Assessment of ship structure under fatigue loading: FE benchmarking and extended performance analysis

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Abstract: This paper presents a numerical procedure based on the finite element (FE) method using ANSYS Workbench software to analyze fatigue phenomena in ship structures. Fatigue failure prediction is used as a stress–life approach, when the stress is still in a linear area. This condition is frequently referred as high-cycle fatigue. Five geometric shapes taken from midship points on the structure of a ship are sampled. There are four types of materials: HSLA SAE 950X, medium-carbon steel, SAE 316L, and SAE 304L. The types of loading imposed on each sample include three conditions: zero-based, zero mean, and ratio. Mesh convergence analysis is conducted to determine the most effective mesh shape and size for analyzing the structure. The results showed that the configuration of the geometric shapes, materials used, loading schemes, and mean stress theory affect the fatigue characteristics of the structure.

Keywords: Finite element method, fatigue phenomena, ship structure

1 Introduction

Currently, technology advancement, particularly in engineering structures, is accelerating. The complexity of the challenges and the obstacles encountered is also increasing; one of the most frequent issues is material failure, which is caused by various factors. Catastrophic material failures result in material losses, which frequently cause fatalities and damage to the environment. Operational loads commonly induce material failure in structures, and they can be classified into two categories: static and dynamic loads. The most frequently encountered issues are typically related to the presence of dynamic loads, such as unintentional loads/impact loads originating from various sources [1]. However, dynamic loads with a relatively small magnitude (far below the yield point) might also cause failure if they continue indefinitely, which is referred as fatigue failure [2].

Although fatigue failure can occur in any structure, it is more prevalent in those subjected to cyclic loads or operating in harsh environments. Ships comprise facilities that are prone to fatigue due to working in corrosive conditions, being subjected to continuous loading by seawater waves, being impacted by changes in ambient temperature, and being subjected to other mechanical loads. Ships are frequently constructed considering empirical loads (design loads), which reflect the most significant subjected static load. However, the causes and mechanisms of fatigue (induced by dynamic loads) remain unknown and under investigation. Ships are constructed from various highly complex structures and joints, responding differently to fatigue risk [3]. Fatigue failure can begin with cracks in locations with a high-stress concentration. These cracks may propagate in a specific direction and produce a fracture. The used materials, geometric shape, and type of loading are factors to consider when examining the features of a structure. The design must be economically viable and safe to operate under various loading scenarios. Fatigue testing should consider actual field scenarios, where testing and measurement should be performed with various parameter adjustments. Understandably, if the standard experimental procedures are used, the above will be extremely difficult, time-consuming, and prohibitively expensive. The finite element method (FEM) is one of the alternatives to this problem because it is highly effective in identifying and visualising fatigue failure mechanisms.

This study presents some data related to fatigue assessment of the design of ship structures from different perspectives. Previous research [1, 3–7] has identified several
potential issues that can be further investigated. The area in the middle of a ship (Figure 1) experiences the most extreme loading and has the highest risk of fatigue failure. These problems should be further examined, using alternative methods and approaches to provide a complete explanation of the phenomenon of weariness. This problem must be discovered in advance and addressed entirely to avoid future disasters [8–10]. The purpose of this study is to determine the effects of geometric shapes, material types, and loading types on the fatigue phenomenon occurring in ship structures using the hot spot stress approach. This study conducts fatigue analysis of numerous sample hot spots with material and applied load variations. This research is simply a follow-up that aims to fill in the gaps of prior investigations. To the authors’ best knowledge, the FEM has not been tested by highly complex mesh convergence investigations, which is critical for ensuring the accuracy of numerical simulation findings.

Figure 1: Weakest areas on ship structures [4]

2 Literature review

2.1 Pioneer works

Research on the phenomenon of fatigue has a very long history [3, 11]. The first paper on the fatigue phenomenon was written in 1837. A further development, the effect of stress concentration on the failure of axle trains, was introduced in 1842. Systematic fatigue testing methods were developed in 1860. This method was later modified in 1870, and the concept of the stress–cycle curve (S–N) curve and endurance limits are introduced. The linear damage hypothesis discovered in 1945 was still referenced when the stress-life approach was adopted to investigate the current fatigue phenomenon. The 1979 loss of MV Kurdistan and the 1980 sinking of the Alexander L. Kielland platform due to fatigue failure [9] prompted academics to conduct additional research on this phenomenon, particularly in marine structures. The growing importance of fatigue strength in maritime constructions has resulted in its study and design recommendations. In the literature, there are numerous approaches for defining stress and implementing fatigue assessment. The two most accepted methods for the stress analysis of the structure of a ship are the hot spot stress approach and the practical notch stress approach [5].

