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Analysis of Ship Elongation Strength on OSV (Offshore Supply Vessel) Modification Due to the Addition of Moonpool Construction to Support National Maritime Energy Security

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Abstract— *Indonesia's offshore energy sector requires cost-efficient vessels capable of supporting drilling operations without compromising structural safety. The installation of a moonpool on an Offshore Supply Vessel (OSV) provides operational advantages for offshore exploration but may reduce the vessel's longitudinal strength due to discontinuities in the hull structure. The objective of this study is to evaluate the longitudinal structural performance of a modified OSV equipped with a moonpool using analytical calculations and the Finite Element Method (FEM). The structural analysis was conducted under four loading conditions: full-load sagging, operational sagging, full-load hogging, and operational hogging. Longitudinal bending moment, stress, and deflection were calculated and compared with the allowable limits specified by the Indonesian Classification Bureau. The highest analytical stress of 59.631 MPa occurred under full-load hogging, while the corresponding FEM result was 58.486 MPa. The lowest stress was obtained under operational sagging, with analytical and FEM values of 20.030 MPa and 19.645 MPa, respectively. All calculated stresses were significantly below the allowable limit, indicating that the modified structure satisfies the applicable classification requirements. These results demonstrate that the proposed moonpool modification is structurally feasible and can safely support offshore drilling operations while enhancing the utilization of existing Offshore Supply Vessels for Indonesia's offshore energy development.*

Keywords— drilling, finite element method, midship, moonpool, offshore supply vessel

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I. INTRODUCTION

As a country with abundant natural resources, Indonesia is expected to develop technologies that support the utilization of its offshore oil and gas resources. Indonesia's petroleum reserves reached 3.6 billion barrels (0.2% of the world's total), while natural gas reserves amounted to 104.25 trillion cubic feet (1.7% of the world's total) [1]. These resources play a strategic role in the national economy. However, as non-renewable resources, oil and gas reserves continue to decline. Data from the Central Statistics Agency (2024) indicate that Indonesia's crude oil production decreased by 59.26% between 1996 and 2022, primarily due to limited exploration activities and insufficient technological development for offshore resource exploitation [2].

To support offshore exploration, specialized marine infrastructure is required, particularly Offshore Supply Vessels (OSVs), which provide logistics and operational support for offshore platforms [3]. The demand for OSVs continues to increase by approximately 4.3% annually, with an estimated requirement of 235 units for 19 platform types between 2011 and 2015 [4]. Besides OSVs, drillships are widely employed because of their superior mobility and operational flexibility compared with semi-submersible platforms [5]. Drillships are commonly used for offshore drilling and subsea investigations, including Cone Penetration Tests (CPT), vibrocore sampling, and geotechnical surveys [6]. Moreover, the eastern Indonesian waters are characterized by deep-water environments with severe sea states comparable to those of the North Sea under extreme conditions, highlighting the need for robust floating structures capable of maintaining operational safety [7].

Despite the increasing demand for offshore exploration vessels, developing a new drillship requires substantial investment and long construction periods, making it economically challenging for many operators [8]. Consequently, modifying existing OSVs into multi-functional vessels represents a promising alternative [9]. However, incorporating a moonpool into the vessel's midship section introduces a large opening that disrupts hull continuity and may significantly reduce longitudinal structural strength, increase stress concentration, and affect global deformation under operational loading conditions [10]. Existing studies have primarily focused on the structural assessment of conventional OSVs or drillships separately, while investigations into moonpool-induced structural behavior in modified OSVs under different global loading conditions remain limited [11]. This knowledge gap creates uncertainty regarding the structural feasibility of converting conventional OSVs into offshore drilling support vessels [12]. Therefore, this study aims to conduct a comprehensive analysis of the

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ship's longitudinal strength following the modification of the moonpool structure, in order to ensure that structural integrity is maintained in accordance with operational safety standards. [13].

To address this issue, design innovation is proposed by modifying an existing OSV to perform additional drillship functions through the installation of a moonpool at the midship region [14]. Since the modification directly affects the primary hull girder, the structural performance of the vessel must be comprehensively evaluated under representative operational loading conditions and wave-induced longitudinal bending [15]. Using a hydrodynamic integration analysis approach that considers dynamic response and shear force distribution, this study aims to evaluate the significance of structural deformation resulting from the modification [16]. The novelty of this study lies in the integrated structural evaluation of a modified Offshore Supply Vessel equipped with a moonpool using both analytical calculations and the Finite Element Method (FEM) under four representative global loading conditions, namely full-load sagging, operational sagging, full-load hogging, and operational hogging [17]. Unlike previous studies that mainly examine conventional hull structures or local moonpool effects, this research provides a comprehensive assessment of global longitudinal strength, stress distribution, and hull deflection based on the allowable criteria specified by the Indonesian Classification Bureau (BKI) [18]. The findings offer a practical engineering basis for the structural feasibility of converting conventional OSVs into cost-effective offshore drilling support vessels, thereby contributing to more efficient utilization of existing fleets for Indonesia's offshore energy exploration [19]. Furthermore, the results of this analysis will validate whether the moonpool structure modifications comply with the allowable stress limits established by the Indonesian Classification Bureau to ensure the vessel's operational safety [20].

I. METHODS

A. Literature Studies

A comprehensive literature review was conducted to establish the theoretical basis for longitudinal strength analysis of Offshore Supply Vessels (OSVs) with moonpool modifications. Classification rules issued by the BKI [18], ABS [21], DNV [22] were reviewed to determine the procedures for calculating longitudinal bending moment, section modulus, allowable stress, and structural assessment criteria. The reviewed literature also provided the foundation for validating the analytical approach and finite element modeling adopted in this study.

B. Analysis of Existing Conditions

At this stage, existing conditions need to be considered to determine the urgency of this ship in supporting the country's mission in terms of energy security. This analysis includes areas of natural resource potential, environmental conditions, and the plan the ship will be used.

C. Ship Data

The principal particulars of the existing Offshore Supply Vessel (OSV) selected as the baseline model are summarized in Table 1. The vessel was subsequently modified by incorporating a moonpool at the midship region to accommodate offshore drilling operations while maintaining the original hull dimensions.

TABLE 1.
SHIP SPECIFICATIONS

General		
Ship Name	M/V Australis	
Ship Type	OSV + drilling (modified)	
Main dimension	Information	Unit
Length O.A.	58,30	m
Length W.L.	55.60	m
Length B.P.	52.00	m
Breadth	13,80	m
Depth	5,50	m
Draft (Max)	4,8	m
Displacement	2445	ton
Speed (Vs)	12	knots
Moonpool	3.06 x 2.80	m

D. Longitudinal Strength Analysis

In this stage, an analysis of the elongation strength of the ship is carried out. This analyst will later produce output, namely the stress due to the maximum bending moment that occurs on the ship and the deflection that occurs [23]. In this analysis, there are several loading scenarios, which are as follows:

TABLE 2.
CHARGING CONDITIONS

No.	Wave	Charge
1	Sagging	Full
		Operational
2	Hogging	Full
		Operational

In the calculation of the longitudinal strength of the nantin ship, the weight distribution and buoyancy along the ship will result in a load curve. From this load curve, the shear force and bending moment on the ship will be calculated. In addition, this study also calculates the moment of cross-sectional inertia of the ship at midship. Therefore, cross-section construction data on the ship at midship is needed.

