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Standardized scalable feet solutions for offshore transformers

Department of Mechanical Engineering

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Abstract

This thesis investigates the design and feasibility of a standardized and scalable support interface (feet system) for offshore power transformers. Offshore transformer installations are exposed to complex combined load conditions, including static mass, platform-induced accelerations, internal electromagnetic vibrations, and transport-related loads. Existing feet solutions are often non-standardized, installation-intensive, and prone to issues such as bolt bending, misalignment, and inconsistent vibration performance. The purpose of this work is to develop a modular feet system that improves structural reliability, vibration control, and installation efficiency across a range of transformer sizes.

The study adopts a mixed-methods engineering approach combining literature review, industry input, analytical modelling, and finite element analysis. Two representative transformers within a mass range of approximately 20–85 tonnes are analysed to evaluate the performance of the proposed solution. Numerical simulations are conducted using finite element models, including stress, bolt, modal, and transport analyses, while analytical calculations are used to verify model behaviour and validate key results.

The results demonstrate that the proposed modular feet design provides adequate structural integrity under all investigated load cases in accordance with offshore standards. Stresses in structural components remain within allowable limits, and bolted connections function effectively as friction-based joints when properly preloaded. Modal analysis indicates that the system achieves sufficient separation between natural frequencies and dominant transformer excitation frequencies, enabling effective isolation of high-frequency vibrations while maintaining stability under low-frequency offshore motions. Transport analyses further confirm that dedicated transport supports ensure safe load transfer during handling and shipping operations.

The findings show that the developed feet concept resolves key limitations of existing solutions by reducing bolt bending sensitivity, improving load distribution, and enabling a more standardized and repeatable installation and manufacturing process. The design also supports scalability across different transformer sizes while maintaining consistent performance characteristics. Although the results confirm engineering feasibility, further validation through

experimental testing, measurements and detailed fatigue analysis is recommended before full-scale implementation.

Table of contents

1	Introduction	6
1.1	Background	8
1.2	Problem statement	11
2	Theoretical Background	16
2.1	Offshore/Marine Transformer Design Basics	16
2.2	Internal Vibrations	20
2.3	Existing Solutions	26
3	Methodology	34
3.1	Standards and regulations	37
3.2	Design Requirements	44
3.3	Load cases	45
3.4	Design Tools and analysis	49
4	Design & Analysis	53
4.1	Different foot element options	53
4.2	Foot and beam design	55
4.3	Structural analyses setup	70
4.3.1	Stress analysis setup	73
4.3.2	Bolts analyses setup	75
4.3.3	Modal analysis setup	78
4.3.4	Transport analysis setup	78
4.4	Analytical tools	80
5	Results & Discussion	84
5.1	Results	84
5.1.1	Light transformer	86
5.1.2	Medium Transformer	111
5.1.3	Compliance with standards and goals	138
5.2	Discussion	138
5.2.1	Stress and deformation	138

5.2.2	Bolt performance and optimization	141
5.2.3	Modal performance	144
5.2.4	Transport performance	146
6	Conclusion & Future Work	148
6.1	Future work	153
	References	155

Use of Artificial Intelligence

This thesis made limited use of artificial intelligence (AI) tools to support the research and writing process. Microsoft Word's built in tools were utilized for language refinement, including sentence structuring, grammar improvement, and clarity enhancement. In addition, AI (Copilot) was occasionally used to assist in identifying general background information and guiding the exploration of relevant topics.

All technical content, engineering analysis, interpretations, and conclusions presented in this work are the sole responsibility of the author. The use of AI did not replace critical thinking, independent verification, or adherence to academic standards. All sources of factual information have been verified against reliable scientific literature, standards, and technical references, and are cited accordingly in the thesis.

1 Introduction

Transformers are essential components of modern electrical power systems, enabling efficient transmission and distribution of electrical energy through voltage transformation. In offshore applications, transformers serve as critical interfaces between generation systems, power electronics, and end-use equipment. Unlike onshore installations, offshore transformers are subjected to complex environmental and operational conditions, including platform motion, vibration, corrosion, and constrained installation environments. (Ziomek, 2013; Heathcote, 1998) Offshore applications are considered to include wind, marine or oil/gas rig instalments.

The increasing deployment of offshore wind farms, floating production systems, and electrified marine assets has intensified the technical requirements imposed on transformer support structures. In these environments, transformer support systems must withstand combined static and dynamic loads, including gravitational forces, wave-induced accelerations, transport loads, wind loads and internally generated electromagnetic vibrations. At the same time, they must ensure reliable load transfer, maintain structural integrity, and minimize vibration transmission to surrounding structures. (Siemens Energy, 2026; RWE, 2024)

Existing transformer feet solutions in offshore applications are typically project-specific, installation-intensive, and not optimized for combined loading conditions. These limitations often result in issues such as bolt bending, misalignment during installation, and inconsistent vibration behaviour. Furthermore, the lack of standardization increases engineering effort and introduces uncertainty in early project phases.

This thesis addresses these challenges by investigating the feasibility of a standardized and scalable transformer feet system for offshore applications that integrates structural design, vibration control, and bolted joint verification within a unified engineering framework. The objective is to develop a modular support concept capable of providing reliable structural performance, controlled vibration behaviour, and improved installability under realistic offshore conditions. The proposed solution is evaluated using a structured verification and validation methodology combining finite element analyses (FEA) and analytical calculations, ensuring its applicability to realistic offshore load conditions and transformer configurations.

Hitachi Energy

Hitachi Energy, a global energy-technology company headquartered in Zurich, Switzerland, is partnering on this thesis. The company, officially incorporated on 30 June 2021, focuses on electrification and power-grid infrastructure, leveraging ABB's former Power Grids business and Hitachi Ltd.'s technology and ownership. Its current structure developed through the launch of the Hitachi–ABB Power Grids joint venture in July 2020, the evolution to the Hitachi Energy brand in 2021, and the transition to full ownership under Hitachi after ABB divested its remaining stake in 2022.

Hitachi Energy's offering spans core technologies across the power value chain, with a strong emphasis on high-voltage equipment, transformers, reactors, grid integration systems, grid automation and lifecycle services. In transformers, the company provides a broad portfolio (including liquid-filled and dry-type transformers) and supports these with services aimed at maintaining reliability and performance over the asset lifecycle. It also positions digital and monitoring capabilities as part of its transformer ecosystem (e.g., digital management and condition-based approaches), supporting modern grid and industrial asset-management needs.

In offshore and marine markets, Hitachi Energy's role extends to electrification solutions for vessels, ports, wind farms and offshore industrial assets, where reliability, compactness, and harsh-environment readiness are critical. For marine vessels, Hitachi Energy offers transformers designed specifically for marine installation constraints (limited space, demanding safety requirements, and reduced noise and vibration needs) that are approved against various global marine classification standards. For ports and terminals, the company provides shore-to-ship power and "smart port" solutions that connect vessels to the port grid at berth, reducing local emissions as well as noise and vibration, and enabling more integrated electrification and automation of port power infrastructure. In offshore oil and gas, Hitachi Energy provides a power delivery system that brings electricity from shore (or generators) to fixed and floating oil and gas rigs.

Within offshore wind projects, electricity moves to the shore and Hitachi Energy's offshore contributions to wind farms can be framed around grid-connection, wind turbine

transformers and offshore substation infrastructure. The company highlights High-Voltage Direct Current (HVDC)-based offshore connections, particularly for wind farms far from shore, where offshore converter stations and compact offshore station solutions support efficient transmission and grid integration.

1.1 Background

Offshore and marine transformers are specialized power components that are adapted to the difficult conditions of the sea environment. In floating offshore substations and rigs, transformers are mounted on deck structures. These experience 6-degree-of-freedom motions and sustained inclinations.

In fixed wind, export voltage levels and the quest for weight reduction create compact, high-performance packages that must still withstand transport, lifting, and storm loading. In floating wind, the same transformers must tolerate inclination and motion while keeping vibration and noise within acceptable bounds (DNV, 2010). In ports and shipyards, the rise of shore-power systems brings large receiving transformers and cable-management equipment on or near decks, adding operational loads and access constraints to be handled by the supporting structure (Dow, 2006).



Figure 1 Substation platform at the Baltic Sea 2 offshore wind site. (Froese, 2016)

Even when the surrounding structure is stiff, transformers are not “quiet” machines. Their cores expand and contract with each half cycle of the magnetic flux (magnetostriction), and windings see electromagnetic forces that vary with current, together producing a tonal vibration signature typically dominated near twice the grid frequency, around 100 Hz on 50 Hz systems and 120 Hz on 60 Hz systems, with additional harmonic content above. This vibro-acoustic excitation wants to travel into the supporting structure, where it can amplify if local modes sit near the tones. The feet design therefore has to balance stiffness (to keep natural frequencies away from excitation frequency), damping or isolation (to limit transmissibility), and practicality for marine operations. The challenge intensifies on electrically propelled vessels and converter platforms, where power-electronic drives alter current waveforms and introduce harmonic forces that can broaden the excitation spectrum; in those cases, the feet and surrounding grillage must be checked against both narrow-band tones and broader dynamic inputs. (Fang et al. 2024; Lehtonen, 1996)

Feet are the structural and vibro-acoustic interfaces that connect a transformer tank to a deck, grillage, or foundation, carry a disproportionate share of the reliability burden. They must transmit dead weight and operational loads without excessive deflection; withstand transportation, and installation events; and resist long-term corrosion, all while enabling maintenance access and, where required, vibration isolation. For fixed offshore substations, deck accelerations from wind and waves propagate into the transformer supports; for floating assets, six-degree-of-freedom motions and sustained tilt make relative movement control and bolt-preload retention more demanding, and suppliers of floating-ready transformers explicitly call out these motion/acceleration constraints. (API, 2014; Braithwaite, 2019; NORSOK, 2022)

Because installation realities at sea can exceed in-place operating levels, transformer feet must also be engineered for temporary but severe load cases during load-out, sea transport, and lifting. (DNV, 2015) On floating assets, the combination of motion and periodic inspection access constraints pushes designers toward interfaces that are both stiff (to control relative motions) and maintainable, with clear paths for inspection of weld toes and for replacing shims or isolators if required.

Corrosion is an equally decisive lifecycle constraint. Marine atmospheric and splash exposures concentrate attack at edges, bolt holes, and crevices, exactly the places most common in feet, soleplates, and chock interfaces, so coating system choices, edge-retentive primers, stripe coats, and surface preparation quality have outsized effects on maintenance intervals. Supplier system guides widely used in offshore sectors detail coating families and thickness schemes tailored to these details, and they have increasingly been adopted by offshore wind owners seeking extended maintenance-free periods. Where dissimilar metals (for example, stainless fasteners in painted carbon-steel plates) or marine splash zones are involved, designers often introduce insulating sleeves/washers or ensure electrical continuity to manage galvanic couples, aligning the mechanical details of the feet with corrosion-control intent. (NORSOK, 2022)

Weight, footprint, geometric imperfections and access further shape the interface. Offshore developers and naval architects push for smaller, lighter transformers to free up deck space and crane capacity, but lighter tanks and higher flux densities can increase vibro-acoustic sensitivity, forcing greater care in how loads are introduced into grillage through feet and shims. Transformer manufacturers for marine service emphasize compact, modular designs and specialized cooling (e.g., water-cooled dry types) precisely because space is limited and noise/vibration expectations are stringent in accommodation-adjacent machinery rooms; at the interface, that translates into thermal clearances, drainage, and protected cable routes that do not bypass any isolation the designer introduces. (Hitachi, 2026) On the ship-to-shore side, shore-power case histories show that the receiving transformer and associated equipment can add double-digit megawatt capacity to a berth, with cable-management systems applying non-negligible operational loads, again a reminder that the supporting feet must be robust not only for steady running but also for handling. (Dow, 2006)

Pulling these threads together, offshore and marine transformers are needed wherever voltage adaptation, isolation, or conversion bridges are required between generation, propulsion, auxiliaries, and the grid, whether on a fixed jacket in the North Sea, a floating platform in deep water or the main deck of a cruise ship in port. They bring with them predictable vibro-acoustic tones around 100/120 Hz, auxiliary cooling vibrations, the prospect of harmonic forces from converters, and the inevitabilities of motion, shock, corrosion, and tight access. A credible foot design therefore behaves like a small system in its

own right: it must place loads into the deck without local buckling or unacceptable deflection; keep modal frequencies comfortably away from tonal excitation or, where space allows, incorporate well-damped isolation; provide replaceable shims or snubbers for transport and survival cases; and embed corrosion control into geometry and coating details that can actually be fabricated and inspected offshore. Addressing those constraints early, armed with the physics of transformer excitation, realistic transport/installation envelopes, and the practicalities of maintenance, installation and manufacturability, turns the interface from a hidden risk into a durable, certifier-friendly part of the power system.

1.2 Problem statement

Currently transformer feet solutions used in offshore and marine environments (fixed platforms and floating structures) are non-standard, installation-intensive, and often insufficient for combined load cases (static mass, dynamic motions, vibro-acoustic excitation, transport, and installation). This leads to recurring issues such as misalignment and bolt shear/bending, rework, and schedule risk. The thesis addresses the design and feasibility of a standardized, scalable feet system for light to medium sized power transformers (~20–85 t) that ensures structural integrity, vibration control, corrosion resistance, and safe, repeatable installation across offshore use cases, with an option to use different modular anchoring/anti-vibration options as per customers' demands.

Research questions

The thesis is guided by the following research questions (RQs):

- RQ1 (Load Cases & Motions): What combined static and dynamic load cases, including 6-DoF platform motions and transformer-induced vibrations, govern the structural design of transformer feet?
- RQ2 (Vibration Control): Which dynamic stiffness and damping characteristics minimize vibration transmissibility from the transformer across relevant excitation spectra offshore, and how can these be realized in modular feet subcomponents?
- RQ3 (Bolt/Shear Integrity): How can the feet design eliminate, reduce, or robustly tolerate shear/bending failure modes observed in current solutions, while maintaining assembly practicality and manufacturability?

- RQ4 (Selection & Standardization): What decision rules enable consistent module selection across transformer sizes and offshore scenarios?

Objectives

The overarching objective of this thesis is to design and validate a standardized, scalable, modular feet system for offshore transformer installations specifically oil rigs, floating substations, and marine vessels that achieves reliable structural behaviour, controlled vibration transmission, and well-planned manufacturability and installation.

1. Architecture and Modularity: Develop a pre-engineered feet module architecture with interchangeable subcomponents suitable for the defined transformer mass envelope and offshore deck interfaces.
2. Structural Integrity: Establish load paths and connection designs that maintain bolt integrity and bending capacity under combined static mass and dynamic motions, including transport and maintenance handling.
3. Vibro-Acoustic Performance: Define and achieve dynamic stiffness/damping targets that limit vibration transmissibility at transformer operating frequencies (50/60 Hz) and dominant offshore excitation bands.
4. Implementation Deliverables: Deliver a reusable design library (models, drawings, selection criteria) and a manufacturing guide.

Scope

This thesis focuses on the feet system and their mechanical interfaces to offshore decks on oil rigs, floating substations, and marine vessels.

The scope encompasses:

- Mechanical design of feet modules and its supports structures, anchoring provisions, isolation/anti-vibration elements, and transportation elements.
- Consideration of offshore motions and combined load cases (operational sea states, transformer-induced vibrations, transport/handling).
- Feet system manufacturability

- Installation processes, jacking sequences, tolerance controls, error-proofing and QA steps.

Limitations

The following aspects are outside the thesis scope or treated only as boundary conditions:

- Redesign of transformer active parts (core, windings) and the full deck/foundation redesign beyond interface specifications.
- Detailed cost benchmarking and supply-chain optimization (economic implications may be discussed qualitatively).
- Full-scale offshore trials: validation is primarily via analysis and structured reviews with manufacturers.
- Land-based seismic-only applications (referenced for contrast where helpful but not pursued as primary design drivers).
- Vibration loads induced by non-transformer-machinery

Assumptions

The development and evaluation of the proposed transformer feet system are based on several simplifying assumptions that define the scope of the problem. The transformer mass range is limited to approximately 20–85 tonnes, with representative configurations assumed to capture typical offshore applications. Offshore operating conditions, including platform motions and acceleration load cases, are treated as known inputs derived from standards-based design scenarios rather than site-specific measurements. The supporting foundation is assumed to provide sufficient global stiffness within defined limits, and its behaviour is represented through simplified boundary conditions in the analyses. Furthermore, external vibration sources from surrounding equipment are not explicitly considered, and the focus is placed on internally generated transformer vibrations and their transmission through the support system. These assumptions enable a consistent and scalable design approach while maintaining a manageable level of modelling complexity.

Methods & Data

This thesis employs a mixed-methods engineering approach to develop and evaluate a standardized transformer feet system for offshore applications. The methodology combines a literature and standards review, practitioner input, analytical modelling, and numerical simulation to ensure both theoretical grounding and practical relevance. The aim is to establish a structured and reproducible framework for translating design requirements into validated engineering solutions.

Relevant offshore and marine engineering standards, as well as scientific literature on structural dynamics, vibration transmission, and bolted joints, are reviewed to define the governing physical principles and design constraints. In addition to literature sources, internal engineering experience and practitioner input are utilized to identify common challenges associated with existing transformer feet solutions, including installation complexity, bolt performance, and vibration behaviour. This combined knowledge base is used to define design requirements and guide the development of the proposed concept.

The design process is supported by both analytical and numerical methods. Analytical models are used to describe fundamental load paths, fastener mechanics, and vibration transmissibility, providing reference values and physical interpretation for key response quantities. Numerical analyses are performed using finite element methods to evaluate the structural, dynamic, and transport performance of the proposed design under representative offshore load cases. The analyses include stress and deformation evaluation, bolt behaviour assessment, modal analysis, and transport load verification.

To ensure the reliability of the results, a structured verification and validation (V&V) approach is applied. Numerical results are cross-checked against analytical calculations, simplified mechanical models, and previously validated analyses to confirm their physical consistency. Model behaviour is further assessed through equilibrium checks, contact status evaluation, and sensitivity to load cases. This approach ensures that the simulation results are not only numerically stable but also physically meaningful and suitable for engineering decision-making.

Data used in this study are derived from a combination of internal documentation, external studies, engineering analyses, and defined load cases based on offshore design standards. Where applicable, assumptions and simplifications are explicitly stated to maintain transparency and reproducibility. Although experimental verification is not included within the scope of this work, the methodology is designed to provide a robust basis for future verification through testing and industrial implementation.

2 Theoretical Background

Rapid offshore electrification, spanning fixed and floating wind, oil & gas topsides, and marine vessels, has pushed power transformers into dynamic, space- and weight-constrained environments where their support interfaces must safely transmit vertical and horizontal actions through the life-cycle: manufacturing, land transport, sea transport, installation, operation, and maintenance.

Offshore transformer design differs from onshore design mainly because offshore environments are far harsher and more constrained. Offshore units must withstand constant humidity, salt spray, and corrosive marine conditions, requiring marine-grade coatings, sealed enclosures, and corrosion-resistant materials. They also face continuous vibration, shock, and platform motion, demanding reinforced mechanical structures and vibration-resistant components, loads that onshore units rarely encounter.

Space on offshore platforms is limited, so offshore transformers must be lighter and semi-compact compared to onshore transformers. However, electrical safety zones around the transformers are the biggest bottleneck regarding compactness. (Guidry et al., 2022; Reyes et al., 2014)

Offshore platforms frequently operate in hazardous zones, so transformers must comply with strict marine safety certifications. Because access is difficult and expensive, offshore transformers must be extremely reliable, require minimal maintenance, and support remote monitoring. By contrast, onshore transformers operate in milder environments, have easier maintenance access, and face fewer mechanical and regulatory constraints.

2.1 Offshore/Marine Transformer Design Basics

Foundation levelness and interface control

Bolted interfaces dominate the compliance of equipment–foundation systems; their contact stiffness and damping depend on flatness/roughness, clamping force, and contact pressure distribution rather than nominal plate thickness alone. Analytical–experimental work shows that deviation from flatness reduces fatigue strength by locally consuming bolt preload to “close the gap,” shifting stress into fewer threads and increasing prying action, effects that

accelerate crack initiation in flange-type joints (Söderlund, 2017). Models that embed joint surface stiffness (normal/tangential) informed by rough-surface contact mechanics reproduce measured joint behaviour more accurately than ideal rigid-contact assumptions; uneven pressure distribution around bolt patterns is a key driver of frequency-dependent joint stiffness and damping (Zhao et al., 2016; Lu et al., 2019). In practice, this literature supports specifying explicit flatness/levelness for the foot–deck interface and prioritizing stiff chocks or machined shims to preserve bolt preload and limit joint-induced flexibility. Foundations for offshore transformers are difficult and expensive to get perfectly flat which means shims or even solutions like RotaChock mounting plates are required (RotaChock, 2026). For this thesis no levelling shim models were added due to their negligible effect on numerical analyses, but they are largely used on bolted interfaces like offshore deck foundations.

Global stability under offshore conditions

From a structural-dynamics viewpoint, an offshore transformer supported on feet behaves as a multi-body system excited by deck accelerations and slow/fast platform motions. In offshore hydrodynamics, platform responses are commonly partitioned into low-frequency (LF), wave-frequency (WF), and high-frequency (HF) ranges (Table 1).

Table 1 Frequency classes for offshore environments (DNV, 2010)

Frequency class	Typical frequency range	Main load sources	Structural response characteristics
Low-frequency (LF)	Much lower than wave peak frequency	Mean wind and current loads; slowly varying wave drift forces; large scale platform motions (surge, sway, yaw)	Quasi static or slowly varying global response; inertia effects generally small; governed by overall stiffness and equilibrium
Wave-frequency (WF)	Around dominant wave frequencies (~0.04–0.5 Hz)	First order wave excitation forces; wave induced drag and inertia	Dynamic global response; potential resonance with structural natural frequencies; significant for fatigue and serviceability
High-frequency (HF)	Higher than dominant wave frequencies	Structural vibration; springing and ringing; local impact type wave effects	Local dynamic response; high accelerations; critical for local stresses, vibrations, and fatigue sensitive components

The WF band covers the first-order, wind-wave-induced motions and loads, i.e., the portion of the spectrum where incident ocean waves carry most of their energy and excite the platform at (or near) the wave peak frequency. DNV groups platform response explicitly in these three bands and identify WF as the region associated with first-order wave periods (with LF linked to slow-drift and HF to local/elastic responses). On offshore platforms, the

dominant vibration content sits in the wave-frequency band, typically $\sim 0.04\text{--}0.5$ Hz (periods $\sim 2\text{--}25$ s). (DNV,2010)

Physically, the WF band arises from linear wave–structure interaction: incident irregular seas, well represented by Pierson–Moskowitz (fully developed wind sea) or JONSWAP (fetch-limited North Sea) spectra, drive platform motions through frequency-dependent response amplitude operators (RAOs). Because the ocean wave spectrum peaks at periods of several seconds to a few tens of seconds, the largest first-order forcing typically lies in that range, and thus WF motions dominate the global accelerations measured on decks and topsides. (Stewart, 1997)

Textbooks and recommended practices describe this chain from environment $\rightarrow$ wave spectrum $\rightarrow$ RAO $\rightarrow$ response. Faltinsen’s *Sea Loads on Ships and Offshore Structures* develops the theory and applications of linear wave-induced motions (and their spectral treatment), showing how platform responses in heave/roll/pitch/surge/sway/yaw are primarily excited at wave frequencies. Chakrabarti’s *Hydrodynamics of Offshore Structures* presents the same framework and provides analytical and numerical tools for evaluating WF responses for both large-volume and slender structures.

Joint mechanics

Foundational texts and reviews consistently advocate long, elastic grip-length bolts to maintain clamping under cyclic shear, with the joint acting as a dissipative, pressure-dependent element whose impedance grows with pretension (Harris et al. 2010; Den Hartog, 1985). In bolted joints subject to transverse vibration, self-loosening is governed by microslip at the interface; controlled “Junker” experiments quantify preload decay versus cycle count. A Junker experiment is a standardized vibration test used in joint theory where a preloaded bolted joint is subjected to controlled lateral displacement perpendicular to the bolt axis while the clamping force is continuously monitored. The experiment demonstrates that preload loss is primarily caused by micro-slip at the contact interfaces, which reduces friction and enables incremental bolt rotation, rather than by axial vibration alone, underscoring that geometry, surface conditions, and locking strategy matter more than nominal fastener strength (Nassar et al. 2025). Friction between clamped parts can be increased through solutions such as friction shims (3M, 2023) . Together, these results argue

for (i) maximizing elastic clamp length and maximizing the stiffness of clamped parts at the feet, and (ii) validating the chosen anti-loosening concept under transverse vibration representative of service.

Transportation and sea-state loads

During lifting and voyages, the dominant excitations are broadband ship motions and occasional shocks; hydrodynamic theory predicts low-frequency, large-amplitude components (heave/roll/pitch/surge/sway/yaw) with velocity-squared drag effects contributing to damping and viscous loads on appendages (Faltinsen, 1990/1993). These motions produce quasi-static inertial load factors at the transformer centre of gravity that must be transmitted through the feet; the resultant cyclic load path is sensitive to whether the feet–deck interface is frictional (load transfer by bearing) or relies inadvertently on bolt shearing. The high mounting position of transformers on offshore rigs often further amplifies these inertial load factors. For design studies, seakeeping texts show why RAO-consistent acceleration spectra are preferable to single-value “g-factors,” especially for fatigue of weld toes in foot plates and for cumulative preload loss in fasteners (Faltinsen, 1990; Chakrabarti, 1987). However, RAO-consistent acceleration spectra is not a requirement in validating structural integrity as per offshore standards, such as DNV which utilizes acceleration loads on transformer centre of gravity.