A study on the measurement of the fatigue of ship structures was undertaken in [12] on a perpendicular joint consisting of a plate connection with a beam (shell–solid coupling). This study showed that finite element analysis may be used quite well to predict structural responses to fatigue loading, particularly in locations of high-stress concentration. Because the metrics tested and compared in this study are displacement-related, other processes are required to determine fatigue life parameters, fatigue damage, and fatigue safety factor. Subsequently, in [5], a comparative study was conducted on various vessels using more advanced approaches. The model under consideration is an integral part of the 4900 PCTC ship. The study established a method to determine the hot spot stress by examining global models. The areas most affected by a fatigue load were subsequently used to create a local model with mesh refinement in the crucial region. Loading scenarios were modelled in various ways, all of which involve using the same type of material. Based on the results of worldwide studies on models, there are other locations that may be exploited as research hot spots. In a study, the use of nonlinear time-domain hydrodynamic models of container ships (DNV-class) [13] showed that the selected material affects the structural response under various loading schemes. The study compared HT32-grade steel with various materials. Different wave heights resulted in several different loading patterns. The fatigue properties of the structures in response to this loading pattern were diversified; however, other hot spots remain to be studied. The fatigue cracks propagation approach can be used to analyse the fatigue characteristics in a shell structure, which is the magnitude of the crack propagation caused by dynamic stresses on ships [14–16]. This technique is predicated on the initial assumption of a crack in specimens derived from several possible sources. This study found that variations in loading scenarios, such as the tangle force and he wave frequency, substantially af-
fected the rate of fracture propagation. This method cannot describe fatigue prior to the onset of cracks.

2.2 Fundamental theory

Fatigue is the irreversible damage of items due to the stress–strain variations caused by external factors [17]. According to [18], fatigue failure occurs in four stages: (1) nucleation of cracks, (2) structurally dependent crack propagation, (3) crack propagation, and (4) failure. Numerous factors can influence fatigue resistance including the type of applied load, material used, mechanical properties, manufacturing techniques, surface roughness, operating temperature, environment, microstructure state, residual stress, corrosion, and crack initiation [19–22].

Metals can be classified according to their uniaxial properties, which include engineering properties and actual characteristics. Engineering properties are types of characteristics used to compute the cross-sectional area and the length of a sample in its original configuration. In comparison, stress–strain factors are calculated using the immediate space and size of a sample loading process. Engineering stress (Eq. (1)) is fundamentally different from actual stress (Eq. (2)).

\[
S = \frac{P}{A_0} \quad (1)
\]

\[
\sigma = \frac{P}{A} \quad (2)
\]

Above, \(P\) represents the axial tension stress, \(A_0\) denotes the initial cross-sectional area of the sample, and \(A\) represents the instantaneous cross-sectional area of the sample. When assessing a structure, the actual stress, which is affected by the cross-sectional variation, is employed. When materials are tested for strength in the laboratory, the shape of the specimen is highly simple, allowing dimensional changes to be easily observed and quantified directly. However, dimensional changes are exceedingly difficult to detect in complex structures, such as ship structures. Each piece has its unique distribution of stress. Expectedly, doing trials on a complete design will require a significant amount of resources. This is because the cost of specialised sensors for stress stamping is high, and the arrangement is relatively intricate. Consequently, an alternate approach for analysing the strength of a structure is to employ an FEM-based software. The FEM is a numerical technique for solving mathematical problems that include specified boundary conditions. In principle, when used to solve a problem, a space model is separated into multiple portions of the domain referred as up elements; this procedure is called meshing. Each element contains various stress components, which can be determined using interpolation and extrapolation concepts. When conducting fatigue analysis using the stress–life approach, each element must be searched for its corresponding stress value and subsequently compared to the fatigue data in the form of an S–N curve.

Different types of stresses can be utilised to forecast fatigue age, including axial stress (\(S_x\) or \(S_y\)) and shear stress (\(S_{xy}\)). Using the von Mises equation, these three types of stresses can be converted into normal or equivalent stress (Eq. (3)). This stress component is used in this study because it encompasses all other stress components. Thus, for the simulation of fatigue due to cyclic loads, the maximum stress (\(S_{\text{max}}\)) and the minimum stress (\(S_{\text{min}}\)) must be obtained, and subsequently Eq. (4) is used to determine the ratio.

\[
\sigma_{eqv} = \sqrt{S_x^2 + S_y^2 - S_xS_y + 3S_{xy}^2} \quad (3)
\]

\[
R = \frac{S_{\text{min}}}{S_{\text{max}}} \quad (4)
\]

The fatigue data obtained from laboratory test processes shown in S–N curves become input variables. Typically, these data are collected at a mean of zero or \(R = -1\) (Figure 2a). If the fatigue data are to be utilised to study an issue under a zero-based loading condition (\(R = 0\) or \(R = \infty\) or with ratio \(R > 0\) (see Figure 2b and Eq. (4)), then the mean stress must be corrected. Various theories can be employed, including the Goodman (England, 1899) (Eq. (5)), Soderberg (the USA, 1930) (Eq. (6)), Gerber (Germany, 1874) (Eq. (7)), and ASME elliptical (Eq. (8)) theories [23–26].

\[
\frac{S_{\text{Alternating}}}{S_{\text{Endurance limit}}} + \frac{S_{\text{mean}}}{S_{\text{ultimate}}} = 1 \quad (5)
\]

\[
\frac{S_{\text{Alternating}}}{S_{\text{Endurance limit}}} + \frac{S_{\text{mean}}}{S_{\text{yield}}} = 1 \quad (6)
\]