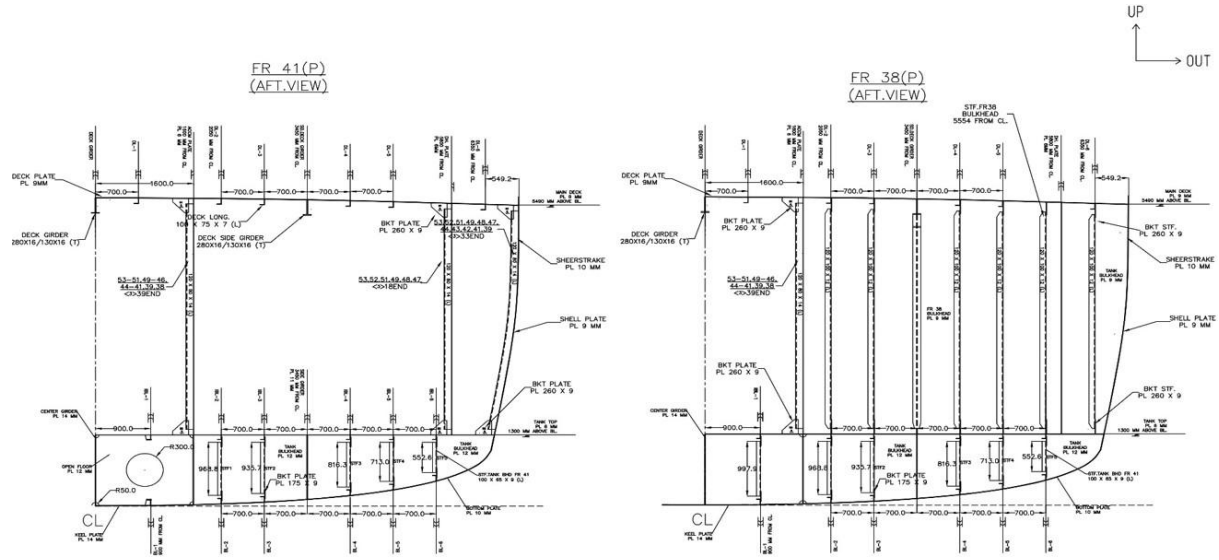


Figure 1. Construction Midship Ship

In the calculation of the elongated strength of the ship, the distribution of the load can be notated by the equation adopted from [13]:

$$load = W - B \quad (1)$$

After calculating the load distribution, the calculation is continued to obtain the value of the sheer force along the ship by means of a simple integration using the equation adopted from [13]. After obtaining the shear force value, the calculation is continued to obtain the maximum bending moment value in the area along the ship. The result of the shear force integration will be obtained again with the following equation.

$$P' = \frac{dS'}{dx} = \frac{d^2M}{dx^2} \quad (2)$$

This result will later be obtained the maximum bending moment value. The moment of inertia is calculated on the cross-section of the ship's midship, including the moment of vertical and horizontal inertia to obtain the value of the distance of the neutral axis of the axis. Calculations are taken on several profiles and longitudinal plates of the ship located in the midship area, then the calculation is carried out using tabulation based on the following equations to get the value of the [24]. Neutral axis distance to the baseline:

$$zna = \frac{\Sigma\, Zi.\, Ai}{\Sigma\, Ai} \quad (3)$$

As for getting the moment of inertia, use the following equation:

$$Iy = \Sigma\, Ioy + \Sigma\, zi^2.Ai - Zna^2.\Sigma Ai \quad (4)$$

In addition, to obtain the result of deflection along the ship due to bending *moment*. using the equation used to obtain the angle of the curve and used to make the deflection curve based on [13]:

$$\int_0^x dy(x) = L.\sum \frac{M}{EI}(x).\frac{1}{2}L\frac{M}{EI}(x) \quad (5)$$

This result will later be used to calculate the final value, namely the stress of the *bending moment* and the position of the neutral axis with the equation:

$$\sigma = \frac{BM}{w} = \frac{BM\, x\, y}{I} \quad (6)$$

The maximum stress yield on all loads will be compared to the values from the FEM based calculation.

E. 3D Modeling of Midship Ships

After knowing some of the maximum stress values obtained from the theoretical calculations, the temporary hypothesis is that there will be a maximum stress in the midship part of the ship due to a large bending moment in the area [25]. Therefore, modeling was carried out on the midship, namely the middle body parallel area of the ship and it is in this part that there is the construction of the moonpool hole. This modeling will be used for analysis inputs in FEM software to obtain the maximum stress value of numerical calculations [26].

F. Finite Element Method (FEM)

From the results of manual calculations, the maximum bending moment that occurs at each station on the ship is taken the highest value for the input of ship strength analysis using the finite element method

(FEM). From these two results, it will be analyzed and compared between manual calculations and FEM calculations to check the results of stress suitability and the results of the deflection of the ship after being loaded [27]. Meshing is done with the guidelines of the rules and software capabilities of the author. so that the analysis process continues to run smoothly and in accordance with the guidelines.

II. ANALYSIS AND DISCUSSION

A. Analysis of Existing Conditions

A.1. Natural Resources Potential Areas

Based on data taken and processed from the Ministry of Energy and Mineral Resources, one of the largest potential oil and condensate reserves in Indonesia is East Java Province, with a total of 588 proven drilling points, 225 possible to be found again, and 30 hope points.

