0 g vertical acceleration is the limit for degradation of passenger safety limit. At values above 1 g, level 4 catastrophic effect level starts. These phenomena can be controlled by specially designed shock absorbing seats, which are defined and specified in reference [12].

In this section, as suggested in Table 14, a 1.0 g vertical acceleration was applied to the vessel’s centre of gravity. The stress results are shown in Figures 15 and 16. The maximum stress value was calculated as 78 MPa, which is under the fatigue endurance limit of AA5059.
Figure 15. Total of a 1.0 g vertical acceleration applied to FE model and stress results.

Figure 16. Total of a 1.0 g vertical acceleration applied to FE model and stress results.
Table 14. Maximum acceleration criteria as per International Code of Safety for High-Speed Craft.

| Effect                          | Criteria Not To Be Exceeded                                                                 | Comments                                                                 |
|---------------------------------|---------------------------------------------------------------------------------------------|--------------------------------------------------------------------------|
| LEVEL 1                          | Maximum acceleration measured horizontally                                                   | 0.008 g: Elderly person will keep balance when holding                   |
| MINOR EFFECT                    |                                                                                             | 0.15 g: Mean person will keep balance when holding                       |
| Moderate degradation of safety   |                                                                                             | 0.15 g: Sitting person will start holding                                 |
| LEVEL 2                          | Maximum acceleration measured horizontally                                                   | 0.25 g: Maximum load for mean person keeping balance when holding        |
| MAJOR EFFECT                    |                                                                                             | 0.45 g: Mean person fails out of seat when not wearing seat belts         |
| Significant degradation of safety|                                                                                             |                                                                          |
| LEVEL 3                          | Collision design condition (calculated)                                                     | Ref. 4.3.3                                                               |
| HAZARDOUS EFFECT                | Maximum structural design load, based on vertical acceleration at centre of gravity         | Ref. 4.3.1                                                               |
| Major degradation of safety      | Risk of injury to passengers: safe emergency operation after collision.                     | 1.0 g: Degradation of passenger safety                                   |
| LEVEL 4                          |                                                                                             | Loss of craft or fatalities                                              |
| CATASTROPHIC EFFECT             |                                                                                             |                                                                          |

4. Discussion and Results

This study was carried out to calculate the undamaged life span of a semi-swath-type vessel under several loading scenarios in the initial design phase. To calculate the stress and hot spot areas caused by the wave loads with different angles of attacks, the Maestro Marine finite element analysis software was utilised. Calculations were performed for five different loading cases, and the results are presented separately at the end of each loading case. The results are summarised and shown in Table 15.

Table 15. Loading scenarios summary table.

| Loading Scenarios | Total Number of Cycles | Predicted Life Without Fatigue Life (Years) | Maximum Stress Values (MPa) |
|-------------------|------------------------|---------------------------------------------|-----------------------------|
| LS1               | 4,007,064              | 12.5                                        | 116                         |
| LS2               | 2,835,315              | 17.6                                        | 178                         |
| LS3               | 2,293,175              | 43.47                                       | 221                         |
| LS4               | 3,176,912              | 31.44                                       | 220                         |
| LS5               | infinite               | infinite                                    | 78                          |

As shown in Table 15, higher sea states offer a longer undamaged life span. As a result of the calculations, the obtained stress values and the number of cycles are inversely proportional. When the encounter frequency is reduced by the result of the longer waves, which are observed in higher sea state conditions, the calculated stress values increased significantly, resulting in the exceedance of the fatigue endurance limits. The operator must reduce the vessel’s speed to limit the maximum stress value under the fatigue endurance limit or the vessel must be designed structurally to stand against the highest stress values caused by the highest sea states.