![Figure 2: Loading conditions: a. Zero-mean; b. Ratio](image-url)
The magnitude of fatigue life can be determined using the Palmgren–Miner linear damage hypothesis (Eq. (9)), where denotes the number of stress range cycles caused by various factual stressors \( S_i \) \( (1 \leq i \leq k) \) and \( N_i \) represents the number of cycles required to cause the failure of the alternating constant stress, \( S_i \) (S–N curve). Failure occurs when cumulative damage (D) exceeds one.

\[
\frac{S_{\text{Alternating}}}{S_{\text{Endurance limit}}} + \left( \frac{S_{\text{mean}}}{S_{\text{ultimate}}} \right)^2 = 1 \tag{7}
\]

\[
\left( \frac{S_{\text{Alternating}}}{S_{\text{Endurance limit}}} \right)^2 + \left( \frac{S_{\text{mean}}}{S_{\text{yield}}} \right)^2 = 1 \tag{8}
\]

\[
D = \sum_{i=1}^{k} \frac{n_i}{N_i} \tag{9}
\]

The research is conducted in several stages, and each stage has its role. Literature studies are conducted to ensure that the used methods follow existing scientific rules. Prior research is referred to define the input variables and the fundamental assumptions. Material properties are obtained from the results of laboratory tests with recognised standards. Numerical methods are validated before use. One approach involves benchmark analysis procedures. The shapes and magnitudes of the meshes for all geometries are different; therefore, convergence studies are conducted to choose the most optimal mesh. These stages are discussed in more detail sequentially in this section.

### 3.1 Assessment procedures

A literature study is conducted to understand the phenomenon of fatigue in ship structures. It identifies several problems related to fatigue in the construction of ships and the methods and the approaches that can be used to investigate fatigue. The method used is the FEM with CAE simulation software, and the technique used is the stress–life approach, which is applied to the hot spot stress area. Before use, this method is validated to replicate previous research, and subsequently the results are compared. If the error is relatively small and meets certain criterion, then this method is valid and worth using.

Geometric models are built using ANSYS Modeller, a CAD program integrated with ANSYS Workbench. Furthermore, the material properties obtained by the literature study are inputted into the software manually. Subsequently, the process of meshing divides a geometry into several small domains called elements. The mesh size is determined to achieve convergence of calculation value and time efficiency. Following this, the boundary condition is determined to set the placement of the pedestal and the loading location. Fatigue analysis is conducted using the fatigue tool in ANSYS Workbench. The parameters that can be varied are fatigue damage, fatigue life, safety factor, and biaxiality indication. The values of some output parameters are subsequently analysed and related for inference. The research scheme in this study is shown in Figure 3.

### 3.2 Engineering three-dimensional models

The studied samples are taken from several locations on the structure of a ship that can represent the most prone parts to fatigue failure. Furthermore, these samples are formed into three-dimensional models, following a limit condition applied based on the loading conditions that occur in the field. Areas with high-stress concentrations (hot spot stress) typically have a short fatigue life.
Figure 4: Geometry locations on midship section model [27]

a. Bracket Plate  
b. Hatch Coaming Plate

c. Hatch Coaming Plate

d. Bottom Stiffened Panel

e. Side Stiffened Panel

Figure 5: Geometry model: a. Geometry 1: Hatch coaming model [16]; b. Geometry 2: Perpendicular joint [12]; c. Geometry 3: Stool joint [12]; d. Geometry 4: Bottom-stiffened panel [28]; e. Geometry 5: Side-stiffened panel [29]
Based on a literature study on the global models of ships, it is found that the hull and the midship (Figure 1) are the most vulnerable areas to failure owing to stress concentration. Therefore, five types of sample models are taken in the above sections representing the hot spot areas on a ship structure (Figure 4). One model is used to investigate the effects of the applied material, loading scheme, and mean stress correction theory on the fatigue behaviour that occurs in the structure. When one variable is examined, the other variables are considered constant.

Boundary conditions are applied to the built geometric models (Figure 5), as shown in Figure 6. Based on a literature study, the magnitude of the load, the position of the support, and other configurations are adjusted to the actual needs.

Some other parameters that are not set in this study are left constant or follow the default settings of ANSYS software. Some basic assumptions that need to be highlighted are that the approach used in this study is the stress–life approach and the stress is still in the linear area below...
the material yield point. The model is in perfect condition, there is no initial crack, and the smoothness of the surface is considered uniform. Mesh convergence is conducted for each model using different mesh sizes. One method to determine the range of mesh sizes to be used is based on the element length to thickness (ELT) ratio. According to [7], ELT values of 5–10 can analyse a complex structure. A small ELT ratio implies that the analysis result is close to the actual value, and consequently the computing time is long. The meshing technique used is local meshing. Mesh in the area of interest is refined thrice smoother than the global mesh. The stress that appears subsequently represents the hot spot stress, whose value is more significant than the nominal stress.

4 Materials

Four types of materials are used in this study: high-strength low-alloy steel (HSLA) SAE 950X [30], medium-carbon steel [31], SAE 316L [32], and SAE 304L [33]. The material properties used as input parameters are listed in Tables 1 and 2 and shown in Figure 7. HSLAs are relatively new materials and are still not as well-known as SAE 316L and 304L for applications in marine structures. Although medium-carbon steels have good mechanical properties, they are rarely used in marine structures with some consideration.