2.2 Internal Vibrations

External vibrations from nearby machinery are unpredictable and rarely disclosed by customers, thus this thesis focuses on controlling internal transformer vibrations and trust other manufacturers to do the same. Managing these internal vibrations is crucial, especially for transformers operating near people. Internal vibrations fall into two main categories: magnetostrictive and electromagnetic.

Magnetostriction Vibrations

Magnetostriction is the field-induced elastic deformation of ferromagnetic materials. In power transformers, the laminations of the grain-oriented electrical-steel (GOES) core change length as the magnetic flux reverses, creating periodic strains that act as a distributed

excitation for the core structure (He et al., 2021; Fang, Ye, & Chen, 2024). This is the commonly audible “hum” that can be heard near transformers.

Magnetostriction can be approximated with:

$$\varepsilon_m \approx kB^2 \quad (1)$$

Where ε_m is the magnetostrictive induced strain, k is the material-dependent magnetostrictive coupling coefficient and B is magnetic flux density.

Because the magnetostrictive strain depends on the square of the magnetic flux density, the dominant vibration frequency is typically twice the supply frequency (100 Hz at 50 Hz systems; 120 Hz at 60 Hz), with higher harmonics shaped by material properties, joint geometry, clamping, and assembly tolerances (He et al., 2021; Siemens Digital Industries Software, 2021).

Recent experimental and modelling work clarifies that (i) the level of magnetostriction varies with material grade and texture (GOES anisotropy), (ii) core seams and T-joints often exhibit elevated local responses, and (iii) the modal characteristics of the core-frame assembly strongly influence the final vibration and radiated sound (He et al., 2021; Marks & Vitolina, 2017). Comprehensive reviews (Fang et al., 2024) and multi-physics studies (He et al., 2025) further show that magnetostriction interacts with Maxwell forces (especially under load and harmonics), leading to a combined excitation spectrum where no-load (core) and load-dependent (winding/leakage field) components both matter.

The transmission path typically follows: core laminations → core clamps / yokes → tank walls and stiffeners → foot elements and bolts → beam/foundation. Along this path, local resonances (e.g., tank panel modes) may amplify or attenuate specific harmonics, while joints and contacts (bolted interfaces, foot-to-beam plates) determine how much of the dynamic force “leaks” to the support structure (IEC, 2020; CIGRE, 2024).

Electromagnetic Vibrations

Under load, a transformer's windings carry alternating current that interacts with the leakage magnetic field in the window. The resulting Lorentz force density $f = J \times B$, where J is current density and B is magnetic flux density, generates radial and axial components that are pulsating at twice the network frequency and contain higher harmonics set by the current/flux waveform. These forces drive the windings into small but broadband vibrations even in normal service; during faults they rise with the square of current, producing large short-duration mechanical stresses. Numerous coupled electromagnetic–structural studies and standards document the mechanism and the dominant role of axial Lorentz forces in exciting the active part and the oil-coupled acoustic field inside the tank (distinct from core-steel magnetostriction, which dominates no-load hum). (Witczak et al. 2023; Jingzhu et al. 2015)

Once excited, the vibration energy propagates through the oil to the tank panels, which become the principal radiator; from there it flows into the support system via the base frame and feet. Measurements and full 3-D FE models indicate that winding-origin axial forces couple efficiently to tank modes, and the resulting pressure/structural waves reach the mounting points with strong components near 100/120 Hz (twice line frequency) and harmonics. This dynamic content at the feet appears as cyclic compression/shear in pads and alternating tension–shear in bolts, superimposed on static gravity and transport preloads. Under abnormal currents (e.g., external short circuits), the same Lorentz mechanism produces transient radial compression and axial compression/expansion in windings, with corresponding short-circuit impact that can transmit to the feet; transformer short-circuit design/verification therefore checks electromagnetic force paths and structural margins (e.g., per IEC 60076-5) to prevent excessive stress or local yielding that would amplify the transmitted vibration. (Kozupa et al. 2018; Witczak et al. 2023; Wang et al. 2023; Solanki 2022)

Table 2 Differences between electromagnetic vibration and magnetostrictive vibration

Vibration type	Magnetostriction	Electromagnetic
Transformer part	Occurs mainly in the core	Occurs mainly in windings & structures
Operation	Present at no-load	Dominant under load
Dependency	Material-dependent	Geometry and current-dependent

Mitigation and damping

Studies consistently report that 100/120 Hz dominates no-load (i.e. output side of the transformer is not connected) vibration, with measurable content at 200–600 Hz and beyond depending on material anisotropy, joint design, and modal shapes (He et al., 2021; Fang et al., 2024). Under load, electromagnetic forces in windings and stray-field coupling can produce additional tones and broad-band components that may rival or exceed core contributions (IEC, 2020; CIGRE, 2024).

Frequency ratios are used in numerous calculations regarding damping and isolation. It is calculated as follows:

$$r = \frac{\omega}{\omega_n} \quad (2)$$

Where ω is excitation frequency (For internal vibrations: $2 \cdot$ line frequency) and ω_n is the structures natural frequency. The natural frequency of a mechanical system is determined by the balance between stiffness and inertia. For a single-degree-of-freedom system, it is given by

$$\omega_n = \sqrt{\frac{k_{eq}}{m}} \quad (3)$$

where m is the mounted mass (including oil) and, k_{eq} is the directional equivalent stiffness of the feet set in the axis of interest.

For a mounted transformer idealized as a single-degree-of-freedom (SDOF) base-excited system, the force transmissibility is

$$T(r) = \frac{\sqrt{1 + (2\zeta r)^2}}{\sqrt{(1 - r^2)^2 + (2\zeta r)^2}} \quad (4)$$

where $\zeta = c/c_{cr}$ is the equivalent viscous damping ratio. This means that when $T < 1$ less vibration is transmitted to the support than is applied by the excitation. From this we can formulate that when $T = 1$ the frequency ratio is $r = \sqrt{2}$. When r is above $\sqrt{2}$ true vibration isolation can be reached and below that motion and force is amplified.

The acceleration transmissibility is:

$$T_a(r, \zeta) = r^2 T(r, \zeta) \quad (5)$$

which describes the ratio between the acceleration response of a vibrating mass and the acceleration of its supporting base and is therefore directly related to inertia-induced internal forces. Unlike force transmissibility, which tends to zero at high frequency ratios and enables true vibration isolation, acceleration transmissibility approaches unity as excitation frequency increases, indicating that mass acceleration follows base acceleration at high frequencies. Consequently, while damping reduces acceleration levels near resonance, it does not provide complete acceleration isolation.

These relations are the standard basis for specifying isolation in machinery mount design and are reproduced in vendor engineering notes and vibration primers used in projects (mass–spring–damper SDOF, transmissibility curves, degree of isolation). (Thomson, 1993; Rao, 2018)

Damping and isolation are important to distinguish from each other, as damping primarily influences vibration amplitude near resonance by dissipating vibrational energy and limiting peak response. However, damping does not affect the frequency ratio at which vibration isolation occurs. At frequencies above $r = \sqrt{2}$, increased damping can degrade isolation performance by introducing additional velocity-dependent forces. Consequently, damping

represents a design trade-off: higher damping improves resonance control, while lower damping enhances high-frequency vibration isolation. (Xiao et al., 2012)

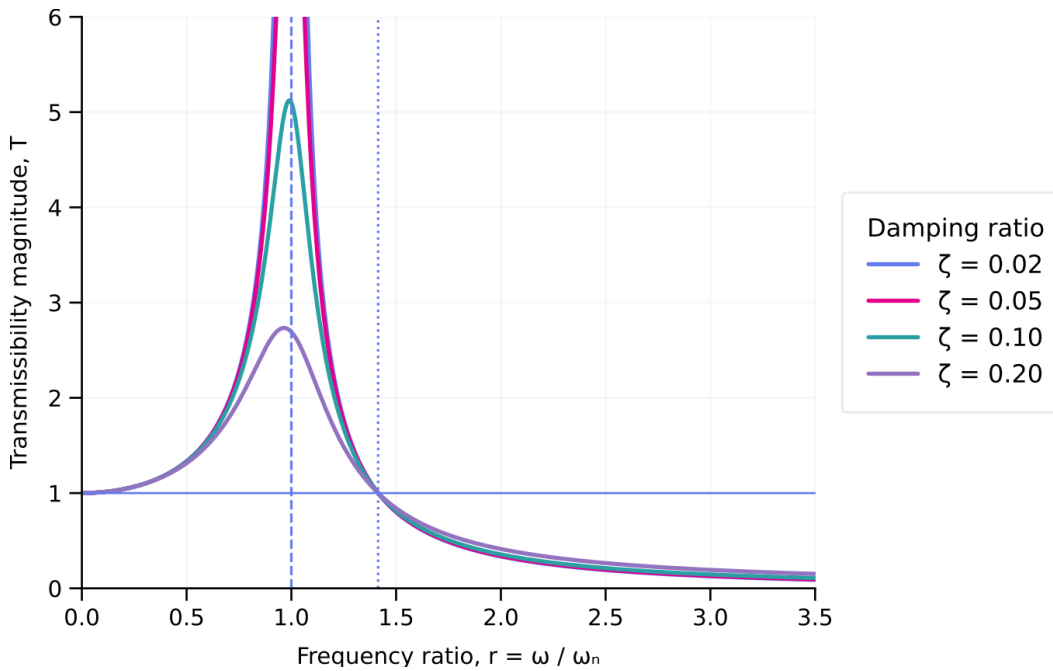


Figure 2 Calculated transmissibility magnitude of a single-degree-of-freedom system as a function of frequency ratio for different damping ratios. Calculated using Equation (4).

Closed-cell polyurethane elastomer bearings such as Sylodyn® HRB-HS provide high dynamic load capacity, low creep, and well-documented dynamic moduli and loss factors, enabling predictive tuning of the support's natural frequency and damping (Getzner, 2025). While solutions like Sylodyn are good at limiting resonance amplification making it a better damper, closed-cell elastomers tend to change in stiffness and damping over time.

(Barszczewska-Rybarek et al., 2022) Natural rubber is characterized by lower stiffness and relatively low damping, enabling efficient vibration isolation at frequencies well above resonance but leading to pronounced resonance amplification.

However, with excitation frequencies considerably lower ($r < 0.5$) than the natural frequency, as in the case of sea born vibrations, the response is mainly governed by system stiffness, and damping does not provide a significant improvement (Figure 2) (Roa, 2017) Due to the viscoelastic nature of rubber, mechanical energy dissipated through hysteresis over a wide frequency range, including low-frequency and quasi-static loading conditions. Rubber mounts help manage motion, stress, and fatigue in offshore systems, even if full isolation isn't possible. (Harris et al., 2010) In the stiffness-controlled regime (sea born vibrations),

softer elements increase displacement and decrease forces while harder elements reduce displacement but increase force. (Nasif et al., 1985) Due to its relatively low stiffness, rubber elements usually need more space compared to most closed-cell solutions. Closed-cell polyurethane elastomers combine solid elasticity with gas-cell compressibility, resulting in higher stiffness and increased damping. This reduces resonance response and improves load-bearing capacity in compact geometries, albeit at the expense of reduced isolation efficiency at higher frequency ratios. (Harris et al., 2010)

The practical significance is twofold:

1. Feet isolation can be effective for transformer born vibrations. The primary tones are well above the low-natural-frequency range attainable with heavy-duty elastomeric bearings under transformer masses (often 5–15 Hz or similar, depending on thickness and grade), so transmissibility can be pushed into the isolation region. (Getzner, 2025).
2. Sea born vibrations cannot be isolated due to their low frequency (0.04-0.5 Hz). These need to be managed with damping, tuning stiffness and avoiding resonance. (Harris et al. 2010)

The design goal is to achieve a frequency ratio of at least $r = \frac{\omega}{\omega_n} \geq \sqrt{2}$ at the dominant excitation tones (e.g., vessel heave/roll/pitch/surge/sway/yaw harmonics, magnetostriction or electromagnetic vibration). In practice offshore, teams often aim for $r \approx 2$ (or higher at critical tones) to give margin against variability in stiffness, mass, and damping.

2.3 Existing Solutions

Hitachi Energy has developed and deployed its own feet solutions to combat vibration in offshore environments. However, these solutions have their disadvantages and constraints. As a part of this thesis a survey was conducted regarding existing offshore feet solutions and their installation processes.

Standard offshore solution

The standard offshore solution has a simple beam structure that is welded on Sylodyn based damping feet elements. It is cheap to manufacture and relatively easy to install. However, for harsh offshore environments its problems outweigh its positives.

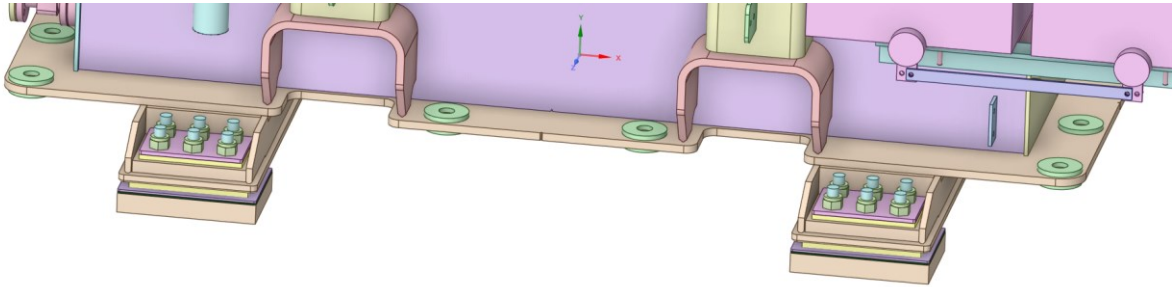


Figure 3 Standard offshore solution (Hitachi Energy)

The foot element consists mainly of two metal plates, a pad plate and a foot plate, a Sylodyn pad and six stud bolts that keep them together (Figure 3). This arrangement works well for vertical loads, but as Sylodyn is a relatively soft elastomer it deflects more over time and gives way when swaying is added.

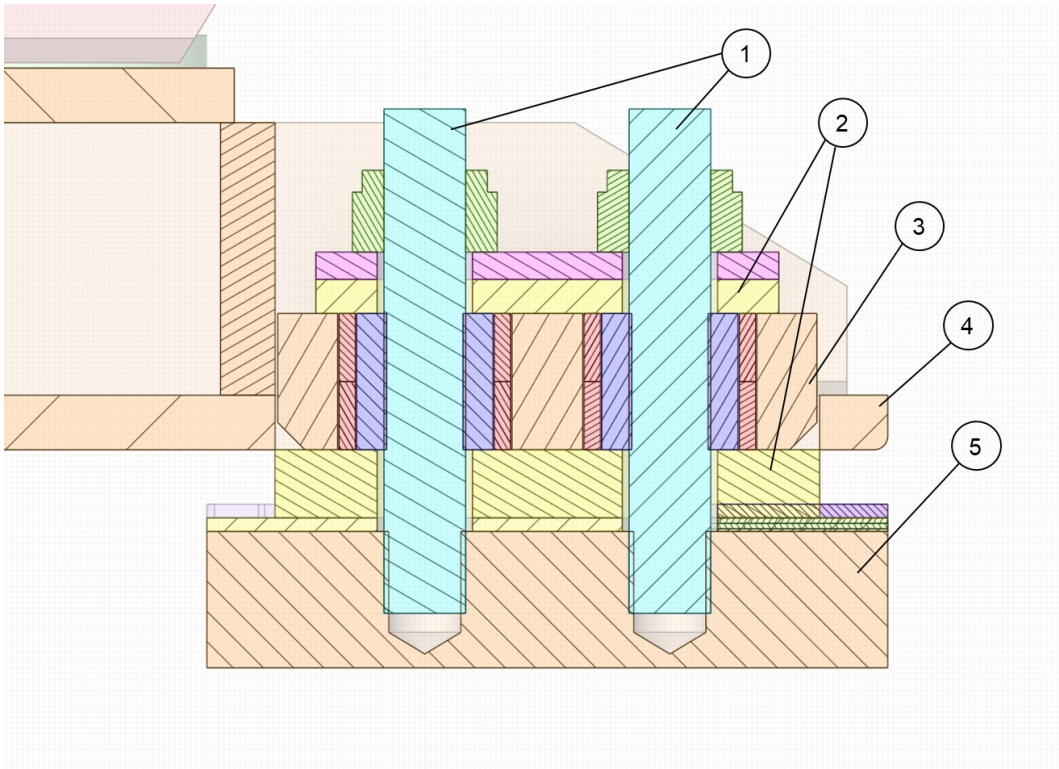


Figure 4 Cross-section of standard offshore solution foot element. 1. Stud bolts, 2. Sylodyn pads, 3. Foot pad, 4. Bottom plate of beam structure, 5. Pad plate (Hitachi Energy)

Wave movements produce a horizontal load that is applied straight to the stud bolts seen in Figure 4. In a normal metal to metal bolt connection this wouldn't be a problem because correctly pretensioned bolts would create enough friction force to counteract the horizontal load, but here because there is a soft material between the plates elastic displacements are generated. This leads to increased bending fatigue in the stud bolts, which in turn means increased maintenance as bolts need to be tightened to lessen fatigue or replaced completely. As the Sylodyn compresses and becomes harder over time and when the bolts are tightened it will also lead to unpredictable damping properties.

Shear stoppers

Shear stoppers are the next step in the evolution of the standard offshore solution. It shares much of its architecture with the previous solution but adds vastly enhanced shear/bending capability with the cost of size, complexity and ease of installation.

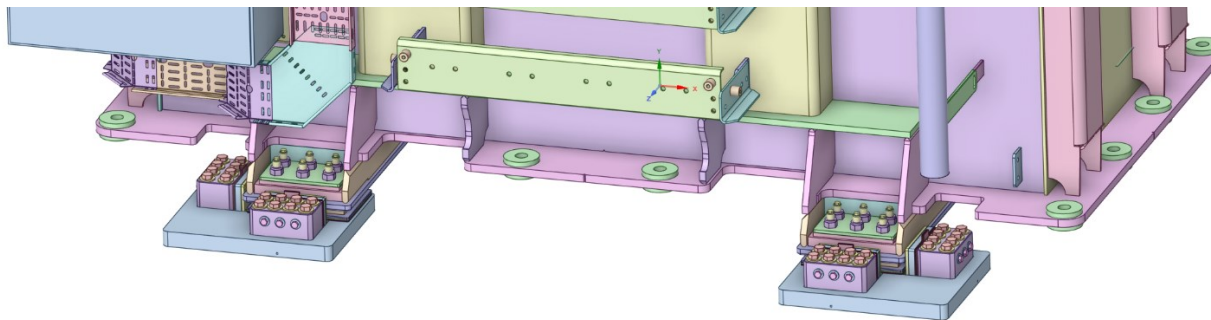


Figure 5 Shear stopper solution (Hitachi Energy)

Shear stoppers utilize the same Sylodyn-based damping beneath the feet and are further equipped with shear stopper blocks positioned on the outer sides, separated by an intermediate copolymer layer. Copolymer layers on the side are backed up by stopper blocks that are bolted down to the pad plate. The blocks restrict foot plate movement, reducing stud bolt bending fatigue, but their hardness limits vibration performance. In addition to having its damping capabilities reduced, it is also large and requires more time to install which can be a meaningful factor on tightly scheduled dockyards with limited lifting equipment.

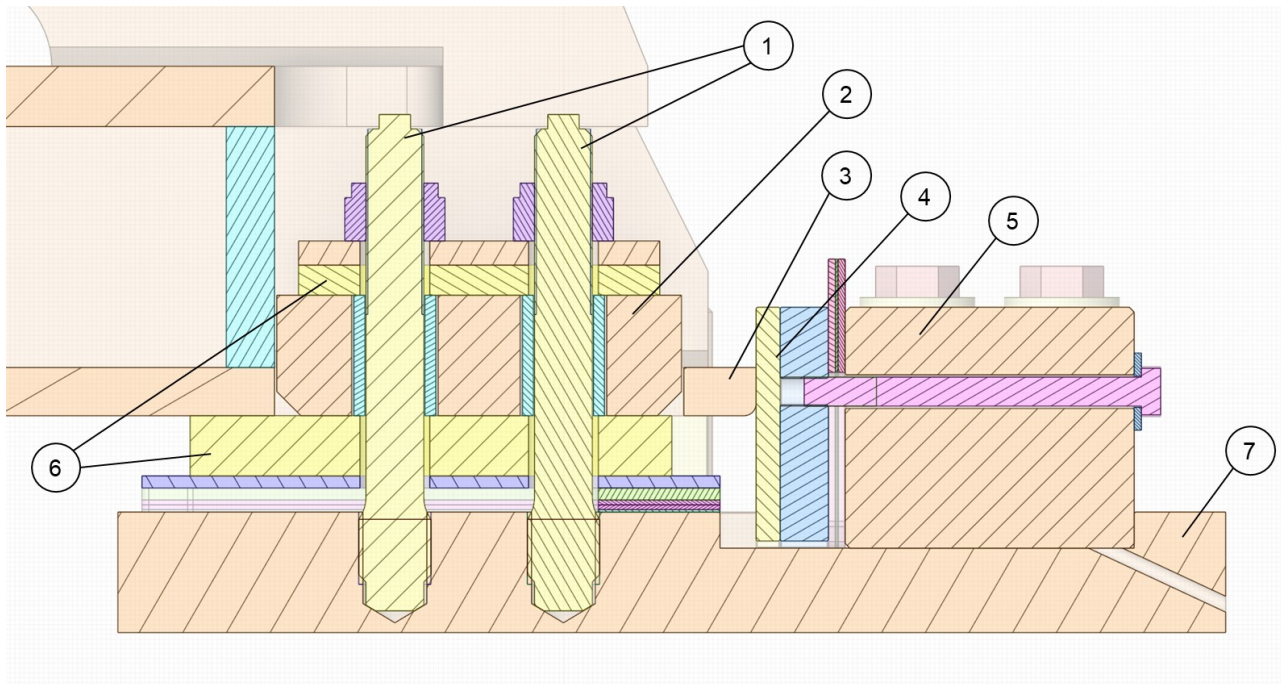


Figure 6 Cross-section of shear stopper. 1. Stud bolts, 2. Foot plate, 3. Bottom plate of beam structure, 4. Copolymer layer, 5. Stopper block, 6. Sylodyn pads, 7. Pad plate. (Hitachi Energy)

In terms of installation, the standard offshore solution and shear stopper follow a similar plan. The feet are assembled into the feet pads while the transformer is jacked up, when ready it is lowered on the foundation and the pad plates are tack welded to stay in position while the feet are disassembled and jacked up again. Pad plate is then welded to the foundation and when it has cooled down the transformer can be lowered back down and feet reassembled.

Shear pin solution

The shear pin solution is a newer solution to combat the issue of swaying. It has a slightly different beam structure to the others and has a large, centred pin that is designed to take horizontal forces away from the stud bolts that keeps it secure to the pad plate. Similarly to the others its main dampening component is a Sylodyn pad between the feet pad plates.

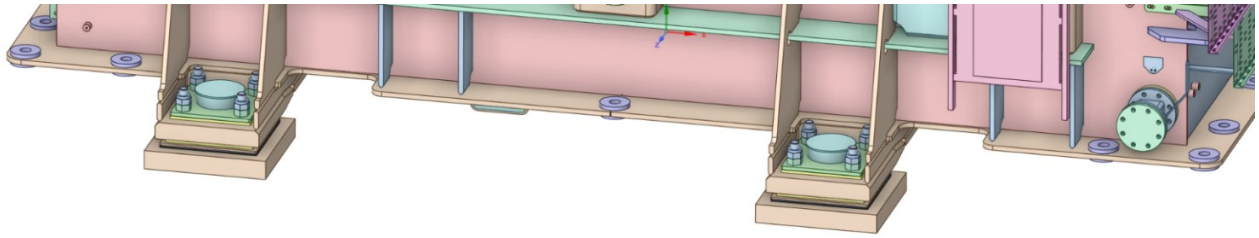


Figure 7 Shear pin solution (Hitachi Energy)

Between the pin and the feet pad there is a sleeve made of Polyoxymethylene Copolymer (POM-C) to prevent metal to metal contact which would be detrimental to the damping performance of the foot. POM-C is a semicrystalline engineering plastic with excellent machinability and high thermal and dimensional stability. Above the sleeve and pin there is a cap to keep the sleeve secure and protect it from Ultraviolet light which it is sensitive to. The shear pin has an extremely tight fit with both the upper and lower parts, the lower part needs to be installed using liquidized nitrogen.

