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RETROFIT SOLUTIONS TO ACHIEVE 55% GHG REDUCTION BY 2030

Technical report with drawings

WP 4 – Technology Demonstration
Task 4.6 – Definition and design of standardized reinforcement solutions for rigid sails WAPS systems
D4.7 – Technical report with drawings
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Acronyms

Acronym	Description
AoA	Angle of attack of wind to a foil
BV	Bureau Veritas
CII	Carbon Intensity Indicator
CCSS	Classification Societies
DNV	Det Norske Veritas
EEDI	Energy Efficiency Design Index
EEXI	Energy Efficiency Existing Ship Index
EDW	Equivalent Design Wave
FEM	Finite Element Modelling
GA	General Arrangement.
GHG	Green-House Gas
IACS	International Association of Classification Societies
IMO	International Maritime Organization
LNG	Liquefied Natural Gas
LPG	Liquefied Petroleum Gas
PU	Public
RES	Renewable Energy Sources
WAPS	Wind-assisted Propulsion Systems

Executive Summary

Maritime transport plays a pivotal role in the European Union's economy, being one of the most energy-efficient way of transport. Despite this, maritime transport also poses a significant challenge, due to its substantial contribution to Green-House Gas (GHG) emissions, which may have an important impact on climate change. Projections indicate a potential increase up to 130% in maritime emissions by 2050 compared to 2008 levels, posing a challenging obstacle to the goals outlined in the Paris Agreement (2015).

In response to this challenge, innovative solutions such as Wind-assisted Propulsion Systems (WAPS) have emerged as promising way for reducing emissions and enhancing energy efficiency in maritime transport. Among these solutions, the eSAIL® system developed by B4B stands out for its potential to revolutionize vessel propulsion using wind power.

The goal of this report is to examine and define the diverse requirements for the structural integration of rigid sail systems on various types of merchant vessels. This work aims to address the complexities associated with the reinforcement needs for WAPS, providing a comprehensive framework that supports agile structural integration across different ship typologies. By thoroughly analysing the variability in vessel designs and the associated regulatory landscapes, this report seeks to outline practical approaches that enable ship-owners to streamline the integration process, thereby reducing costs and lead times across shipyards, class societies, and naval engineering firms.

This report delves into the criteria and steps necessary to define the foundation of a WAPS, with a particular focus on identifying commonalities and variances in reinforcement requirements based on ship type. It addresses the challenges of standardization by exploring the impact of ship-specific parameters on structural loads and foundation design. The document also proposes a set of minimum common requirements and typical reinforcement morphologies that can serve as adaptable solutions tailored to the needs of individual vessels, rather than imposing rigid, standardized designs that may not be suitable for all scenarios.

Ultimately, the aim of this work is to equip stakeholders with the knowledge and tools needed to achieve efficient and reliable structural integration of WAPS. By prioritizing adaptable design practices and emphasizing the importance of customization based on real-world operational conditions, the report provides a roadmap for optimizing both the economic and environmental performance of ships incorporating rigid sail systems. This approach not only enhances the viability of WAPS technology across a broader range of vessel types but also supports the industry's ongoing efforts to reduce emissions and improve the overall sustainability of maritime operations.

This approach not only advances efforts to mitigate GHG emissions in maritime transport, but also propels the transition towards a more sustainable and resilient maritime industry in the face of climate change.

1 Introduction

Maritime transport plays an essential role in the EU economy and is one of the most energy-efficient modes of transport. Despite this, it contributes significantly to the rising levels of greenhouse gas (GHG) emissions. Approximately 2.9% of all emissions worldwide produced by human activity came from shipping in 2018, accounting for 1,076 million tons of CO_2.

According to projections, by 2050 these emissions might rise by as much as 130% with respect to the 2008 levels. The goals of the Paris Agreement in 2015 (a global framework to prevent disastrous climate change by limiting global warming to far below 2°C and pursuing efforts to restrict it to 1.5°C) may be undermined if the impact of shipping activities on climate change increases as predicted. At the EU level, maritime transport accounts for 3 to 4% of total CO_2 emissions, or more than 124 million tons of CO_2 in 2021 [1].

To drastically reduce Green-House Gas (GHG) emissions from international shipping, effective global measures are needed. In July 2023, the International Maritime Organisation (IMO) committed new targets for GHG emission reduction, to be developed and adopted in 2025. The EU action to make sure that maritime transport plays its part in achieving climate neutrality in Europe by 2050 is an essential step in incentivising the necessary reductions.

The International Maritime Organization (IMO) has established a set of specific mandatory measures, such as the CII (Carbon Intensity Indicator), EEXI (Energy Efficiency eXisting ship Index), and EEDI (Energy Efficiency Design Index) to monitor and reduce emissions from international shipping. In July 2023, IMO committed new targets for GHG emission reductions to be developed and adopted in 2025. The IMO strategy aims to reduce of 20% of the GHG emissions from international shipping by 2030, of 70% by 2040 and NET-ZERO by 2050, compared to 2008 [2].

A detailed road map to decarbonize the fleet is becoming more and more important, as emission regulations harden. Switching to renewable fuels will not be sufficient to meet these restrictions. Only when combined with other technologies, such as WAPS, the GHG emission performance will be reduced enough, and the business asset will remain attractive. In order to comply with these regulations, ship-owners and ship operators do not necessarily have to make the full investment all in one go, rather they have the possibility to split it up according to the regulatory compliance targets. Furthermore, at the end of a ship's commercial life, the wind propulsion systems can also be transferred to other vessels.

The objective of this deliverable is to examine and define the structural integration requirements for rigid sail systems across various types of merchant vessels. This work aims to provide a flexible framework that accommodates the diverse design and regulatory conditions found in the maritime industry. By doing so, it seeks to streamline the integration process, enabling ship-owners to reduce associated costs and lead times in collaboration with shipyards, class societies, and naval engineering firms.

The report outlines the key criteria and procedural steps for defining the foundation of a WAPS, considering the eSAIL® developed by bound4blue as a case study. It highlights the challenges of standardization due to the wide variability in ship designs and operational contexts. Instead of pursuing rigid standardized solutions, the work concludes with the development of minimal common requirements and adaptable reinforcement morphologies, offering a practical path forward for integrating WAPS in a variety of vessel types.

It is estimated that wind energy reduces fuel consumption by 10-30%, depending on whether the weather conditions are favourable during the journey. More interestingly for a sector that is moving towards decarbonisation, WAPS may reduce CO_2 emissions by a percentage between 10% and 60%, depending on the technology used [3].

There are currently four types of WAPS whose level of maturity and development is different:

1. Kites: they are systems similar to those used for kitesurfing, adapted to be installed on two vessels. They are connected to the bow and operate at high altitude, performing an 8-shaped trajectory, capturing the wind, and pulling the vessel itself. The operation requires a launch and recovery system for the kite, as well as a complex autonomous operation of the kite in flight.

2. Flettner rotors: they are large cylindrical mechanical sails that rotate to create a propulsive force for the boats. This technology was developed by Anton Flettner during the 1920s. They exploit the Magnus Effect, which is an aerodynamic phenomenon for which a spinning object in an air flow develops a pressure difference on one side with respect to the other one, as a result of the local velocity field induced by the body rotation. Such a situation generates the aerodynamic forces of Lift (L) and Drag (D). In the case of Flettner rotors, the spinning object is a cylinder located vertically on the vessel deck. The aerodynamic characteristics are dependent on the spin speed of the rotor, so a rotation shall be constantly maintained.

3. Flexible and rigid sails: they are airplane-like wings, with a very similar working principle. They have an aerodynamic cross-section (aerofoil) that, when exposed to an airflow, produces lift, because of the pressure difference between the upper and lower sides. The amount of lift is manly related to the incidence of the aerofoil with respect to the wind direction, commonly indicated as the Angle of Attack (AoA). Their operation is based on ensuring that the aerofoil is correctly oriented with respect to the wind (appropriate AoA), with no additional requirement for power. It is a **passive system**, which typically does not offer a significant aerodynamic performance. In fact, the lift coefficient is generally around 1.5, resulting in larger and heavier systems for equivalent savings. Due to the large size and weight, the number of sails per vessel is limited, since they 1) require a large deck space, 2) impose limitations of visibility for the crew and of the on-board cameras, 3) require strong/heavy hull reinforcement at the installation points and 4) have a large impact on the cargo capacity. Finally, another challenge arises from the fact that most sails must be lowered and stored when not in use, due to their side mounting, especially when loading or unloading cargo.

4. Suction sails: they are based on the use of **boundary layer active control systems** to avoid the detachment of the air flow from the surface. The airflow around a thick aerofoil or at a large AoA typically detaches, generating a separation area, typically turbulent, which results in a large drag force. If a suction area is located at the detachment point and a small amount of boundary layer is aspirated, the flow remains attached to the aerofoil. This results in a significant increase in the lift coefficient, well above the values obtained from a passive wing-sail, reaching values even higher than 6. In addition, a movable flap is installed, to generate asymmetry, increase lift, and cover the suction area not in use. The operation is somewhat equivalent to adjusting the AoA to the wind direction and setting the appropriate aspiration to the wind conditions. The aspiration requires a low power consumption. Finally, another advantage compared to passive wing-sails is that the system can be tilted instead of folded.

In the context of this deliverable, the focus will be on the suction sails developed by B4B, referring to as **eSAIL®**, the commercial name given by the company. Therefore, some of the strategies and procedures may not be 100% suitable for other WAPS technologies and will have to be adapted.

2 Preliminary analysis

Report technical content.

The purpose of sails is to catch the energy from the wind and transmit a forward thrust into the vessel to reduce fuel consumption. Therefore, WAPS induce local loads into the ship structure, which must be reinforced accordingly. Due to the prototype nature of merchant vessels, it is not possible to fully standardize this reinforcement, which shall always be tailored. Some common aspects have been found in the different type of merchant vessels studied, which are:

- Bulk Carriers
- General Cargo – short sea shipping Coasters
- Ro-ro (roll on, roll off)
- Liquid Bulk Tankers (chemical, crude oil, oil products)
- Gas Tankers (LPG or LNG)

Tankers and Ro-Ro vessels have commonly free space on exposed decks. Tankers have higher restrictions to prevent explosions and solutions must be found. Ro-ro vessels have commonly the navigation bridge on the forward end of the ship, which provides some interesting advantages for the integration.

Bulk carriers and general cargo ships require access to hold hatch covers for cargo operations. This may create restrictions to the positioning of WAPS on board, in order to avoid interferences with ship loading operations.

2.1 Basic considerations

The fundamental considerations and critical engineering parameters necessary for the structural integration of rigid sail systems on merchant vessels include:

i. Analysis methodology
   - Classification Society (CCSS) of the Vessel
   - Combination Loads defined by the CCSS.
   - Inertial loads induced by Accelerations due to ship motions (ship specific)
   - Wind loads induced by the WAPS.
   - Global ship structure and local scantling (ship specific)

ii. Designs optimized and validated by FEM tools

iii. Interface requirements.

iv. Preliminary analysis to classify different ship types to define operational restrictions to minimize negative effects on ship loading operations.

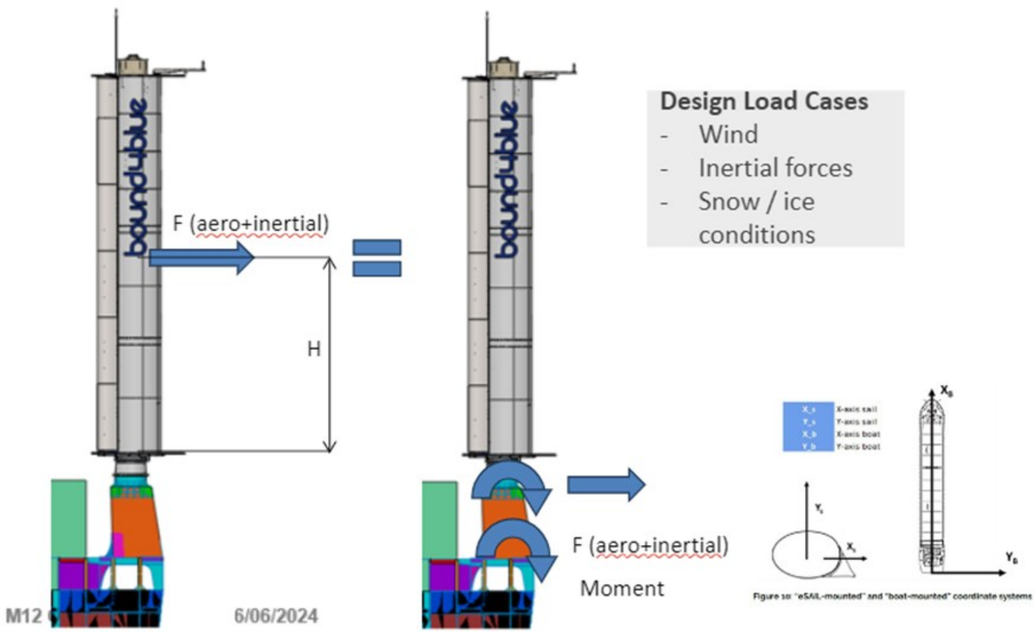
2.2 Analysis methodology

Each CCSS has at this moment different criteria on the structural approach to WAPS loads and structural assessment (Figure 1). In general, all of them require both a strength and fatigue analysis. Both must be considered in the design of the ship foundation.

The following procedure is suggested:

1. Determine tentative positions for WAPS on board.
2. Use CCSS rules for calculation of inertial loads based on the ship specific data, such as Length, Draught, metacentric height and position on board.
3. Calculate wind loads.
4. Calculate ice, snow and green sea loads, if applicable.
5. Determine the worst-case load combination and use it to dimension the structure. To do so:
    a. Typically, wind and waves are superimposed in three different directions:
        i. Head Sea with maximum forward acceleration.
        ii. Beam Sea with maximum transverse acceleration.
        iii. Oblique Sea with maximum oblique acceleration.
    b. Combination of other loads such as green sea, ice and snow or others are subject to each project and CCSS criteria.
6. Dimension loads at the interface WAPS-/ ship structure to give them as an input for FEM calculation.
7. Based on loads, make a 3D (CAD) model of preliminary structure.
8. Optimization process and final model definition.
9. Approval from owner and main stakeholders.
10. Submission for approval at CCSS.

With the load magnitude and direction, a preliminary structure must be aligned with ship primary structure.