Koksal [34] previously completed an FEM fatigue analysis on notched cantilever beams. The research material was a structural steel, with the mechanical parameters listed in Table 3 and the S–N curve shown in Figure 8c. The dimensions of the geometric model are 1000 × 100 × 75 mm; on one side of the beam, a notch with a large angle of 90° and a depth of 25 mm is created (Figure 8a). Cantilever beams

Table 1: Chemical composition

| Type of material | C  | Cr  | Mo  | Si  | Mn  | S  | P  | Ni  | Cu  | Co  | N  | V  | Nb |
|------------------|----|-----|-----|-----|-----|----|----|-----|-----|-----|----|----|----|
| HSLA SAE 950X    | 0.23 | 0.9 | 1.35 | 0.05 | 0.04 | 0.15 | 0.04 | 0.15 | 0.04 |
| Medium-carbon    | 0.44 | 0.04 | 0.02 | 0.23 | 0.57 | 0.16 | 0.24 | 0.002 |
| SAE 316L         | 0.29 | 17.54 | 2.38 | 0.5 | 1.8 | 0.012 | 0.032 | 12.6 | 0.26 | 0.18 | 0.077 |
| SAE 304L         | 0.02 | 18.5 | 0.049 | 1.78 | 0.011 | 0.014 | 9.78 |

Table 2: Mechanical properties

| Type of material | Modulus of elasticity (MPa) | Yield strength (MPa) | Ultimate strength (MPa) |
|------------------|-----------------------------|----------------------|-------------------------|
| HSLA SAE 950X    | 204700                      | 437                  | 485                     |
| Medium-carbon    | 210000                      | 490                  | 710                     |
| SAE 316L         | 200000                      | 283                  | 592                     |
| SAE 304L         | 190200                      | 277                  | 572                     |

Table 3: Mechanical properties of structural steel (ANSYS)

| Modulus elasticity (GPa) | Ultimate tensile strength (MPa) | Poisson’s ratio | Yield Strength (MPa) |
|-------------------------|--------------------------------|----------------|---------------------|
| 200                     | 460                            | 0.3            | 250                 |
Table 4: Verification results

| Value                        | Previous test (33) | Re-simulation result | Error (%) |
|------------------------------|--------------------|----------------------|-----------|
| Min fatigue life             | 90,700.00          | 89,948.00            | 0.83      |
| Min safety factor            | 0.60824            | 0.60686              | 0.20      |
| Maximum equivalent alternating stress | 141.72          | 142.04               | 0.20      |

Figure 8: a. Cantilever beam design [34]; b. Boundary condition; c. S–N curve of structural steel (ANSYS)

are intended for a minimum cycle life of $10^6$. Their one end is stationary, whereas the other end is loaded with a force of 10 kN in the direction of the z-axis (Figure 8b).

Based on a convergence study, a 4-mm mesh produces 90,322 elements and a single processing step requires 20 s for computation. A tetrahedral mesh with Jacobian ratio $=1$ is chosen as the mesh shape. The applied loading is a zero-based force of 10 kN. After applying a cyclical load, the remaining settings are kept as default or constant. Table 4 lists and Figure 9 shows the verification findings.

Figure 9: Validation results: benchmarking with previous test [35]
Assessment of ship structure under fatigue loading

Figure 10: Mesh convergence tests: a. Nominal stress; b. Hot spot stress

Figure 11: Size 15-mm mesh quality: a. Aspect ratio; b. Skewness
6 Mesh convergence study

The shape and size of a mesh significantly influence simulation results. Generally, a small mesh size implies that the analysis results are close to the actual condition [36]. Concurrently, a small mesh size implies numerous formed elements. Consequently, the computing time becomes long, and in some cases, the computer devices used cannot deal with these calculations [37, 38]. Thus, a mesh convergence study is needed to choose the most suitable shape and size of a mesh. When the simulation results meet the criteria and the time required for one simulation is relatively short, the mesh is considered appropriate. Simulation results are considered convergent if the changes in the mesh size do not affect the values of the tested parameters. ANSYS software provides several forms of meshes, including hexahedron, pyramid, prism, and tetrahedral. A hexahedron-shaped mesh is considered ideal because it produces relatively small elements and is the form most easily analyzed by a program solver [39]. However, this mesh cannot adjust a shape well, particularly for a complex geometry. Although it produces relatively larger elements, a tetrahedron mesh can be used for complex geometries. Therefore, this type of mesh is chosen for this study.

A convergence test is conducted on Geometry 2 by continuously changing the mesh, following which the stress values in all changes are compared. Two meshing techniques are applied: global and local meshing. Global meshing is used to examine the nominal stress on one intact structure. In comparison, local meshing is used to search for the stress hot spots in critical areas. The stress magnitudes of the stress hot spots are above the nominal stress because the distribution investigated covers a more detailed scope. Based on the test results (Figures 10a and 10b), a 15-mm mesh is determined as the best to solve this first case. In addition to no change occurring in the stress value, a computing time of less than 4 s is the most relevant consideration. A mesh with a smaller size is not chosen because it requires a longer computing time. Although a mesh with a larger size tends to produce values far below the criteria or away from the actual conditions, the mesh quality test shows that the inspection area has a good aspect ratio and skewness. The aspect ratio (Figure 11a) is the most extensive with the smallest mesh. The aspect ratio approaching the value of 1 is better because the governing equation becomes more superficial and makes it easier for solvers to find the final result. The data are expected if the skewness is in the range of $-2$–$2$. As shown in Figure 11b, the skewness magnitude is still in the field of $0.059686$–$0.99432$, which indicates that the mesh has a good quality.