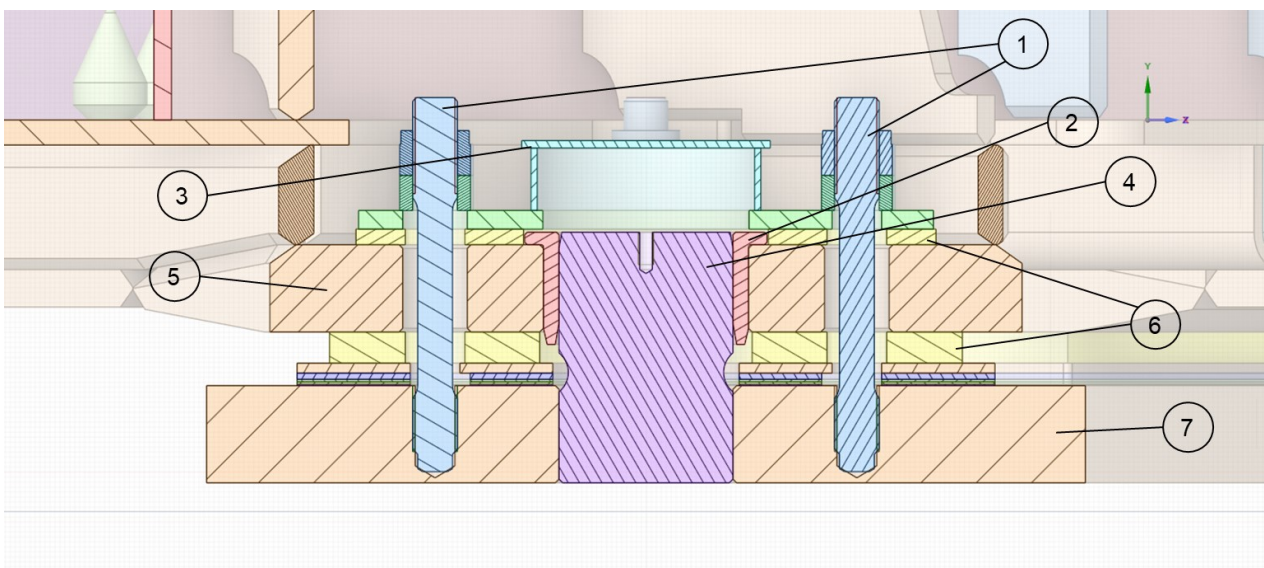


Figure 8 Cross-section of shear pin solution foot element. 1. Stud bolt, 2. POM-C sleeve, 3. Cap, 4. Shear pin, 5. Foot pad, 6. Sylodyn pads, 7. Pad plate. (Hitachi Energy)

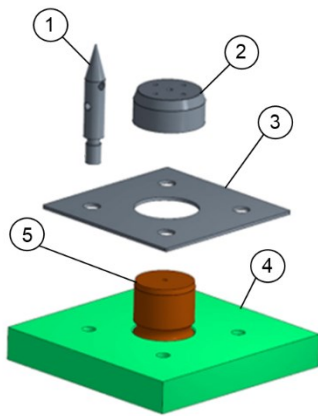


Figure 9 Explosion view of shear pin solutions installation guides. 1. Guide pin, 2. Guide cap, 3. Sylodyn, 4. Shear pin, 5. Pad plate. (Hitachi Energy)

Installation is planned in a way in which guiding pins and a guiding cap are used to force the foot plates to align correctly when the transformer is lowered on them (Figure 9). At this stage, the Sylodyn pad is positioned on the pad. After this the stud bolts and POM-C sleeve should fit perfectly. However, because the sleeve fits so tightly, the guiding cap must also be extremely tight. This makes it challenging to accurately align it with the pin hole while ensuring that the guide pins pass through their designated openings. The guiding cap has been previously taken off for the installation, but this means that a new looser sleeve had to be used leaving more clearance in the shear pin. Furthermore, this installation process requires substantial resources to execute

effectively, which may prove challenging in a busy dockyard environment.

Table 3 Main advantages and disadvantages of each design

Type	Disadvantages	Advantages
Standard offshore	High levelness is needed for bolts to work correctly, Bolts bend during horizontal load Does not fulfil DNV requirements Questionable vibrational performance Low pretension results in loosening of bolts	Good damping
Shear stoppers	Bulky, cumbersome to install, small amount of bolt bend, vibration properties unverified Questionable vibrational performance Low pretension results in loosening of bolts	Can handle high shear forces
Shear pin	Difficult to install, unverified vibration properties Questionable vibrational performance Low pretension results in loosening of bolts	Can handle heavy vertical and horizontal loads

Across all solutions there are three main problems, either the bolts are vulnerable to bending fatigue, or they have a stiff connection in their structure which can conduct vibrations into the foundation or vice versa. And finally, the installation processes are cumbersome and require on-site welding. These are the main structural properties which need to be resolved in the standardized scalable solution.

3 Methodology

The methodology chapter describes the methodological framework used to develop and verify the offshore transformer feet design that includes the feet elements, fastener interfaces, manufacturability and foundation assumptions. The aim is to provide a repeatable, and auditable description of how design requirements are translated into engineering decisions, how critical loads cases are defined and how the structural and vibro-mechanical performance is assessed. Because offshore applications combine stringent safety expectations with harsh environmental and operational conditions, the methodology emphasizes conservative assumptions where uncertainty is high, and a structured verification and validation approach. This verification and validation approach includes FEA with complementary analytical hand calculations to support design decisions and to verify numerical result.

Objectives of the new designs are to provide a pre-engineered feet module architecture that (I) handles the additional loads of an offshore environment, (II) has the vibro-acoustic performance needed to achieve isolation at operating frequencies, (III) is relatively easy to install and (IV) is easily implementable on variable transformer sizes and weight ranges. All are achieved with careful and iterative designing, and both (I) and (II) are verified to have succeeded through various structural and modal analyses. (III) and (IV) success are harder to quantify, but both are evaluated, (III) is done through comparing the expected installation process of the new design to older solutions and (IV) by assessing the commonality between the designs. Additionally, the design needs to follow applicable standards, for an offshore environment.

As said in the introduction chosen transformer mass envelope is 20-85t. Two different transformers were chosen, to represent the opposing ends of the mass envelope. Both transformers were originally designed to use older feet designs and thus been analysed with them. This provides an already validated base analysis to compare results with and verify that the analyses work as intended. New feet solutions are designed to be usable on these but also be scalable to different sizes of transformers within the envelope, while satisfying as many design goals as possible.

Light transformer

For the lower end of the mass envelope a transformer with a mass of 24t was chosen, of which 5.9t is oil and 11.5t the active part. The transformer, as shown in Figure 10, was originally designed to use shear stoppers as feet as it is deployed on a wind farm substation located in the North Sea.

The transformer is 2.2 meters tall from bottom to lid and the bottom is 3.2 meter long and 1.56 meter wide.

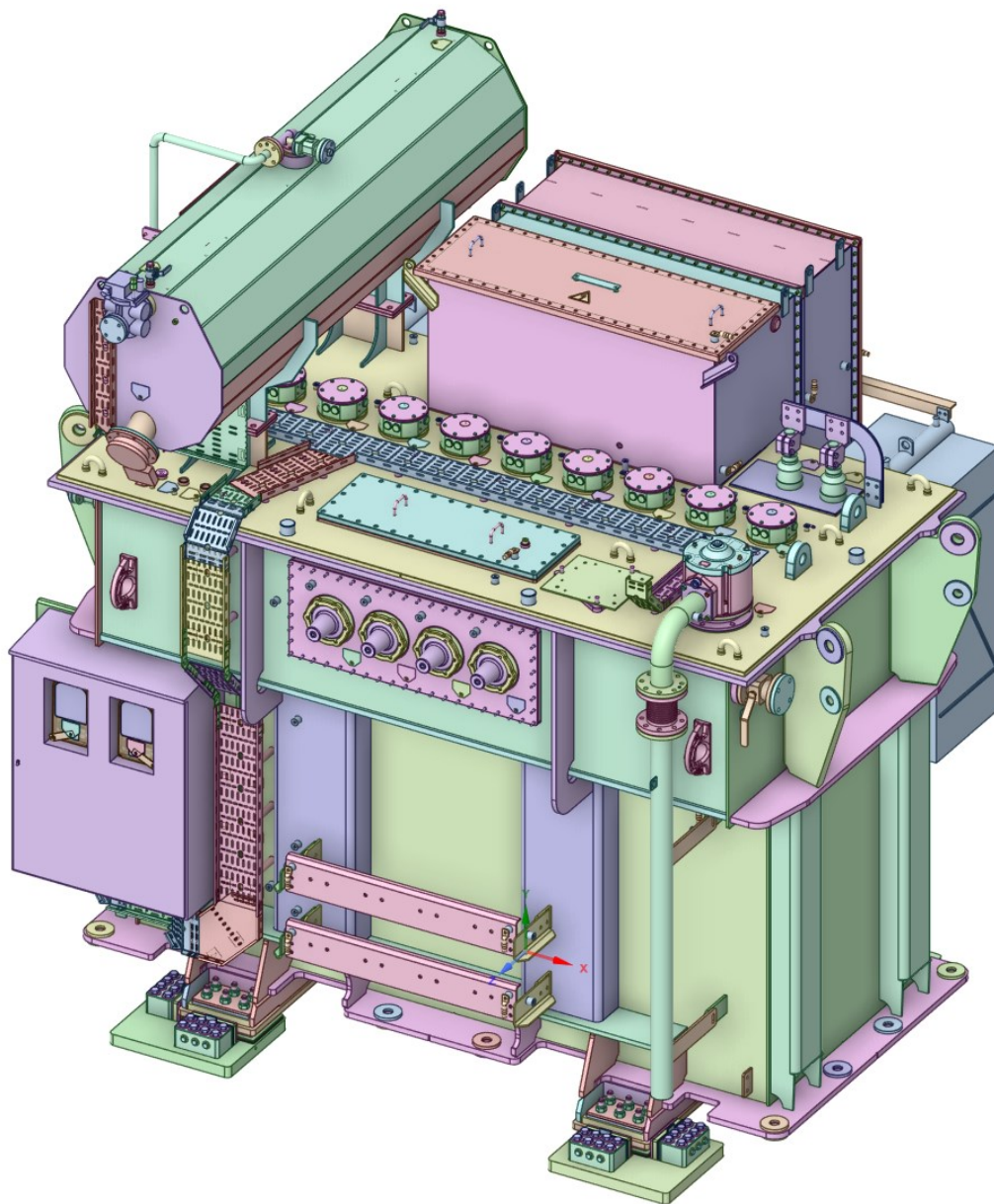


Figure 10 Light transformer (Hitachi)

Medium transformer

As for the other end of the envelope, a transformer with a mass of 83t was chosen seen in Figure 11. Its internal oils mass of around 21.2t and the active part comes in around 43.3t. This transformer used the shear pin solution as its anchoring and is deployed on a large Floating Production, Storage and Offloading (FPSO) vessel. FPSOs are used to in the oil and gas industry to process and store oil until it can be offloaded into a tanker. These vessels may themselves receive gas or oil straight from existing well centres or from a drilling platform. The bottom plate of the transformer is 2.54 meter wide and 5.01 meter long, with the cover being at the height of 3.6 meters relative to the bottom.

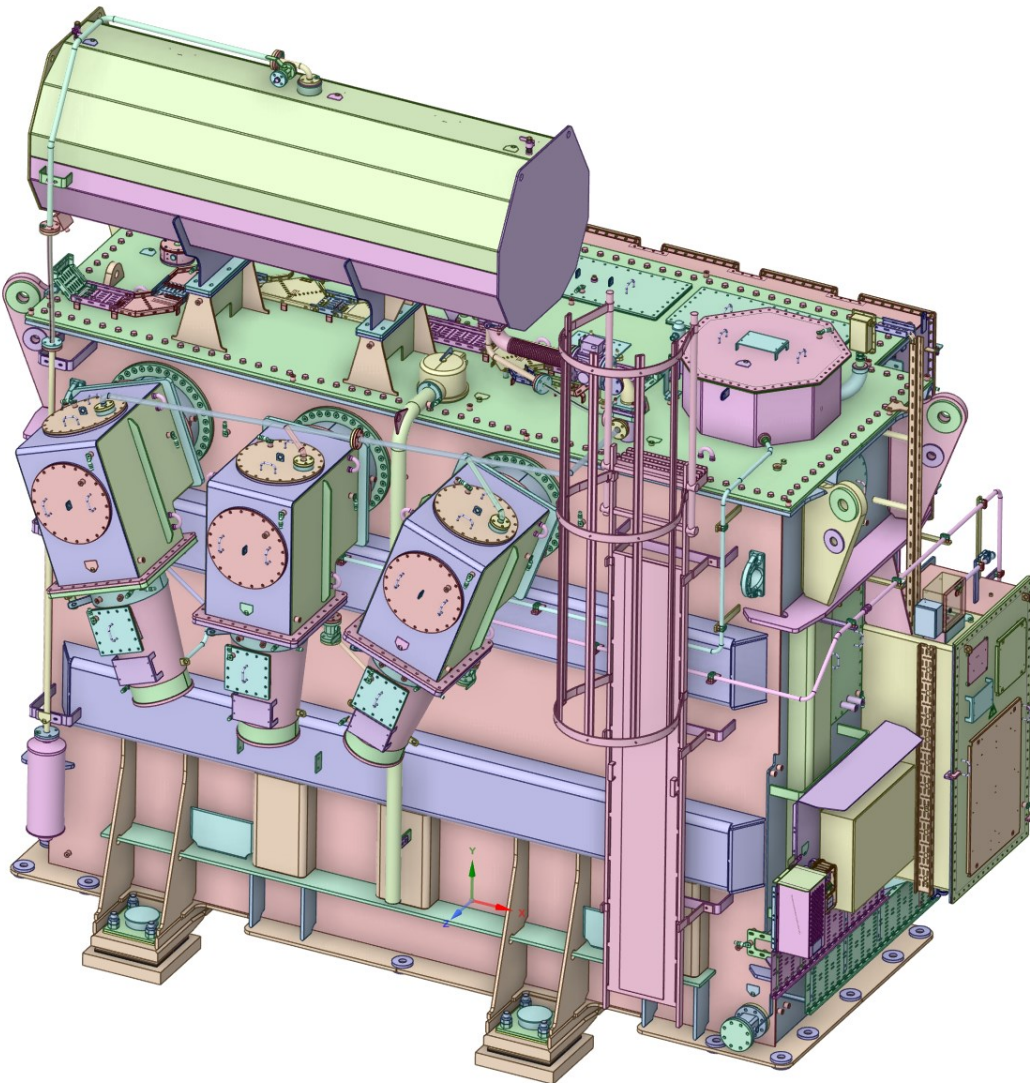


Figure 11 Medium transformer (Hitachi)

Table 4 Relevant information of both transformers

	Light transformer	Medium transformer
Mass (t)	24	83
Operating environment	Wind farm substation	FPSO-vessel
Dimensions (m)	3.2 × 2.2 × 1.56	5.01 × 3.6 × 2.54

3.1 Standards and regulations

Strict standards and regulations govern the design and qualification of the transformer support interfaces and their fastened connections. These documents are used to set acceptance criteria, define representative environmental and operational envelopes, and verify that the design is up to the task it was design for. For transformers there are multiple different standards and methods to achieve validity, but only a relatively small amount of them fulfilled. Offshore transformer design and verification sit at the intersection of electrical product standards, marine/offshore structural rules, lifting/transport codes, and hazardous-area certification schemes. In practice, projects apply a stack of standards that map to distinct verification activities:

- (i) electrical design/manufacturing/testing (IEC 60076 series)
- (ii) structural design and global/local integrity for lifting, sea transport, and in-place operation (DNV-ST-E273, DNV-OS-C101, VDI 2230)
- (iii) lifting device verification and operational handling (DNV-ST-E273)
- (iv) intermodal transport packaging/accelerations (CTU Code)
- (v) seismic qualifications where applicable (IEC 62271-207 and/or IEEE 693)
- (vi) explosive-atmosphere compliance (ATEX/IECEX).

Of these the most important standards for this thesis are (ii) structural design and lifting standards.

Standards are implemented to ensure compliance with local laws and regulations while accommodating customer preferences. DNV standards are often preferred by European offshore customers.

Environmental loads (DNV-RP-C205)

DNV-RP-C205 (Environmental conditions and environmental loads) gives a detailed framework for calculating the environmental loads in the areas the transformer is going to be transported to and operated. Environment is defined with significant wave height, wave periods and design sea state, additionally currents and wind speed are used if relevant. These are converted into platform motions using simplified motion models, RAOs or conservative motion envelopes. Acceleration at the installation location is calculated in vector form:

$$\mathbf{a} = \mathbf{a}_{\text{trans}} + \boldsymbol{\gamma} \times \mathbf{r} + \boldsymbol{\omega} \times (\boldsymbol{\omega} \times \mathbf{r}) \quad (6)$$

Where $\mathbf{a}_{\text{trans}}$ is linear acceleration (Surge, sway and heave), $\boldsymbol{\gamma}$ angular acceleration (rad/s^2), $\boldsymbol{\omega}$ is angular velocity (rad/s) and $\mathbf{r}$ is the position vector from reference point to transformer.

Loads are in the form of body accelerations or inertial forces that are used in compiling load cases to be analysed and verified according to DNV-OS-C101. These loads are provided by the client, thus for the validation process to work correctly it is required for the client to pass on accurate information regarding the conditions the transformer needs to bare.

Lifting and Sea Transport (DNV-ST-E273)

DNV-ST-E273 addresses the structural design and verification of portable offshore units with particular emphasis on transportation and offshore handling. The standard requires that realistic transport and lifting scenarios are defined and that corresponding inertia loads are applied to the structure for verification. Transportation loads are typically represented by equivalent static accelerations acting on the unit mass and permanent equipment, with load magnitudes justified using recommended practices such as DNV-RP-C205 or project-specific transport analyses. As a result, transportation load cases often govern the structural dimensioning of offshore transformer skids and support interfaces. Maximum allowable von Mises stress for transportation is calculated with:

$$\sigma_{transport,allow} = \alpha \cdot f_y \quad (7)$$

Where α is transport stress utilization factor and f_y is material yield strength.

In-place operation (DNV-OS-C101)

DNV-OS-C101 (Design of offshore steel structures, LRFD (Load and Resistance Factor Design)) is used for the structural design methodology, load and resistance factors, detail categories, fatigue classes, and acceptance criteria across ULS/FLS/SLS/ALS with acceptance criteria based on a linear-elastic strength approach (yield-stress limit). The structural validation in this work is primarily based on the Ultimate Limit State (ULS). ULS verification ensures that the structure can withstand extreme combinations of environmental, gravitational, and inertial loads without collapse, loss of load-carrying capacity, or instability. This represents the fundamental safety requirement for offshore structures, where failure consequences are severe and intervention possibilities are limited. Other limit states, such as Serviceability Limit State (SLS) and Fatigue Limit State (FLS), address functional performance and long-term durability respectively, but do not ensure survival under extreme loading. Accidental Limit State (ALS) verifications are intended to demonstrate robustness under rare abnormal events and are therefore considered supplementary to, rather than a replacement for ULS verification. The ULS analysis uses different load cases in a 3D FEM environment to verify structural stability. These load cases each have their own accelerations as per the OS-C101-standard.

DNV-OS-C101 does not prescribe fixed acceleration numbers. Instead, it requires you to derive the characteristic accelerations from a load-effect analysis (hydrodynamic/structural), using the project environmental basis (normally 100-year for ULS) and the combination rules in C101, with environmental inputs and methods taken from DNV-RP-C205. You then apply these computed accelerations at the transformer location for the ULS checks.

Design loads must not induce nominal von Mises stresses in the base material that surpass the yield strength divided by the relevant material resistance factor. Nevertheless, localised peak stresses often exceed this threshold; thus, plastic strain criteria are necessary in these regions. The analysis of bolts and welds is conducted in accordance with DNV-OS-C101,

applying the relevant material resistance factors. Maximum allowable von Mises stress in ULS cases is calculated with

$$R_{eHd} = \frac{R_{eH}}{\gamma_m} \quad (8)$$

Where R_{eH} is characteristic material resistance (MPa) and γ_m is material partial safety factor.

For welds in this thesis, all beams and plates are designed with chamfers and an assembling order to facilitate full penetration welding, which can, according to OS-C101 be considered as of the same part, thus welds do not need to be individually analysed.

Bolt joints

Where the mounting concept uses bolted interfaces, e.g., transformer feet to deck, tie rods, or auxiliary frames, VDI 2230 provides the systematic calculation method for high-duty bolted joints. The guideline formalizes the load-path model (spring model of bolt vs. clamped parts), assembly preload determination (torque/tension, scatter, embedding losses), utilization checks under external load (separation/slip) and fatigue verification for alternating loads. It also prescribes friction assumptions, clamp length ratios, and treatment of eccentric loading and multi-bolt patterns by decomposition into calculable sub-geometries. Applying VDI 2230 to offshore transformer feet ensures that preload, reserve against joint opening, and slip safety are demonstrated for the operational combinations defined by DNV (lifting, transport, in-place), while the capacity of the surrounding structure is checked to DNV-OS-C101.

VDI 2230 operates differently compared to more common standards like EN1993-1-8. For preload calculations the bolt is regarded as an elastic spring compared to Eurocodes strength element. Preload is determined so that the bolt connection becomes frictional without bolt utilization factor v rising above 1 even in a worst-case scenario. A nominal assembly preload is calculated, assuming mean tightening factor, tool accuracy and friction coefficients, using bolt specific dimensions, half of maximum embedding and utilization factor. From this nominal preload a maximum and minimum preload is determined using various factors, like tightening scatter, unfavourable conditions, suitability for high-reliability joints, variable friction conditions and different magnitude of embedding. Additionally, a probability factor of 90% is applied to both ends of the preload envelope, common for

structural joints. Minimum preload is used in analyses to check for worst-case slippage resistance, and maximum preload is used to check whether the bolt stress remains below $R_{p0.2}$, which is the minimum 0.2% proof yield strength of bolt material. For this thesis preload for bolts is calculated using internal calculation tool made by Hitachi Energy.

Combined stress check including internal and external stresses is done by combining all loads and comparing them to $R_{p0.2}$ and utilization factor v :

$$v \cdot R_{p0.2} = \sqrt{\sigma_w + 3 \cdot \tau_{max} + \sigma_{vths}} \quad (9)$$

Where σ_w is normal stress, τ_{max} is torsional stress and σ_{vths} is thermal expansion stress. Temperatures used in this thesis are 40 °C for operating and 20 °C for assembly. The Hitachi Energy bolt calculator automatically calculates the resulting forces, which are negligible ($F_{vths} \ll 1$ kN) for all bolts used because of the minor temperature difference and short bolt length. Therefore, these forces can be disregarded in stress checks.

Normal stress includes axial stresses and bending stress from external sources:

$$\sigma_w = \frac{F_s}{A_s} + \frac{M_b}{W_b} \quad (10)$$

Where F_s is axial loads, A_s is bolt stress area, W_b is bending modulus and M_b is external bending stress. Axial loads include preload (F_M) and external axial load (F_{SE}):

$$F_s = F_M + F_{SE} \quad (11)$$

Bending modulus (W_b) is calculated using:

$$W_b = \frac{\pi \cdot d_s^3}{32} \quad (12)$$

Where d_s is calculated using:

$$\frac{d_2 + d_3}{2} = d_s \quad (13)$$

Where d_2 is the pitch diameter of the bolt thread and d_3 is the minor diameter of the bolt thread. This is valid for normal shank bolts (not for reduced shank diameter bolts). External axial loads (F_{SE}) and external bending stresses (W_b) are extracted from the FEA results.

Torsional stresses are calculated using:

$$\tau_{max} = \frac{M_G}{W_P} \quad (14)$$

Where W_P is the polar moment of inertia and M_G is the thread moment. Polar moment of inertia can be calculated with either plastic or elastic properties. VDI recommends using plastic and it is calculated with:

$$W_P = \frac{\pi \cdot d_s^3}{12} \quad (15)$$

The thread moment is calculated using:

$$M_G = F_M \cdot \left(\left(\frac{P}{2\pi} \right) + \left(\frac{d_2 \cdot \mu_{G \min}}{2 \cdot \cos \frac{\alpha}{2}} \right) \right) \quad (16)$$

Where P is pitch of thread, $\mu_{G \min}$ is minimum friction coefficient on thread and α is flank angle of thread.

All equations combined the combined stress check formula becomes:

$$v \cdot R_{p0.2} = \sqrt{\left(\frac{F_M + F_{SA}}{A_S} + \frac{M_b}{\left(\frac{\pi \cdot d_s^3}{32} \right)} \right)^2 + 3 \left(\frac{F_M \cdot \left(\left(\frac{P}{2\pi} \right) + \left(\frac{d_2 \cdot \mu_{G \min}}{2 \cdot \cos \frac{\alpha}{2}} \right) \right)}{\frac{\pi \cdot d_s^3}{12}} \right)^2} \quad (17)$$

Vibrations

Offshore transformers are subject to both internally generated electromagnetic excitation and externally induced dynamic loads arising from platform motion, wind, wave action, and, where applicable, seismic effects. While no single standard defines general numerical vibration limits for normal transformer operation, vibration-related requirements are addressed collectively through IEC 60076 for transformer mechanical behaviour, IEC 61892 and DNV-OS-D201 for offshore environmental suitability, and IEEE 693 for dynamic and

resonance-related qualification of substation equipment. Numerical criteria exist only for special cases (seismic qualification, environmental testing, transport), not continuous operational vibration.