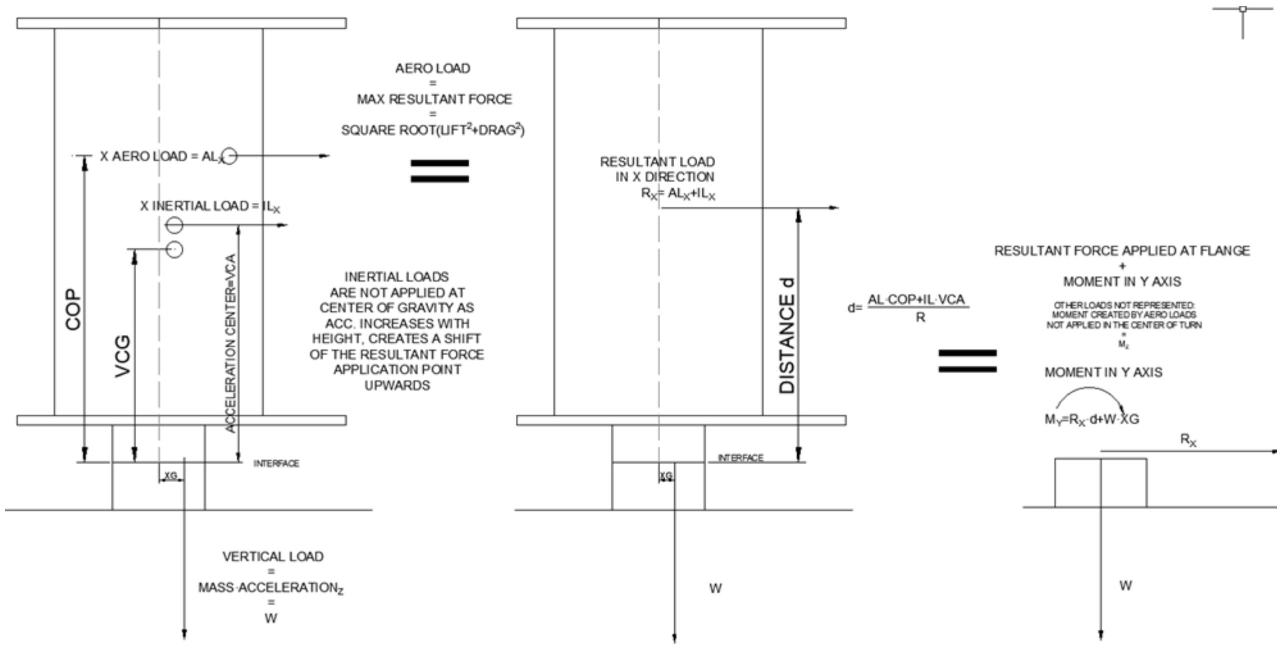


Figure 1: Loads application and simplification.[1]

Aerodynamic and inertial loads are applied at varying heights, generating distinct moments. Consequently, the interface provides the foundation designer with a combined total force and an equivalent moment as input. The new foundation, retrofitted to the ship, must withstand these forces and moments. While some aspects are standardized, the existing structure must be reinforced to effectively absorb and distribute these loads into the primary structure without creating stress concentrations ("hot spots"). However, this reinforcement cannot be standardized, as deck layouts vary for each ship.

For better compatibility, the sign criteria of forces and moments is taken the same as given by CCSS (Figure 2).

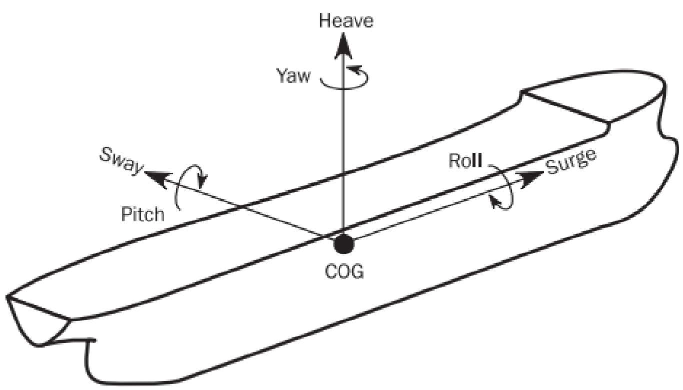


Figure 2: Definition of sign criteria for ship motions and forces [4].

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Several commonalities have been identified depending on the type of ship under consideration. These shared characteristics provide valuable insights into the unique requirements and challenges associated with different vessel types. The primary findings and key conclusions drawn from this analysis are detailed in the following sections.

The foundation must be designed to accommodate all conditions, which essentially fall into two categories:

- regular conditions with maximum operating loads;
- extreme environmental conditions, out of operation loads (surviving condition).

The operating loads for regular conditions are defined by WAPS manufacturer.

Surviving conditions are defined by CCSS and up to this moment, different criteria is found. For example, survival wind speed is defined as 70 knots by Bureau Veritas [5] (see Table 2) and 100 knots by DNV [4] and ClassNK [6] Depending on WAPS characteristics like weight, area, lift and drag, tilting or retracting mode, and combination loads defined by CCSS, the more restrictive loads could be regular or extreme. The combination of loads specified by DNV [4] is presented in Table 1.

Table 1: Load combinations defined by DNV – ST-0511 [4].

LOADS	**Regular Loads (ULS)**	**Extreme loads (ULS)**						**Fatigue loads (FLS)**	
Load combination case	**R1**	**E1**	**E2**	**E3**	**E4**	**E5**	**E6**	**F1**	**F2**
Wind	WR	WE	WE	WE[2]	WE			WF	
Inertia	IR		IE	IE				IF	
Ice and snow				SE					
Temperatures					TE				
Green sea and spray						GE			
High Frecuency									HF
Others	OR						OE	OR	

where WR are the Wind Regular loads, IR the Inertia Regular loads, and OR are the Other Regular loads; WE are the Wind Extreme loads, IE are the Inertia Extreme loads, SE are the Snow and ice Extreme loads, TE are the Thermal Extreme loads, GE are the Green Extreme sea loads, and OE are the Other Extreme loads; WF are the Wind Fatigue loads, IF are the Inertia Fatigue loads, HF are the High frequency Fatigue loads, and OR are the Other loads.

Table 2: Load factors defined by BV NR-206 2024 [5].

	NORMAL ENVIRONMENTAL CONDITIONS AT SEA				
	Elementary Design loads				
	Wind loads		Inertial loads (1)		
	Lift and drag loads induced by wind (2)	Reaction forces on mast (3)	Mass of element	Ice added-mass	Acceleration
System Condition	Wind Speed defined by the designer		Ship motions and accelerations defined by the designer according		

2 In load combination E3, the wind speed defined as 'WE', may be reduced to 26 m/s, similar to the value underlying the IMO weather criteria for intact stability.

			to Section 4.3.3,Item c of 206-NR_2024-01		
In Operation / intact	1.3 (4)	1.1	1.05 (5)	1.0	1.3
In operation / accidental(6)	1.1	1.1	1.05 (5)	1.0	1.1
	EXTREME ENVIRONMENTAL CONDITIONS AT SEA				
	Elementary Design loads				
	Wind loads		Inertial loads (1)		
	Lift and drag loads induced by wind (2)	Reaction forces on mast (3)	Mass of element	Ice added-mass	Acceleration
System Condition	Wind speed 70 knots as defined in section 4.2.1, item b) of 206-NR_2024-01		Ship motions and accelerations as defined in section 4.3.3, item b) of 206-NR_2024-01		
In Operation / intact	1.0	1.0	1.05 (5)	1.0	1.1
In operation / accidental(6)	1.1	1.1	1.05 (5)	1.0	1.1
	EXTREME ENVIRONMENTAL CONDITIONS AT HARBOUR				
	Elementary Design loads				
	Wind loads		Inertial loads (1)		
	Lift and drag loads induced by wind (2)	Reaction forces on mast (3)	Mass of element	Ice added-mass	Acceleration
System Condition	Wind speed 100 knots as defined in section 4.2.1, of 206-NR_2024-01		Ship motions and accelerations as defined in section 4.3.3, item b) of 206-NR_2024-01		
Out of Operation / intact	1.0	1.0	N.A.		

(1) Longitudinal, vertical and transversal inertia loads induced by ship motions.
(2) Lift and drag loads induced by wind and their distribution induced by apparent wind with gusts effect, specifying the associated.
combination of the wind propulsion system configurations and the wind angle of attack.
(3) Reactions forces on mast as defined in accordance to [Section 4.1.1] Item b) of the 206-NR_2024_01.
(4) When automatic release systems approved by the Society are provided to avoid wind overloads on the wind propulsion systems, a value of 1.15 may be considered.
(5) When weight report including sufficient margin is available, a value of 1,0 may be considered.
(6) Accidental loads are to be defined on a case-by-case basis by risk analysis (HAZID or HAZOP), if requested. Provided risk mitigation are taken, lower value of α may be considered by the Society on a case-by-case basis.

As the manufacturer aims to produce all units in the most standardized manner possible, the design of the WAPS system structure is typically developed to accommodate the worst-case scenario for any type of CCSS and any potential position on the ship for which it is intended to be placed. This approach ensures that the system is robust enough to handle the most demanding conditions across various ship types. As a result, the WAPS structure is generally larger and more oversized than necessary, while the ship foundation can be optimized and refined for the specific vessel.

To complete the analysis, Inertial loads are calculated with a procedure given by each CCSS. For example DNV [4] requires the use of an "envelope acceleration" method, and three different scenarios to be evaluated as shown in Table 3. The design accelerations shall be equal or higher than the envelope accelerations of the ship on which the WAPS is installed.

Table 3: Envelope acceleration combinations given by DNV [4].

Load combination	$a_{x\text{-}u}$ (2)	$a_{y\text{-}u}$ (2)	$a_{z\text{-}u}$
Head sea 1	$a_{x\text{-}env}$	0	$a_{z\text{-}env\text{-}pitch}$
Head sea 2 (1)	$a_{x\text{-}env}$	0	(-) $a_{z\text{-}env\text{-}pitch}$
Beam sea 1	0	$a_{y\text{-}env}$	$a_{z\text{-}env\text{-}roll}$
Beam sea 2 (1)	0	$a_{y\text{-}env}$	(-) $a_{z\text{-}env\text{-}roll}$
Oblique sea 1	0.6 $a_{x\text{-}env}$	0.6 $a_{y\text{-}env}$	$a_{z\text{-}env}$
Oblique sea 2 (1)	0.6 $a_{x\text{-}env}$	0.6 $a_{y\text{-}env}$	(-) $a_{z\text{-}env}$

(1) Load combination is only applicable for uplift conditions.
(2) The horizontal accelerations shall be applied in the direction(s) giving maximum response.
Note: "$a_{x\text{-}env}$" refers to envelope acceleration in the x-axis, with similar notation applied to the y-axis ("$a_{y\text{-}env}$") and z-axis ("$a_{z\text{-}env}$"). When roll or pitch is mentioned at the end, it specifies that the envelope acceleration in the z-axis is caused by roll or pitch movements. For the table headers, "$a_{x\text{-}u}$" denotes the acceleration in the x-axis, where "u" represents interchangeable terms such as "env" (envelope).

Notwithstanding, Bureau Veritas [1] has a different approach and uses the "equivalent Design Wave Concept". This procedure analyses twelve different dynamic cases at varying loading conditions, resulting in a total of 24 cases when both directions are considered. The definition of these cases is shown in Table 4. A minimum of two loading ship conditions are required, ballast or full loaded. This means that at least 48 dynamic cases are obtained, from which the maximum values are taken.

Table 4: Dynamic load cases defined by BV [1].

EDW	Load case	Definition
HVM	HVM1	Head sea with maximum negative vertical bending moment (sagging) amidships. (3)
	HVM2	Head sea with maximum positive vertical bending moment (hogging) amidships. (3)
HVMf (1)	HVMf1	Head sea with maximum negative vertical bending moment (sagging) at 0.25 L from Aft End.(3)
	HVMf2	Head sea with maximum positive vertical bending moment (hogging) at 0.25 L from Aft End.(3)
FVM (2)	FVM1	Following sea with maximum negative vertical bending moment (sagging) at 0.25 L from Aft End.(3)
	FVM2	Following sea with maximum positive vertical bending moment (hogging) at 0,25 L from Aft End.(3)
BR	BR1-P	Beam sea with maximum negative roll motion.
	BR2-P	Beam sea with maximum positive roll motion.
	BR1-S	Beam sea with maximum positive roll motion.
	BP2-P	Beam sea with maximum positive roll motion.
BP	BP1-P	Beam sea with maximum negative hydrodynamic pressure at the waterline amidships.
	BP2-P	Beam sea with maximum positive hydrodynamic pressure at the waterline amidships.
	BP1-S	Beam sea with maximum negative hydrodynamic pressure at the waterline amidships.

	BP2-S	Beam sea with maximum positive hydrodynamic pressure at the waterline amidships.
OHM	OHM1-P	Oblique sea with maximum negative horizontal bending moment amidships.
	OHM2-P	Oblique sea with maximum positive horizontal bending moment amidships.
	OHM1-S	Oblique sea with maximum positive horizontal bending moment amidships.
	OHM2-S	Oblique sea with maximum negative horizontal bending moment amidships.
OHS (2)	OHS1-P	Oblique sea with maximum negative horizontal shear force and torsion at 0.75 L from Aft End.
	OHS2-P	Oblique sea with maximum positive horizontal shear force and torsion at 0.75 L from Aft End.
	OHS1-S	Oblique sea with maximum positive horizontal shear force and torsion at 0.75 L from Aft End.
	OHS2-S	Oblique sea with maximum negative horizontal shear force and torsion at 0.75 L from Aft End.
OVA (2)	OVA1-P	Oblique sea with maximum negative vertical acceleration at L from Aft End.
	OVA2-P	Oblique sea with maximum positive vertical acceleration at L from Aft End.
	OVA1-S	Oblique sea with maximum negative vertical acceleration at L from Aft End.
	OVA2-S	Oblique sea with maximum positive vertical acceleration at L from Aft End.
(1) Applicable to fatigue assessment only. (2) Applicable to strength assessment only. (3) The vertical shear force is also maximised with the dominant load component		

CCSS regulations for WAPS indicate allowable stress values depending on calculation method. Regulations differentiate between FEM with fine/coarse mesh and analytical calculations. According to the Scantling check in shell elements made of steel and aluminum the yield criteria must be calculated as follows and checked.

The Von Mises equivalent stress (σ_{eq}) must be in compliance with the following formula (Eq. 1)

$$\sigma_{eq} \leq \frac{R_y}{\gamma_m \gamma_R} \qquad \text{(Eq. 1)}$$

where:

R_y [N/mm^2] is the minimum yield stress, defined in NR206 DT R02 [5] (section 3);

γ_m is the material factor and equals to 1.02;

γ_R is the rresistance factor and is equal to

1.3 for direct calculation;

1.2 for finite element calculation in standard mesh areas;

1.1 in fine mesh areas.