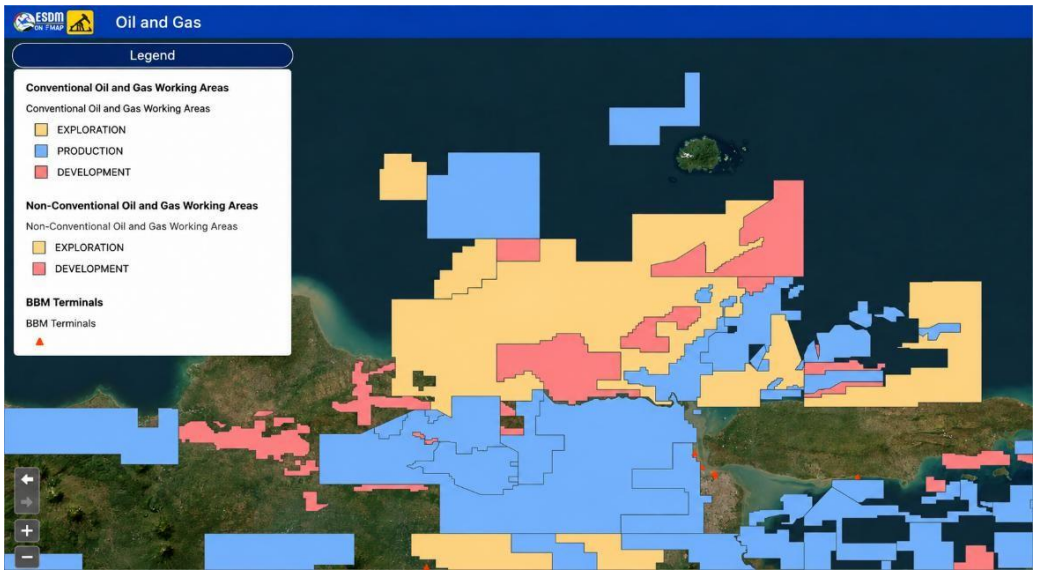


Figure 2. Oil and Gas Location Distribution

It can be seen in Figure 2 above It can be seen that the conventional oil and gas working area is marked with orange which is still in the exploration stage. This area will later be used as a ship operational area for offshore drilling and supply activities.

A.2. Determination of Drilling Position

The determination of this Drilling Position is used to be considered in ship modification with the addition of the Ship Operational Area drilling platform.



Figure 3. Ship Operational Area Plan

It can be seen from the figure above that it can be explained that the ship's operational area plan is in the northern sea area of Tuban-Bawean, for the maximum depth of ship operations, which is ±65 mLWS. Then for the plan to dock the ship is at Tanjung Perak Port.

A.3. Environmental Conditions

In planning the construction and operation of the ship, it is necessary to pay attention to the environmental conditions of the ship that will operate. The following are the environmental conditions in the ship's operational area plan.

TABLE 3.
ENVIRONMENTAL CONDITIONS OF OPERATIONAL AREAS

No.	Condition	Value	Unit
1	Max. Wave Height	1,65	m
2	Max. Sea Depth	390,00	m
3	Max. Wind Velocity	5,45	m/s
4	Max. Wave Velocity	0,20	m/s

A.4. Drilling System Specification

The drilling system specifications obtained from existing ship data are used to obtain the weight value of the system, because later in analyzing the strength of the ship also considers the weight of the components and the weight point.

TABLE 4.
DRILLING SYSTEM *SPECIFICATION*

	Specifications	Information
1	Core diameter	80mm diameter core (maximum 2.8 m length)
2	Max. torque.	9.600 Nm
3	Max speed.	280 rpm
4	Dynamic load max.	40 tons
5	Static load max.	50 tons
	Mud	4.000 L Capacity, Storage 8.000 L

B. Longitudinal Strength Analysis

B.1. Full Ship Weight Distribution

To calculate the weight of a full ship, it takes several tank weights, as well as the weight of components that are on the ship but can be moved (non-permanent). The calculation of tank weight is calculated using the help of maxsurf stability software, so that the calculation results are more accurate. Before calculating the weight of the full vessel. An empty vessel weight is required which is calculated using the method of the LR rules (lloyd Register) using the calculation of the curve M x a. The results of the calculation of empty vessels are added to the weight of the tank. The following is a distribution of the full weight of the ship.

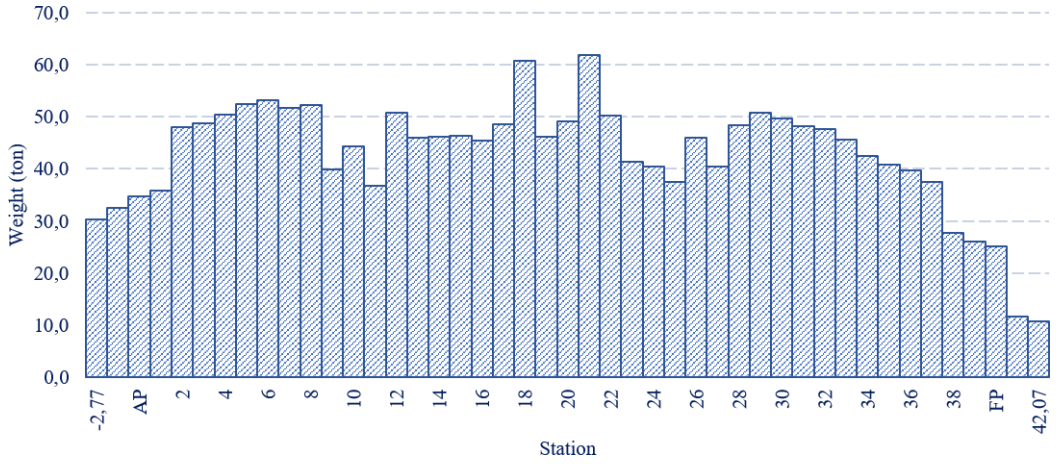


Figure 4. Distribution of Full Cargo Vessels

From the weight distribution, the full weight of the ship was obtained a value of 2250.30 tons spread along the ship.

B.2. Operational Load Weight Distribution

In the planning of ship weight during operational conditions, that is, when the ship drills, some weight will change according to these conditions. Some drilling components will be deposited at 1 point and the dynamic load added from the drilling machine will be added. With the loading scenario, the drill pipe is centered at the drilling center with the maximum depth condition (600 m) in the state of the pipe filled with mud at a diameter of 80 mm with calculations attached.

The dynamic load of the drilling machine is according to existing specifications, the quantity of fuel is adjusted to fuel consumption according to the longest shipping route and the volume of the ballast tank is adjusted so that the ship is in stable condition. After the above conditions have been taken into account, some component weights are calculated along with the weight of the previous empty ship.

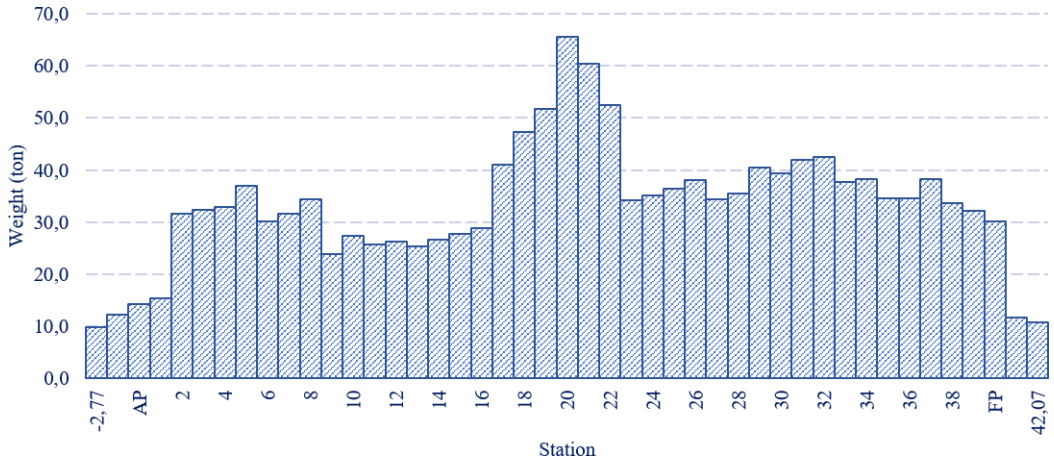


Figure 5. Heavy Distribution of Operational Cargo Vessels

From Figure 5 Above it can be seen that the weight distribution in the middle of the ship tends to be larger. This is due to the weight of the drilling components located in the center of the ship. From this distribution, the value of the ship weight was obtained as 1.905,65 tons.