The FEA can be conducted to demonstrate compliance with needed transformer and offshore standards, which primarily specify functional and performance-based requirements rather than explicit numerical vibration limits. Modal analyses are used to identify the natural frequencies and mode shapes of the transformer support structure and to verify sufficient separation from dominant excitation frequencies associated with transformer operation and offshore environmental loading in line with IEC 60076 and IEC 61892. Static and equivalent dynamic stress analyses were applied to verify mechanical integrity and load-bearing capacity of the transformer feet and supporting structure, addressing requirements from DNV-OS-D201 and IEC 60076. Furthermore, the dynamic response of the structure can be evaluated using concepts consistent with IEEE 693 to assess amplification behaviour and ensure that vibration-induced stresses remain within acceptable limits.

Materials

International standards applicable to offshore transformers, such as IEC 60076, IEC 61892, and DNV-OS-D201, do not prescribe specific construction materials but instead define functional requirements related to mechanical strength, corrosion resistance, fatigue performance, and suitability for offshore environmental conditions. In practice, these requirements are typically satisfied by using structural carbon steels, such as S355 or equivalent, for transformer tanks, support frames, and feet, combined with qualified offshore corrosion protection systems corresponding to ISO 12944 corrosion class CX. Critical fasteners and anchor bolts are commonly realized using galvanized high-strength steels or stainless-steel grades (A4 or duplex) to ensure long-term resistance to corrosion and vibration-induced fatigue.

Where vibration isolation or damping is required at the interface between the transformer and supporting structure, elastomeric materials such as natural rubber or closed-cell polyurethane are commonly employed due to their well-documented dynamic properties and offshore durability. Compliance with the applicable standards is therefore demonstrated

through appropriate material selection, protective systems, and structural verification rather than adherence to predefined material grades.

3.2 Design Requirements

Building on this, the design requirements are presented to clarify the guiding principles applied throughout the work, such as robust load paths, manufacturability, maintainability, corrosion and fatigue considerations and a pragmatic balance between mass, stiffness, and damping. All features presented in Table 5 are treated as mandatory.

Table 5 Design requirements

Main Requirements	Goals
Structural integrity	1. Withstand respective operational load cases
	2. Withstand sea transportation load cases
	3. In-place bolt connections are frictional
Installability	4. 2x200mm spaces per long side for hydraulic jacking
Anti-vibration	5. Isolation from 100/120 HZ transformer vibrations
	6. Avoid resonance with dominant sea vibrations (0.04-0.5 Hz)
Manufacturability	7. All welds are full penetration weldable

Ensuring structural integrity is essential when designing a feet system. The system must be capable of withstanding all ULS load scenarios effectively. Bolt connections play a key role in maintaining this integrity and must be carefully examined. It is recommended that all bolt

connections are designed to be frictional to prevent shear/bending forces. Components are checked for maximum allowable von Mises stresses which are calculated using formulas from Chapter 3.1 according to their respective standards and factors from Table 6.

Table 6 Safety constraints used in verifying compliance.

Parts	Origin	Type	Factor
Structural steel parts (In-place)	DNV-OS-C101	Partial material safety factor	$\gamma_m = 1.15$
Structural steel parts (Sea transportation)	DNV-ST-E273	Transport stress utilization factor	$\alpha = 0.85$
Bolts	VDI 2230	Utilization factor	$v < 1$

Installability and manufacturability must be considered. The transformer requires four hydraulic jack locations: each long side needs two ~200mm-wide spaces free of beams or pad plates, positioned as to ensure stability during jacking. From a manufacturability standpoint the structure needs to be full penetration weldable to ensure high fatigue resistance, predictable structural behaviour and reduce sensitivity to weld orientation.

In the case of anti-vibration, the feet system should be able to isolate the transformer from the higher operating vibrations and avoid resonance with dominants sea vibrations. In principle this mean that the natural frequency of the transformer should be around 5-15 Hz and have some damping capability.

3.3 Load cases

In addition to permanent self-weight load, the feet design is governed by worst case accelerations that are formulated based on DNV-OS-C101 and DNV-RP-C205.

The two offshore transformers considered in this thesis are characterized by different masses and are intended for different operational environments. Consequently, the acceleration load cases applied in the structural analyses were selected based on their respective real-world usage conditions. Offshore design standards require acceleration loads to be representative

of the dynamic response of the specific system rather than prescribed as uniform values. The use of different acceleration load cases ensures that each design is evaluated under relevant and representative service conditions, while the underlying analysis methodology and acceptance criteria remain consistent.

Load cases are calculated based on location of the transformer (elevation from sea level), stability of the vessel/platform and projected wave and wind loads, all of which are provided by the customer. In addition, offshore transformers have sea transport loads that are usually noticeably higher than in-place loads. They are based on the same factors as in-place loads, but the loads are calculated based on the vessel transporting the transformer to its site. Table 7 shows the load cases used in this thesis, where Y is in the vertical direction, X is in the longitudinal direction and Z is in the transversal direction.

Table 7 Accelerations used to analyse in-place ULS performance for light and medium transformers.

Load case	Light transformer			Medium transformer		
	Accelerations (m/s ²)			Accelerations (m/s ²)		
	X	Y	Z	X	Y	Z
1	4.905	14.715	0	1.924	13.19	3.757
2	4.905	9.81	0	1.924	6.43	3.757
3	0	14.715	4.905	1.924	13.19	-3.757
4	0	9.81	4.905	1.924	6.43	-3.757
5	-4.905	14.715	0	-1.924	13.19	3.757
6	-4.905	9.81	0	-1.924	6.43	3.757
7	0	14.715	-4.905	-1.924	13.19	-3.757
8	0	9.81	-4.905	-1.924	6.43	-3.757

Sea transportation

For sea transportation the two transformers were analyzed using load cases prepared according to DNV-ST-E273. Transformer acceleration varies by location on the ship, and horizontal loads change with transformer direction in the vessel. Vessel rolling causes higher acceleration than pitching because of the vessel's shape.

Table 8 Sea transport accelerations applied.

Load case	Accelerations (m/s ²)			Case
	X	Y	Z	
1	0	19.62	11.772	Roll
2	0	0	11.772	Roll
3	0	19.62	-11.772	Roll
4	0	0	-11.772	Roll
5	9.81	19.62	0	Pitch
6	9.81	0	0	Pitch
7	-9.81	19.62	0	Pitch
8	-9.81	0	0	Pitch
9	4.905	19.62	9.81	Quartering
10	4.905	0	9.81	Quartering
11	-4.905	19.62	9.81	Quartering
12	-4.905	0	9.81	Quartering
13	4.905	19.62	-9.81	Quartering
14	4.905	0	-9.81	Quartering
15	-4.905	19.62	-9.81	Quartering
16	-4.905	0	-9.81	Quartering

3.4 Design Tools and analysis

For verifying performance of the design during aforementioned load cases, 4 different analyses are conducted using FEA. Stress analysis, bolt analysis, sea transport analysis and modal analysis are completed following the flow chart in Figure 12. PTC Creo 8 is used to model and assemble the design and Ansys SpaceClaim is used to simplify and prepare the model for analysis. The analyses themselves are conducted using Ansys Mechanical 2025 R2.

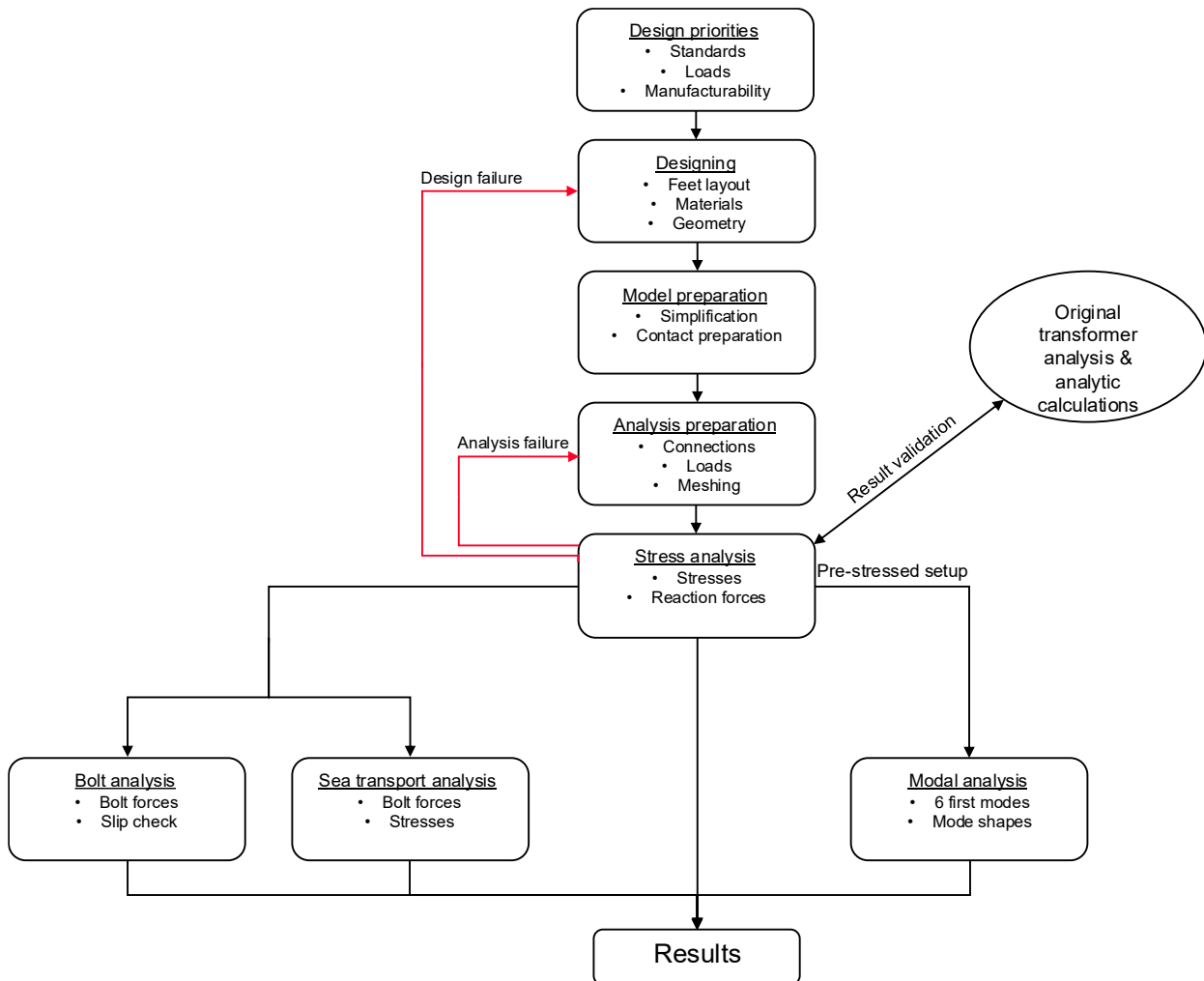


Figure 12 Flow chart of Finite Element Analysis process

Stress analysis is completed first and compared to the results from the original analysis for that transformer and simple force calculations. If the forces don't seem feasible or the design performs badly it is sent back to analysis preparation or designing phase. In-place stress is required to be measured only in stress analysis as it includes the point of highest stress located on the centre bolt surface and if the design has stresses above allowable limits in this analysis, it will be refined. When stress analysis is run successfully it can be used to run

modal analysis and to provide a base for bolt and sea transport analyses. Preload calculations done with Hitachi Energy bolt calculator give the minimum tightening load that are then used by both bolt and sea transport analyses. Maximum external axial load is determined first without any preload on the bolts, slippage and bending moments are analysed with preload active. Getting a good frictional bolt connection between stressed surfaces is extremely important for bolt integrity for high fatigue situations. All analyses that include bolts are run with the minimum preload and checked for slippage. For in-place load cases, if slippage is not observed the joint is satisfactory and can be checked according to Equation 17 with external axial and bending force combined with the maximum possible preload. This verifies that the bolt joint works as intended even in extreme cases. For sea transportation both highest and lowest preloads were analysed separately.

The specific purpose and values measured and evaluated in each analysis are listed in Table 9.

Table 9 Table of FEA to be conducted and what they measure (KPI).

Analysis	Purpose	KPI
Stress analysis (ULS)	Evaluate forces projected on feet elements	Stress on beams, Deformation of springs, Reaction forces on fixed supports
Bolt analyses (High/low preload)	Confirm that bolt connections hold	Bolt bending moment, Bolt axial forces, Connection slippage
Modal analyses	Deduce critical frequencies	First modes, Mode shapes, Effective mass
Transport analysis	Verify that the design can withstand sea transportation	Bolt bending moment, Bolt axial forces, Connection slippage, Stress on transport supports

To ensure that the results presented in this thesis are credible and suitable for engineering decision-making, the numerical work is supported by a structured verification and validation (V&V) approach. Key outputs from the finite element models are cross-checked using analytical hand calculations, previous analyses and simplified mechanics-based estimates to confirm magnitudes, trends, and governing load paths. In addition, model sanity checks are systematically applied, including equilibrium verification through reaction force balance, monitoring of contact status (separation/sliding) to confirm intended load transfer, and pretension verification to ensure that bolt loads and joint behaviour are represented

consistently. Together, these measures form an auditable V&V framework that links simulation outputs to fundamental mechanics and independent checks, thereby increasing confidence in the validity of the analysis results and the resulting design recommendations. V&V measures taken to ensure realistic results listed in Table 10, additionally contact tool in Ansys was utilized to check for realistic connections.

Table 10 Central V&V measures taken per analysis to verify and validate result.

Analysis	Verification	Validation
Stress and deformation	Sum of downward reaction forces are equal to the mass of the transformer; Compared to previous analyses	Rubber elements deform as predicted; Reaction forces are reasonable
Bolt	Preload shows in axial forces and is consistent	Friction takes most of the shear forces
Modal	No outstanding frequencies or inconsistent effective mass results	Single degree of freedom comparison with hand calculated frequency
Sea-transport	Sum of downward reaction forces are equal to the mass of the transformer; Bolt loads are plausible	Load case sensitivity; Load path verification

4 Design & Analysis

This chapter presents the development of the proposed transformer feet system and the analyses conducted to evaluate its performance under the defined design conditions. The design process is based on the requirements established in the previous chapters, including structural integrity, vibration performance, and installation feasibility in offshore environments. A modular feet concept is developed through iterative design, considering load paths, manufacturability, and compatibility with different transformer sizes. Chapter 4.1 and 4.2 addresses RQ3 and RQ4 by guiding the design to be as standardized as possible and providing a possible solution for eliminating excessive bolt shear, while Chapter 4.3 declares how RQ1, RQ2 and RQ3 are addressed with the different analyses. The performance of the design is assessed using FEA, including stress and deformation, bolt integrity, modal response, and transport load cases. These analyses provide the basis for verifying that the proposed solution meets the relevant design criteria and behaves as intended under combined static and dynamic loading conditions.

4.1 Different foot element options

To tackle the main problems of the existing solutions, bolt shear/bending and vibration issues, a search for manufacturers specializing in mounting equipment was conducted. Three different interesting manufacturers were chosen and contacted. All of them had the same basic shape designed to combat bolt shear forces.

VCE Techniek – GM 6000

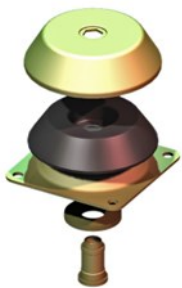


*Figure 13 GM 6000.
(VCE Techniek, 2014)*

VCE techniek from the Netherlands manufactures an anti-vibration foot solution that has the maximum static load capacity of 4500 kg and a natural frequency of 5,5 Hz (Figure 13). The cone shape of the foot allows the rubber part to take the horizontal loads. The center bolt can be tightened to the internal foot threads without any damping material in between meaning that the horizontal loads can be negated with friction connecting the foot plate and the “hat” of the foot. Additionally, the foot can be supplied with a height adjustment feature. (VCE Techniek, 2014)

Paulstra V318-D



Figure 14 V318-D. (Paulstra, 2026)

Paulstra from France manufactures a heavier foot with the load range of ~250-7000 kg (Figure 14). It uses a stainless-steel wire mesh as cushioning which makes it highly durable. However, it has a higher natural frequency compared to rubber based anti-vibration elements at around 11-17 Hz. (Paulstra, 2026)

Vibratec HD Marine

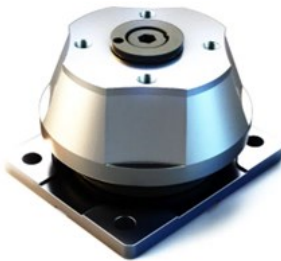


Figure 15 HD Marine. (Vibratec, 2026)

HD Marine (Figure 15) is a foot manufactured by Vibratec from Sweden. Like GM 6000, HD Marine uses natural rubber but has a more cylindrical shape compared to the cone shape of GM 6000. With its hardest natural rubber option, it has a load range of ~3000-6600 kg and maximum shock load it can withstand at 150 kN. Its natural frequency is 8-12 Hz. (Vibratec, 2026)

For this thesis VCE Techniek GM 6000 was ultimately chosen as the vibration element to be used, because of its low natural frequency and economic implications. As all search results found only relatively low load solutions, VCE Techniek was asked to model a possible future GM 12 000 foot with a static load capacity of 10 000 kg and 50% size increase to all dimensions. Relevant information that was given by VCE Techniek for the thesis:

- 3D model of GM 6000
- Spring coefficients used to represent the feet elements (Table 11)
- Commenting on analysis results

Due to the cone shape of the foot element two different spring coefficients were needed, horizontal and vertical (Table 11). Each element has three springs where two are horizontal and one is vertical.

Table 11 Spring coefficients provided by VCE Technik.

Model	Horizontal spring coefficient (N/mm)	Vertical spring coefficient (N/mm)
GM 6000	5625	6250
GM 12 000	9680	10 300

4.2 Foot and beam design

The number of feet was first calculated to be six for the light transformer, using information from the previous analyses on the transformer. This design was deemed to be too demanding for the feet by the manufacturer due to excessive deflection and was thus increased to eight. After eight feet was found to be satisfactory through analyses, it was used to choose the amount of feet for the medium transformer by scaling it with the increased carrying capacity and mass of the GM 12 000 and transformer mass respectively. Thus, the medium transformer was modelled with 12 feet modules and the light transformer with eight feet, with their respective rubber element sizes. This means that the light transformer should have four beams and medium transformer six. For six beams to fit on the bottom of the medium transformer “double beams” are needed where two rubber elements share a foot plate and a horizontal beam, meaning the actual number of beams goes down to four.

Table 12 General information about new feet solutions on both transformers. Masses for transformers are from general arrangement drawings.

	Light transformer	Medium transformer
Mass (t)	24	83
Operating environment	Wind farm substation	FPSO-vessel
Dimensions (m)	3.2 × 2.2 × 1.56	5.01 × 3.6 × 2.54
Rubber elements	8	12
Number of beams	4	4 (2 single beams and 2 double beams)
Static capacity of rubber elements (t)	4.5	10
Percentage of static capacity used on mass	67 %	69 %

The new anchoring design utilizes the same materials (S355) as earlier solutions, as they are effective, readily available, and familiar to manufacturers.

Beam structure

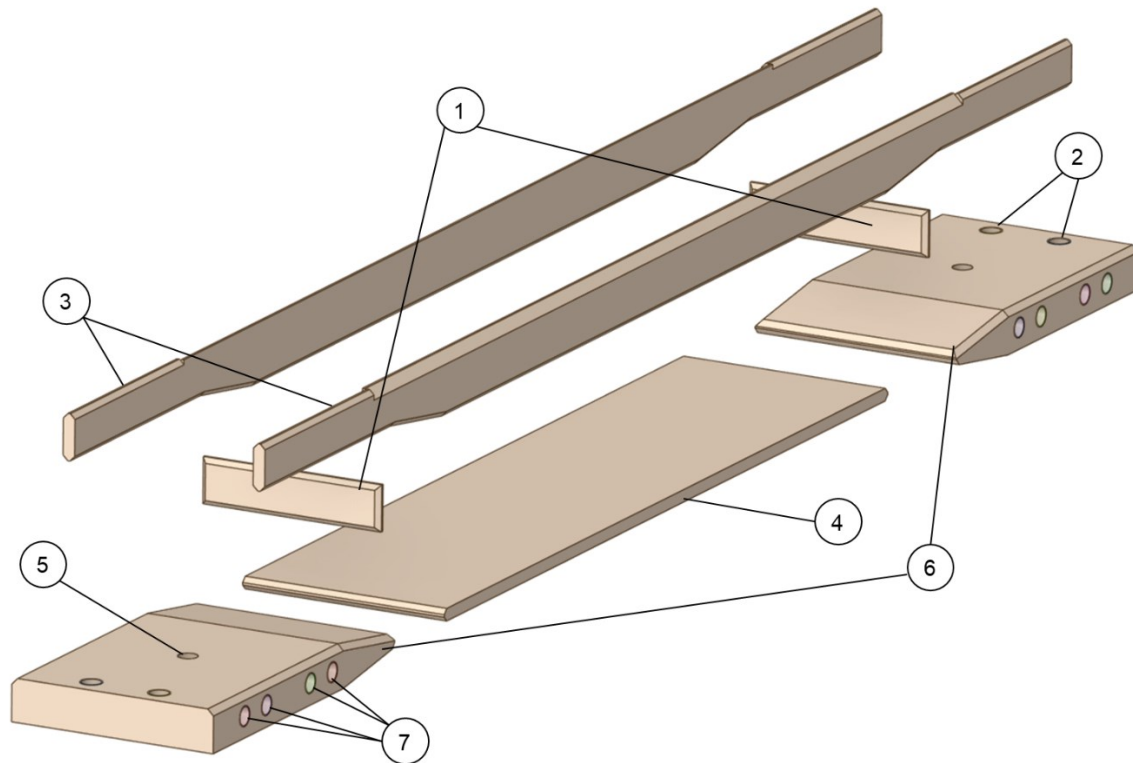


Figure 16 Single beam structure. 1. Insulation plate, 2. Jacking bolt holes, 3. Beam plates, 4. Bottom plate, 5. Centre bolt hole, 6. Foot pads, 7. Transport support holes.

Firstly, a standard beam structure was designed so that it could be easily duplicated and used where feet are needed. The single beams are composed of two perpendicular beam plates, one bottom plate and a foot and insulation plate on both ends. For both light and medium transformers, the beam structures utilize same plate thicknesses apart from on the foot pads. This design assures robust structural performance and easy implementation procedure across the whole mass envelope. As Figure 17 indicates the model includes chamfers to facilitate full penetration welding. The middle part of the beam plates upper surface has a single sided chamfer as it is welded to the transformer tank floor after the beam itself is assembled and is welded only from one side thus needing a larger chamfer to achieve full penetration. These welds are in compression and are thus not as critical as the other welds that are in tension which are all chamfered and welded from both sides.

As discussed in Chapter 2.2, more space is required for rubber elements than with closed-cell options. To address this, a “double beam” solution was designed.

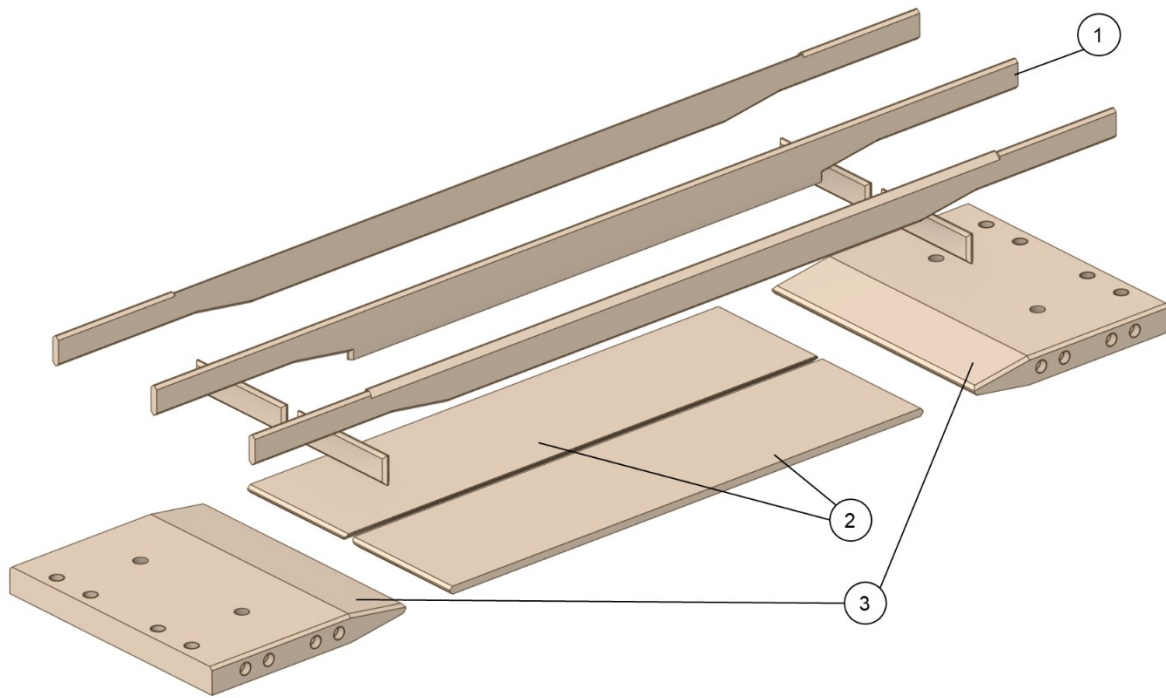


Figure 17 Double beam. 1. Middle beam plate, 2. Chamfered bottom plates, 3. Double foot pads.