At the same time, for the same parameter, DNV [4] uses a slightly different terminology:

$$\lambda_y \leq \lambda_{f\ perm}$$

Where:

λ_y is the yield utilization factor;

$\lambda_y = \frac{\sigma_{vm}}{\sigma_{yr}}$ is used for shell elements and solid elements;

$\lambda_y = \frac{|\sigma_{axial}|}{\sigma_{yr}}$ is used for beam and rod elements;

σ_{vm}[N/mm^2] is the von Mises stress, (see for definition [4], section 4.1.1.3);

σ_{axial}[N/mm^2] is the axial stress, (see for definition [4], section 4.1.1.4);

σ_{yr}[N/mm^2] is the reference yield strength, (see for definition [4], section 4.1.1.1).

$\lambda_{f\ perm}$ is the permissible fine mesh yield utilization factor as given in Table 4, Table 5, Table 6 and in [4].

For DNV, if size of mesh used is defined as “fine”, normally below 50x50 [mm], then the stress is allowed to be above the yield for some hot spots. If fatigue analysis is performed, this stress concentration is allowed to reach even higher levels. Table 5 can be used for direct comparison.

Table 5: Comparison between yield utilization factors.

Permissible Yield Utilization Factors			**DNV**		**BV**
			Without Fatigue Check	With Fatigue Check	
Fine Mesh (smaller 50x50)	Not Adjacent to Weld	Regular Loads	1.22	1.46	0.91
		Extreme Loads	1.53	1.84	
	Adjacent to Weld	Regular Loads	1.08	1.30	
		Extreme Loads	1.35	1.62	
Coarse Mesh		Regular Loads	0.72		0.83
		Extreme Loads	0.9		
Direct Calculation		Regular Loads	0.63		0.77
		Extreme Loads	0.77		

As already shown in Table 1 and Table 2, loads may or may not include an increased safety factor. This factor is adjusted based on whether it pertains to operational loads or extreme environmental conditions, such as survival conditions when the system is out of operation.

Since BV NR206 [5] applies magnifying factors to the loads, whereas DNV-ST-0511 [4] does not, the allowable stress values are significantly different. Therefore, corrections need to be made for a thorough and accurate examination.

2.3 Example of foundation design loads

A comparison between the approaches/rules outlined by the two above-mentioned classification societies is made here, to illustrate the effects on the design loads. This is the starting point for any optimization work on the foundation design.

Also, for same ship, two different positions are given to emphasize on the importance of load definition for any optimization of structure to be retrofitted on board.

2.3.1 Characteristics of the ship under examination

An oil product cargo tanker is considered for analysis. The key characteristics of the ship, which are necessary to estimate its motion and accelerations, is outlined in Figure 3 (screenshot from Marine Software calculation tool from Bureau Veritas, based on [5]).

Ship

Length	L =	174,80	m
Breadth	B =	32,20	m
Navigation Notation		Unrestricted	
Ship with Bilge keel		Yes	

Loading Condition - Full Load

Draught	T_{FL} =	12,891	m
Block coefficient	$C_{b\text{-}FL}$ =	0,812	
Waterplane coefficient	$C_{w\text{-}FL}$ =	0,913	
Metacentric height	GM =	2,478	m
Radius of Gyration	kr =		011 m

Loading Condition - Ballast

Draught	T_{Ball} =	7,319	m
Block coefficient	$C_{b\text{-}Ball}$ =	0,772	
Waterplane coefficient	$C_{w\text{-}Ball}$ =	0,833	
Metacentric height	GM =	7,707	m
Radius of Gyration	kr =		014 m

Maximum Sea State for **Normal** Conditions: Hs less than 7.0m

Figure 3: Base ship main dimensions.

The positions selected for the WAPS are the forward most and outermost sides of the ship, chosen to achieve the maximum acceleration results in a worst-case scenario.

2.3.2 Acceleration results: position 1 (forward)

As previously mentioned, CCSS designs must account for two distinct conditions:

a) operational loads;
b) non-operational loads in extreme environments (survival loads).

Since the forces generated under extreme environmental conditions are significantly greater than those during regular operation, the focus of this discussion will be on survival loads. The foundation must be designed to withstand these larger forces effectively.

To ensure accurate analysis, calculations are performed using two distinct approaches, which are outlined below:

1. using BV 467 [1];
2. using DNV-RU-SHIP [7].

According to BV [1], a minimum of three cases needs to be calculated:

- maximum longitudinal acceleration (with transversal and vertical acceleration for the case that maximizes the longitudinal one);
- maximum transversal acceleration (with longitudinal and vertical acceleration for the case that maximizes the transversal one);
- maximum vertical acceleration. (with longitudinal and transversal acceleration for the case that maximizes the vertical one).

Acceleration results at sail center of gravity are shown in Table 6.

Table 6: Acceleration at forward position[3] and dynamic cases occurrence according to BV [1].

	Description	EDW/Load	$a_{x\text{-env}}$ [m/s^2]	$a_{y\text{-env}}$ [m/s^2]	$a_{z\text{-env}}$ [m/s^2]
Extreme scenario	Load case #1 with maximum longitudinal acceleration	OVA1-S-Ballast	2.887	-1.723	-6.739
	Load case #1 with minimum longitudinal acceleration	OVA2-S-Ballast	-2.887	1.723	6.739
	Load case #1 with maximum transversal acceleration	BR2-P-Ballast	0.000	7.285	-2.109
	Load case #1 with maximum transversal acceleration	BR1-S-Ballast	0.000	7.285	0.128
	Load case #1 with maximum vertical acceleration	OVA2-P- Ballast	-2.561	-1.723	6.989
	Load case #1 with minimum vertical acceleration	OVA1-P- Ballast	2.561	1.723	-6.989

Env. = Envelope

BV-WAPS NR-206 [1] include a magnifying factor for accelerations, given in Table 2. For extreme cases the value is 1.1, but for regular operating conditions this factor is 1.3. This guideline also accounts for a mass increased factor of 1.05 (i.e, 1.05·* 1.1 = 1.155). To compare accelerations (and induced inertial loads) between BV [1] and DNV [4], both these factors should be taken into consideration, as DNV do not account for them.

Following DNV, there are three load cases:

- head sea (maximum longitudinal acceleration, with no transversal acceleration);
- beam sea (maximum transversal acceleration, with no longitudinal acceleration);
- oblique sea (a combination of the two previous according to the factors defined in Table 4).

To visualize, a representation of both values is given in Figure 4 and in Table 7.

Table 7: Accelerations for forward position according to DNV and comparison with BV results.

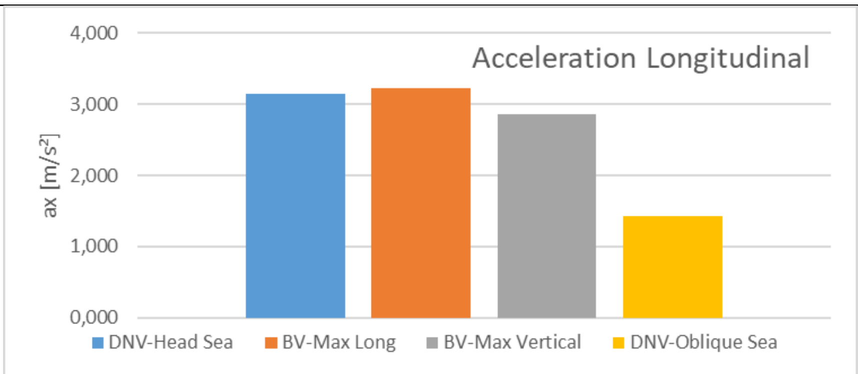
Description		$a_{x\text{-env}}$ [m/s^2]	$a_{y\text{-env}}$ [m/s^2]	$a_{z\text{-env}} + g$ [m/s^2]
DNV	Head Sea	3.151	0	19.882
	Beam Sea	0	9.003	14.188
	Oblique Sea	1.891	5.402	19.982
BV·1,155	Maximum longitudinal	3.221	1.922	17.324
	Maximum transversal	0.000	8.126	12.159
	Maximum vertical	2.857	1.922	17.603

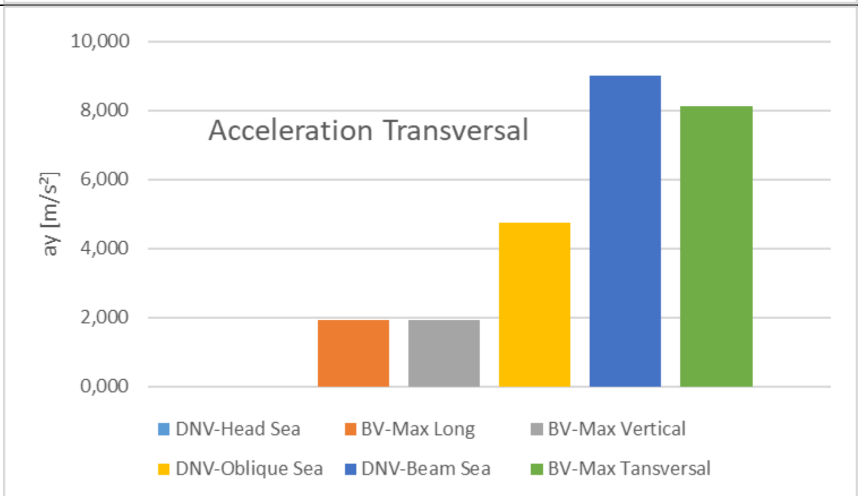
[3] Calculation position: x=163.00 [m]; y= 8.22 [m]; z= 36.61 [m].

DNV/BV	Maximum longitudinal	0.98	0.00	1.15
	Maximum transversal	-	1.11	1.17
	Maximum vertical	0.66	2.81	1.14

Env. = Envelope

The case defined by DNV as oblique sea is very similar to the obtained case in BV as OVA-S. However, this is used to compute maximum longitudinal acceleration. For BV, there is a second OVA (oblique sea condition) case that maximizes vertical acceleration, which also happens in DNV. This is why oblique sea from DNV is compared to maximum vertical acceleration case from BV.





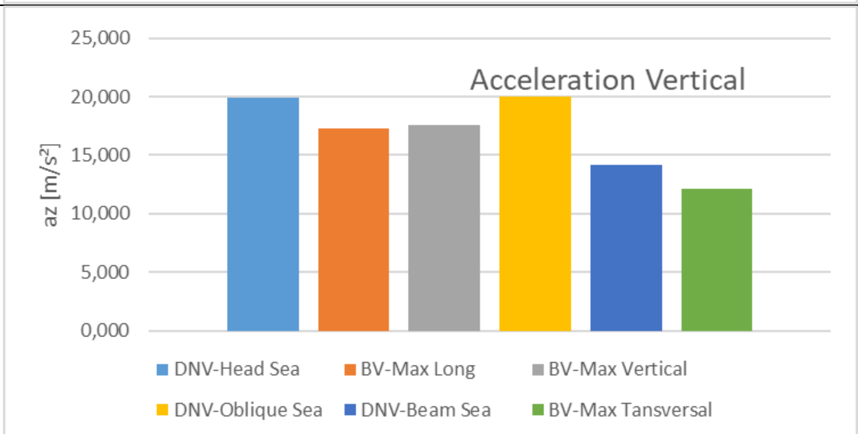


Figure 4: Graphs showing accelerations in three directions according to [4] or [5].

As can be seen, when considering the magnifying factors from BV, in the horizontal plane for transverse and longitudinal cases acceleration values are similar for DNV and BV, with differences are within a 11%.

When looking at vertical accelerations, DNV always throws greater values with differences exceeding 15%.

Oblique case is not good to make a straight comparison but a root main square average of the two. Then the difference is 70 % greater for DNV over BV.

2.3.3 Acceleration results: position 2 (midship)

A second installation position is analyzed for the same ship, in particular the midship location along the centerline being selected. This area experiences less severe ship motions compared to other positions, acceleration results are detailed in Table 8.

Table 8: Acceleration at midship position[4] and dynamic cases occurrence according to BV [1].

	Description	EDW/Load	$a_{x\text{-env}}$ [m/s²]	$a_{y\text{-env}}$ [m/s²]	$a_{z\text{-env}}$ [m/s²]
Extreme scenario	Load case #1 with maximum longitudinal acceleration	OVA1-S-Ballast	2.724	0.223	-1.304
	Load case #1 with minimum longitudinal acceleration	OVA2-S-Ballast	-2.724	-0.223	1.304
	Load case #1 with maximum transversal acceleration	BR2-P-Ballast	0.000	7.285	-1.118
	Load case #1 with maximum transversal acceleration	BR1-S-Ballast	0.000	7.285	1.118
	Load case #1 with maximum vertical acceleration	OVA2-P- Ballast	0.000	-2.786	2.667
	Load case #1 with minimum vertical acceleration	OVA1-P- Ballast	0.000	2.786	-2.677

Env. = Envelope

As observed, the dynamic cases vary depending on the longitudinal position on board. In this instance, the maximum vertical acceleration corresponds to a beam scenario, meaning there are no significant accelerations in the longitudinal direction. A similar table than the one included in the previous section Table 7) is provided for comparison (Table 9).

Table 9: Accelerations at midship position according to DNV and comparison with BV results.

Description		a_{x_env} [m/s²]	a_{y_env} [m/s²]	a_{z_env} + g [m/s²]
DNV	Head Sea	3.151	0	14.207
	Beam Sea	0	9.003	13.949
	Oblique Sea	1.891	5.402	14.207
BV·1,155	Maximum longitudinal	3.039	0.249	11.261
	Maximum transversal	0.000	8.126	11.054
	Maximum vertical	0.000	3.108	12.793
DNV/BV	Maximum longitudinal	1.04	0.00	1.26
	Maximum transversal	-	1.11	1.26

[4] Calculation position: x: 87.40 [m]; y: 0.00 [m]; z: 36.61 [m].

	Maximum vertical	-	-	1.11

Env. = Envelope

Similar conclusions can be drawn, with observed differences in the main longitudinal and transverse directions remaining under 11%. However, the vertical direction now shows a significantly greater discrepancy, with DNV indicating 26% higher acceleration compared to BV. Another valuable analysis is the comparison of accelerations at the forward and midship positions.

For DNV, when applying the envelope acceleration method, the longitudinal position of the sails has no impact on longitudinal (x) and transverse (y) accelerations; only the height (z) influences the results.