B.3. Distribution of Ship Buoyancy

Buoyancy calculations show the distribution of buoyancy on the ship. In this analysis, 2 wave conditions were used, namely sagging and hogging. The wave height is adjusted to the wave height of the shipping area on the ship, which is 1,65 m. To make calculation easier, and accuracy, the help of maxsurf stability software is used. Which is reflected in the curve of area. At every 40 stations the ship has a displacement value input.

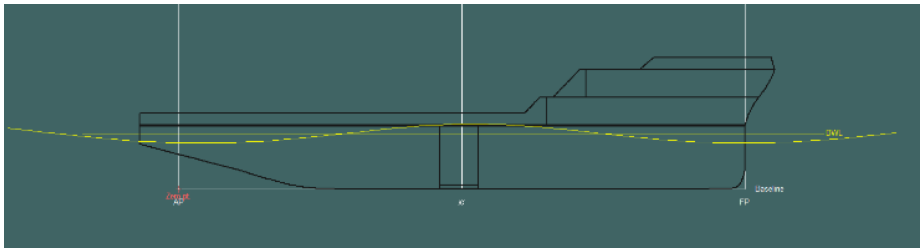


Figure 6. Condition Hogging

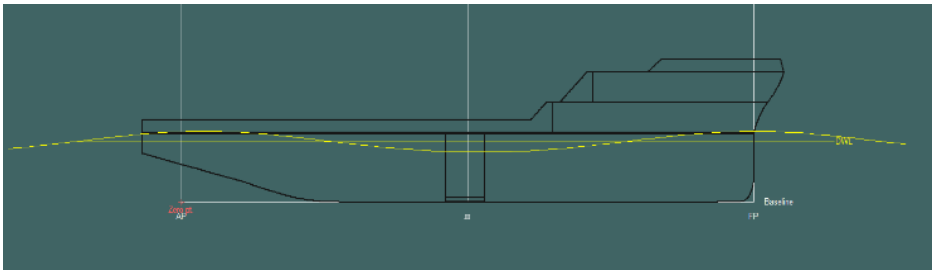


Figure 7. Condition Sagging

The displacement of the ship when exposed to the waves is calculated using the simpson method such as several things that must be considered [13] such as the size of the ship's displacement is equal to the weight of the ship calculated with a tolerance of ≤ 1% and the central point of buoyancy is located on one vertical plane with the weight point of the ship. The following is the distribution of ship buoyancy from 2 wave conditions:

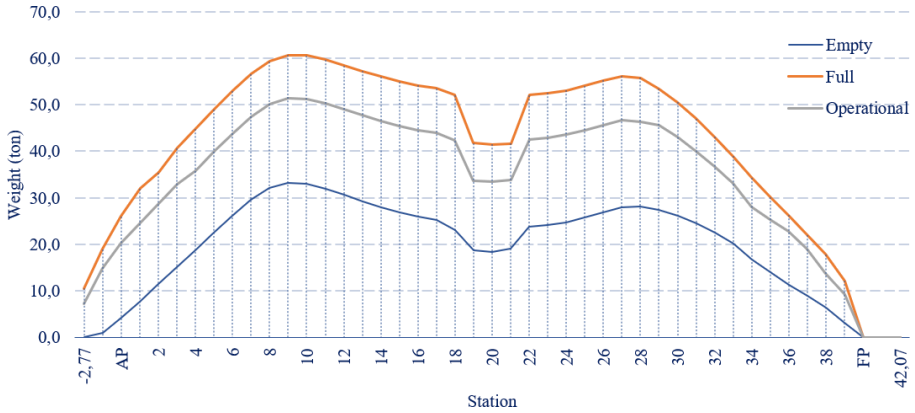


Figure 8. Distribution Buoyancy Conditions Sagging

During the sowing wave conditions, the histogram shows 2 area values in front and rear of the larger ship. This is due to the different wave height with the wave saving conditions. The midship area still shows the area with a smaller value. Because there is a hole in the hull of the ship.

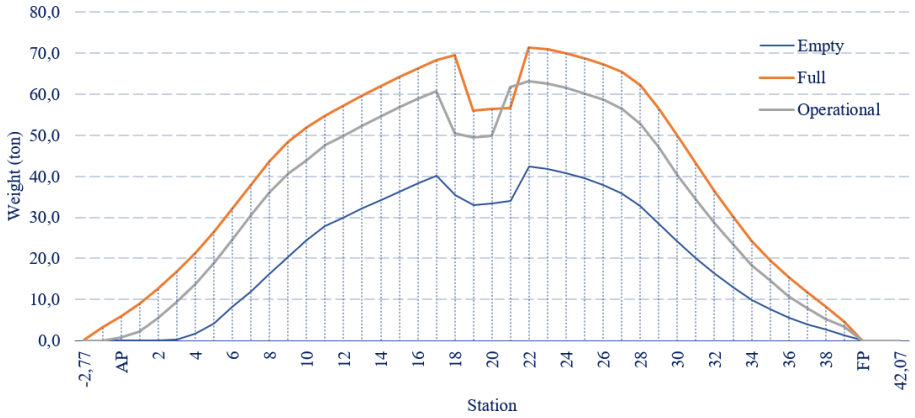


Figure 9. Distribution Buoyancy Conditions Hogging

During the hogging wave condition, a large area value is found in the midship of the ship. This is because there is 1 peak hogging wave in the midship of the ship, but the area in that part has a small value due to the hole.