Although it follows the same basic design, it is adapted to include two rubber components on each foot pad. The middle beam plate has a slightly extended underside that fits in between the two bottom plates with the centre lower edges chamfered.

The beam structure changes its length according to the size of the transformer, maintaining a certain distance from the sides of the transformer to the rubber elements. Spacing of the beams also changes with the number of beams needed and centre of gravity. Additionally, beams with the GM 12000 naturally have larger foot plates and are thus wider compared to the lighter GM 6000. These beams stiffen the transformer bottom plate and nullify any deformation due to mass of the active part or oil.

Planned beam assembly process

Both beams need precise assembly. Using four jigs per transformer streamlines and ensures accuracy in the process.

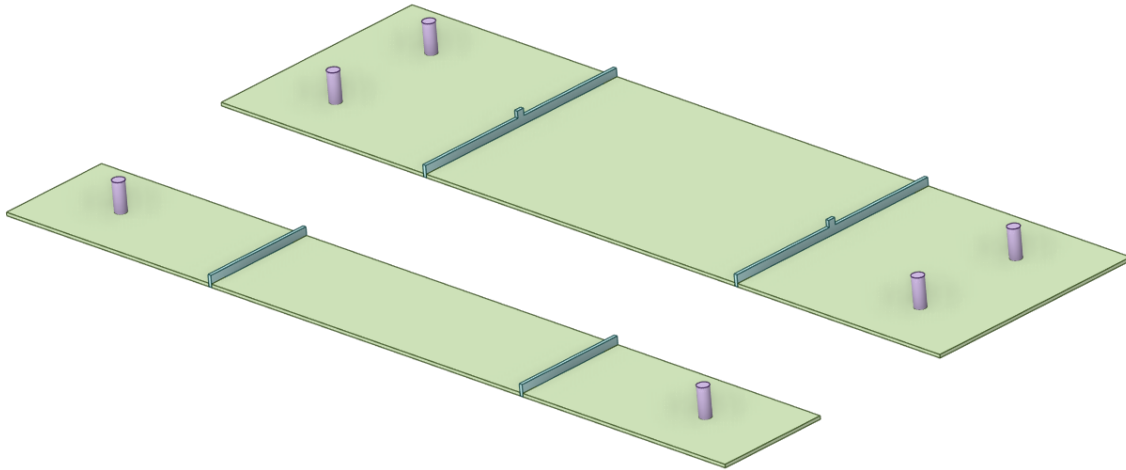


Figure 18 Jigs for a single beam and a double beam

Single beams require a simple jig that holds the two foot pads at the right distance from each other and a small support bracket that holds the bottom plate at the required height. Jigs include a base plate, two support brackets, and either two or four pins, depending on whether they are for single or double beams. Support brackets fit into base plate cutaways to improve error proofing and accuracy. The components are assembled using the jig (Figure 18) and welded together, after which the beam is welded to the bottom of the transformer tank.

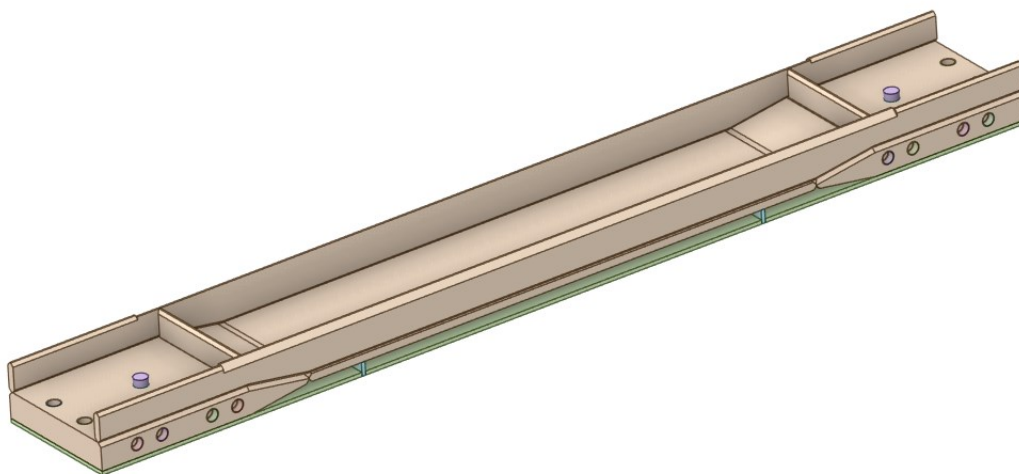


Figure 19 Single beam on a jig. (Medium Transformer)

The double beam is complex due to its central beam. Except for the middle beam plate, all components are assembled and tack welded on the jig. The jig leaves room for the middle beam plate.

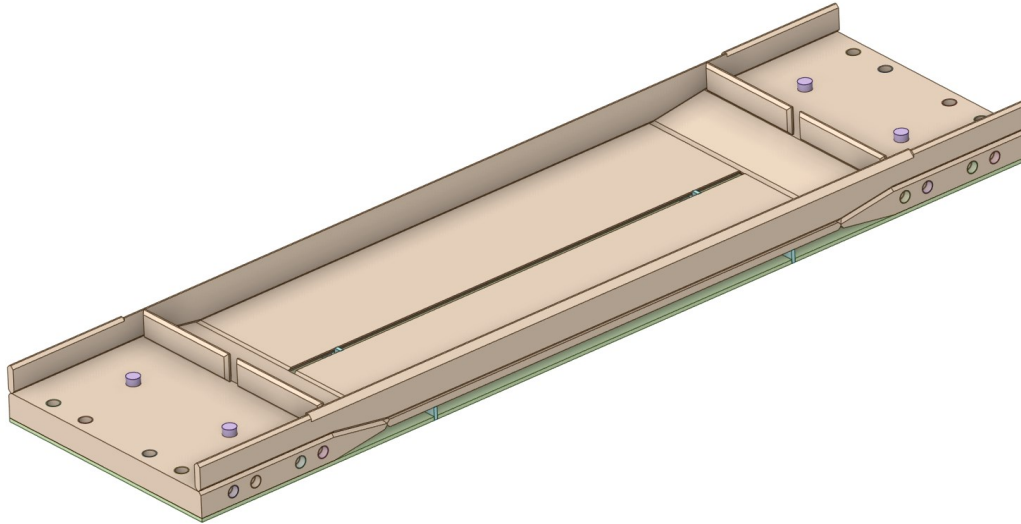


Figure 20 A half-ready beam on a double beam jig. (Medium Transformer)

Following this, the final jig is used. A base plate fitted with pins ensures each beam is correctly spaced longitudinally to the transformer seen in Figure 21. Before securing the half-ready beam structure to the jig, the centre beams are positioned in their slots. The beam assembly is then aligned with the guiding pins on the bottom of the transformer. After confirming the jig is properly set, the centre beam is tack welded to the transformer bottom. Next, both the jig and half-ready beams are lifted off so that the centre beam can be fully

penetration welded to the transformer bottom. Afterward, the other beams are completed and welded to the transformer bottom.

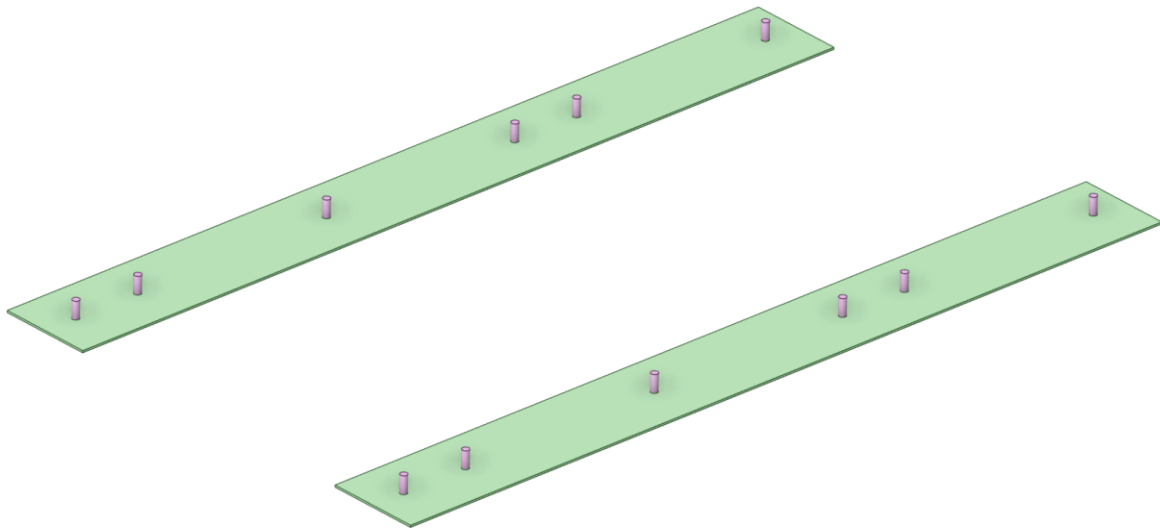


Figure 21 Longitudinal jig. (Medium transformer)

These jigs should be order bound, as beam length and spacing differ by project, despite a consistent main structure. Jigs and welding plans were reviewed and accepted by Hitachi Energy's tank subcontractor. The jigs are order bound as different projects have different dimensions and masses.

Foot design

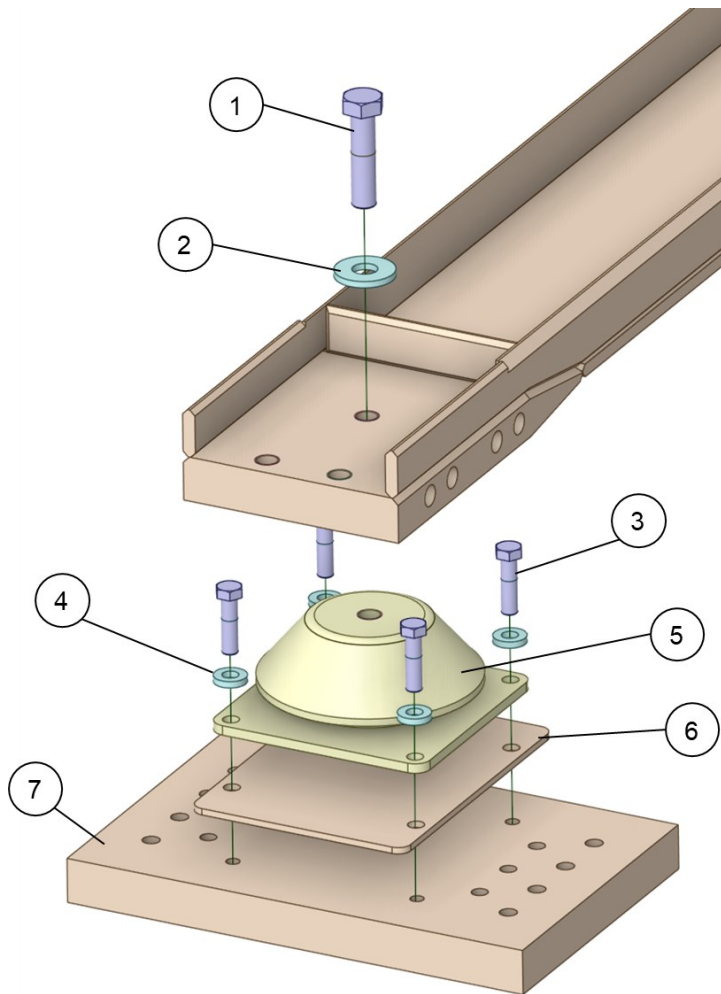


Figure 23 Explosion view of foot module. (Medium transformer). 1. Centre bolt, 2. Centre washer, 3. Element bolts, 4. Element washers, 5. Rubber element, 6. Shim plate, 7. Pad plate.

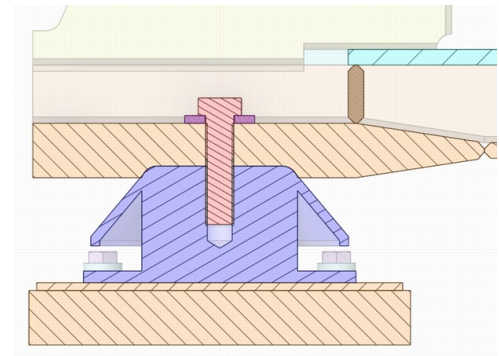


Figure 22 Cross-sectional view showing the embedding of the rubber element. The rubber element is a simplified model of GM 12000.

Embedded under the foot plate is the rubber element. Holding it on the top is a single centre bolt and four smaller element bolts at the bottom. All bolts smaller than M36 are on heavy DIN 7349 washers to maximize clamp length, M36 screws have DIN6340 washers. Partially serving the same purpose is also the shim plate underneath the rubber element, in addition to clamp length it provides a cover for the pad plate against the jacking bolts above. The shim can be replaced easily if it suffers damage from the jacking bolts. These bolts offer valuable flexibility during installation, as they help control the deflection of the rubber element when transport supports are being installed or removed. While the jacking bolts are sturdy enough to raise the entire transformer under stable conditions, their optimal use is in combination with lifting jacks, to hold the transformer at a specific height as transport supports or rubber elements are fitted.

The double beam foot design resembles the single beam but uses two rubber elements and their associated components.

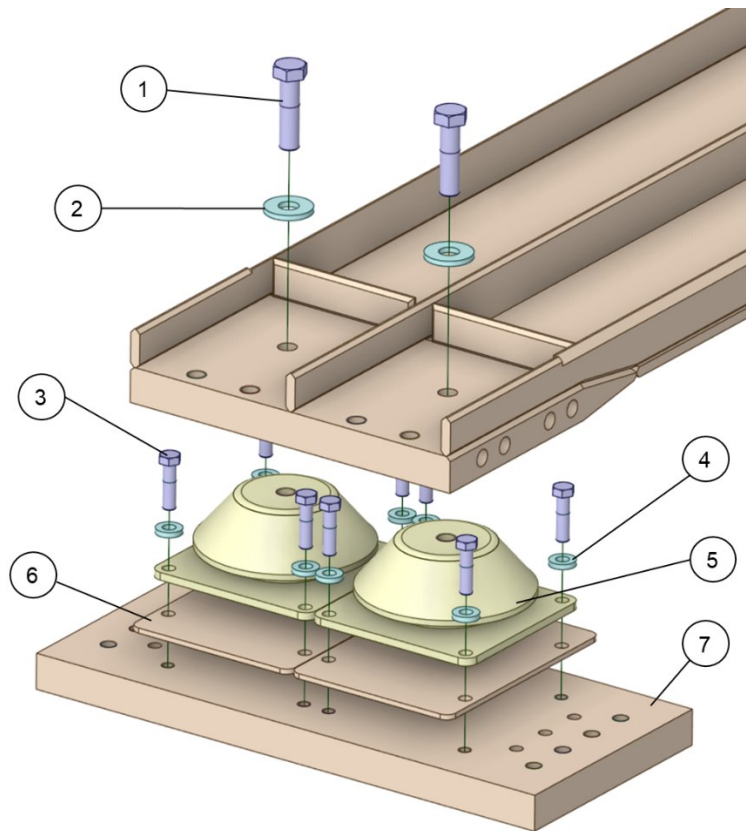


Figure 24 Explosion view of double beam with rubber elements. 1. Centre bolt, 2. Centre washer, 3. Element bolts, 4. Element washers, 5. Rubber element, 6. Shim plate, 7. Pad plate. (Medium transformer)

Sea transportation

Since sea transportation imposes up to 33% more load on the transformer compared to on-site loads, either the number of feet can be increased, or a transport specific solution can be designed. Because horizontally stable rubber elements take up significant space, sea transport supports were developed for this purpose. Transportation is a short period of time compared to the in-place operational lifetime of the transformer and thus fatigue is less of an issue, and designing is focused more on usability.

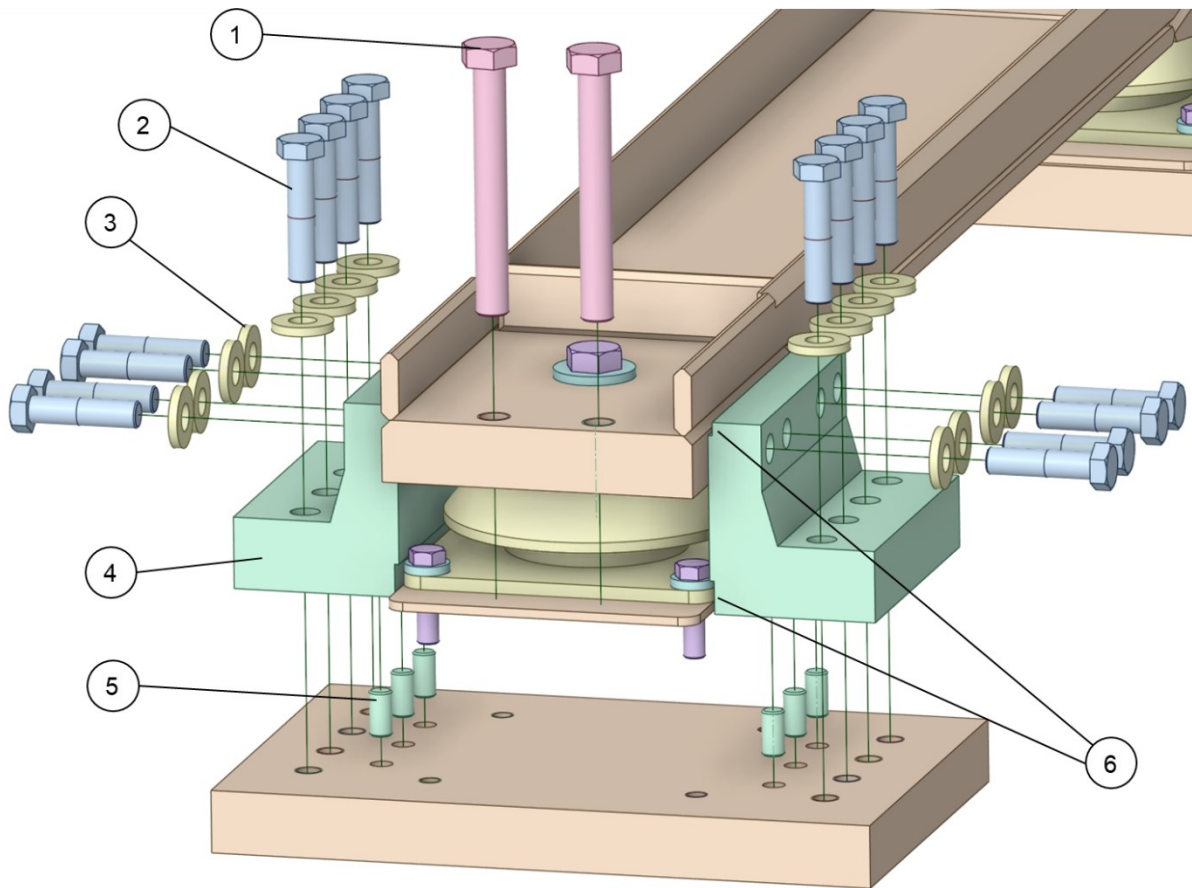


Figure 25 Explosion view of transport supports. 1. Jaking bolts, 2. Transport support bolts (TS-bolts), 3. Transport support washers (TS-washers), 4. Transport supports, 5. Shear pins, 6. Cut aways. (Medium transformer)

The support itself is a L-shaped part with 3-4 holes for screws both horizontally and vertically and 2-3 holes for shear pins. The number of holes depends on the mass of the tank. TS-bolts are designed to clamp the support to foot pad and pad plate in order to take the weight off the rubber element. The supports are supposed to take all loads and thus the rubber elements and its associated part could be completely absent during sea transport if preferred. Shear pins are designed to reduce shear loads on the vertical bolts although they are needed only in extreme shock load conditions. TS-washers lengthen clamp distance, promoting frictional contact between components, however this is not enough to facilitate a friction-type joint and thus friction shims are used for the upper bolt joint. Small cut aways are machined off transport supports to give more space for the rubber element and shim plate at the bottom, and higher clearance for the pad plate-beam plate weld.

Bolts scale together with the rubber element size. Bolt sizes are listed in Table 13. A4-80 grade bolts are used for in-place bolts due to their high resistance to salt water, as well as their high tensile strength and elongation capacity during preloading. For transportation 10.9

bolts are needed to secure the joint during the increased magnitude of the transport load cases.

Table 13 Bolts used in each design.

Model	Bolt	Use case
Light transformer	M24 (A4-80)	Centre bolt Jacking bolts
	M24 (10.9)	TS-bolts
	M16 (A4-80)	Element bolts (GM 6000)
Medium transformer	M36 (A4-80)	Centre bolt Jacking bolts
	M30 (10.9)	TS-bolts
	M24 (A4-80)	Element bolts (GM 12 000)

Installation process

Once the transformer with its designed feet has been delivered to the site, installation can begin. The customer should already have a prepared foundation with marked locations for the pad plates. Depending on the situation, the transformer may be moved or lifted onto the foundation. Once positioned, the pad plates are tack welded to the foundation and separated from the rubber elements so that the transformer can be jacked up. With the transformer raised, the pad plate can be securely welded to the foundation without exposing the rubber element to heat damage from welding. After welding is complete, the transformer can be sunk down and attached to the pad plates, and the welds can be treated with corrosion protection. This finalizes the installation process.

Material selection

Steel parts are manufactured from S355 steel and coated with the CX system per ISO 12944. Manufacturing the feet structure from structural steel grade S355 in combination with a CX corrosion protection system represents a well-established and DNV-compliant solution. S355 provides adequate strength, ductility, and weldability for welded offshore support structures governed by Ultimate Limit State criteria. Corrosion resistance is achieved through a high-performance coating system appropriate for severe offshore environments, consistent with the corrosion protection philosophy outlined in DNV offshore standards, where coated carbon steel is preferred over material substitution. The structural verification is therefore based on the mechanical properties of the base steel, while durability is ensured through certified coating systems and inspection-based maintenance strategies. (DNV, 2023) The rubber used in the rubber elements is susceptible to degradation from UV light; however, this does not pose a concern as it is contained within the element itself.

Table 14 Yield stresses for relevant materials.

Material	Utilized Yield Stresses (MPa)
S355(16 - 40mm)	$R_{eH} = 345$
S355(40 - 63mm)	$R_{eH} = 335$
A4-80	$R_{p0.2} = 600$
10.9	$R_{p0.2} = 900$

Overview

Below in Figure 26 is the medium transformer assembled with the new feet solution. Above the bottom plate of the tank are similar side supports that were used in the older designs. In future project where new feet are used tank accessories need to be adjusted to accommodate these supports. Tank bottom plate is also customized to work specifically with the current feet layout and centre of gravity. These steps are easily done in preliminary design and should not harm adaptation of the new solution. The beam structure is customized based on the width of the transformer while the rubber element is chosen based on mass and

length of the transformer with the GM 6000 being used with ~12-40t transformers and GM 12 000 used between ~40-90t if the length of the specific transformer can accommodate all required rubber elements.