7 Results and discussion

The data that are successfully captured are processed and grouped based on the investigated variables. The information displayed as contour images and graphs shows the relationships among the variables.

7.1 Effect of geometry shape

It can be observed from Figure 12a that Geometry 1 under loading produces a maximum stress of 252.47 MPa, which is still below the stress yield. Stress concentration occurs in the connection area and spreads to the plate area. A minimum stress of 1.2082 MPa occurs in the support area, which experiences the slightest moment. Geometry 2 (Figure 12b) has a stress of 289.76 MPa, which is relatively larger than Geometry 1. Geometry 3 (Figure 12c) experiences almost the same stress as Geometry 2 because Geometries 2 and 3 have similar adjacent ship structures and boundary conditions. However, Geometry 3 can distribute the stress better than Geometry 2. Geometry 4 (Figure 12d) experiences a very high stress, which amounts to 312.67 MPa in the neck area. Geometry 5 (Figure 12e) is under a very small stress of 188.83 MPa. The received load is well channelled throughout the surface of the model. Thus, the structure is subsequently strengthened by stiffeners.

First, the amount of stress in the structure is inversely coupled with the predicted fatigue life. A large value of the stress implies a short fatigue life. It can be proven from Figure 13d that Geometry 4, which experiences the highest stress, has a very short fatigue life. In contrast, Geometry 5 (Figure 13e), which is under little stress, has a good fatigue life. The second point to note is the relation between the stress concentration and the minimum fatigue life. Based on literature studies, fatigue failure typically begins with an initial crack. Initially, Geometries 2 and 3 (Figures 13b and 13c) seem safe, where the red contoured area is relatively small. However, the stress exceeds the endurance limit, where based on calculations, the corresponding area will fail in specific cycles. Failure typically begins with a fatigue crack. Although very small in size, such a crack is very dangerous because it can propagate very rapidly.

Figure 14 shows the safety factor of each geometry. Please note that ships are structures designed with a high level of security. Therefore, to ensure their ability to be qualified to withstand repeated loading, the fatigue safety factor is calculated based on a design life of $10^9$ cycles. Areas with a fatigue safety factor of 1 are expected to survive when undergoing 1 billion cycles. Geometry 1 (Figure 14a) has a
uniform minimum distribution of safety factors with values below 1, i.e., the areas with red contours are predicted to fail before they can undergo 1 billion cycles. As shown in Figure 14d, Geometry 4 has the structure with the lowest level of security, whereas Geometry 5 has the safest form of the system.

Total deformation (Figure 15) presents the deformation distribution in the structure due to the loading process. Geometries 2 and 3 (Figures 15b and 15c) are deformed by 3.3328 and 3.4809 mm, respectively. Both these forms undergo the most significant deformation among all conditions. This indicates that both structures are deformable, and the different geometries are rigid. However, the deformations have no significant effect on the phenomenon of fatigue failure. When compared with Figure 13, it can be observed that the contours of the total deformation are different from the contours of the fatigue life. Areas that experience maximum deformation do not necessarily have

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**Figure 12:** Alternating stress results: a. Geometry 1: Hatch coaming model [16]; b. Geometry 2: Perpendicular joint [12]; c. Geometry 3: Stool joint [12]; d. Geometry 4: Bottom-stiffened panel [28]; e. Geometry 5: Side-stiffened panel [29]
Figure 13: Fatigue life results: a. Geometry 1: Hatch coaming model [16]; b. Geometry 2: Perpendicular joint [12]; c. Geometry 3: Stool joint [12]; d. Geometry 4: Bottom-stiffened panel [28]; e. Geometry 5: Side-stiffened panel [29]
Figure 14: Safety factor results: a. Geometry 1: Hatch coaming model [16]; b. Geometry 2: Perpendicular joint [12]; c. Geometry 3: Stool joint [12]; d. Geometry 4: Bottom-stiffened panel [28]; e. Geometry 5: Side-stiffened panel [29]
Figure 15: Total deformation results: a. Geometry 1: Hatch coaming model [16]; b. Geometry 2: Perpendicular joint [12]; c. Geometry 3: Stool joint [12]; d. Geometry 4: Bottom-stiffened panel [28]; e. Geometry 5: Side-stiffened panel [29]
a short fatigue life. The deformation is still in the elastic area, which is a condition that allows the structure to return to its original shape after the load is removed. This is supported by the evidence that the stress (Figure 12) is still below the material yield point.