B.4. Ship Load Distribution

After calculating the weight distribution and buoyancy, the values of the two distributions are used to calculate the load distribution with the equations discussed earlier. The load distribution can be seen in the following figure:

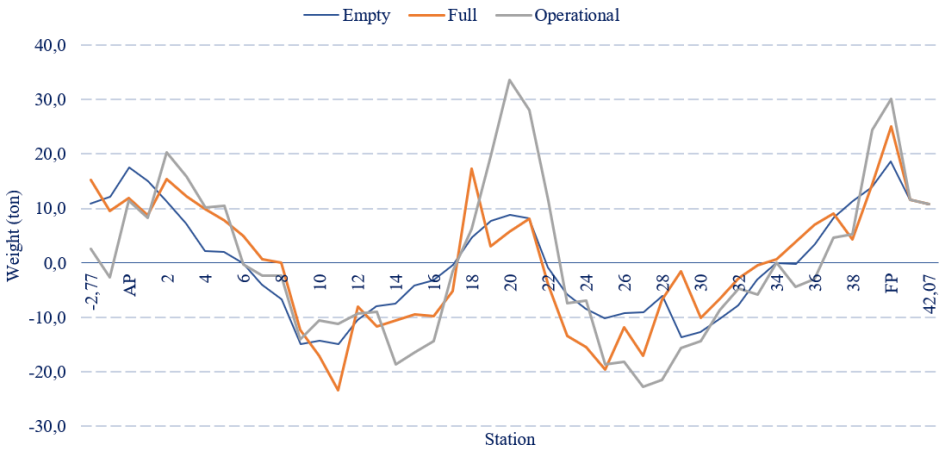


Figure 10. Condition Load Distribution Sagging

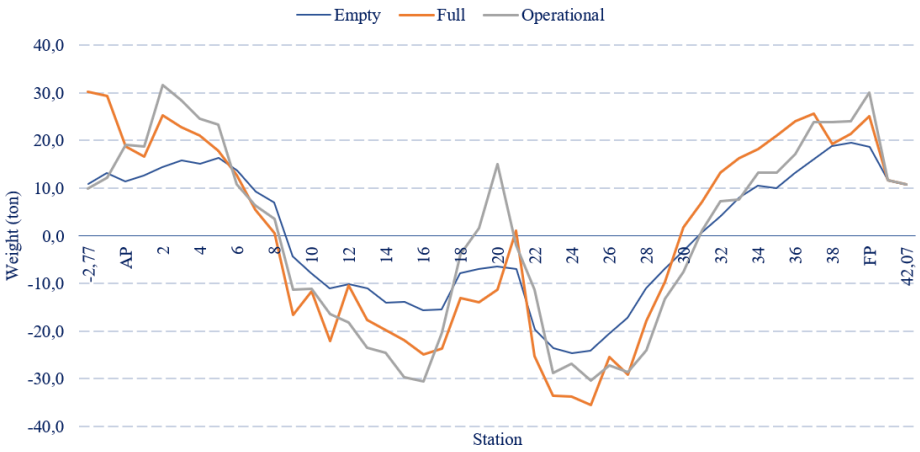


Figure 11. Condition Load Distribution Sagging

B.5. Calculation of sheer force and bending moment

Shear force or shear force on a ship occurs due to the presence of varying loads on each part along the ship. A simple integration of the load distribution curve (p') which is the accumulation of force increments starting from the end of the ship to the end limit of the other ship must be zero. Curve checks can be performed [13]. If p' is zero, S is the maximum or minimum point and the deflection point occurs in the curve M, if p' is the maximum point, the inflection point occurs in the S curve and If S is zero, M is the maximum or minimum. From these several conditions, the following is a graph of sheer force and bending moment in 2 charge conditions:

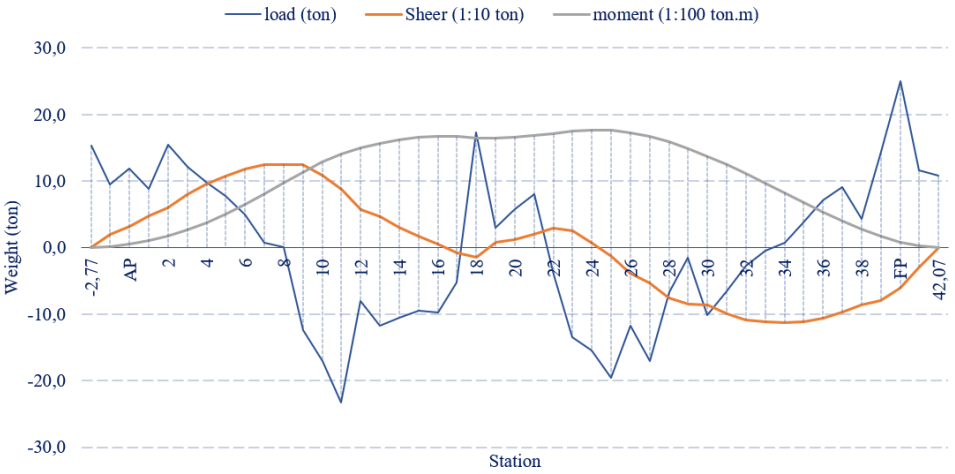


Figure 12. Full condition strength chart Sagging

In the condition of the ship full on the sagging waves, the ship's bending moment chart experiences a bending moment with a maximum value of 1765.72 ton.m.

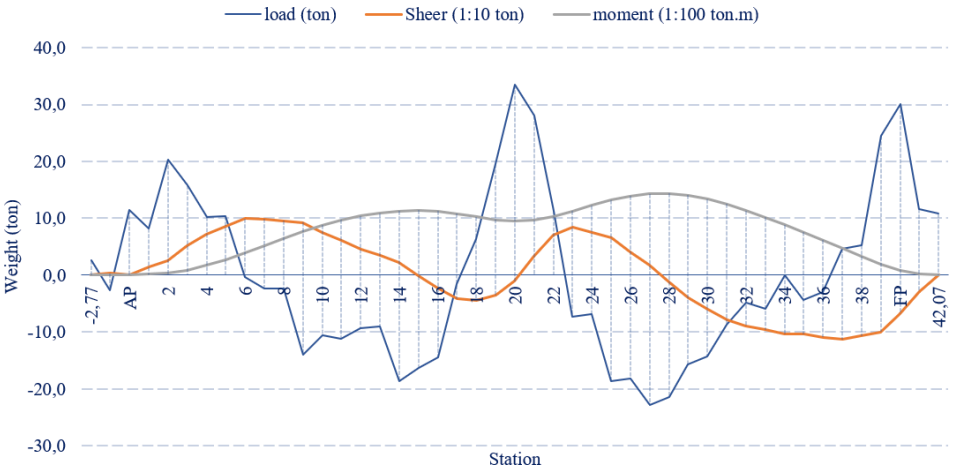


Figure 13. Graph of the strength of Operational conditions Sagging

In operational ship conditions on sagging waves, the ship's bending moment chart has a maximum value of 1.435,82 ton.m. The bending moment when the sagging condition shows a curve of 2 peaks caused by the weight and buoyancy values have a straight proportional value. The weight distribution is centered on the drilling point, as well as the same shape of the sagging wave contour, but the majority of the maximum value occurs in the midship area.