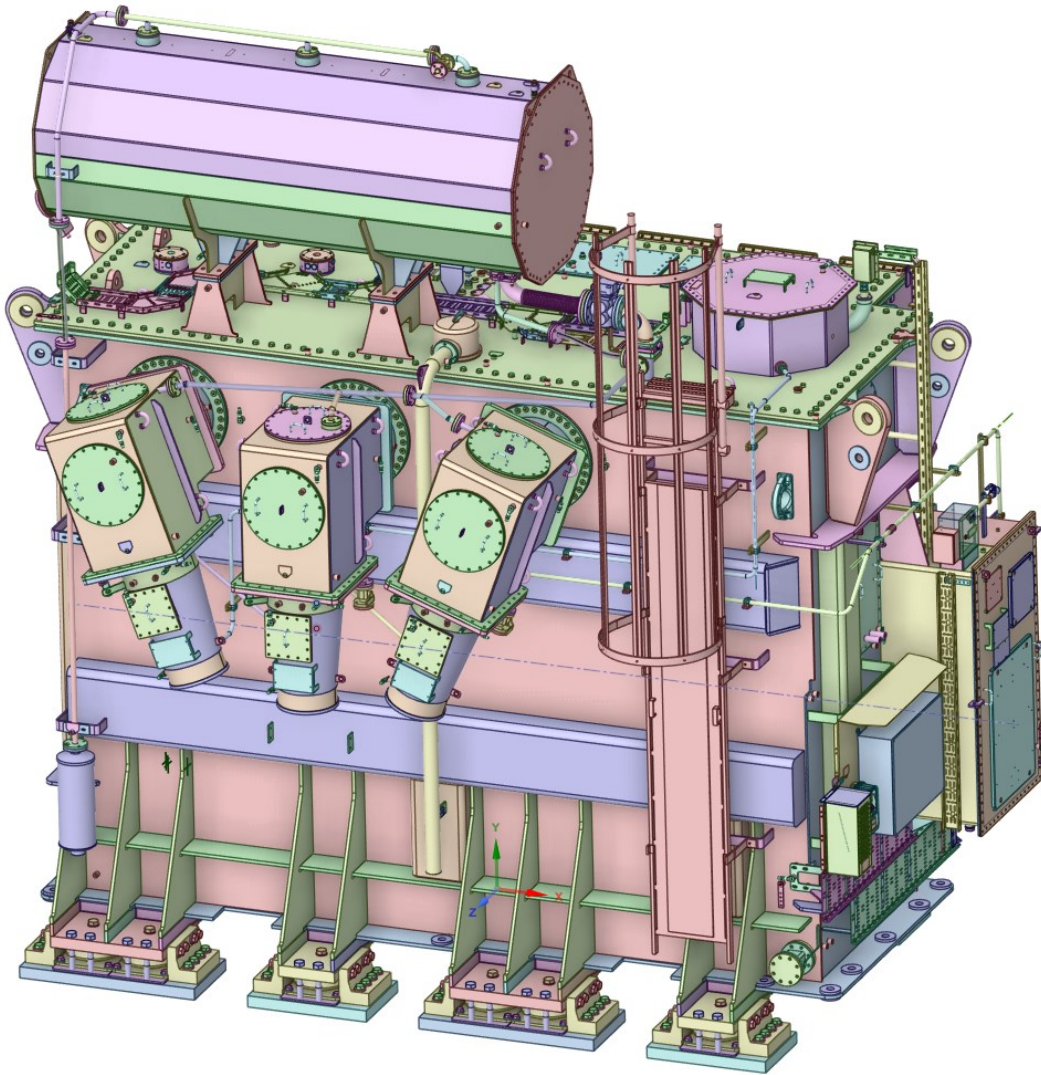


Figure 26 Medium transformer assembled with the new feet solution system including transport supports. Square cut-aways in tank bottom are for lift jacks required by goal 4.

The new feet tackle main requirements in a structured and clear way (Table 15). Structural integrity in the given mass envelope is accounted for with over sizing critical parts, designing additional supports and using high grade materials. Ease of installation is improved by having a modular rubber element with lower possibility of having alignment problems in addition to having room for needed jacking space. Vibrational issues are mitigated by choosing a low natural frequency element and stiff paths for vibrations to travel between the tank and

foundation. Lower frequency vibrations are managed with rubbers viscoelastic properties minimizing impact and avoiding resonance peaks. Manufacturing is made easier with error-proofing tools such as order bound jigs, while the customer gets a more accurately dimensioned product which helps with designing foundations. With jigs and an assembling plan weld can be full penetration welded for maximum strength.

Table 15 Table that shows how main requirements were accomplished with the design.

Main Requirements	Goals	Accomplished with
Structural integrity	1. Withstand respective operational load cases	Thick S355 plates used, < 70% of rubber element static capacity used on mass
	2. Withstand harsh sea transportation load cases	Transport supports to handle loads during sea transportation
	3. In-place bolt connections are frictional	High and accurate preload
Installability	4. 2x200mm spaces per long side for hydraulic lifting jacks	“Double beams” to save space if needed
Anti-vibration	5. Isolation from 100/120 HZ transformer vibrations	Low natural frequency rubber elements
	6. Avoid resonance with dominant sea vibrations (0.04-0.5 Hz)	Natural frequency above resonance peak and rubbers force dissipation properties
Manufacturability	7. All welds are full penetration weldable	Chamfers, Welding jig and assembly plan

4.3 Structural analyses setup

As mentioned, an already completed and verified analysis of the transformers with their old feet designs is used as a base for structural analysis models. The new and old designs are merged in Ansys Workbench in a way where only model data is transferred as shown in Figure 27. Old feet designs are suppressed, and new designs are added in its place for the main analysis.

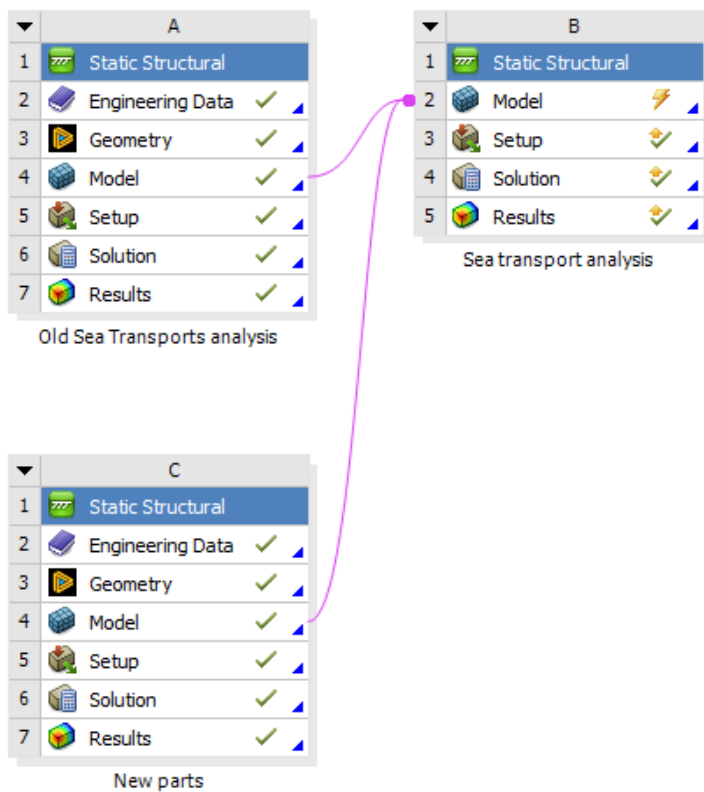


Figure 27 Analysis structure inside Ansys Workbench.

Because these transformers themselves have previously been analysed and verified to withstand the calculated load cases, the analysis can be focused on only newly modelled elements and interfaces. For steel-to-steel non-welded contacts a frictional coefficient of $\mu = 0.15$ was chosen as it is commonly used for interfaces in structural and bolted-joint analyses as a conservative design value for steel-to-steel. Although higher friction coefficients may be measured for clean and dry steel surfaces, friction is highly sensitive to surface condition, contamination, lubrication, and corrosion. Design standards therefore adopt lower-bound values that remain valid under unfavourable conditions. Values around $\mu = 0.15$ correspond to experimentally observed friction for steel surfaces with light lubrication or contamination

and are recommended by bolted-joint design guidelines such as VDI 2230. Using this conservative value ensures that friction-dependent resistance is not overestimated and that structural safety does not rely on uncertain interface conditions. Additionally, when friction shims are used a friction coefficient of 0.45 is used.

Internal forces due to oil movement inside the transformer tank are modelled using various forces and pressures. Hydrostatic pressure that is tied with the load-specific downward acceleration is applied to the walls. The inertial effect of oil is modelled as forces on the walls and floors of all oil-filled tanks. The magnitudes of the forces are calculated with $F = m \cdot a$ where a is the load-case-dependent acceleration and m is the mass of oil in the tank.

Simplifications made for analyses:

- Plates that did not connect to bolts were simplified as shell structures
- Bolts were modeled as beams and not solid bolts
- Springs were used to simulate rubber structures
- Oil loads were simulated with load-case-dependent hydrostatic pressure and inertia forces on tank walls ($F = m \cdot a$)

Coarse meshing was used for most of the tank surfaces while connections and bolt interfaces were meshed with finer mesh. Mesh sizes were defined based on engineering practise. If any meshing related issues were observed in the results, the mesh or part connections were refined until the results were deemed satisfactory. Computational resources posed a moderate limitation throughout the analysis, as evidenced by the relatively coarse mesh sizes. Since different FEM platforms use various names for meshing methods, Table 16 provides a general overview of methods used in Ansys Mechanical. Mesh sizes and methods per part/surface are found in analysis setup chapters below, as well as the main purpose of each analysis in relation to research questions.

Table 16 General information about meshing methods used in Ansys mechanical.

Method	General name	Dimensionality	Typical Elements	Main Purpose
Sweep	Swept/mapped hexahedral meshing	3D	Hex / Wedge	Best-quality structured solid mesh
MultiZone	Multi-block structured hex meshing	3D	Hex-dominant (some wedges/pyramids)	Structured mesh for complex solids
Hex-Dominant	Hex-dominant hybrid meshing	3D	Hex + wedge+ pyramid + tetrahedral	Fallback when hex is difficult
Tetrahedrons	Unstructured tetrahedral meshing	3D	Tetrahedral	Robust automatic meshing of complex geometries
Sheet Body	Shell element meshing	2D	Shell elements	Thin structures
Line Mesh	Beam element discretization	1D	Beam elements	Bolts, beams, frame members

4.3.1 Stress analysis setup

Table 17 Purpose and KPI's of stress analysis.

Analysis	Purpose	KPI
Stress analysis (ULS)	Evaluate forces projected on feet elements	Stress on beams, Deformation of springs, Reaction forces on fixed supports

Stress and deformation analysis is done to calculate the maximum von Mises stress for the beam structures and both deformation and reaction forces on the rubber elements. This analysis is used to point out the governing loads for the design regarding RQ1. Accelerations used are In-place accelerations from Table 7 applied on their respective transformers. As mentioned, the rubber elements were modelled using three springs with spring coefficients equivalent to those of the real element sent by the manufacturer. In this analysis the shim plate was bonded to the pad plate. Transport supports and unneeded details were suppressed for the analysis as shown in Figure 28.

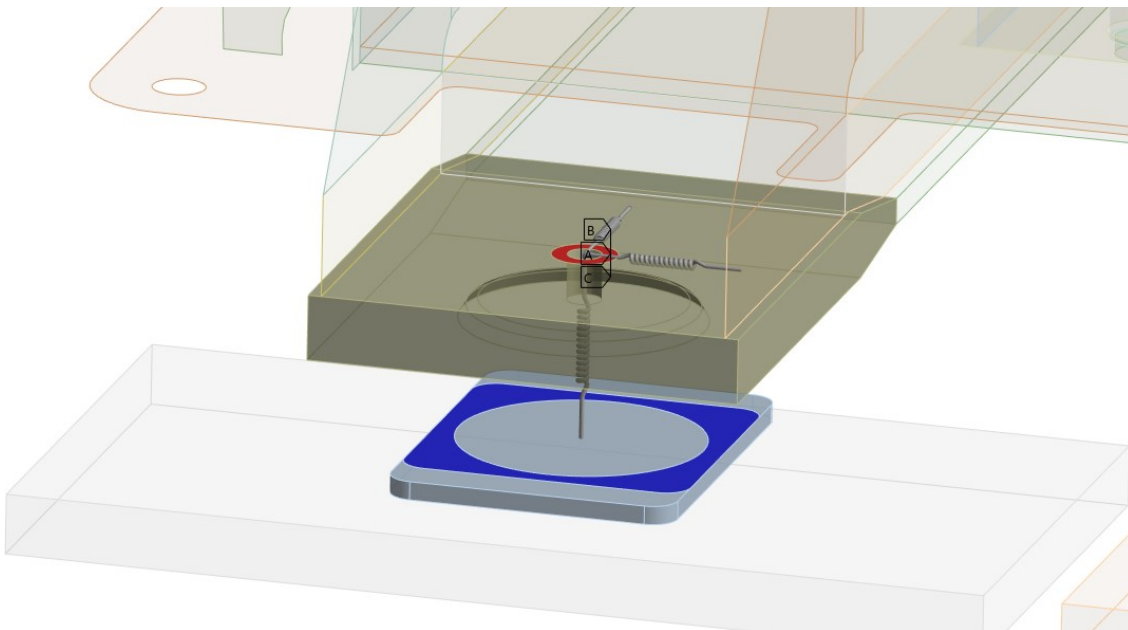


Figure 28 Example of how rubber elements were modelled.

Table 18 and Table 19 show the mesh sizes and methods for different parts and surfaces. Tank mesh was generalized as a range it includes hundreds of parts, and it has been proven to work in its previous analyses using its original feet. The meshing was designed to capture global stiffness and local stress distribution while maintaining computational efficiency. For solid structural components sweep and MultiZone were used where geometry permitted as these provide superior accuracy compared to other methods.

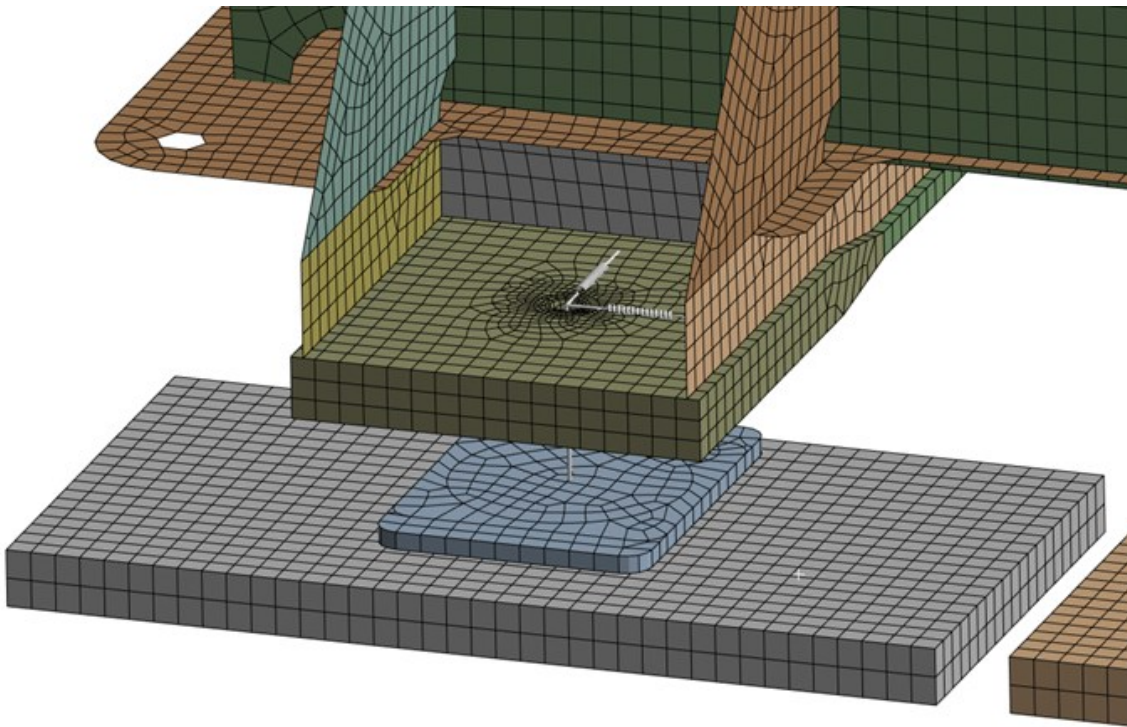


Figure 29 Meshed foot of Light transformer for stress and deformation analysis.

Table 18 Light transformer stress and deformation analysis mesh. This is used as base mesh for Light transformer analyses.

Part (Special surface)	Mesh size	Method
Feet pads (Washer)	20 mm (5mm)	Hex-dominant
Beam structure	20 mm	Sheet body
Pad plates	20 mm	Sweep
Shim plates	20 mm	MultiZone
Tank	~5-100 mm	MultiZone/Sweep

Table 19 Medium transformer stress and deformation analysis mesh. This is used as base mesh for Medium transformer analyses.

Part (Special surface)	Mesh size	Method
Feet pads (Washer)	45 mm (5 mm)	Hex-dominant
Beam structure	45 mm	Sheet body
Pad plates	45 mm	MultiZone
Shim plates	30 mm	Tetrahedrons
Tank	~5-100 mm	Sweep/MultiZone/Hex-dominant

4.3.2 Bolts analyses setup

Table 20 Purpose and KPI's of bolt analysis.

Analysis	Purpose	KPI
Bolt analyses	Confirm that bolt connections hold	Bolt bending moment, Bolt axial forces, Connection slippage

In the bolt analysis bolt holes were added and washer projections were used for accurate preload management. The element bolts were modelled using beam elements with assigned cross-sectional properties, enabling efficient representation of axial and shear load transfer while maintaining numerical robustness. This approach is consistent with force-based bolt verification methods, such as VDI 2230, and is suitable for global structural assessment, although local thread and bearing stresses are outside the scope of the model. Rubber elements were modelled similarly as in stress analysis. The shim plates contact with the pad plate was set to frictional to test for slippage. The centre bolt was not modelled as reaction

force results are assumed to represent the load on the centre bolt and can be used to analytically calculate performance of the bolt.

The bolt analysis was run with both no preload to gather accurate axial loads and with preload for slippage and shear loads.

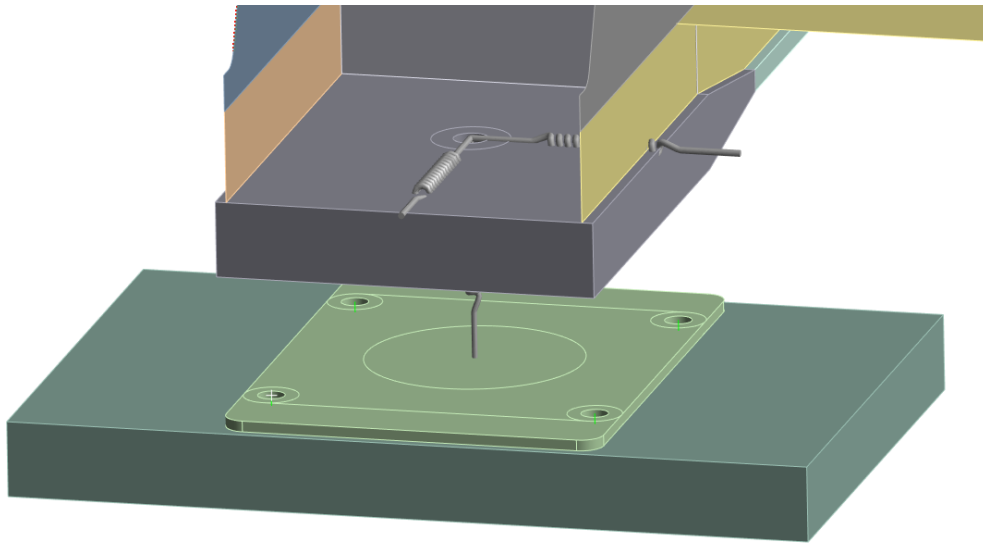


Figure 30 Example of how washers and beams were modelled in bolt analyses.

The bolt analysis meshing incorporated numerous elements from stress analysis, with additional refinement applied to load surfaces to accurately and consistently reflect the mechanical behaviour of the bolted joints (Table 21, Table 22). Bolts were modelled using beam elements, discretized along the bolt axis, with cross-sectional properties defining stiffness and load-carrying capacity. This approach enables accurate evaluation of axial forces, shear forces, and utilization ratios while avoiding the computational expense associated with fully solid bolt modelling.

Results from this analysis are used to address RQ1 in conjunction with the stress analysis to better understand the governing offshore loads. Additionally, the bolt analysis is utilized to evaluate if the design effectively reducing bolt bending/shearing (RQ3).

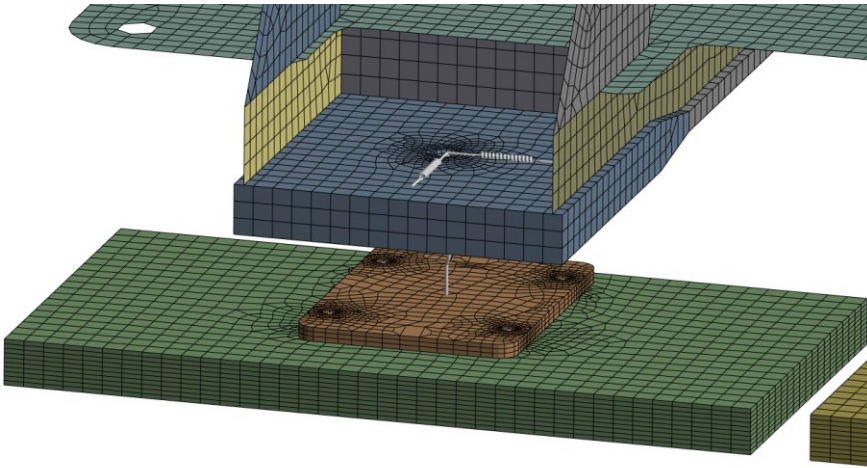


Figure 31 Meshed foot of Light transformer for bolt analysis.

Table 21 Additional surface/beam sizing's needed for light transformer bolt analysis.

Beam/Surface	Mesh size	Method
Shim plate washers	5 mm	MultiZone
Pad plate holes	5 mm	MultiZone
Element bolts	3 mm	Line mesh

Table 22 Additional surface/beam sizing's needed for medium transformer bolt analysis.

Beam/surface	Meshi size	Method
Shim plate washers	5 mm	Tetrahedrons
Pad plate holes	5 mm	MultiZone
Element bolts	3 mm	Line mesh

4.3.3 Modal analysis setup

Table 23 Purpose and KPI's of modal analysis.

Analysis	Purpose	KPI
Modal analyses	Deduce critical frequencies	First modes, Mode shapes, Effective mass

The modal analysis was performed assuming undamped behaviour, in order to extract the natural frequencies and mode shapes governing the dynamic response. (Den Hartog, 1985) Modal analyses were performed using the same base analysis as the stress and deformation analysis. The modal analysis employed a mesh topology consistent with the stress and deformation analysis model. Modal analysis is used to evaluate the performance of the rubber elements and to assess whether the assumptions and element choices made serve real benefits in controlling vibrations (RQ2).

4.3.4 Transport analysis setup

Table 24 Purpose and KPI's of transport analysis.

Analysis	Purpose	KPI
Transport analysis	Verify that the design can withstand sea transportation	Bolt bending moment, Bolt axial forces, Stress on transport supports

Transport analysis was conducted using transport supports and applying sea transportation accelerations. The rubber element and its accessories were removed as shown in Figure 32, and the main interests are slippage and bolt forces. All interfaces to the transport supports are modelled as frictional, bolts are modelled similarly as in bolt analysis and pins are modelled as solids in the light transformer and as beams in the medium transformer as the

modelling was incapable of converging during analysis. The upper bolt joint utilizes friction shims and thus has a friction coefficient of 0.45 compared to other steel-to-steel interface's 0.15.

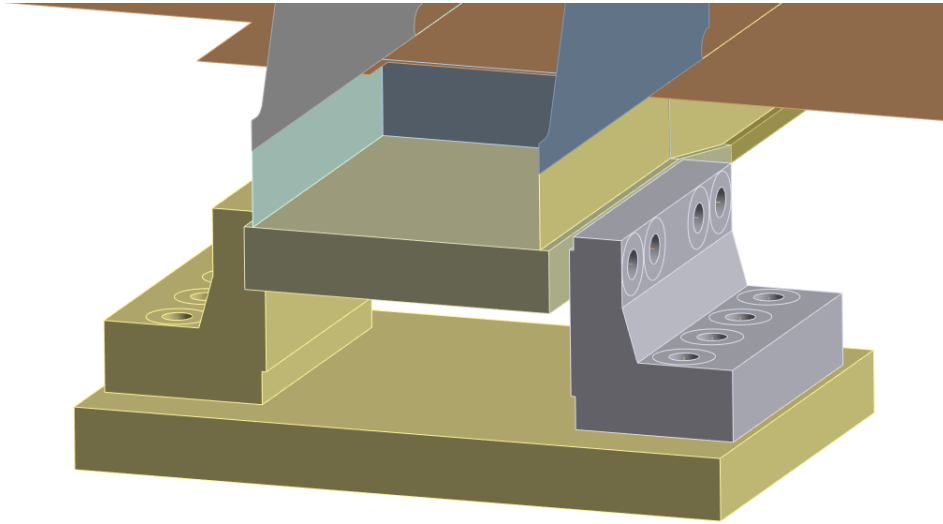


Figure 32 Example of how transport support was modelled for analysis.

For the sea transport analysis, the mesh was tailored to represent inertia-dominated load cases arising from accelerations during transportation but kept relatively close to the base mesh from stress analysis. MultiZone was prioritized when possible and hex-dominant where it was not possible. (Figure 33)

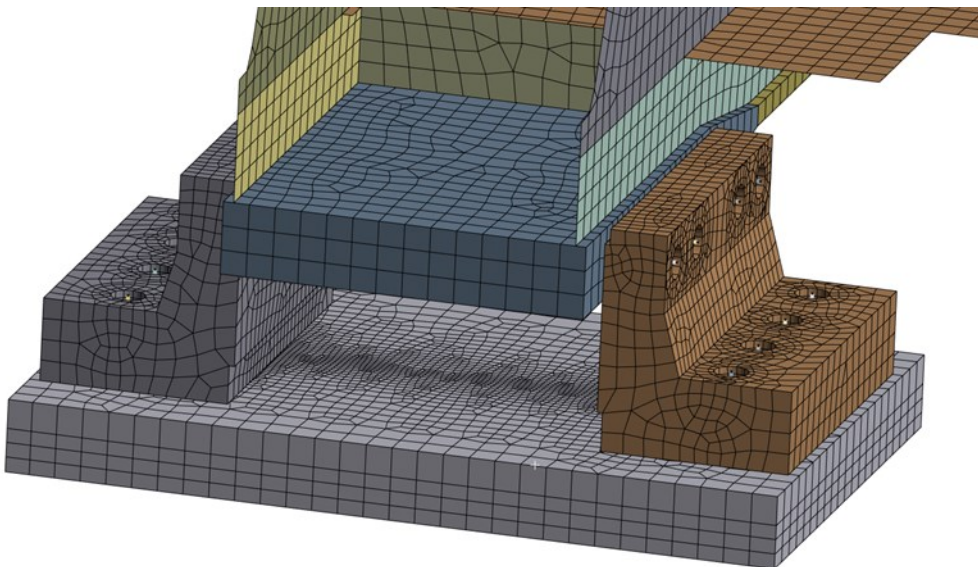


Figure 33 Meshed foot of Light transformer for transport analysis

Table 25 Meshing for Light transformer sea transport analysis. All not mentioned are the same as in stress and deformation analysis.