Figure 16 compares the performance of all geometries under fatigue loading. Each geometry presents a unique response. Geometry 4 has the weakest characteristic or a very short fatigue life, whereas Geometry 5 has good fatigue resistance. In Figure 17a, Geometries 4 and 1 show slight deformation, whereas Geometries 2 and 3 undergo the most extensive deformation. Figure 17b shows that all geometries are still in critical areas, where the minimum safety factor is below 1. However, Geometry 5 is relatively safe because it has the highest safety factor. Figure 17c presents that a large damage occurrence implies a short fatigue life.

Figure 16: Relationship between alternating stress and fatigue life of each geometry

Figure 17: Fatigue results: Maximum deformation, minimum safety factor, maximum damage, and minimum life
7.2 Effect of material used

Geometry 2 is adopted as a sample to investigate the effect of the type of material used on the fatigue force on the structure. In addition to material variations, other variables that affect this property, such as the magnitude of loading, surface smoothness, stress corrosion factor, and loading scheme, are considered constant. SAE 316L and 304L materials are marine-grade steels, i.e., they meet standards for forming marine structures. The HSLA 950X material is a steel that is not yet commonly used in the marine structure industry. Despite its high toughness and corrosion resistance, the HSLA is considered less effective owing to its reasonably high production cost. Although a medium-carbon steel has good strength, it is not recommended for marine structure applications. It can be observed that the HSLA 950X (Figure 18a) material has a minimum fatigue life of 2.257e5 cycles, which is longer than those of the SAE 316L and 304L materials. The used medium-carbon steel (Figure 18b) has the highest fatigue resistance of all types of considered materials. However, it should be noted that corrosion can shorten the age of fatigue because it can trigger the appearance of an initial crack. Of the four types of materials, only the carbon medium has no corrosion resistance properties. The medium-carbon and SAE 316L materials have better stress distributions than HSLA 950X and SAE 304L.

Based on the contours of the safety factor (Figure 19), it is determined that the SAE 316L material has the lowest security, followed by SAE 304L. The HSLA 950X and medium-carbon materials have a higher safety factor than the specified criteria limit. Thus, none of the four materials

![Figure 18: Fatigue life: a. HSLA 950X; b. Medium-carbon; c. SAE 316L; d. SAE 304L](image)
is entirely safe and will potentially fail in a given cycle. Figures 20 and 21 show that the sequence of materials with the highest to the lowest fatigue resistance is medium-carbon steel, HSLA 950X, SAE 304L, and SAE 316L. Other factors that must be considered are the working environment conditions. The medium-carbon material will be brittle at a low temperature. Under a relatively small impact, the load alone can destroy the structure in these conditions. HSLA 950X, SAE 316L, and 304L are laboratory-tested and proven to withstand low temperatures. More research on this phenomenon needs to be conducted in the future.
Figure 21: Fatigue results: Maximum deformation, minimum safety factor, maximum damage, and minimum life for each material used.

Figure 22: Alternating stress results: a. Fully reversed; b. Zero-based; c. Ratio; d. Relevance with fatigue life.
7.3 Effect of load type

Three types of loading are compared (Figure 22). At the same amount of load, varying stresses will be produced if the type of loading is different. As can be observed, the loading ratio type only produces an alternating stress of 216.74 MPa. Similarly, when the exposure is changed to fully reversed, the stress appears to be 289.76 MPa. When the geometry is subjected to the highest loading ratio, the generated stress increases several folds to 650.21 MPa. This loading ratio type occurs when the bending and tensile forces have different values. There are some materials whose tensile strengths are higher than the compressive strength and vice versa. Thus, a material can be safe under a zero-based or fully reversed loading type, whereas it is fatal under the loading ratio. Therefore, the selection of materials needs to consider the type of loading that the structure may experience. The fatigue life material will certainly be affected by the type of loading.

The loading type does not affect the geometry deformation (see Figure 23). The fully reversed, zero-based, and ratio type loading produce the same total deformation of 3.3328 mm. The loading ratio of 2 with a scale factor of 1 does the most damage. A large damage implies a short fatigue life. When a geometry is subject to zero-based loading, the safety factor values produce the safest conditions among all types of loading.

7.4 Effect of mean stress correction theory

The data presented in the S–N curves were obtained from the results of the fatigue tests conducted in the laboratory. Fatigue tests are commonly conducted on uniaxial testing machines with a fully reversed loading scheme. These data cannot be used as the basis for analysing fatigue phenomena in structures that undergo non-fully reversed loading because the structural responses present different characteristics relative to the data shown in S–N curves. Therefore, the stress that emerges under non-fully changed conditions must be corrected using a mean stress theory. Mean stress theories have different effects. For example, in this study, the loading scheme is changed to zero-based loading to show differences. Figure 24 shows the maximum stress of each mean stress theory used. Under the same conditions, the Goodman, Soderberg, Gerber, and ASME elliptic theories correct the maximum stress to 206.6 MPa, 216.74 MPa, 159.08 MPa, and 153.57 MPa, respectively. First, it is noticeable that the Goodman and Soderberg theories produce almost the same stress values, whereas the Gerber theory
values are close to the results of the correction by the ASME elliptic theory. Based on literature studies, the Goodman and Soderberg theories are very suitable for analysing brittle materials, whereas the stress on ductile materials can be suitably corrected using the Gerber and ASME elliptic theories.