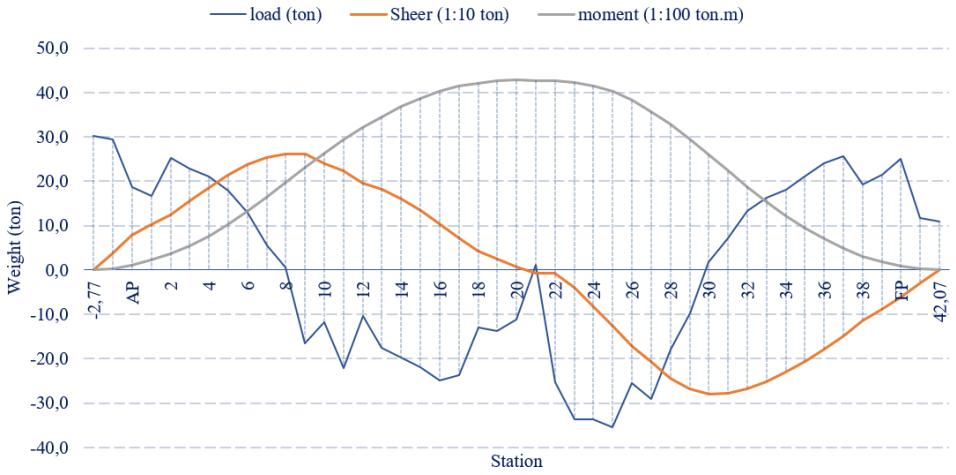


Figure 14. Full Condition Strength Chart Hogging

In the condition of a full ship on hogging waves, the ship's bending moment chart has a maximum value of 4.274,63 ton.m.

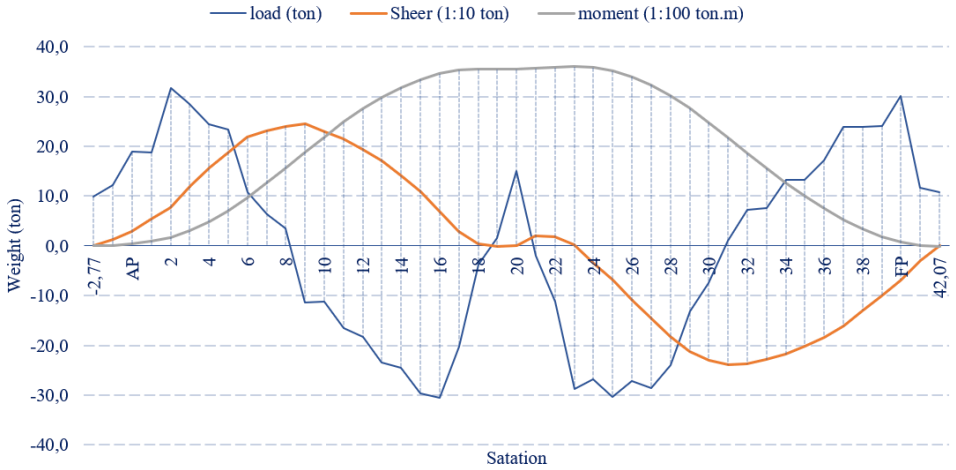


Figure 15. Graph of the strength of Operational conditions Hogging

Meanwhile, in operational ship conditions on hogging waves, bending moment has a maximum value of 3.602,74 ton.m. load distribution when hogging conditions in the midship tend to be smaller because the waves have a buoyancy value centered on that part. The amount of load will decrease drastically due to the difference between weight and small buoyancy. So that the maximum bending moment occurs in that part. This value will later be used to obtain the maximum stress that occurs at each pre-planned loading condition.

B.6. Calculation of The Moment of Inertia

From the equation that has been discussed earlier, the calculation of the moment of inertia in the ship's midship section, several variables are known as follows:

ΣA = 352920 mm²

Σz.A = 882497585.5 mm³

$$zna = \frac{882497585.5}{352920}$$

$$zna = 2500.55\ mm$$

Vertically known:

ΣyA = 1172668326 mm³

So:

$$yna = \frac{1172668326}{352920}$$

$$yna = 3322.75\ mm$$

After obtaining the vertical distance of the point of weight to the baseline, the next step is to calculate the cross-sectional modulus of the ship's midship. The modulus of midship cross-section according to BKI 2025 vol. II sec 5 C.2.1 [28] must not be less than:

$$Wmin = k\ .\ c_0\ .L^2.B\ .(Cb + 0.7).10^{-6}$$

Where:

k = material factor (A32) which is 0.74

c0 = wave coefficient

$$c0 = \left(\frac{L}{25} + 4.1\right).Crw \qquad \text{for L} < 90\ \text{m}$$

Crw = Sailing Range Coefficient

= 0.90

$$c0 = \left(\frac{52}{25} + 4.1\right).0.9 = 5.56$$

So:

$$Wmin = 0.209\ m^3$$

From the BKI guidelines vol. II 2025 sec. 5.2.3 the moment of inertia of the horizontal midship midship must not be less than:

$$Iy = 3.10^{-2}.W.\frac{L}{k} \qquad [m4]$$

So:

$$Iy = 3.10^{-2}.0.209.\frac{52}{0.74}$$

$$Iy = 0.437\ m^4$$

Thus, the moment of inertia value Horizontal and Vertical that is

Known:

ΣI_{0y} = 1673363175333.3 mm^4
Σz^2A = 3797867648495.8 mm^3
ΣI_{0z} = 76343321230.54 mm^4
Σy^2A = 5849573158900 mm^3

So:

Moment of inertia with respect to the horizontal axis:

$$Iy = 1.758\ x\ 10^{12}\ mm^4$$

$$Iy = 1.758\ m^4 \qquad \text{Meet}$$

Moment of inertia with respect to the vertical axis:

$$Iv = 2.029\ x\ 10^{12}\ mm^4$$

$$Iv = 2.029\ m^4 \qquad \text{Meet}$$

The Modulus of Cross-sectional Ship *midships* can be calculated in accordance with the guidelines of BKI 2025 vol. II sec. 5.4.1 as follows:

$$Wbott = \frac{Iy}{e} \qquad (7)$$

Where:

e_B = the neutral axis distance to Base Line
e_B = 2500.55 mm

so:

Cross-sectional modulus to bottom :

$$Wbott = \frac{1.758}{2.5}$$

$$Wbott = 0.703\ m^3 \qquad \text{Meet}$$

B.7. Stress Calculation

After calculating the moment of inertia, the last step is to determine the maximum stress value of the ship due to the bending moment that occurs using the formula that has been described earlier, however, the stress value must not exceed the permissible stress required by BKI. With the formula that has been explained earlier, the value of the permit stress is as follows:

Known:

x = length Watchdog Middle Body
x/L= 0.17

$$Cs = 0{,}5 + \frac{5}{3}.\frac{X}{L} \qquad \text{to } 0 \le X/L < 0.30$$

$$Cs = 0{,}5 + \frac{5}{3}.0.17 = 0.775$$

k = material factor (A32) which is 0.74

so:

$$\sigma p0 = 18.5\frac{\sqrt{52}}{0.74}$$

$$\sigma p0 = 178.8\ Mpa$$

$$\sigma p = 0.775\ x\ 178.8$$

$$\sigma p = 138.6\ Mpa$$

So the construction permit stress is 138.6 Mpa

From the equation discussed earlier, the maximum stress result will later be used as a comparison variable with the FEM calculation. The stress results are as follows:

TABLE 5.
RESULTS OF SHIP TENSION THEORY CALCULATIONS

No.	Condition	Max Moment (ton.m)	Max Stress (Mpa)
1	Sagging - Full	1765.721	24.632
2	Sagging - Operational	1435.826	20.030
3	Hogging - Full	4274.635	59.631
4	Hogging - operational	3602.741	50.258

The analytical assessment revealed that the full-load hogging condition generated the greatest longitudinal stress, with a maximum value of 59.631 MPa, whereas the operational sagging condition produced the lowest stress of 20.030 MPa. This behavior indicates that hogging imposes a more critical longitudinal bending effect on the hull girder because the wave crest is positioned around the midship region, where the moonpool opening interrupts the continuity of the hull structure and reduces its effective sectional stiffness. As a result, higher stress concentrations develop in the vicinity of the modified midship structure.