This study shows that an undamaged life span assessment is strongly dependent on the sea region, specific wave characteristics, angle of attack, operating hours in sea states, and vessel building material. Structural improvements, especially in hot spot regions, would significantly increase the undamaged life span. As highlighted at the beginning of this study, more dense and fine mesh modelling is required to achieve more accurate results in fatigue analysis. Following this study, and after revealing the hot spot regions, the mentioned locations can be remapped and meshed with the smaller size elements to obtain more accurate and precise results. Stress singularities should be handled carefully in structural and fatigue analyses, especially in hot spot regions. Stress singularities are expected to occur at the sharp re-entrant corners and point restraints. St-Venant’s principle can be applied to overcome these singularities, or some geometric nonconformities, such as sharp and re-entrant corners, can be removed in the following analysis.
The evolution of naval vessels towards high-speed crafts subjected to severe sea conditions has promoted an increasing interest in lightweight, high-strength materials. Due to its strength and weight characteristics, aluminium has been proven especially suitable as the construction material for hull structures and other vessel parts. However, fatigue in aluminium naval crafts needs to be effectively addressed for the proper life cycle assessment [28]. In addition, sea loads on the vessel due to ship motions are not the only source of fatigue cracks on high-speed vessels. Another common and most crucial fatigue cause for high-speed aluminium vessels are water jets and supercavitating propellers. Generally, the fatigue cracks on high-speed aluminium vessels occur near the transom and around the water jets and propellers [29]. In this study, propulsion systems and their fatigue effects are not discussed.

The LS4, the most realistic operation condition, can be recreated based on real-time data obtained from ship logbooks to create more accurate loading conditions and results. The LS4 loading condition also reveals that the vessel operator should avoid operating the vessel in short wavelengths at high speeds, which may exceed the fatigue endurance limit. Hence, to increase the vessel’s undamaged life span, the operator must know the relation between the vessel’s relative speed and maximum stress values to avoid exceeding the fatigue endurance limit. The operator must reduce the vessel’s speed to limit the maximum stress under the fatigue endurance limit, or the vessel must be designed structurally to stand against the highest stress values caused by the highest sea states.

The encounter frequencies, one of the most influential parameters in fatigue life calculation, are also dominantly effective in motion sickness incidence (MSI) analysis. Sinusoidal low-frequency translational oscillations may trigger motion sickness, especially in high-speed crafts. As Table 2 reveals, wave encounter frequencies differ between 0.275 Hz and 0.102 Hz. Based on the research of Golding JF et al., 8 of 12 subjects faced moderate nausea(seasickness) at 0.1 Hz. Twelve of 12 subjects experienced seasickness at 0.2 Hz, and 7 of the 12 subjects experienced nausea [30] at 0.4 Hz. Based on this study, sea state condition three on the Aegean Sea presents a less comfortable voyage since the encounter frequency is 0.205 Hz. The operator should also consider the MSI frequencies during operation and the fatigue phenomena.

The LS5 was created to estimate how long the vessel’s undamaged life would be under operational loads if operated to consider human health and safety as per the International Code of Safety for High-Speed Craft. This scenario clearly shows that the vehicle has an infinite life when operated according to the International Code of Safety for High-Speed Craft since 78MPa is significantly lower than the fatigue endurance limit at \(10^8\) cycles. As Figure 3 suggests, if the used material were AA5083 instead of AA5059, we would not say that the vessel has an infinite fatigue life since the S–N curve of AA5083 still has a tendency to decrease with the number of cycles.

5. Conclusions

In this study, a multihull boat’s fatigue analysis was carried out for different sea states and loading conditions at a vessel speed of 20 knots. The undamaged fatigue life of the naval vessel was calculated for different loading conditions and scenarios. As the results of the analysis and comparison between different scenarios show, when the vessel is operated as per the International Code of Safety for High-Speed Craft, which was analysed at LS5, it offers the most extended undamaged fatigue life. On the contrary, the shortest fatigue life was calculated at LS1 since this loading scenario has the highest number of cycles due to high encounter frequencies and the hull structural stress over the fatigue endurance limit. This study clearly shows that the fatigue life of the vessel is directly and most dominantly related to the operator factor and the selection of the boat’s building material.

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