The stress distribution and the safety factor distribution (Figure 25) for each criterion have the same contours but different values when reviewed. Figure 26 shows that the Goodman and Soderberg theories are very conservative, with which the limit of failure is set far below the results of the lab experiments. The Gerber and ASME elliptic theories set higher failure limits than the other methods, approaching the data shown on the S–N curves. The Goodman and Soderberg theories result in smaller safety factor values than the Gerber and ASME elliptic theories. When used to predict the age of fatigue, the Goodman and Soderberg theories result in a minor period of fatigue.

The mean stress correction theory used does not affect the deformation value in the geometry (Figure 27). The Soderberg theory produces the most damage and a very short fatigue life. The ASME elliptic theory causes a slight damage and leads to excellent fatigue life. Specifically, different means stress theories introduce different characteristics into a geometric structure. Each theory has its advantages and disadvantages, and each depends on the material used.
Figure 25: Safety factor using different mean stress correction theories: a. Goodman; b. Soderberg; c. Gerber; d. ASME elliptic

Figure 26: Relationship between alternating stress and fatigue life using different mean stress correction theories
8 Conclusion

In this paper, the research results on the effects of geometric shapes, material types, loading types, and mean stress correction theories were presented. The research methods were validated by comparison with previous research. A difference in the results of <5% indicates that the technique used in this study is valid and can be accounted. Differences in the geometric shapes and locations of the structure affect the fatigue behaviour. The above study results prove that the bottom-stiffened panel has the highest fatigue risk, whereas the side-stiffened panel has the lowest. The medium-carbon steel material has a high resistance to high-cycle fatigue, which is the loading of fatigue with an intensity below the yield strength of the material. Fatigue characteristics against low-cycle fatigue need to be studied further. However, this material cannot be used for ship structure applications because of its low corrosion resistance. Corrosion can decrease strength because this will trigger the onset of a fatigue crack. The medium-carbon steel is also brittle; therefore, when a rupture occurs, the propagation is expected to have a high speed. Although the HSLA, SAE 316L, and 304L materials have relatively shorter fatigue life, they have good corrosion resistance. These materials are designed according to marine-grade steel standards.

Further research on the fatigue behaviour in low-cycle fatigue needs to be conducted considering other factors such as corrosion resistance and environmental temperature. Compared to fully reversed loading, zero-based loading produces relatively low stresses; therefore, a structure will have a safe fatigue life in this condition. The same nominal weighting ratio type leads to higher stresses, resulting in short-life fatigue. The mean stress theory used does not have a significant difference when the loading type withstood by the structure is the same as in the lab experiment results (fully reversed). The Goodman theory corrects the stress far below the safe limit when the loading scheme is changed to zero-based and ratio. The Goodman and Gerber theories are relatively more conservative than the Soderberg and ASME elliptic theories.