The numerical analysis performed using the Finite Element Method (FEM) produced results that closely matched the analytical calculations, with discrepancies remaining below 2%. This level of agreement confirms the validity of the analytical approach for evaluating the global longitudinal response of the modified vessel, while the FEM analysis provides a more comprehensive visualization of local stress distribution and concentration around the moonpool opening.

The observed structural response is consistent with findings reported in previous investigations of offshore vessel modifications, which have shown that large hull or deck openings can reduce the longitudinal stiffness of the hull girder but remain structurally acceptable when supported by an appropriate reinforcement design. Accordingly, the present study demonstrates that the proposed moonpool modification complies with the allowable stress limits established by the Indonesian Classification Bureau (BKI), indicating that the modified OSV is capable of maintaining adequate longitudinal structural integrity for offshore drilling operations.

B.8. Deflection Calculation

From the equation that has been discussed earlier, niladeflection is as follows:

TABLE 6.
DEFLECTION CALCULATION RESULTS

No.	Condition	Max deflection (mm)
1	Sagging - Full	15.287
2	Sagging - Operational	11.019
3	Hogging - Full	34.519
4	Hogging - operational	29.784

C. Ship Midship Modeling

The selection of modeling on the midship refers to the guidelines of the ABS rules. In addition, it is in this area that is suspected to have the greatest bending moment. Because the area is the place where drilling machines and equipment are located and operate. The following is a 3D image of the ship's midship:



Figure 16. 3D model Midship Ship



Figure 17. 3D model Moonpool Area

D. Finite Element Method

In the FEM analysis, meshing with a size of 0.11 m was used. This refers to several considerations such as the rules of the rules and the capacity of the device used by the author.

TABLE 7.
MESH SIZE REFERENCE

No.	Source	Mesh Size
1	BKI 2025	Non-explicit (follows ABS/IACS)
2	DNV 2021	0.4 – 1.0 m
3	IACS CSR	0.3 – 0.6 m
4	ABS	0.3 – 1.0 m

Source: Rules BKI dan ABS [29], 2025

The following is a graph of the mesh convergence test.

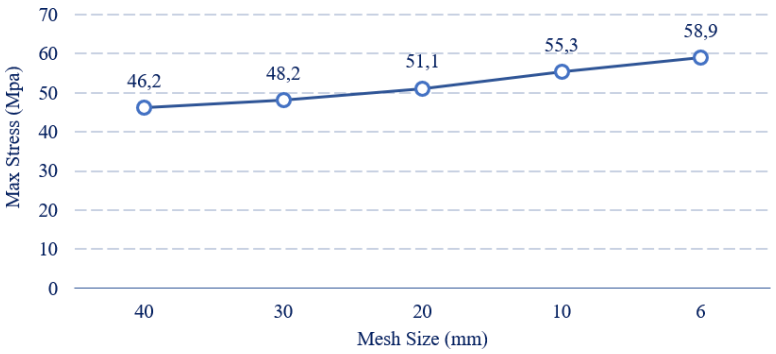


Figure 18. Mesh Convergence Test

In addition, boundary conditions are carried out on the model to provide a loading scenario to produce a stress that is in accordance with the theoretical calculations. The following is an explanation of the boundary condition :

- The PMB (parrel middle body) model is 8.58 m long starting from stations 18-25 at the input on the mechanical APDL
- One of the back ends (A) is clamped (fix support). All Degrees of Freedom (DOF) are locked, translation x, y, z = 0; rotation x, y, z = 0
- The other end (B) is left free to be given a force (F) with a remote force input with the aim of concentrating all the force at 1 point in the Middle with coordinates (x = 6900 mm), y is the position of the neutral axis (y = 2500 mm), z (0 mm) with the direction (↑) for sagging and (↓) hogging and calm water.
- The force input (F) is obtained from the maximum moment value at each load condition which is converted to force by dividing by the length of the PMB (parrarel middle body) of the ship, which is 8.58 m according to the principle of moment equation.

The results of the FEM analysis are as follows:

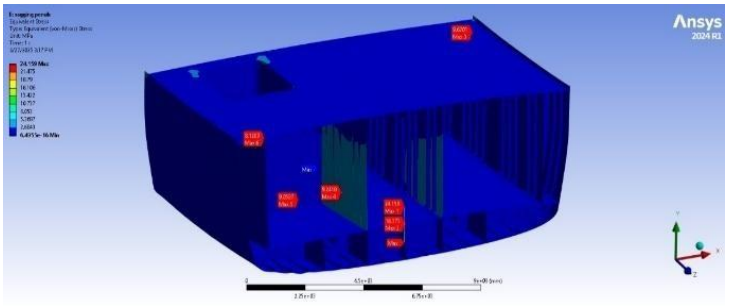


Figure 19. Result stress condition Sagging-full

The maximum stress value in the full load sagging condition was 24,159 Mpa and the minimum stress value was 6,495 Mpa.

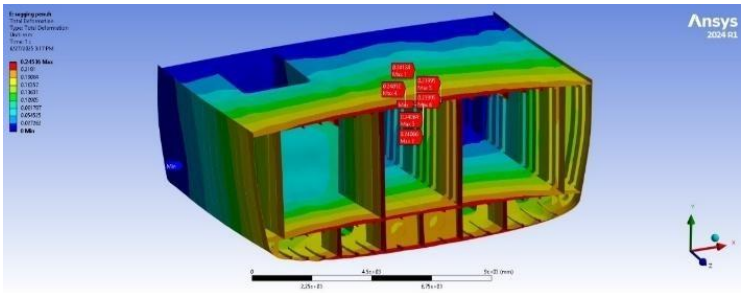


Figure 20. Results of Deflection conditions Sagging-full

The maximum deflection value in the full load sagging condition was obtained on the PMB model of 0.245 mm and the minimum deflection value was 0.027 mm. As for the operational conditions, they are as follows:

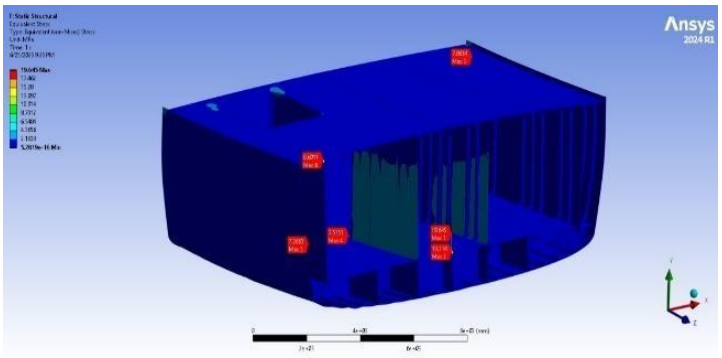


Figure 21. Result stress condition Sagging-Operational

The maximum stress value in the sagging condition was obtained with an operational load of 19,645 Mpa and a minimum stress value of 5,281 Mpa. The maximum deflection value in the sagging condition with the operational load in the PMB model was 0.199 mm and the minimum deflection value was 0.022 mm.