In contrast, BV employs the EWD method, where altering the ship's position causes different dynamic cases to dominate, resulting in variations in maximum acceleration at the same height. For example, in the forward position, longitudinal acceleration was 6% higher compared to the midship position.

For both methods, vertical acceleration increases as the position moves away from midship. Under DNV, the vertical acceleration increase was approximately 40%, whereas for BV, it was more pronounced, reaching up to 54%.

Another notable observation, highlighted through a separate set of calculations, is that longitudinal accelerations rise significantly faster with height under DNV compared to BV. This relationship can be expressed as a linear equation in the form of:

$$a_x = b + k_x \cdot z$$

$$a_y = b + k_y \cdot z$$

Where b represents the independent term, and k denotes the slope, which defines the rate at which acceleration increases with height (z). The independent term aligns with the observations discussed in the previous tables, where no significant variations were noted. However, the slope term shows a considerable impact, particularly on longitudinal acceleration. This relationship is further detailed in Table 10.

Table 10: Slope indicating the rate of increase in acceleration with height.

	DNV	BV	DNV/BV
k_x	0.11	0.07	1.48
k_y	0.14	0.12	1.16

In conclusion, the analysis highlights notable differences in acceleration behavior depending on the classification method and the position on the ship. While longitudinal and transverse directions show relatively small discrepancies, with variations under 11%, the vertical direction exhibits significantly greater differences. DNV results in 26% higher vertical accelerations compared to BV.

The comparison of forward and midship positions further underscores the influence of classification methods. Under DNV's envelope acceleration approach, longitudinal and transverse accelerations remain unaffected by sail position, with height being the sole influencing factor. Conversely, BV's EWD method demonstrates that changes in longitudinal position can shift dynamic cases, leading to differences in maximum accelerations. For instance, longitudinal acceleration is 6% higher in the forward position compared to midship.

Vertical accelerations consistently increase as the position moves away from midship, with DNV reporting a 40% rise and BV showing an even greater increase of 54%. Moreover, longitudinal accelerations are more sensitive to height under DNV compared to BV, as evidenced by the calculated slopes. The slope for longitudinal acceleration is 1.48 times higher in DNV, while the slope for transverse acceleration is 1.16 times greater.

These findings emphasize the importance of considering both classification methods and ship-specific characteristics when evaluating dynamic loads and designing structural foundations for WAPS installations. Detailed insights provided in Table 9 and Table 10 further illustrate the variation in acceleration behavior and highlight the necessity of tailored approaches for accurate assessment and robust design.

2.3.4 Wind loads comparison

Lift and drag loads are calculated as follow:

$$L = \frac{1}{2} \cdot \rho \cdot S \cdot V^2 \cdot C_L$$
$$D = \frac{1}{2} \cdot \rho \cdot S \cdot V^2 \cdot C_D$$

where L is the lift, D is the drag, ρ is the air density, S is the area, V is the speed, and C_L and C_D are respectively the lift and drag coefficients.

Since force is proportional to the square of the wind speed, even a small increase in wind speed can have a substantial effect.

DNV requires considering a minimum apparent wind speed of 100 knots, while BV suggests a minimum of 70 knots of true wind. To calculate the apparent wind speed, the ship's speed must also be factored in. A conservative approach assumes 50% of the ship's speed is added to the true wind speed, as assuming 100% of the ship's speed would be overly optimistic.

For the sample vessel analyzed in this study, the typical speed under ballast conditions, with the main engine operating at 85% load, is 14.6 knots. Using the conservative estimate, half of the vessel's speed (7.3 knots) is added to the true wind speed of 70 knots, resulting in an apparent wind speed of 76.3 knots. Further difference of correcting the speed with height is observed between BV and DNV using the correction formula for DNV for apparent wind speed with Height (Eq. 2).

$$U_{we} = \max\left(50; 44\left(\frac{h_L}{10}\right)^{0.15}\right) \qquad \text{(Eq. 2)}$$

Using the correction formula for true wind speed for BV (Eq. 3):

$$V(z) = \mathrm{V}\left(\frac{z}{10}\right)^{\alpha} \qquad \text{(Eq. 3)}$$

where $V(z)$ [m/s] is the true wind speed at height z; α is the wind shear exponent, generally assumed as 0.14 for normal wind and 0.11 for extreme wind; V is the true wind speed measured at 10 [m] above the sea level (V must not be less than 36.01 [m/s] in extreme environmental conditions at sea and must not be less than 51.44 [m/s] in extreme environmental conditions in harbour); and z [m] is the height above the sea level.

To observe the influence in load, square speeds must be compared in combination with magnifying factors. Even though, in extreme conditions this factor[5] is unity (Eq. 4).

$$\frac{DNV\ Apparent\ Wind\ Speed^2}{BV\ Apparent\ Wind\ Speed^2 \cdot factor} = \frac{102.4^2}{88.2^2 \cdot 1.0} = 1.35 \qquad \text{(Eq. 4)}$$

The comparison between DNV and BV highlights significant differences in the treatment of wind speed corrections with height and their impact on load calculations. Using the correction formula for apparent wind speed in DNV (Eq. 2) and the true wind speed formula for BV (Eq. 3), notable discrepancies emerge. DNV's formula results in a higher apparent wind speed, reflecting a more conservative approach for extreme environmental conditions. When comparing the squared wind speeds in combination with the magnifying factors, the DNV approach yields a load that is 35% higher than BV's. This difference stems from DNV's assumption of more severe wind conditions, particularly at greater heights, compared to BV's rules, which assume less extreme winds.

In summary, DNV's methodology anticipates higher wind loads under extreme conditions, leading to a more conservative design approach. This ensures greater safety margins in situations where exceptionally high winds might be encountered, whereas BV's rules assume a less aggressive wind environment, resulting in lower estimated wind loads.

2.3.5 Total load comparison

The forward position is utilized for calculating inertial loads. Table 11 compares the loads derived using the procedures recommended by DNV and BV. To make the values dimensionless, the DNV loads were divided by the BV loads. As a result, a ratio of 1.35 indicates that the forces and moments from DNV are 1.35 times greater than those from BV, meaning the DNV values are 35% higher than the BV values.

Table 11: Comparison of loads: DNV/BV. [6]

	EXTREME ENVIRONMENTAL OUT OF OPERATION								
	Maximum longitudinal acceleration			Maximum transversal acceleration			Maximum vertical acceleration		
	F_{xb}	F_{yb}	F_z	F_{xb}	F_{yb}	F_z	F_{xb}	F_{yb}	F_z
Aerodynamic Load	1.35				1.35		1.30	1.35	
Inertial Loads	0.97		1.05		1.08	1.24	1.62	1.69	1.03
TOTAL	1.18		1.05		1.17	1.24	1.44	1.51	1.03
	M_{xb}	M_{yb}	M_z	M_{xb}	M_{yb}	M_z	M_{xb}	M_{yb}	M_z
Aerodynamic Moment		1.35	1.35	1.35		1.35	1.35	1.30	1.35
Inertial Moment		1.00	0.76	1.08		0.97	1.47	1.50	1.51
TOTAL		**1.20**	1.07	**1.18**		1.10	1.40	1.39	1.42

[5] Factor represents the percentage difference in obtaining the apparent wind speed based on the methods proposed by DNV and BV (Eq. 2 and Eq. 3). This will result in different aerodynamic loads.
[6] Color ramp indicates the scale of values (green = lower loads and red = higher loads), highlighting discrepancies between the two classification societies.

In conclusion, the oblique case is not particularly relevant for comparison, as previously discussed.

In the horizontal plane:

- inertial loads in the longitudinal and transversal directions are very similar, with differences of less than 8%;
- moments differ between BV and DNV due to variations in the acceleration gradient with height. In DNV, the greater acceleration at the top of the sail shifts the center of acceleration, magnifying the moment arm and, consequently, the resulting moment.

In the vertical plane:

- vertical loads are significantly higher for DNV compared to BV, with a maximum difference of 24%.

2.4 Requirements for different types of ships

The International Association of Classification Societies (IACS) provides common structural requirements, particularly for bulk carriers and tankers, which are governed by the Common Structural Rules (CSR). To illustrate these requirements, images depicting typical structures of various ship types have been sourced from one of its member organizations, Images adopted from [8].

2.4.1 Tankers

The deck structure of chemical and oil tankers is typically located outside the tanks (Figure 5, Figure 6, Figure 7, and Figure 8), allowing for the possibility of welding a foundation on top of the beams and longitudinal girders without impacting the deck plate. This design is crucial for minimizing the need for painting and pressure testing of the tanks. Both tasks - painting and pressure testing - would incur additional costs and time at the yard.

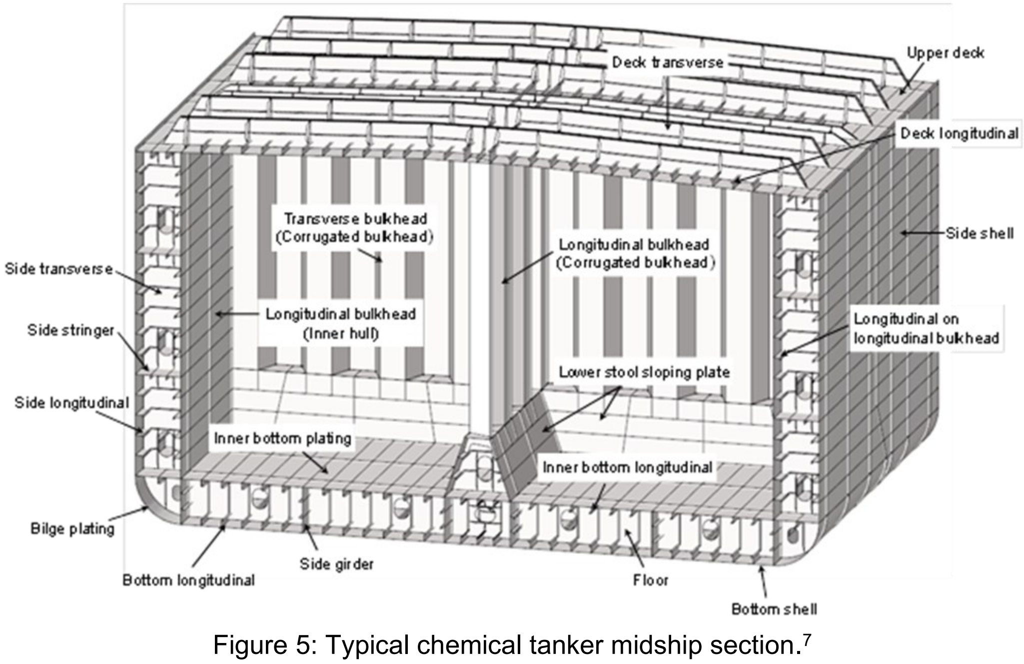


Figure 5: Typical chemical tanker midship section.[7]

[7] The scheme was adopted from [8].

Figure 6: Example of chemical tanker with 4 units WAPS arrangement.[8]

Figure 7: Example of Chemical Tanker with 3 units WAPS Installation.[9]

[8] Reproduced with permission from Odfjell, copyright Odfjell, 2024.
[9] Reproduced with permission from EPS, copyright EPS, 2023.

Figure 8: Example of Oil product tanker with 4 units WAPS units.[10]

2.4.2 Bulk carriers

Different types of dry bulk carriers can be found. In respect to WAPS installation two main types can be distinguished: box shaped bulk carrier (Figure 9) and ore carriers (Figure 10).

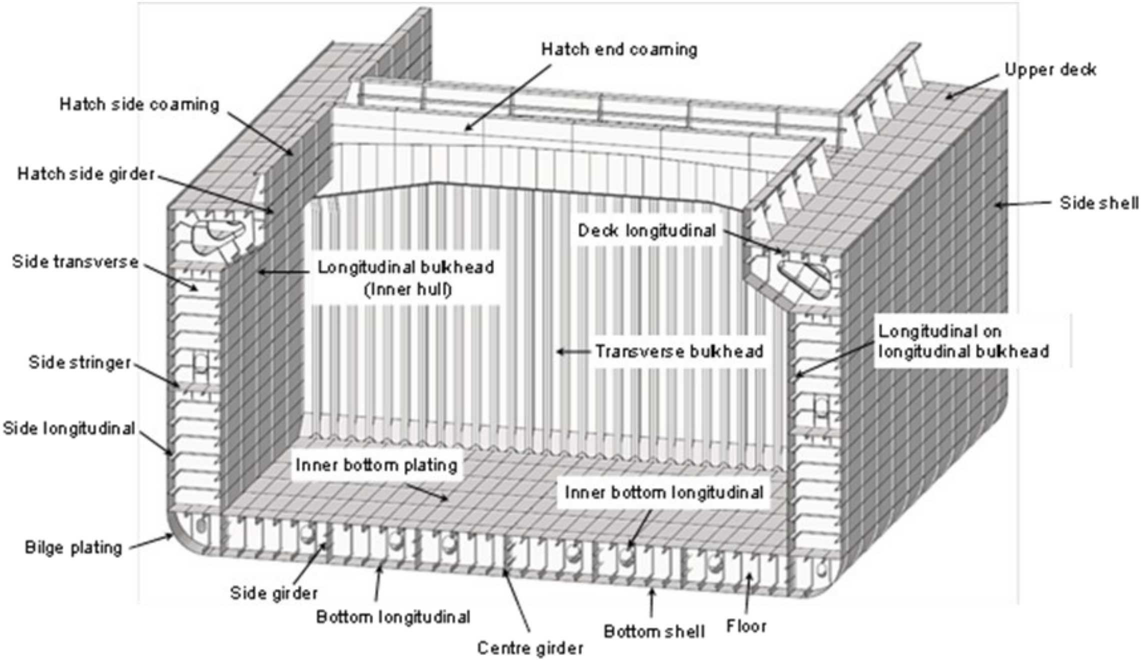


Figure 9: Typical box-shaped bulk carrier.[11]

[10] Reproduced with permission from MARFLET, copyright MARFLET, 2024.
[11] The scheme was adopted from [8].