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References

[1] Li Z, Ringsberg JW, Storhaug G. Time-domain fatigue assessment of ship side-shell structures. Int J Fatigue. 2013;35:276–90.
[2] Xi R, Spiryagin M, Wu Q, Yang S, Liu Y. Fatigue life assessment methods for railway vehicle bogie frames. Eng Fail Anal. 2020;116:104725.
[3] Fajri A, Prabowo AR, Muhayat N, Smaradhana DF, Bahatmaka A. Fatigue analysis of engineering structures: State of development and achievement. Procedia Struct Integr. 2021;33:19–26.
[4] Ringsberg JW, Li Z, Tesanovic A, Knifsund C. Linear and nonlinear FE analyses of a container vessel in harsh sea state. Ships Offshore Struct. 2014;10(1):20–30.
[5] Tasdemir A, Nohut S. Fatigue analysis of ship structures with hinged deck design by finite element method. A case study: fatigue analysis of the primary supporting members of 4900 PCTC Mar. Structures. 2012;25(3):1–12.
[6] Prabowo AR, Bae DM, Sohn JM, Zakki AF, Cao B, Cho JH. Effects of the rebo unding of a striking ship on structural crashworthiness during ship-ship collision. Thin-Walled Struct. 2017;115:225–39.
[7] Prabowo AR, Putranto T, Sohn JM. Simulation of the behavior of a ship hull under grounding: effect of applied element size on structural crashworthiness. J Mar Sci Eng. 2019;7(8):270.
[8] Vukelić G, Vizentin G. Common Case Studies of Marine Structural Failures. Failure Analysis and Prevention. IntTech; 2017. pp. 135–51.
[9] France EJ. The Alexander L. Kielland Disaster Revisited: A Review by an Experienced Welding Engineer of the Catastrophic North Sea Platform Collapse. J Fail Anal Prev. 2019;19(4):875–81.
[10] Sedmak A. Computational fracture mechanics: an overview from early efforts to recent achievements. Fatigue Fract Eng Mater Struct. 2018;41(12):2438–74.
[11] Schütz W. A history of fatigue. Eng Fract Mech. 1996;54(2):263–300.
[12] Osawa N, Hashimoto K, Sawamura J, Nakai T, Suzuki S. Study on shell-solid coupling FE analysis for fatigue assessment of ship structure. Mar Structures. 2007;20(3):143–63.
[13] Li Z, Ringsberg JW, Storhaug G. Time-domain fatigue assessment of ship side-shell structures. Int J Fatigue. 2013;55:276–90.
[14] Alshoaibi AM, Fageehi YA. 2D finite element simulation of mixed mode fatigue crack propagation for CTS specimen. Integ Med Res. 2020;9(4):7850–61.
[15] Božič Ž, Schmauder S, Wolf H. The effect of residual stresses on fatigue crack propagation in welded stiffened panels. Eng Fail Anal. 2018;84:346–57.
[16] Zhang Y, Huang X, Wang F. Fatigue crack propagation prediction for marine structures based on a spectral method. Ocean Eng. 2018;163:706–17.
[17] Bishara M, Horst P, Madhusoodanan H, Brod M, Daum B, Rolffes R. A structural design concept for a multi-shell blended wing body with laminar flow control. Energies. 2018;11(2):1–21.
[18] Hansen PF, Winterstein SR. Fatigue damage in the side shells of ships. Mar Structures. 1995;8(6):631–55.
[19] Boardman B. Fatigue resistance of steels. In: ASM Handbook Volume 1: Properties and Selection: Irons, Steels, and High-Performance Alloys. ASM International; 1987. p. 77–81.
[20] Gaidai O, Storhaug G, Naess A, Ye R, Cheng Y, Xu X. Efficient fatigue assessment of ship structural details. Ships Offshore Struct. 2020;15(5):503–10.
[21] Šlězák T. Fatigue examination of HSLA steel with yield strength of 960 MPa and its welded joints under strain mode. Metals (Basel). 2020;10(2):1–14.
[22] Vaaar J, Kumari A, Frondelius T. Literature review of fatigue assessment methods in residual stressed state. Eng Fail Anal. 2020;110:104379.
[23] Venkatasudhahmar M, Dilipraja N, Mathiyalagan P, Subba SV, Sathyaseelan P, Logesh K. Finite element analysis of fatigue life of spot welded joint and the influence of sheet thickness and spot diameter. Int J Mech Mechatron Eng. 2016;14(06):76–82.
[24] Pastorvic D, Vukelic G, Bozic Z. Coil spring failure and fatigue analysis. Eng Fail Anal. 2019;99:310–8.
[25] Raymond L. Browell PE, Hançq A. Predicting fatigue life with ANSYS workbench: How to design products that meet their intended design life requirements. 2006 International ANSYS Conference. 2006 May 2-4.
[26] Lotsberg I, Sigurdsson G. Hot spot stress S-N curve for fatigue analysis of plated structures. J Offshore Mech Arctic Eng. 2006;128(4):330–6.
[27] Glen IF, Dinovitzer A, Paterson RB, Luznik L, Bayley C. Fatigue Resistant Design Guide For Ship Structures (No. SR-1386). Ship Structure Committee; 1999.
[28] He W, Liu J, Xie D. Numerical study on fatigue crack growth at a web-stifferner of ship structural details by an objected-oriented approach in conjunction with ABAQUS. Mar Struct. 2014;35:45–69.
[29] Okawa T, Sumi Y, Mohri M. Simulation-based fatigue crack management of ship structural details applied to longitudinal and transverse connections. Mar Struct. 2007;19(4):217–40.
[30] Dindinger P. HSLA 345X (SAE950X) Fatigue Test Report. F.D.E. Committee. Ontario, Canada; 2014.
[31] Li C, Dai W, Duan F, Zhang Y, He D. Fatigue life estimation of medium-carbon steel with different surface roughness. Appl Sci (Basel). 2017;7(4):1–11.
[32] Jacquelin B. FHAP. SAE 316L Fatigue Test Report. Ontario, Canada; 1983.
[33] Nachtigall AJ. SAE 304L Fatigue Test Report. Ontario, Canada: F.D.E. Committee, Ontario, Canada; 2012.
[34] Köksal NS, Kayapunar A, Çevik M. Fatigue analysis of a notched cantilever beam using ANSYS workbench. Proceedings of the Fourth International Conference on Mathematical and Computational Applications. 2013 Jun 11-13; Manisa, Turkey. 2013. p. 111–8.
[35] Fajri A, Prabowo AR, Surojo E, Imaduddin F, Sohn JM, Adiputra R. Validation and Verification of Fatigue Assessment using FE Analysis: A Study Case on the Notched Cantilever Beam. Procedia Struct Integr. 2021;33:11–8.
[36] Prabowo AR, Ridwan R, Muhayat N, Putranto T, Sohn JM. Tensile analysis and assessment of carbon and alloy steels using fe
approach as an idealization of material fractures under collision and grounding. Curved Layer Struct. 2020;7(1):188–98.

[37] Prabowo AR, Sohn JM. Nonlinear dynamic behaviors of outer shell and upper deck structures subjected to impact loading in maritime environment. Curved Layer Struct. 2019;6(1):146–60.

[38] Prabowo AR, Laksono FB, Sohn JM. Investigation of structural performance subjected to impact loading using finite element approach: case of ship-container collision. Curved Layer Struct. 2020;7(1):17–28.

[39] Prabowo AR, Sohn JM, Putranto T. Crashworthiness performance of stiffened bottom tank structure subjected to impact loading conditions: ship-rock interaction. Curved Layer Struct. 2019;6(1):245–58.