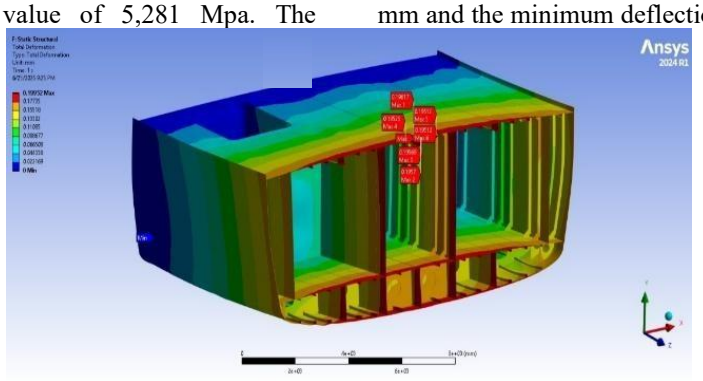


Figure 22. Results of Deflection conditions Sagging-Operational

As for the condition of hogging waves, it can be seen in the following figure:

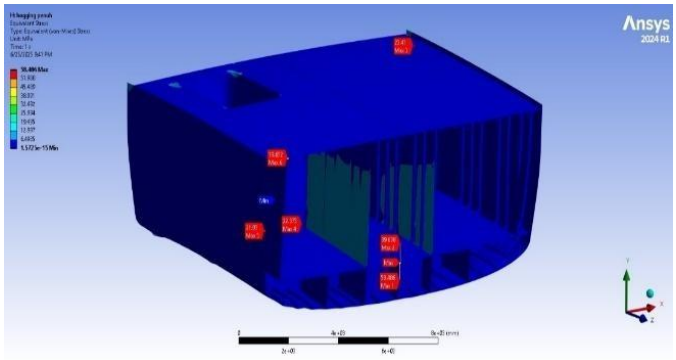


Figure 23. Result stress condition Hogging-Full

The maximum stress value in the hogging condition with a full load was 58,486 Mpa and the minimum stress value was 15,725 Mpa.

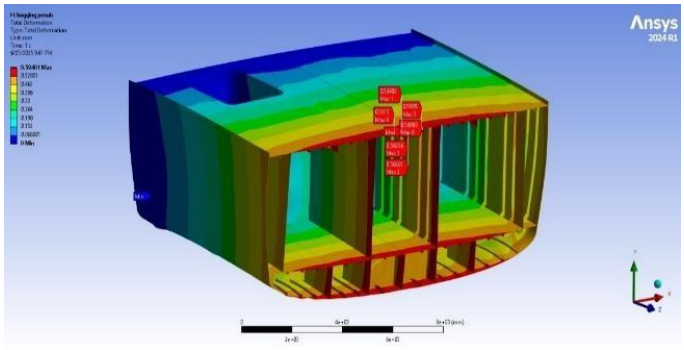


Figure 24. Results of Deflection conditions Hogging-full

The maximum deflection value in the hogging condition with full load on the PMB model was 0.594 mm and the minimum deflection value was 0.066 mm. In addition, the maximum stress value in hogging conditions was obtained with an operational load of 49,243 Mpa and a minimum stress value of 13,253 Mpa.

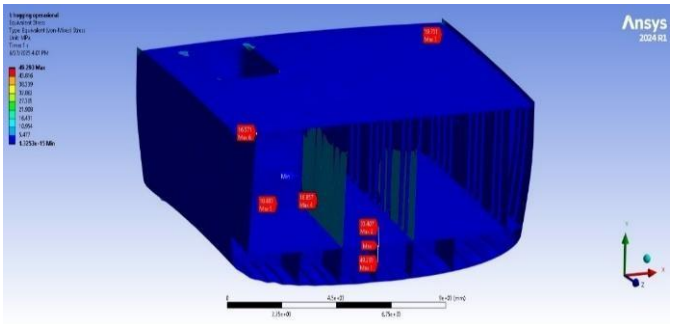


Figure 25. Result Stress condition Hogging-Operational

The maximum deflection value in hogging conditions with an operational load on the PMB model is 0.500 mm and a minimum deflection value of 0.055 mm.

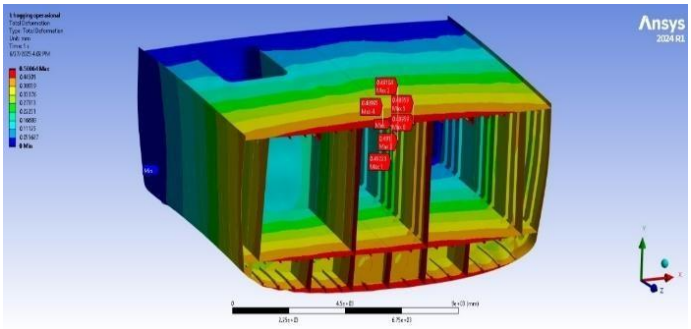


Figure 26. Results of Deflection conditions Hogging-Operational

III. CONCLUSION

From the results of the analysis that has been carried out, several conclusions are obtained as follows:

1. In all loading scenarios, the analysis of the elongated strength of the ship produced the smallest bending moment in the condition of the Operational load sagging wave with a moment value of 1435,826 ton.m at station 20 with a maximum stress value of 20,030 Mpa and a deflection of 11,019 mm, as well as the largest bending moment in the hogging wave condition The full load is loaded with a bending moment value of 4274,635 ton.m at station 20 with a maximum stress value of 59,631 Mpa and a deflection of 34,519 mm.
2. The results of the analysis using the element method showed the smallest stress in the condition of the operational load sagging wave with a maximum stress value of 19,645 Mpa and a deflection at the PMB of 0.199 mm, while the largest stress occurred in the condition of a full load hogging wave with a maximum stress value of 58,486 Mpa and a deflection at the PMB of 0.594 mm.
3. The results of the analysis showed that the calculation of the elongated strength of the ship due to the addition of the moonpool construction had met the criteria in accordance with the BKI permit stress of 138.6 Mpa. From these results, it shows that with the modification of the OSV ship as a drilling vessel, it still meets in terms of longitudinal strength.

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