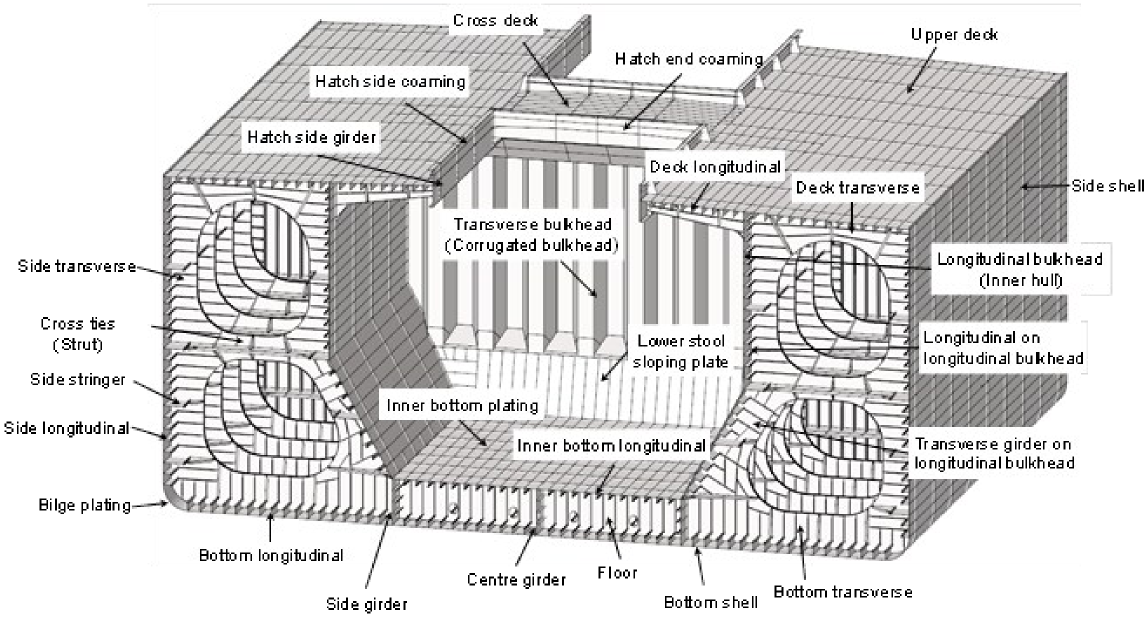


Figure 10: Typical ore carrier midship section.[12]

Many bulk-carrier have single skin (Figure 11) instead of side tanks or void spaces, as shown below.

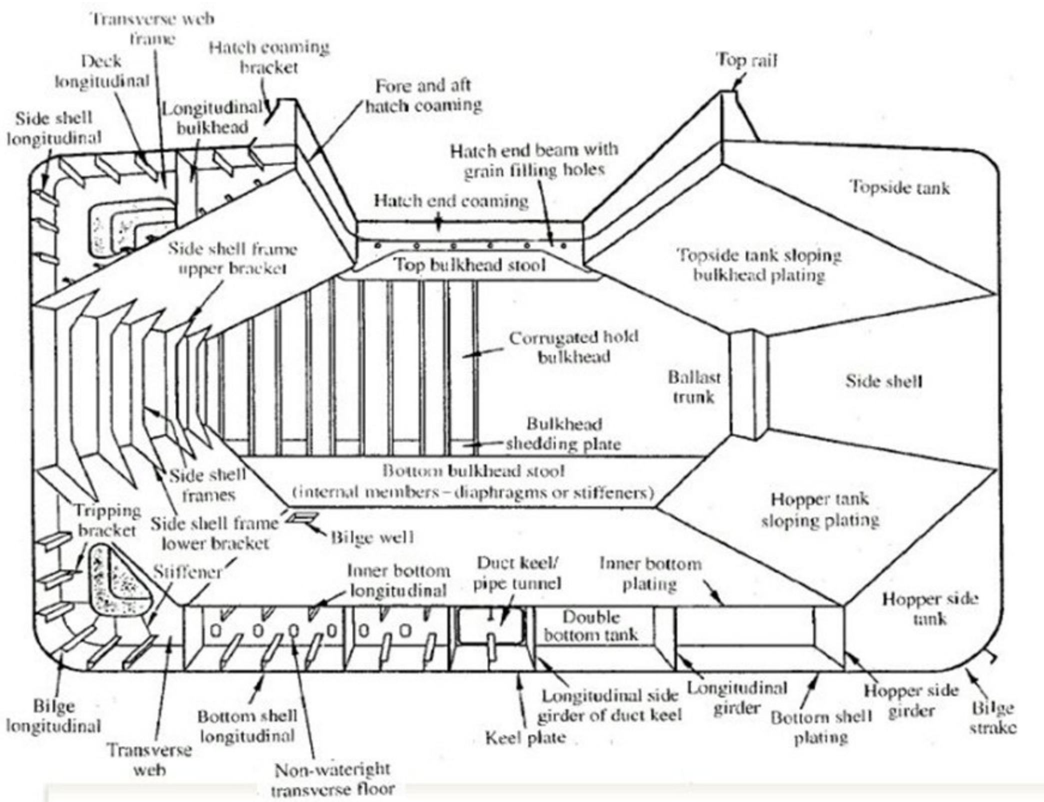


Figure 11: Single skin midship section of bulk carrier.[13]

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[12] The scheme was adopted from [8].
[13] The scheme was adopted from [8].

Generally, free space is available on the deck. To minimize interference with loading operations, WAPS systems should be located between hatches (see Figure 12).

Figure 12: Proposed layout of a bulk carrier WAPS configuration of 4 eSAILS ®.[14]

Figure 13: Example of WAPS installation on a bulk carrier (converted to juice carrier).[15]

[14] Reproduced with permission from Bound4Blue, copyright Bound4Blue, 2024.
[15] Reproduced with permission from Bound4Blue, copyright Bound4Blue, 2024.

Installing WAPS systems on the sides of the vessel can cause a shift in the center of gravity outward, resulting in a permanent list (transversal inclination). However, this list is typically less than 1° of heel, meeting the requirements of the IMO IS 2008 Code. Placing the foundations on the sides also allows for reinforcement in the hopper tank area, without impacting the cargo space.

If foundations are positioned between hatches, a watertight bulkhead is typically found below, which helps absorb vertical loads and reduces the need for additional reinforcement.

Although placing the systems along the centerline might affect loading operations, it offers the advantage of improved visibility (see Figure 12 and Figure 13). For ore carriers, more space is generally available on the sides than along the centerline.

A FEM model was developed to calculate the required steel weight and analyses the stress distribution within the existing structure.

The results are presented in Figure 14. The mesh size used in the FEM calculation is 30 mm for critical points and 200 mm for non-critical areas.

Similarly to general cargo ships, an alternative location is to place sails at the forecastle to avoid interferences with cargo area and loading operations.

Ideally on this type of vessels, structure will be directly welded to main deck without the need for increasing the thickness of the plate, to reduce time and costs.

Depending on cargo operations, three typical arrangements are expected:

- sail in cargo area, between hatches at centerline;
- sail in cargo area, between hatches on one side;
- sail in forecastle.

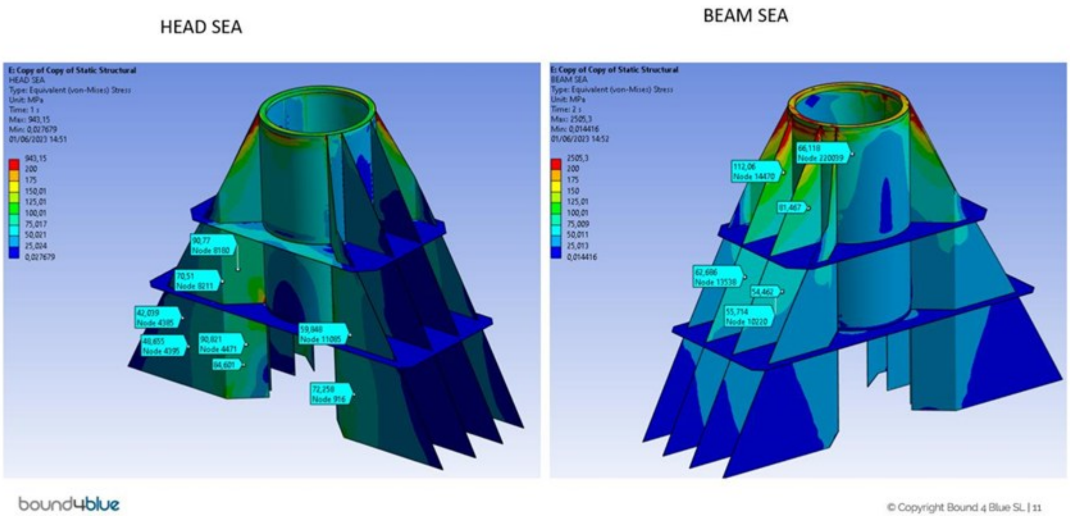

Figure 14: Stress distribution FEM results for ship shown in Figure 16.[16]

As shown in the typical midship section drawings (Figure 15 and Figure 16**Errore. L'origine riferimento non è stata trovata.**), there is one stool at the top of transversal bulkheads. In some cases, some additional reinforcements will be required to strengthen for the additional loads.

[16] Reproduced with permission from Bound4Blue, copyright Bound4Blue, 2024.

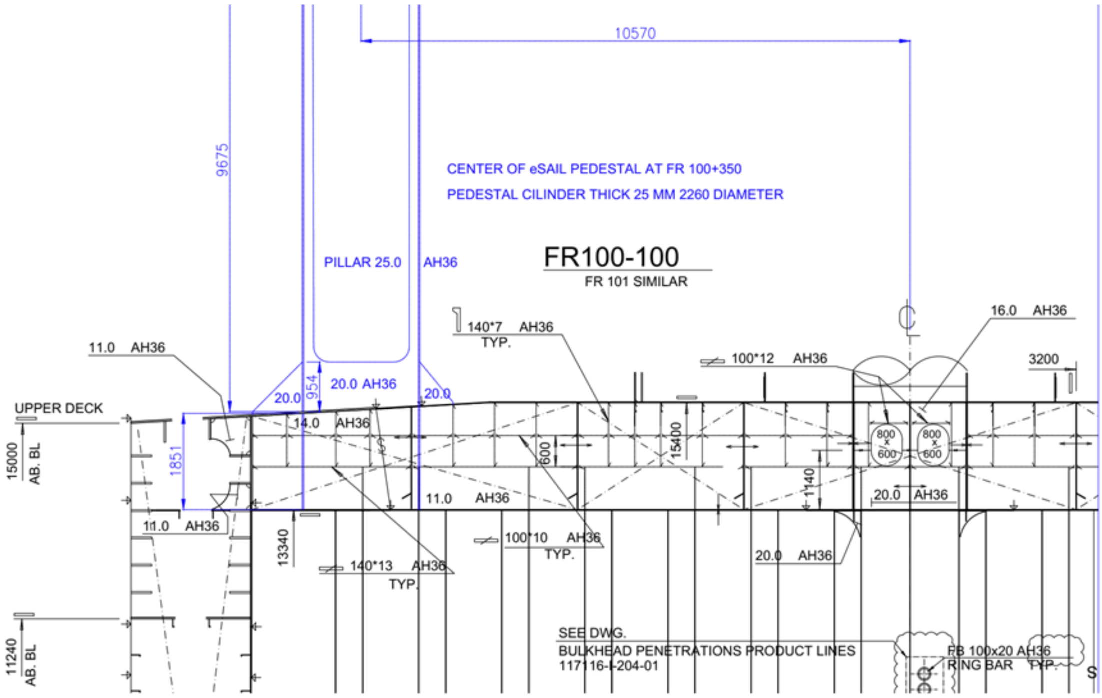

Figure 15: Transversal section of box shaped bulk-carrier with side foundation for eSAIL® on top of transversal bulkhead (between hatches).[17]

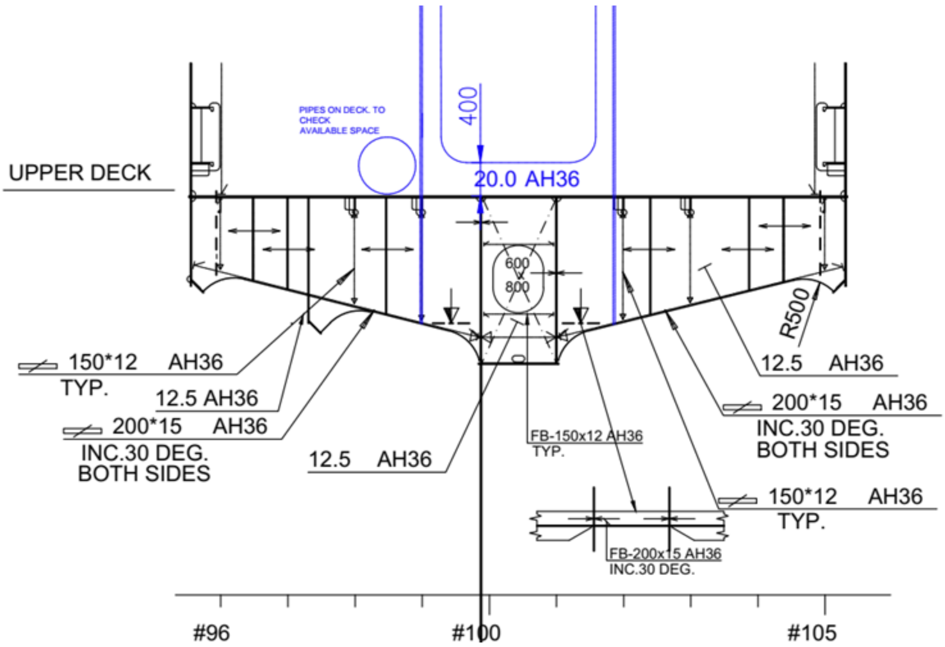

Figure 16: Longitudinal section of box shaped bulk carrier with foundation at centerline.[18]

[17] Reproduced with permission from Bound4Blue, copyright Bound4Blue, 2024.
[18] Reproduced with permission from Bound4Blue, copyright Bound4Blue, 2024.

Handy size bulk carrier many times account for cranes. It is difficult to remove the crane and substitute it for a sail, as the crane has a reason for being there in the first time. But normally kamsarmax and capsize bulk carriers are gear-less.

Figure 17: Typical handy size bulk carrier with cranes.[19]

It is interesting to see if the design can be changed and sails can be added in the same location where the smaller sisters have cranes (Figure 17).

2.4.3 Ro-ro ships

Ro-ro ships (including car carriers and other truck-loaded vessels) present unique considerations for the installation of systems like WAPS. Generally, the exposed deck is not used for cargo (Figure 18 and Figure 19), allowing for open space on deck and reducing the interference with loading operations.

Below this exposed deck, there is typically a hangar equipped with numerous systems, including:

- Ventilation and smoke extraction systems.
- Powerful firefighting systems.
- Fire detection systems.
- Hydraulic piping.
- Lighting.
- Cable trays with large power and control cables.
- Insulation.
- Piping for other systems.