Part (Special surface)	Mesh size	Method
Transport supports (Washer)	20 mm (5 mm)	Hex dominant
Feet pads (Holes)	50 mm (5 mm)	Hex dominant
Pad plates (Holes)	30 mm (5 mm)	MultiZone
Pins	5 mm	MultiZone
Bolts	5 mm	Line mesh

Table 26 Meshing for Medium transformer sea transport analysis. All not mentioned are the same as in stress and deformation analysis.

Part (Special surface)	Mesh size	Method
Transport supports (Washer)	20 mm (5 mm)	Hex dominant
Feet pads (Holes)	25 mm (15 mm)	MultiZone
Pad plates (Holes)	25 mm (5-15 mm)	MultiZone
Pins	5 mm	Line mesh
Bolts	5 mm	Line mesh

4.4 Analytical tools

Analytical tools are used in this thesis to support verification and validation of the numerical models, to provide reference values for key response quantities, and to interpret the finite element results within established mechanical and standard-based frameworks. While detailed stress and deformation patterns are obtained from FEA, analytical methods are applied to assess numerical correctness, physical plausibility, and compliance with applicable design standards.

Sum of Reaction forces

The base analysis (Stress and deformation) is verified by comparing the sum of downward reaction forces to the transformer's mass:

$$m_{eq} = \frac{\Sigma R_{\downarrow}}{a_{\downarrow}} \quad (18)$$

Where $\Sigma R_{\downarrow}$ is the sum of downward reaction forces during a load case, m_{eq} is the equivalent mass associated with the load case and $a_{\downarrow}$ is the applied downward acceleration during the load case. This formula is in accordance with Newton's second law and is commonly used in DNV-RP-C205-style inertia load checks.

Maximum Reaction force on a foot

Reaction forces can be calculated using Newton second law and the formula for moment of a force. Force to a foot from X-component of the load case can be calculated starting with:

$$M = F \cdot r \quad (19)$$

Moment is replaced with the moment from acceleration forces (F_{acc}) on the centre of gravity. r_y and r_x represent the distance to the foot from the centre of gravity in the Y and X directions respectively:

$$F_{acc} \cdot r_y = F_x \cdot r_x$$

$$F_x = \frac{F_{acc} \cdot r_y}{r_x}$$

F_{acc} is calculated using Newtons second law with the acceleration being load case specific:

$$F_x = \frac{a_x \cdot m \cdot r_y}{r_x} \quad (20)$$

Where m is the mass of the transformer.

All load case specific accelerations can be added and divided with the number of feet used (n) to get the estimated force on the foot in question:

$$F = \frac{F_x + F_z + F_y}{n} \quad (21)$$

Of these forces, force from vertical acceleration F_y can be calculated using only Newtons second law. This the whole equation becomes:

$$F = \frac{\frac{a_x \cdot m \cdot r_y}{r_x}}{n} + \frac{\frac{a_z \cdot m \cdot r_y}{r_z}}{n} + \frac{m \cdot a_y}{n} \quad (22)$$

This formula can be used to interpret reaction force results to help understand rubber element utilization.

Element deformation

Deformation at the feet can be checked by calculating the vertical deformation with Hooke's law:

$$\delta = \frac{F_r}{k} \quad (23)$$

Where F_r is the reaction force on a foot, k is the spring coefficient given by manufacturer and δ is the calculated deflection.

Bolt friction

For calculating stresses for centre bolts, axial forces are calculated as the maximum upward reaction force on a single rubber element or as four times larger than the external maximum axial stress of the element bolts if reaction force is not observed for a conservative estimate. Bending stresses are determined by first calculating whether the join is a friction type joint comparing frictional force from minimum preload and the maximum horizontal reaction forces present:

$$F_{s \text{ centre}} = F_{s \text{ max}} - \mu \cdot F_{M \text{ min}} \quad (24)$$

Where $F_{s \text{ centre}}$ is the shear on the centre bolt, $F_{M \text{ min}}$ is the minimum preload, μ is friction coefficient and $F_{s \text{ max}}$ is the maximum external shear force. If the result of this calculation is

$F_{s\ center} < 0$ the shear forces can be ignored. Bending for centre bolt is calculated based on the maximum horizontal forces on a foot and the thickness of the bolt:

$$M_b = F_h \cdot \frac{d}{2} \quad (25)$$

Where F_h is the maximum horizontal force on the centre bolt and d is the nominal diameter of the bolt. (Shingley, 2021) The maximum horizontal force on the bolt is calculated as being 10% of the maximum horizontal force on a foot. 10% is chosen as a conservative estimate as it is the typical amount of shear forces FEA shows even when the connection is a friction-type joint due to elastic calculations made by the software. This shear force is additionally supported by the elastic resilience of the bolt and clamped materials; clamped material has roughly 10% of the elastic resilience compared to the bolt material.

Modal validation

In modal analyses the results are validated using a single-degree-of-freedom spring-mass system with vertical stiffness (Shingley, 2021). First rigid-body modes on elastic supports can be calculated with:

$$f_y = \frac{1}{2\pi} \sqrt{\frac{k_{tot}}{m}} \quad (26)$$

Where f_y is the frequency of first rigid-body mode in Y-direction, k_{tot} is the total vertical stiffness and m is the mass of the transformer. k_{tot} is calculated with the number of feet used (n) and vertical stiffness of a foot in N/m (k_{foot}):

$$k_{tot} = n \cdot k_{foot} \quad (27)$$

Combined these equations are:

$$f_y = \frac{1}{2\pi} \sqrt{\frac{n \cdot k_{foot}}{m}} \quad (28)$$

f_y is compared to the first mode in Y-direction from the modal analysis to calculate the accuracy. As per Ewins an error of ~20–30% is acceptable due to modelling simplifications and mass participation effects.

5 Results & Discussion

This chapter presents and discusses the results obtained from the numerical analyses carried out in this study. The results are evaluated with respect to the defined design objectives and the applicable structural and offshore standards. The chapter covers stress and deformation analyses, bolt integrity verification, modal analysis, and sea transport analysis.

The results of this study are primarily based on FEAs and simplified analytical models, which rely on assumptions regarding material behaviour, load distribution, and boundary conditions. Elastomeric components are modelled using linear spring representations, which do not capture their non-linear and time-dependent behaviour under varying loads. Simplifications in bolt modelling and contact definitions limit the resolution of local stresses and phenomena such as micro-slip or thread behaviour. External vibration sources, long-term fatigue, and environmental degradation effects are outside the scope of this work. Therefore, the results demonstrate engineering feasibility but do not constitute full validation for operational deployment without further experimental verification.

The presented results are organized according to transformer type first and then by analysis type. For each analysis overall results are presented after which they are verified and checked against passing criteria. These results are then interpreted and expanded upon in the discussion chapter.

5.1 Results

The results presented in this chapter are obtained from FEAs and analytical calculations. The analyses include static ULS load cases according to DNV- OS-C101, sea transportation load cases according to DNV-ST-E273, modal analysis, and bolted joint verification according to VDI 2230.

Generalized passing criteria for both transformers per analysis:

- Stress analysis:

1. Stresses within allowable limitations ($\leq \sigma_A$)

- a. S355(16 - 40mm): $R_{eHd} = \frac{R_{eH}}{\gamma_m} = 300 \text{ MPa}$

b. S355(40 - 63mm): $R_{eHd} = \frac{R_{eH}}{\gamma_m} = 291 \text{ MPa}$

2. Deformation and reaction forces on the rubber elements must be confirmed by element manufacturer to be tolerable.

- Bolt analysis:

1. Bolt stresses need to be within allowable limits:
 - a. VDI 2230 check: $v < 1$
2. Contact needs to be frictional and not slipping

- Modal analysis:

1. Modes must be well into the isolation range for internal excitations
 - a. $r_{int} > 2$
2. Modes must not resonate with peak sea born vibrations
 - a. $r_{sea} < 0.5$

- Transport analysis:

3. Bolt stresses need to be within allowable limits:
 - a. VDI 2230 check: $v < 1$
4. Stresses within allowable limitations ($\leq \sigma_{t \text{ allow}}$)
 - a. S355(16 - 40mm): $\sigma_{t \text{ allow}} = f_y \cdot \alpha = 293 \text{ MPa}$
 - b. S355(40 - 63mm): $\sigma_{t \text{ allow}} = f_y \cdot \alpha = 285 \text{ MPa}$

5.1.1 Light transformer

Stress results

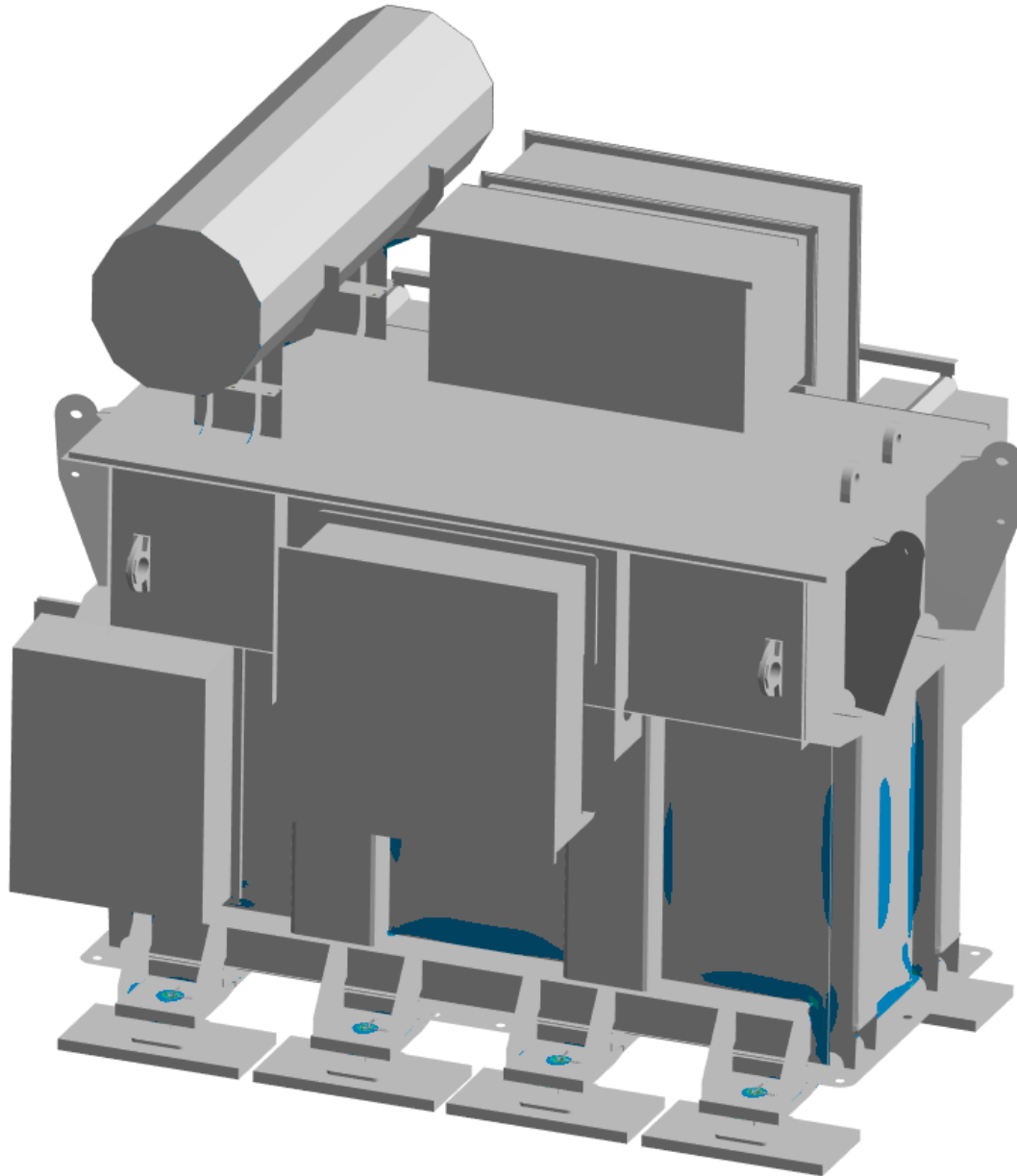


Figure 34 Maximum equivalent (von Mises) stress over time.

Equivalent von-Mises stress was used to assess yielding of the main steel structures. As expected, both maximum peak and nominal stresses were well below the minimum allowable limit of 300 MPa for lighter plates and 291 MPa for the foot and pad plates. With peak maximum von-Mises stress located at the edge of where the washer tightens on the foot plates surface and where the springs are connected to, as expected. In reality this force would

be directed to the bottom of the footplate on a larger area thus the stress would be even smaller than the reported maximum of 239 MPa in Figure 35. Maximum nominal stresses are evaluated as being ~160 MPa as peak forces are due to simplification in the analyses.

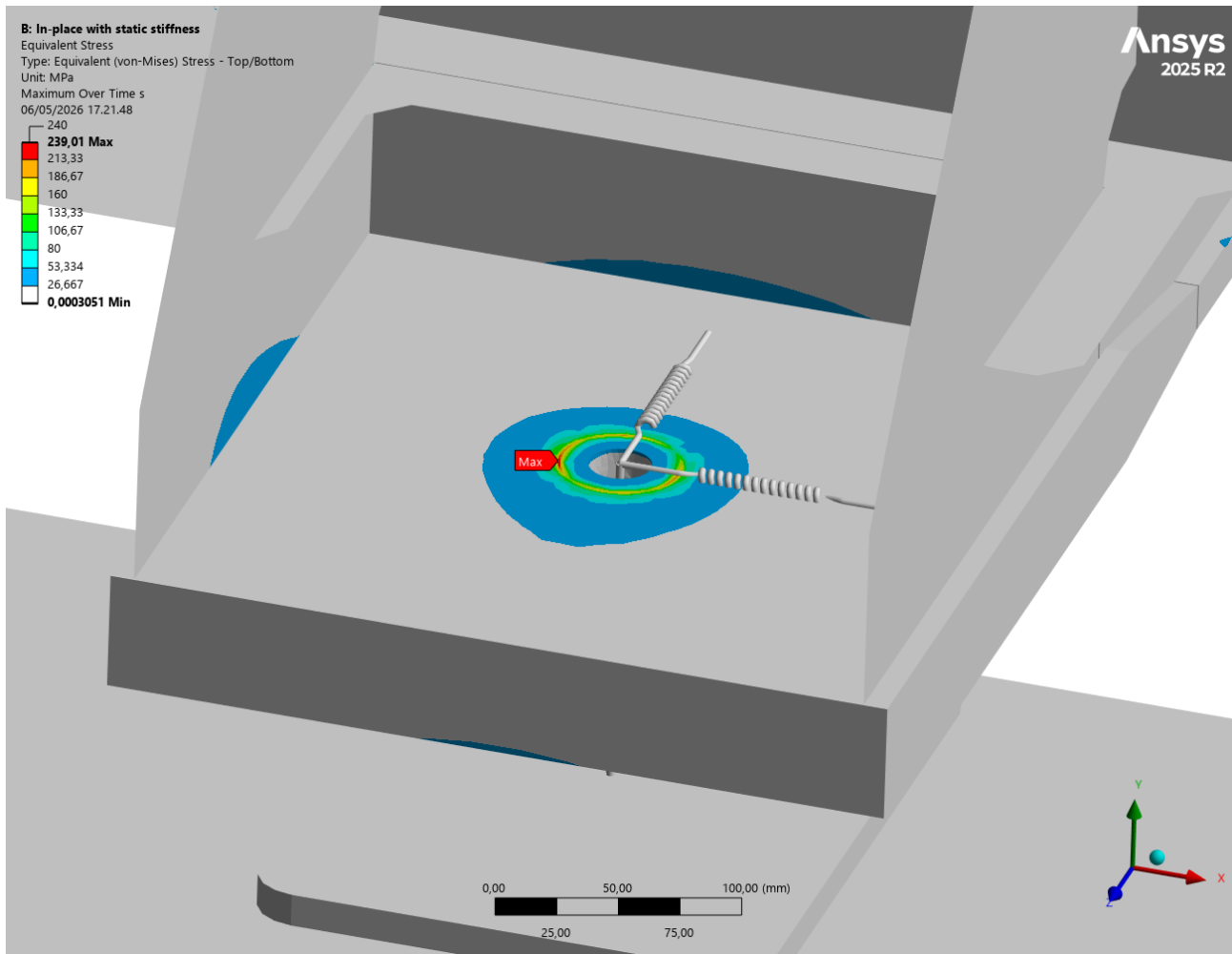


Figure 35 Maximum equivalent (von Mises) stress over time. Point of maximum stress flagged with a red marker.

Deformation results

B: In-place with static stiffness

Total Deformation 2
Type: Total Deformation
Unit: mm
Maximum Over Time s
06/05/2026 19.03.34

23,326 Max
20,734
18,143
15,551
12,959
10,367
7,7754
5,1836
2,5918
0 Min

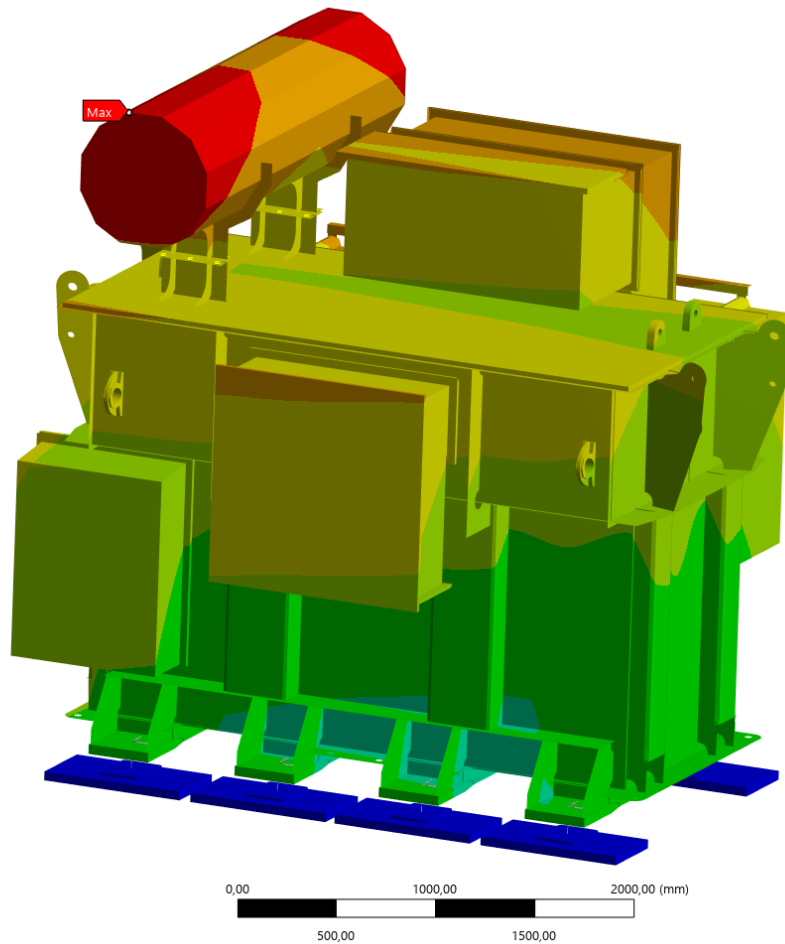


Figure 36 Maximum total deformation over time for whole tank.

Maximum deformation over time indicates that deformation grows farther up the transformer as shown in Figure 36, which is expected as the tank and its accessories are a rigid structure and a vast majority of the deformation happens at the rubber elements.

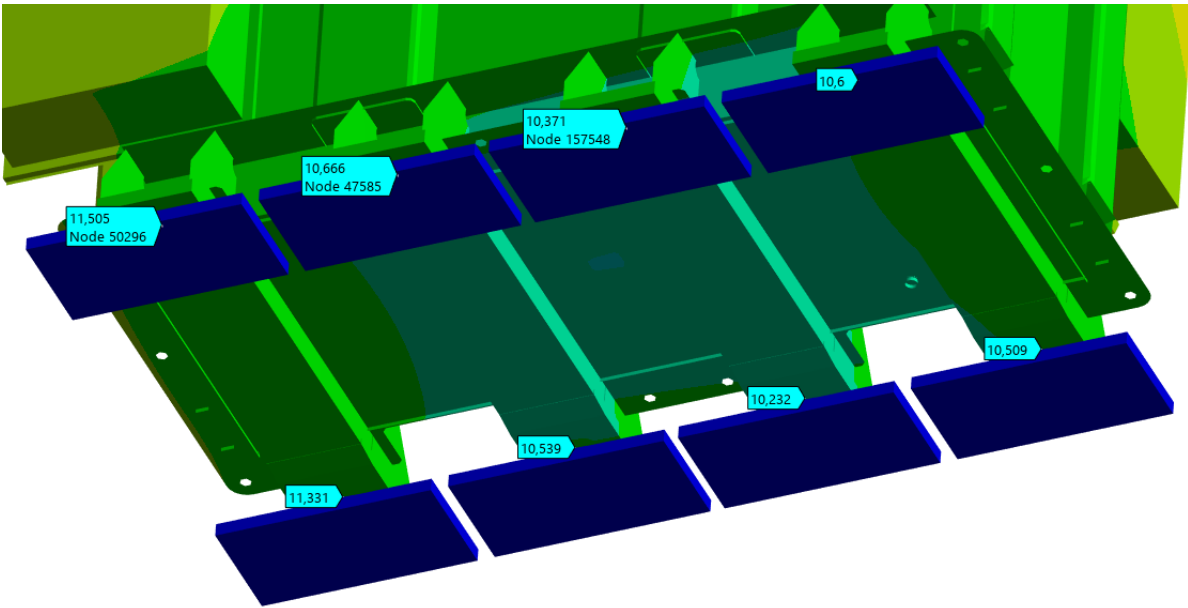


Figure 37 Maximum deformation over time for light Transformer.

Under static (only weight) load the rubber elements deflect ~6.5 mm. During in-place loads deformation results indicate that the springs representing the rubber elements deform 10.2 - 11.5 mm at maximum. Most of the deformation occurs downward, about 11.2 mm at the highest, with a maximum of 2.6 mm deformation horizontally. Vertical deformation is verified with highest vertical force from

Table 27 and spring coefficient from Table 11 using Hooke's law (Equation 23):

$$\frac{70\,200\text{ N}}{6250\text{ N/mm}} = 11.232\text{ mm}$$

The close alignment between calculated deformation using Hooke's law and the observed simulation results confirm that the rubber elements were accurately modelled.

Reaction force results

B: In-place

Solution

Time: 0, ms

13/04/2026 12.25.09

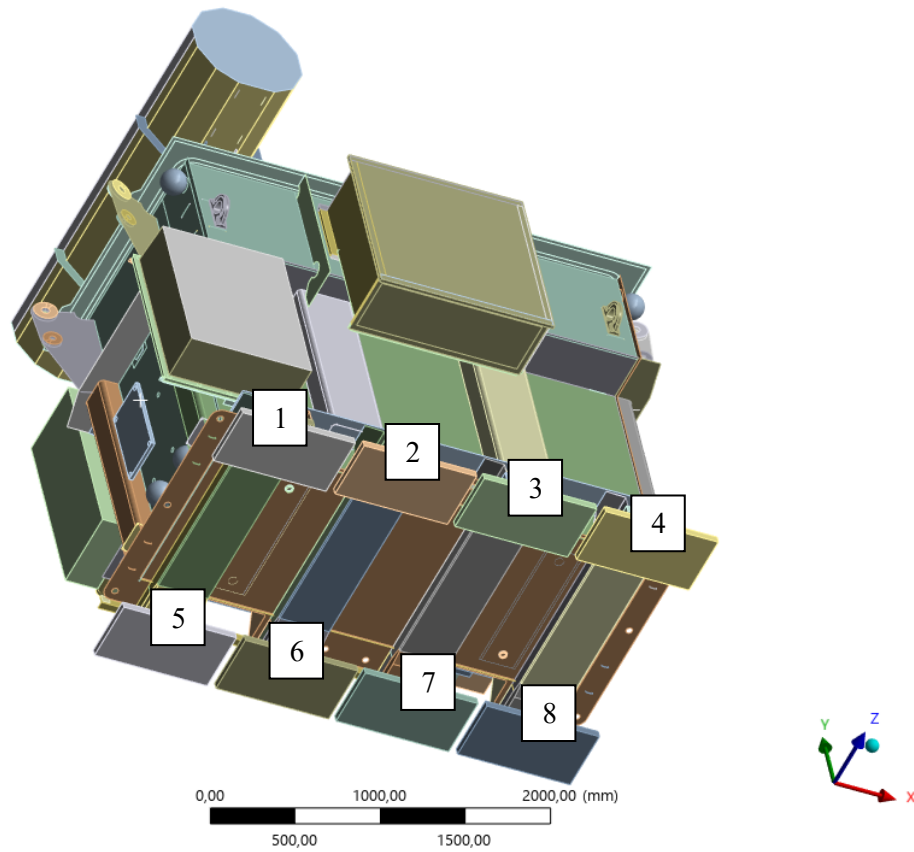


Figure 38 Numbering for reaction force results for Light Transformer.