[19] Reproduced with permission from Tsuneishi, copyright Tsuneishi, 2024.

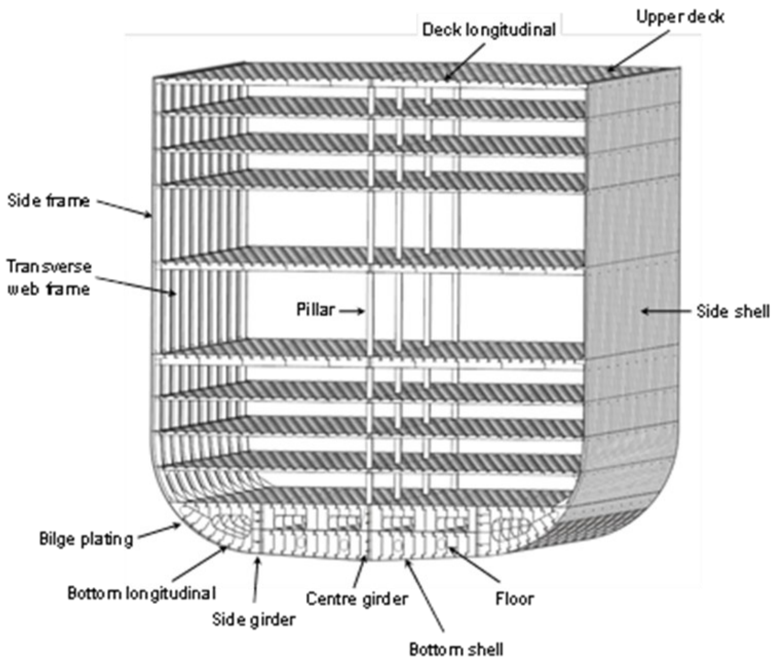

Figure 18: Typical car carrier midship section.[20]

These systems pose significant obstacles when a structural reinforcement is needed below the deck, as considerable time may be required for removal, access work, refitting, and functionality checks. Special attention should be given during conceptual design and feasibility studies to minimize the costs associated with these additional tasks on this type of vessel.

Figure 19: Example of Ro-ro WAPS Installation.[21]

[20] The scheme was adopted from [8].
[21] Reproduced with permission from Bound4Blue, copyright Bound4Blue, 2024.

Loads from internal ramps hanging from deck was combined with sail-induced loads and a FEM model was created for the calculation.

The calculation is achieved with SolidWorks Simulation 2022 standard and results are summarized in Table 12.

Table 12: Parameters used for the simulation.

Mesh summary	
Mesh Solver	static solver
Number of elements	approximately 208500
Number of nodes	approximately 485000
Type of element	triangular shell
Degree of elements	quadratic

A multiscale mesh model (plate elements) with a mesh size of 50 to 700 [mm] has been realized. The size of elements depends on the area of interest where stresses are higher. Outside zone of interest, the size of elements is 700 mm (side shells, deck and primary members that are far away from foundations).

Inside zone of interest, the size of elements is reduced to 100 and to 50 [mm] depending on the precision needed. Results are included in Figure 20 and Figure 21.

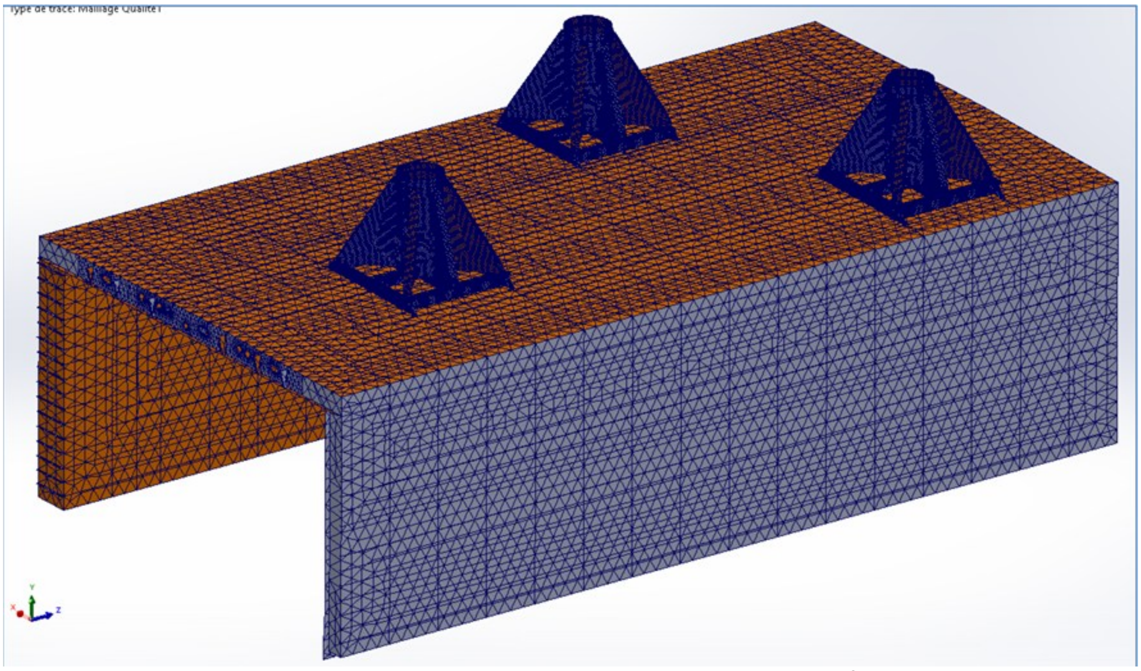

Figure 20: Mesh of foundations installed on deck for three eSAILS® on a ro-ro ship..[22]

[22] Reproduced with permission from Bound4Blue, copyright Bound4Blue, 2024.

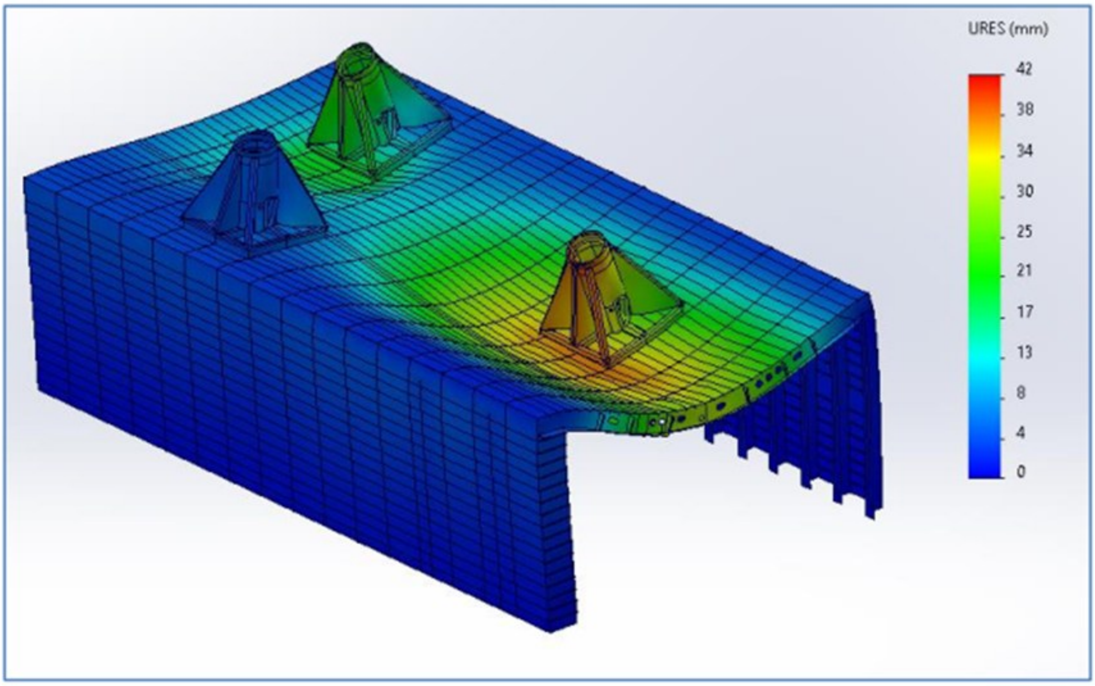

Figure 21: Displacement FEM results of foundations for three eSAILS® on a ro-ro ship.[23]

2.4.4 Gas tankers

Different type of gas tankers can be found, mainly LPG and LNG tankers. While LPG is commonly transported in independent tanks (type A, B or C can be found), LNG can be transported in independent tanks or membrane tanks (Figure 22 and Figure 23).

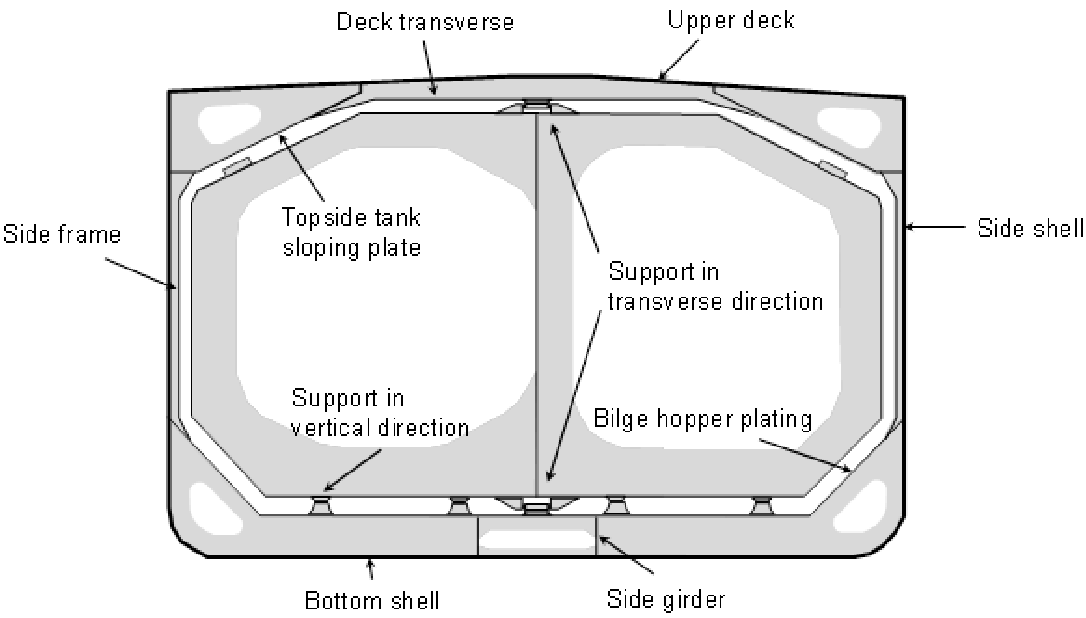

Figure 22: Typical LPG gas tanker, independent tank type A or B midship section.[24]

[23] Reproduced with permission from Bound4Blue, copyright Bound4Blue, 2024.
[24] The scheme was adopted from [8].

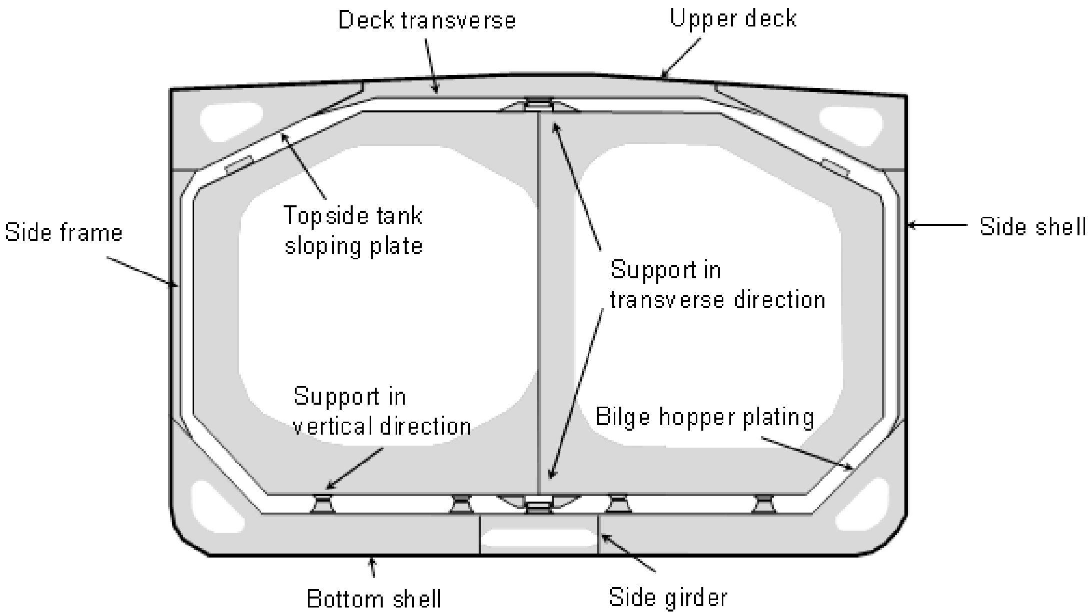

Figure 23: Typical gas tanker with independent tank type C.[25]

Case Study: a LPG tanker (160 m long LOA[26]), illustrated in Figure 24, was modelled to find the optimal deck position for the WAPS, performing a structural analysis. The deck structure (beams, longitudinals, frames, etc.) is located below the deck plates, meaning it is not visible from the outside. Therefore, the pedestals must be welded directly to the deck plates. Access to the underdeck is therefore required to ensure the correct position of the pedestal and the welding areas, ensuring the loads transmission.

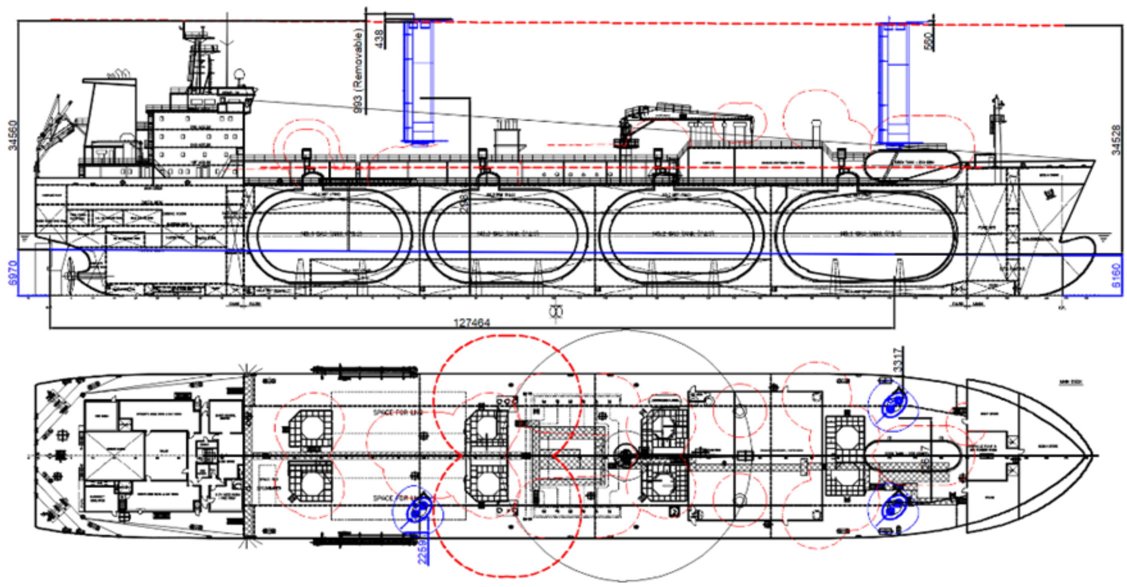

Figure 24: Typical LPG GA and sail arrangement.[27]

[25] The scheme was adopted from [8]..
[26] Length overall (LOA).
[27] Reproduced with permission from Bound4Blue, copyright Bound4Blue, 2024.

It should be noted that air draft was concluded as one of the main restrictions.

Two FEM models were developed to calculate the forward and aft sail structures. The model was meshed with a 50 [mm] element size (fine mesh according to [1]), which was found sufficient for this analysis.

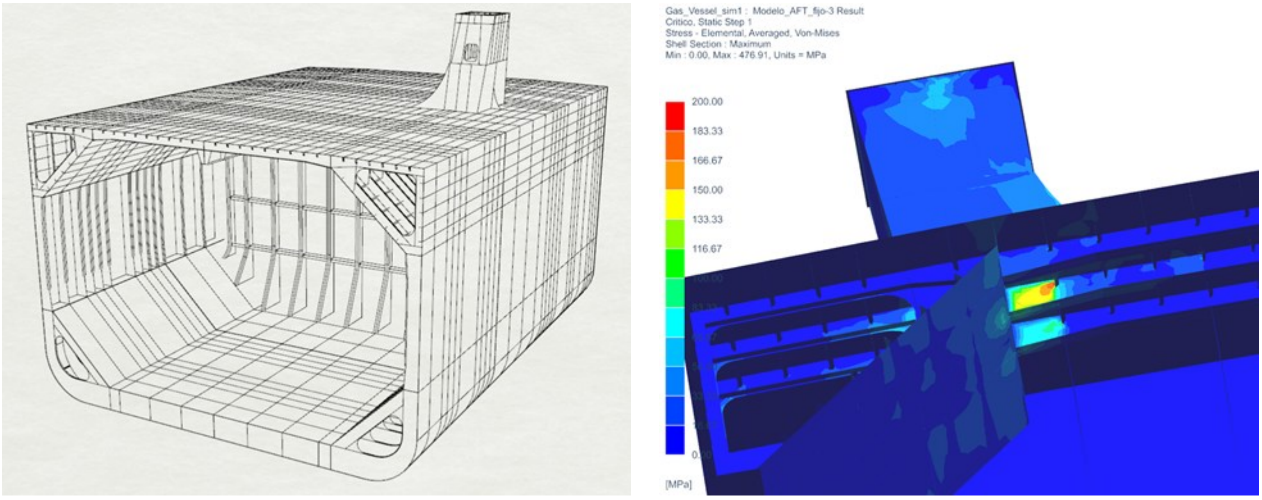

Figure 25: Mesh model and stress FEM calculation results of eSAIL® foundation installed on LPG tanker.[28]

Figure 25**Errore. L'origine riferimento non è stata trovata.** and Figure 26 show the FEM meshed model and the stress results on the eSAIL® foundation installed in the LPG tanker.

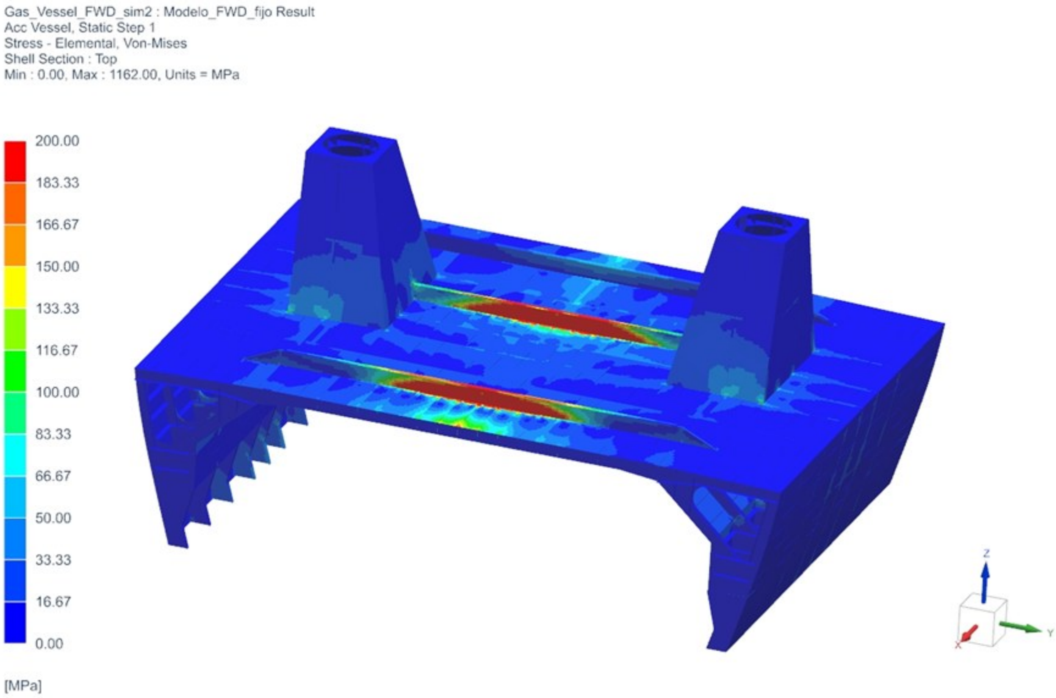

Figure 26: Stress FEM results of typical LPG forward structure on deck.[29]

[28] Reproduced with permission from Bound4Blue, copyright Bound4Blue, 2024.
[29] Reproduced with permission from Bound4Blue, copyright Bound4Blue, 2024.

2.4.5 General cargo

Typically, general cargo has very wide hatches and there is very limited space on deck. Three options are suggested, respectively depicted in Figure 27, Figure 28 and Figure 29.

Case A: forward of cargo area - forecastle.

Figure 27: Example of WAPS installation on forecastle.[30]

Case B: aft of cargo area - Aft of superstructure.

Figure 28: Example of WAPS installation aft of superstructure on a general cargo.[31]

Case C: according to the needs and possibilities of the vessel, foundations could be integrated with side shell like some crane pedestals.

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[31] Reproduced with permission from Bound4Blue, copyright Bound4Blue, 2024.

Figure 29: Typical general cargo ship with side cranes.[32]

For case a, forward of cargo area, free space must be found between mooring winches, anchor windlass and forward light mast (Figure 30).

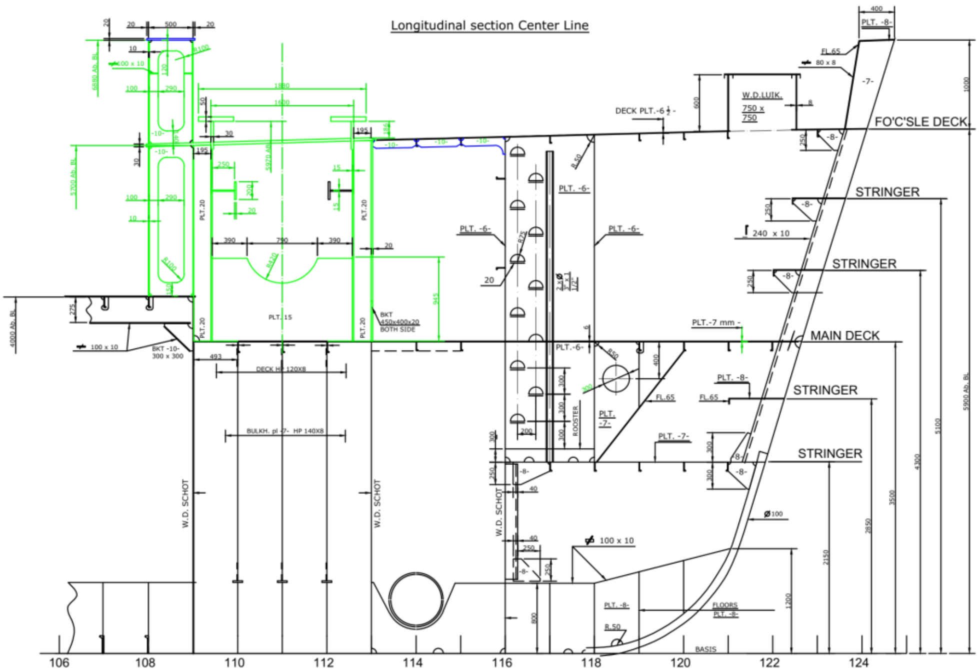

Figure 30: Structure Longitudinal view on a fore castle foundation.[33]

A 3D FEM model was created to calculate and optimize the structure. A global mesh size of 250 mm was used for the hull structure areas far from the win- sail foundation, as these regions do not

[32] Reproduced with permission from Bound4Blue, copyright Bound4Blue, 2024.
[33] Reproduced with permission from Bound4Blue, copyright Bound4Blue, 2024.

experience stress contributions from loads applied to the foundation (Figure 31). Areas near the foundation were modelled with significantly finer mesh sizes of 50 [mm] x 50 [mm], and even smaller plate dimensions of 25 [mm] x 25 [mm].

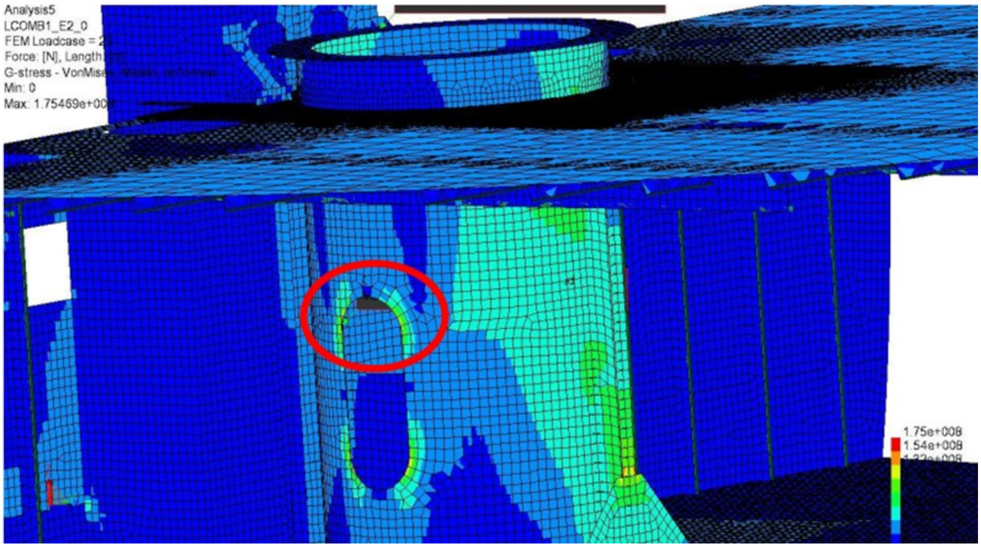

Figure 31: Mesh model for FEM calculation for foundation placed forecastle.

In case b, located aft of the superstructure, special attention must be given to potential aerodynamic interferences from obstructions positioned forward. For example, a FEM model was created for a symmetrical structure designed to support two WAPS units. The model was meshed with a 50 [mm] element size (fine mesh in accordance with [1], which was found sufficient for this analysis. See the figures Figure 32, Figure 33, and Figure 34 for the meshed model and the results.

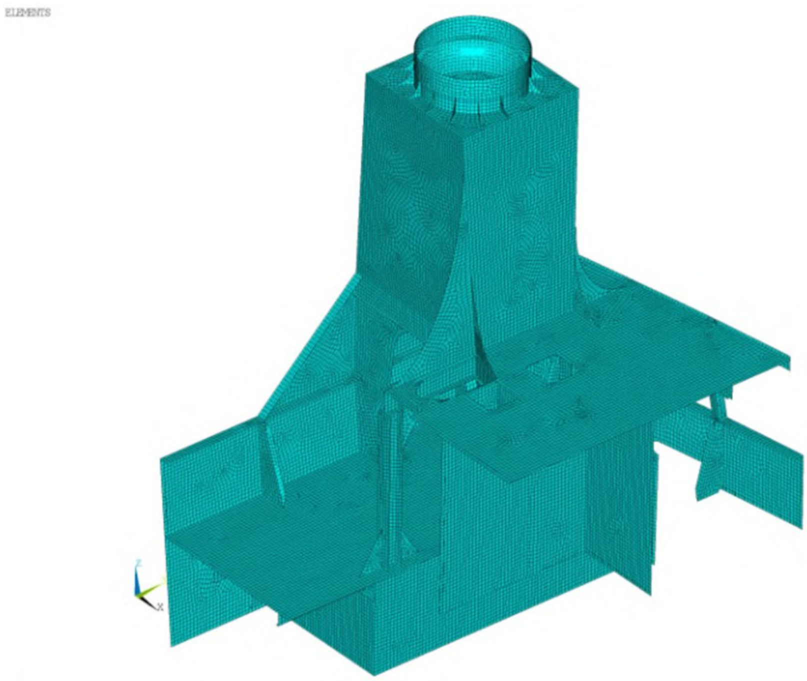

Figure 32: Mesh of a symmetrical structure for two WAPS units.

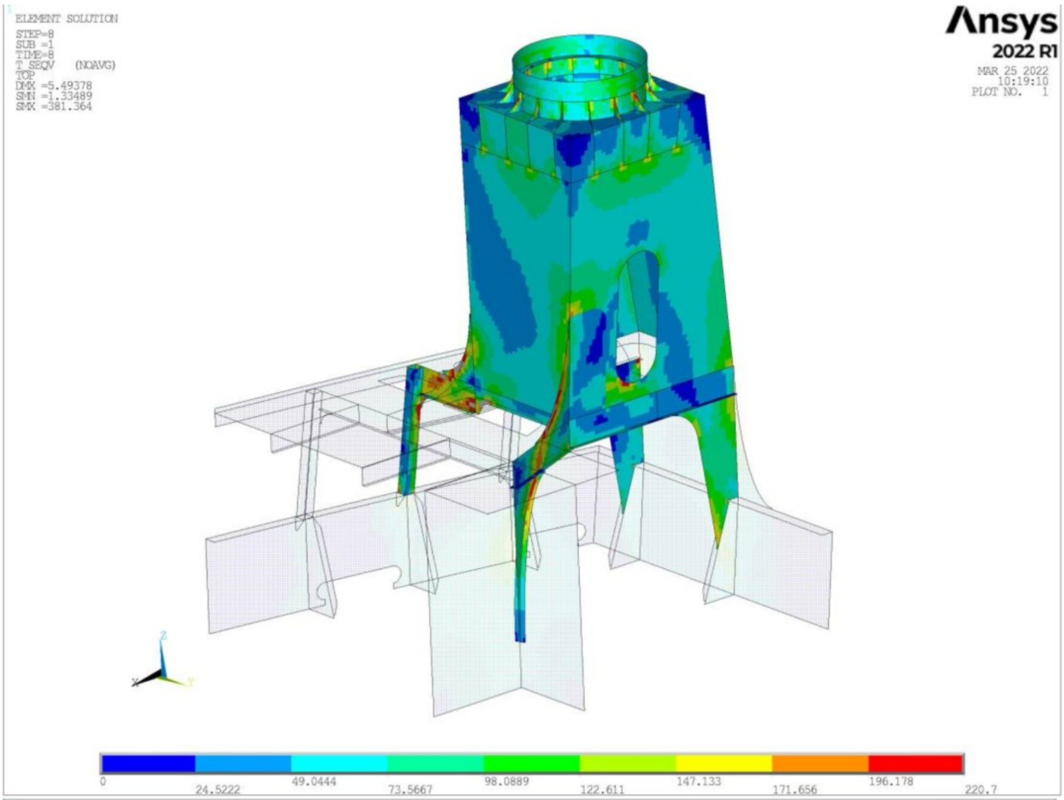

Figure 33: Stress FEM calculation results of a symmetrical structure placed on the aft.

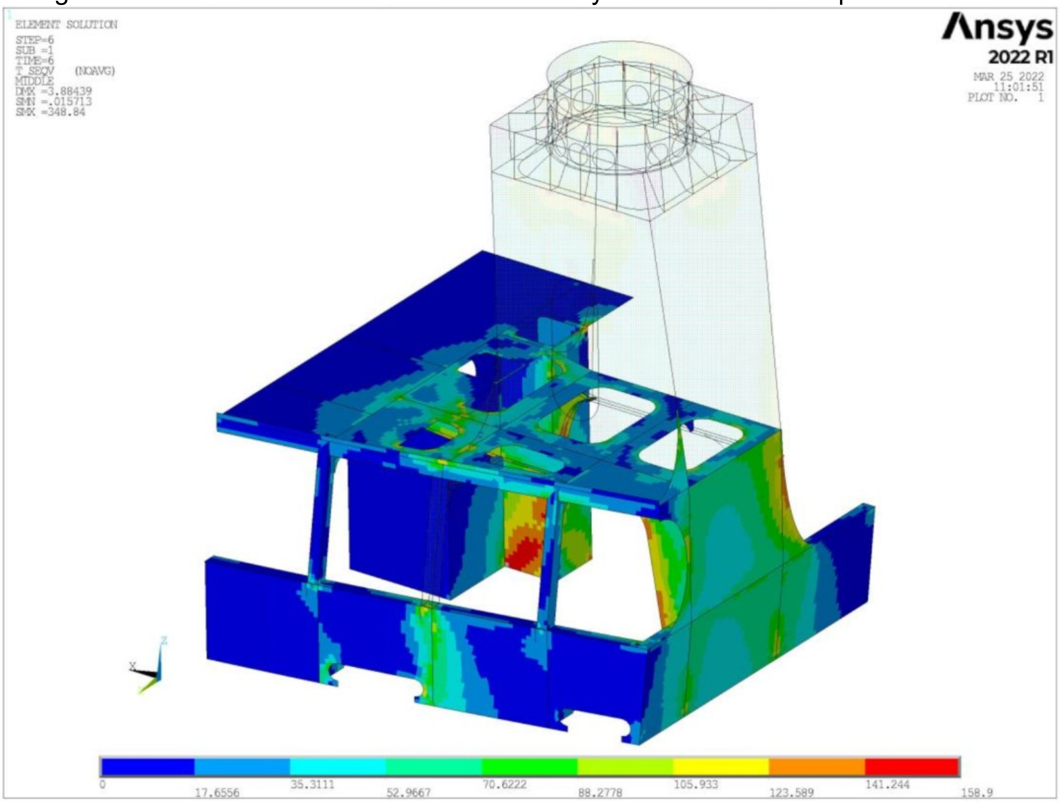

Figure 34: FEM calculation results of structure behind the eSAIL® foundation.

2.5 Interface requirements

The mechanical connection between ship and WAPS structure should meet the following requirements:

- Ease of installation.
- Possibility to remove for maintenance and repairs.
- Effective load transmission from WAPS main structure to ship primary structure.

Typically, WAPS systems have a circular slew bearing at the lower end, to enable orientation to wind direction. Ship structure is normally orthogonal, running in both fore-aft and athwart ships. While in new builds it is easier to build a cylinder integrated into ship main structure, for retrofitting purposes it is easier to adopt the correct footprint and weld on top of the existing primary structure. Some small reinforcements, like brackets, may be required to avoid stress concentrations, but the overall cost is reduced. A transition piece is therefore required from circular shape to rectangular, see Figure 35.

The most common structural interfaces are:

1. bolted connection;
2. welded connection.

Figure 35: Typical transition from circular to rectangular shape.[34]

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2.5.1 Bolted connection manufactured by yard

With this idea, WAPS system manufacturers only provide a drawing to the yard with the flange requirements. It is within the scope of a shipyard to fabricate the complete foundation with such a flange on the end.

A transition piece with bolted connection, like that shown in Figure 36, offers an easy way to be removed in a hypothetical maintenance situation while protecting the slew bearing during transport and installation operation.

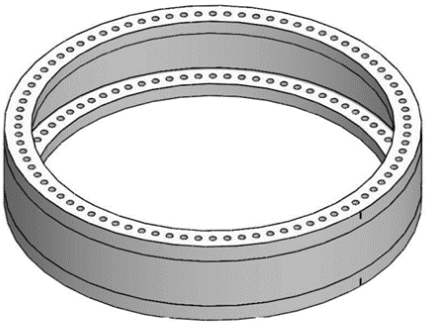

Figure 36: Double flange interface.

The ship flange requires a high degree of flatness and dimensional control to ensure compatibility with WAPS system flanges. Machining is therefore required, and if welding of the flange to the ship structure is not performed with care, deformation due to thermal tension may be excessive and re-machining in situ is required, **increasing costs and time.** Marks for proper installation are given (Figure 37).

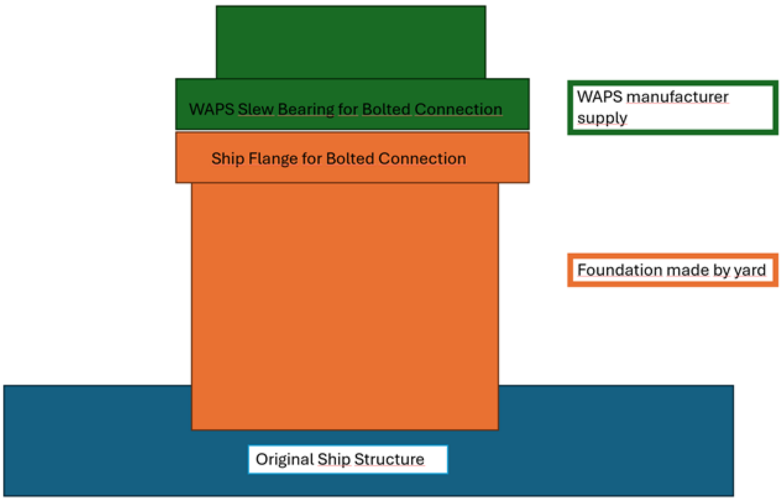

Figure 37: Bolted connection manufactured by yard.

2.5.2 Welded connection

Given the common practice of butt-welding plates or tubes at shipyards, it is recommended that the top part of the foundation be provided by the WAPS system manufacturer as a deliverable item. This piece shall have a flange on top end and open tube on the other end, prepared for welding. The shipyard's scope of work is limited to performing the butt weld on the remaining foundation sections that are welded to the ship structure.

The top part will be manufactured in accordance with the specifications set out by WAPS, ensuring that all requirements are fulfilled and controlled by the manufacturer to guarantee compatibility and reduce the risk of reprocessing at the installation stage. It is the duty of the shipyard to ensure that the pipes to be welded together are of compatible dimensions in order to guarantee the success of the foundation manufacture.

To enable the disassemble, the top part of the foundation still has a bolted connection, as shown in Figure 38, Figure 39, and Figure 40.

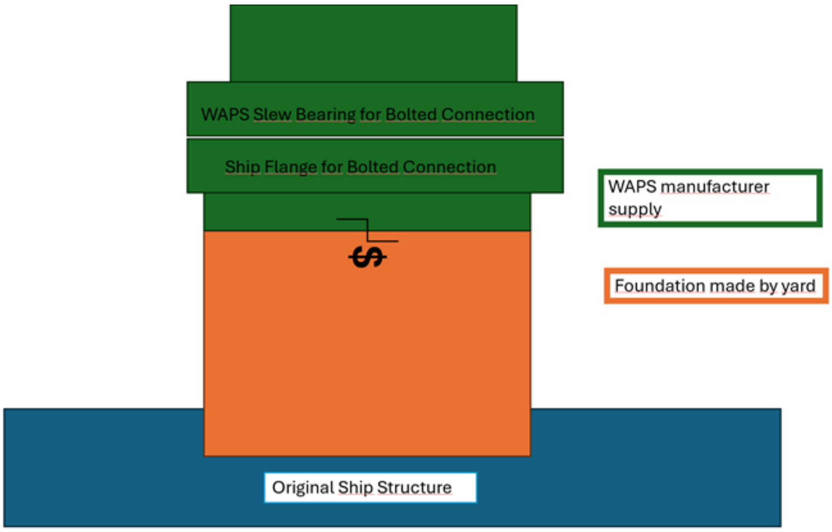

Figure 38: Top part of foundation supply of WAPS manufacturer.

Figure 39: Top part of foundation prepared for welding.

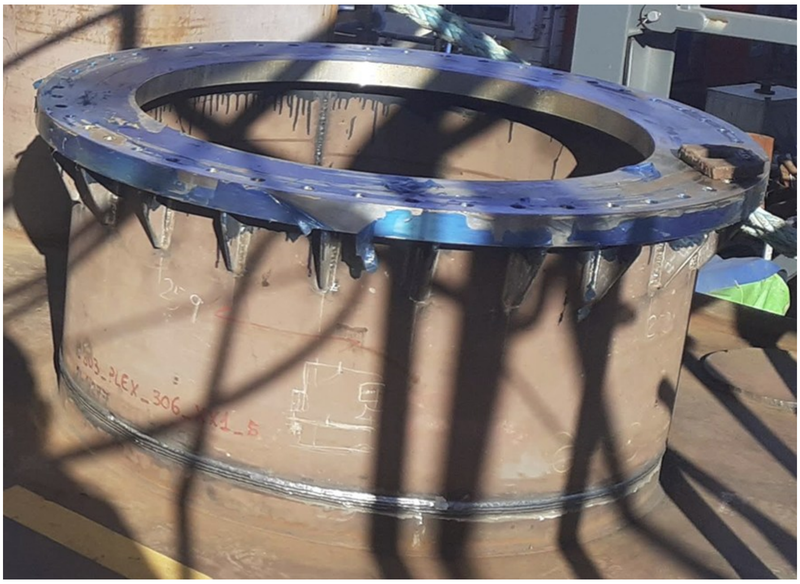
Figure 40: Top part of foundation welded to ship structure.

3 Closing remarks

Despite the initial goal of developing standardized reinforcement solutions based on ship typology, the findings of this project reveal that the considerable variability in vessel designs, coupled with differing regulations, renders the creation of standardized designs unfeasible. As outlined in sections 2.2 and 2.4, the analysis of load calculations and specific requirements across various ship types underscores the significant influence of vessel-specific parameters on the structural demands for reinforcements. This variability is observed not only across different ship classes but also within the same vessel, as illustrated by the notable load differences resulting from varying CCSS rules and positional changes onboard.

These findings highlight the challenges of creating a one-size-fits-all solution. The loads experienced by reinforcements can vary significantly even with minor changes in ship configuration or regulatory requirements. Therefore, any standardized reinforcement design would need to accommodate an envelope of maximum forces and moments to cover all possible scenarios. However, as explored in this study, such an approach would often lead to excessively conservative reinforcement solutions, resulting in increased steel weight, higher costs, greater emissions, and a reduction in payload capacity due to the added deadweight.

Given these limitations, the project has shifted towards establishing a set of minimum common requirements and developing typical reinforcement morphologies. This more flexible approach allows for adaptation to the specific needs of each vessel type, avoiding the imposition of a rigid, standardized design that may not be optimal in all cases. By focusing on shared foundation requirements and creating adaptable base morphologies, this strategy supports tailored reinforcement designs that can be adjusted to the unique structural and operational context of each vessel.

Moreover, the analysis emphasizes the importance of optimizing the life cycle performance of reinforcement solutions. A customized approach, as opposed to a standardized one, aligns better with the goals of reducing life cycle emissions and minimizing costs. Tailored designs allow for precise adjustments based on actual operational loads and structural needs, thus avoiding the overdesign that would result from standardization.

The outcomes of this project advocate for a continued flexible design approach, one that incorporates real operational data and the specific requirements of each ship type. This methodology ensures not only structural integrity and safety but also optimizes the economic and environmental performance of vessels utilizing rigid sails. Moving forward, it is recommended that the maritime industry adopts these findings and embraces adaptable design practices that can address the diverse and evolving landscape of ship designs and regulatory requirements.

In conclusion, while the pursuit of standardized reinforcement solutions remains an attractive objective, the complexity and variety of ship designs necessitate a more tailored approach. Defining minimum common requirements and type-specific reinforcement morphologies offers a balanced pathway, ensuring that reinforcement solutions are both effective and efficient across a wide range of vessel types and operational conditions.

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