Table 27 Reaction forces per foot. Element numbering in accordance with Figure 32.

	Maximum horizontal force [kN]	Maximum vertical force [kN]
Rubber element 1	14.8	70.2
Rubber element 2	14.4	66.4
Rubber element 3	14.2	63.6
Rubber element 4	14.3	64.6
Rubber element 5	14.8	69.2
Rubber element 6	14.4	64.5
Rubber element 7	14.1	62.6
Rubber element 8	14.1	63.8

Reaction forces on the pad plates indicate that the highest vertical stresses can be observed on the two feet right under the compensator, with feet 1 and 5. All rubber components share the task of resisting horizontal forces, ensuring a balanced load across all feet. No lifting force was observed during any load case.

The mass of the modelled transformer is verified using Equation 18. Example with load case 1:

$$\frac{341\,287\text{ N}}{14.715\text{ m/s}^2} = 23\,193\text{ kg}$$

The difference between load cases for this calculation is < 1 kg. Mass of the model is 23 193 kg which is 3.4% lower than the nominal mass of the transformer provided in the general arrangement drawing (rounded up) and 2.2% lower than the same value taken from a verified and validated analysis using the previous design. The discrepancy between the two analyses is due to a different measuring point between the analyses. Previous analysis measured reaction forces under the pad plate while in the new analysis they were measured under the

shim plate to capture the forces acting on rubber element. This mass discrepancy is acceptable.

Bolt analysis results

From Hitachi Energy bolt calculator, the pre-tightening load envelope was calculated to be from 53.3 kN to 78.5 kN. Thus, the bolt analysis was conducted using tightening force of 53.3 kN and 78.5 kN is used to check for utilization factor of the bolt. Additionally, the M24 centre bolt pre-tightening load envelope was calculated to be from 112.9 kN to 173.9 kN.

H: No Preload In-place with bolts

Axial Force 2

Type: Directional Axial Force(X Axis) (Unaveraged)

Unit: N

Solution Coordinate System

Maximum Over Time s

08/05/2026 16.27.06

5105,9 Max

4538,5

3971,2

3403,9

2836,6

2269,3

1702

1134,6

567,32

-9,3786e-39 Min

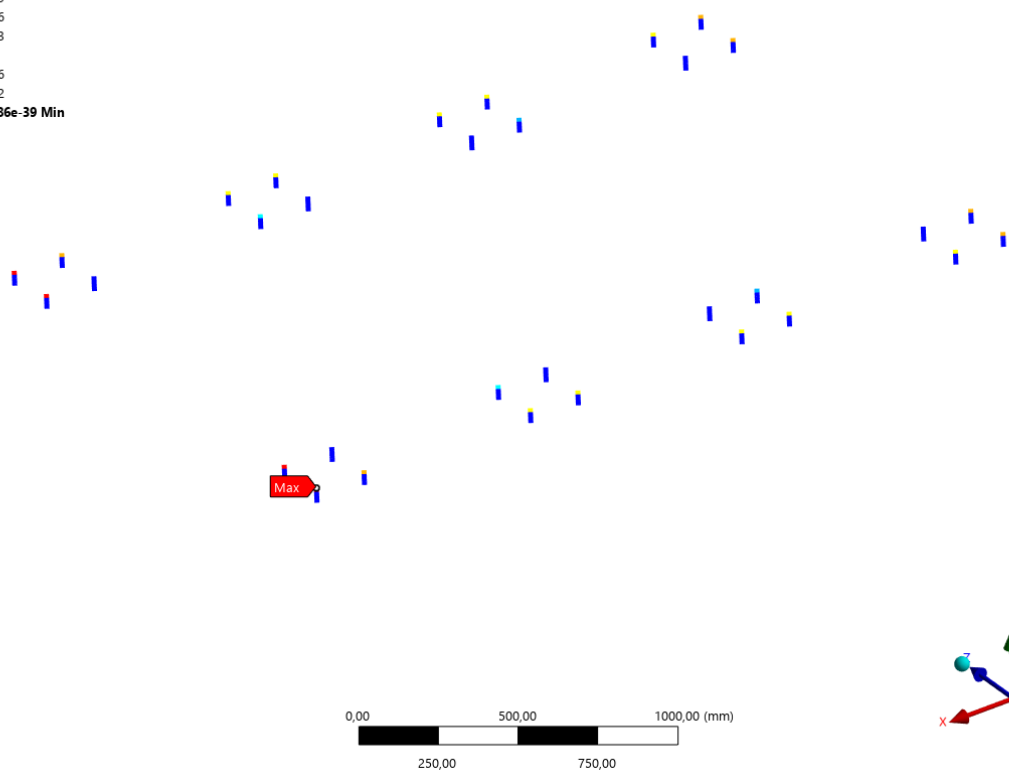


Figure 39 Maximum axial force over time for Light Transformer element bolts.

Axial forces were analysed with no preload present. As shown in Figure 39 the maximum axial force is observed in the corner most bolt, which is to be expected. Axial forces only show up on outer three bolts in each foot as the centre most bolts are always under compression rather than tension like the others. The loaded segments of the bolt beams are observed to

bear substantial loads compared to the lower part, attributable to the modelling approach applied to the beams. The maximum axial force over time is 5106 N.

H: No Preload In-place with bolts
 Axial Force 2
 Type: Directional Axial Force(X Axis) (Unaveraged)
 Unit: N
 Solution Coordinate System
 Maximum Over Time s
 08/05/2026 16.36.21

5105,9 Max
 4538,5
 3971,2
 3403,9
 2836,6
 2269,3
 1702
 1134,6
 567,32
 -9,3786e-39 Min

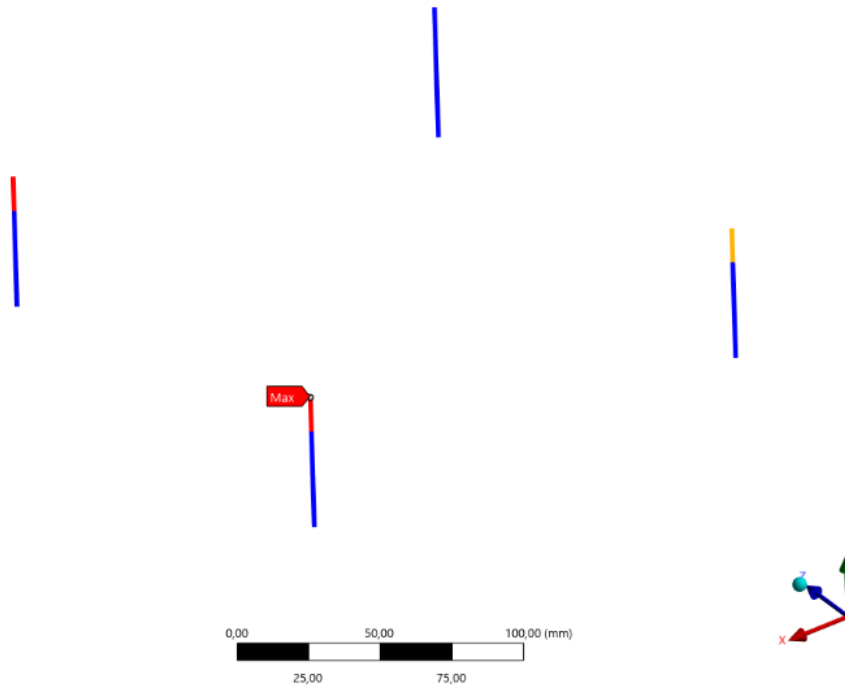


Figure 40 Maximum axial force over time focused on the point of maximum force. (Light Transformer)

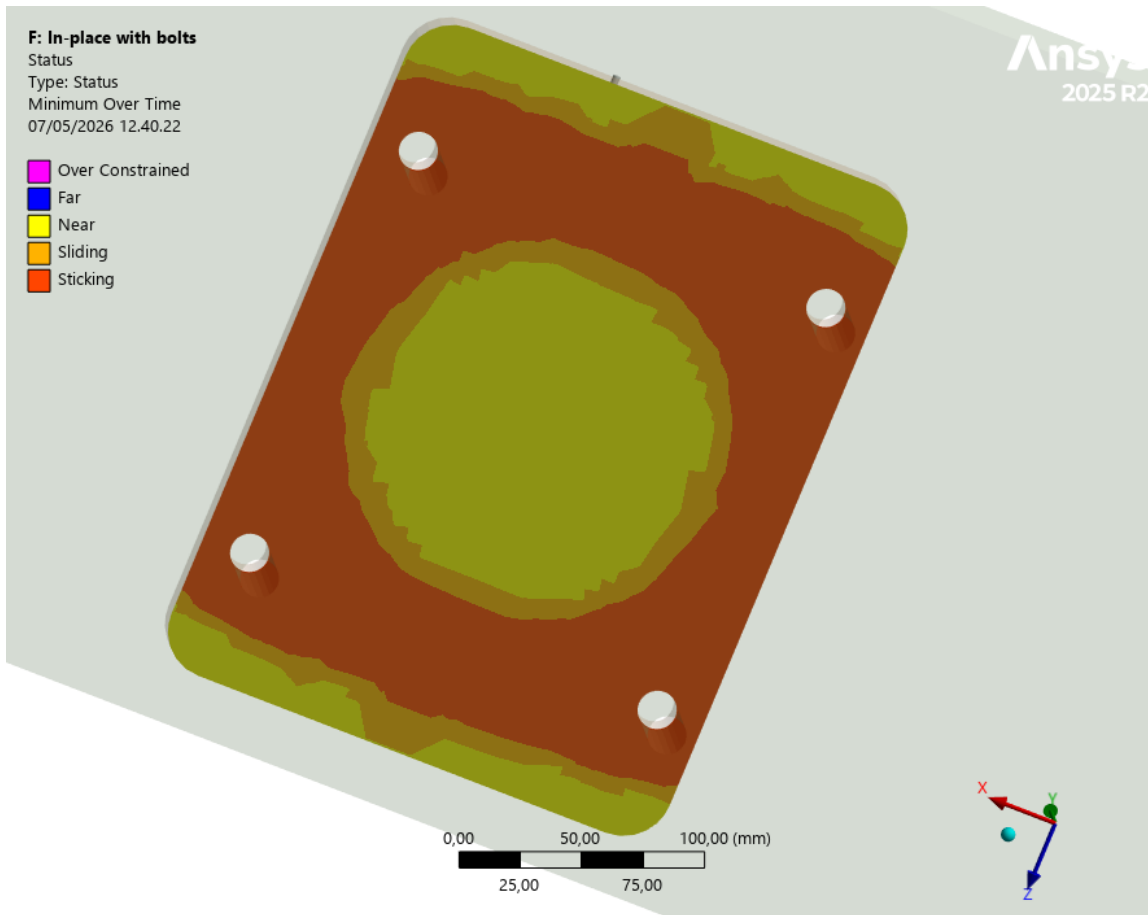


Figure 41 Contact status of minimum friction over time between pad plate and shim plate. (Light transformer)

No slippage was found between the pad plates and shim plates (Figure 41) indicating that the plates are adhering and friction is carrying almost all shear loads as intended.

F: In-place with bolts

Total Bending Moment

Type: Total Bending Moment (Unaveraged)

Unit: N·mm

Maximum Over Time s

11/06/2026 15.25.45

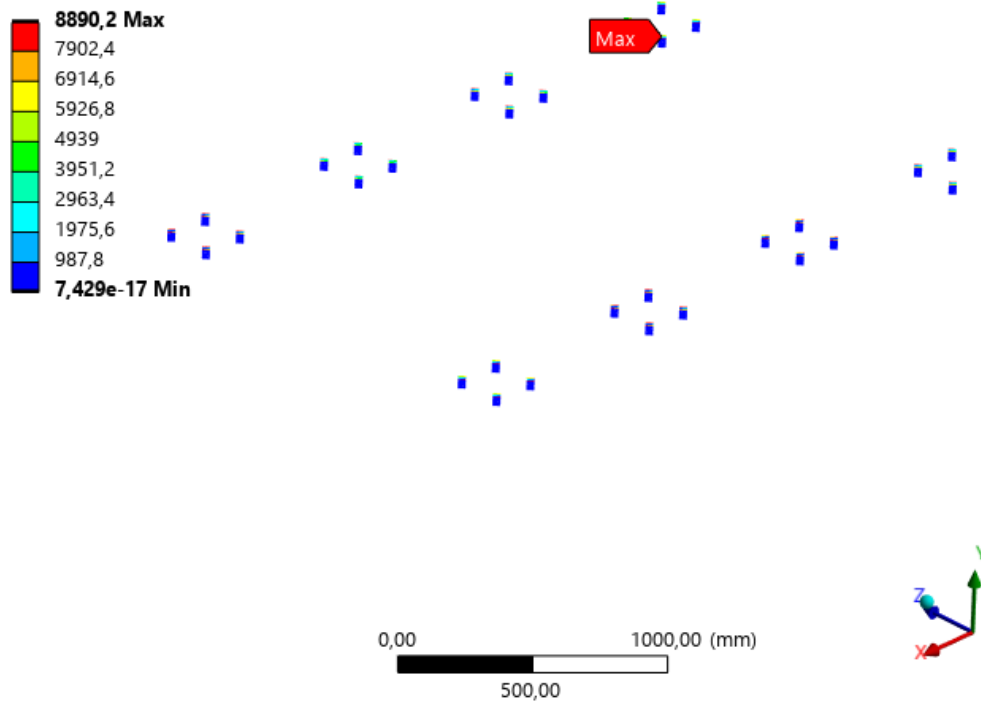


Figure 42 Maximum bending moment over time for Light Transformer element bolts.

Bending moments are more equal among all element bolts compared to axial force. As Figure 42 and Figure 43 indicate the maximum bending moments over time is 8890 Nmm.

F: In-place with bolts

Total Bending Moment

Type: Total Bending Moment (Unaveraged)

Unit: N·mm

Maximum Over Time s

11/06/2026 15.25.49

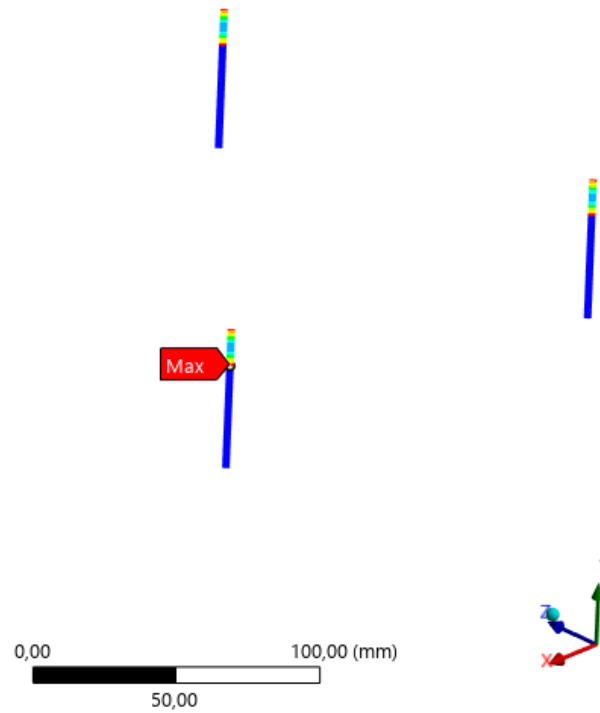
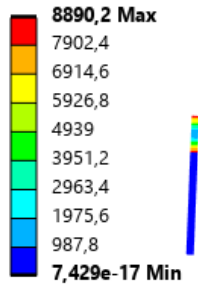


Figure 43 Maximum bending moment over time focused on point of maximum force. (Light Transformer)

Frictional forces acting on the centre bolt are compared to horizontal forces using Equation 24:

$$F_{s \text{ centre}} = 14\,800 \text{ N} - 0.15 \cdot 112\,900 \text{ N}$$

$$F_{s \text{ centre}} = -2135 < 0$$

As the result is below 0, bolt specific bending force is calculated as 10% of the maximum horizontal force:

$$F_{s \text{ centre}} = 0.1 \cdot 14\,800 \text{ N} = 1480 \text{ N}$$

From this bending moment can be calculated using Equation 25:

$$M_{b \text{ centre}} = 1480 \text{ N} \cdot \frac{24 \text{ mm}}{2} = 17\,760 \text{ Nmm}$$

There was no lifting force observed during reaction force analysis and thus centre bolt axial force is calculated as 4 times larger than the maximum axial force on element bolts:

$$F_{SA\ centre} = 4 \cdot 5106\ N = 20\ 424\ N$$

This is a conservative way of calculating axial force.

Combined stress check is done by calculating the utilization factor using Equation 17 and values from Table 28.

Table 28 Bolt properties used for light transformer combined stress check.

Symbol	Element bolts (M16)	Centre bolt (M24)
F_M	78 500 N	173 900N
d	16 mm	24 mm
d_s	14.12 mm	21.19 mm
d_2	14.70 mm	22.05 mm
P	2 mm	3 mm
$\mu_{G\ min}$	0.07	0.07
α	60°	60°
A_S	157 mm ²	353 mm ²
F_{SA}	5106 N	20 424 N
M_b	8890 Nmm	17 760 Nmm

For the M16 element bolts:

$$v \cdot R_{p0.2}$$

$$= \sqrt{\left(\frac{78\,500\text{ N} + 5106\text{ N}}{157\text{ mm}^2} + \frac{8890\text{ Nmm}}{\left(\frac{\pi \cdot 14.12\text{ (mm)}^3}{32} \right)} \right)^2 + 3 \left(\frac{78\,500\text{ N} \cdot \left(\left(\frac{2\text{ mm}}{2\pi} \right) + \left(\frac{14.70\text{ mm} \cdot 0.07}{2 \cdot \cos \frac{60^\circ}{2}} \right) \right)}{\frac{\pi \cdot (14.12\text{ mm})^3}{12}} \right)^2}$$

$$= 589.2\text{ MPa}$$

$$v \cdot R_{p0.2} = 589.2\text{ MPa}$$

$$v = \frac{589.2\text{ MPa}}{600\text{ MPa}} = 0.98 < 1$$

For the M24 centre bolts:

$$v \cdot R_{p0.2}$$

$$= \sqrt{\left(\frac{173\,900\text{ N} + 20\,424\text{ N}}{353\text{ mm}^2} + \frac{17\,760\text{ Nmm}}{\left(\frac{\pi \cdot 21.19\text{ (mm)}^3}{32} \right)} \right)^2 + 3 \left(\frac{173\,900\text{ N} \cdot \left(\left(\frac{3\text{ mm}}{2\pi} \right) + \left(\frac{22.05\text{ mm} \cdot 0.07}{2 \cdot \cos \frac{60^\circ}{2}} \right) \right)}{\frac{\pi \cdot (21.19\text{ mm})^3}{12}} \right)^2}$$

$$= 593.1\text{ MPa}$$

$$v \cdot R_{p0.2} = 593.1\text{ MPa}$$

$$v = \frac{593.1\text{ MPa}}{600\text{ MPa}} = 0.99 < 1$$

The utilization factors for both bolts are less than 1, indicating that the bolts are considered capable of effectively securing the connection. Checking stress with bending moment from low preload analysis and tightening stress from high preload makes this check conservative as, even with frictional-type joints, bending moment is reduced when preload is higher.

Modal analysis results

Table 29 Modal analysis results for Light Transformer

Mode	Direction	Frequency (Hz)	Effective mass (%)		
			X- direction	Y- direction	Z- direction
1	Z	4.27	0.01	0.04	70.54
2	X	4.52	62.17	0.02	0.01
3	Y	8.53	0.05	94.67	0.06
4	Rotate Y	9.96	0.01	0	0.02
5	Rotate Z	12.25	32.51	0.03	.0.02
6	Rotate X	14.09	0.01	0.02	24.09
Sum			94.76	94.76	94.75

The first six modes capture approximately 95% of the effective mass in all direction, indicating that higher modes have negligible contribution to the acceleration-based response. All six natural modes analysed are dominated by rigid-body motion of the transformer on its feet, while structural deformation occurs only at higher frequencies. This behaviour is expected for heavy offshore equipment mounted on compliant interfaces and indicates that the global dynamic response is governed by support stiffness rather than internal structural flexibility.

Validating the results is conducted with calculating f_y as per Equation 28 and comparing it to the analysis result of 8.5314 Hz:

$$f_y = \frac{1}{2\pi} \sqrt{\frac{(6250 \text{ N/mm} \cdot 1000 \cdot 8)}{24\,000 \text{ kg}}} = 7.2644 \text{ Hz}$$

Analysis result has an 14.9% error compared to f_y which is well within acceptable limits. It is typical that the FEA results are higher than the analytical results as FEA captures effects such as partial stiffness of the frame, non-uniform load sharing between feet and effective mass being lower than total mass.

Transport analysis

For transport analysis bolt preload was calculated to be from 180 to 270 kN. Maximum and minimum preloads were both analysed. No preload was used when analysing axial forces on bolts.

Sum of downward reaction forces were calculated using results from load steps that had downward acceleration and Equation 18. The mass of the model comes in at 23 543 kg which is ~1.9% higher than the transformer as per general arrangement drawing. Deviation between highest and lowest masses are <1 kg. Reaction forces indicate load-case-sensitivity and stress results as expected. Loads are transmitted through expected paths (active parts → tank → beams → transport supports) with a slight increase of stress on the side of the compensator.

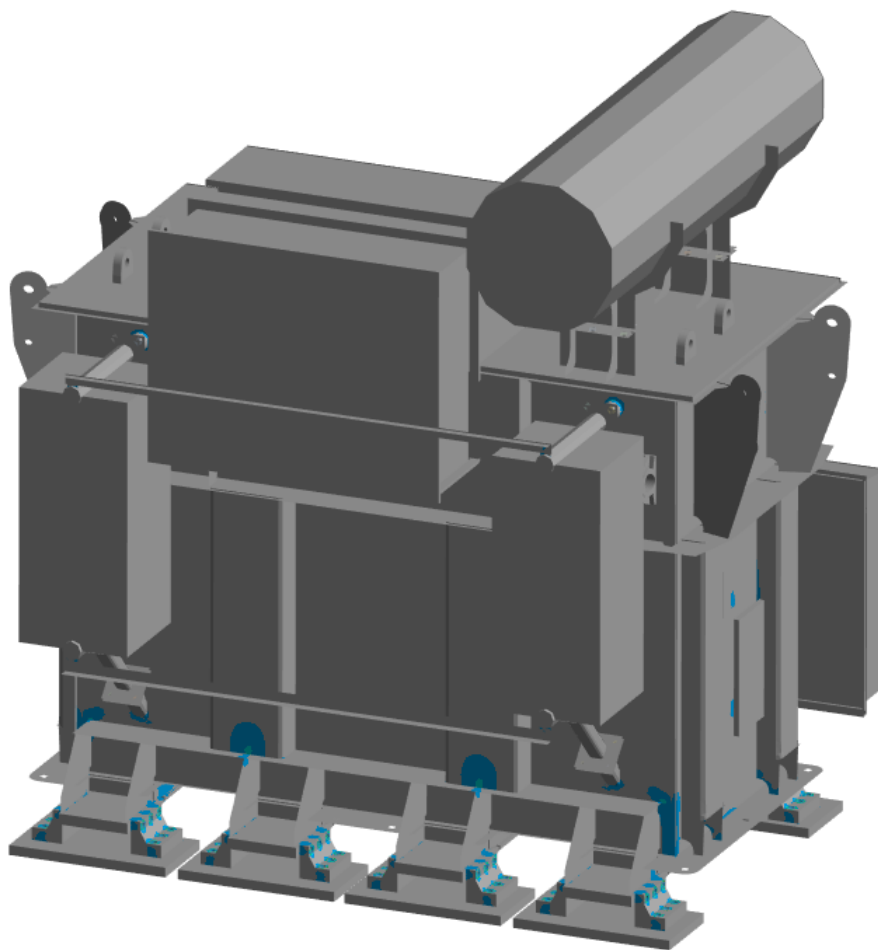


Figure 44 Global stress analysis for light transformer transport analysis.

Global transport analysis indicates stress is most prevalent around the bolt joint areas (Figure 44). Stresses were analysed with bolt preloads active. Figure 46 shows the point of peak maximum stress over time which is 280 MPa at the edge of a bolt hole. The placement of the stress is due to simplifications made in the modelling of the bolt beams where the “thread part” of the beam is connected to the bolt hole. The maximum stress would be at the threads of the screw, but this gives a more efficient model with a conservative peak stress. Maximum nominal stresses are ~180 MPa. The maximum allowable stress for the foot pad is 285 MPa, thus stress is within allowable limits.

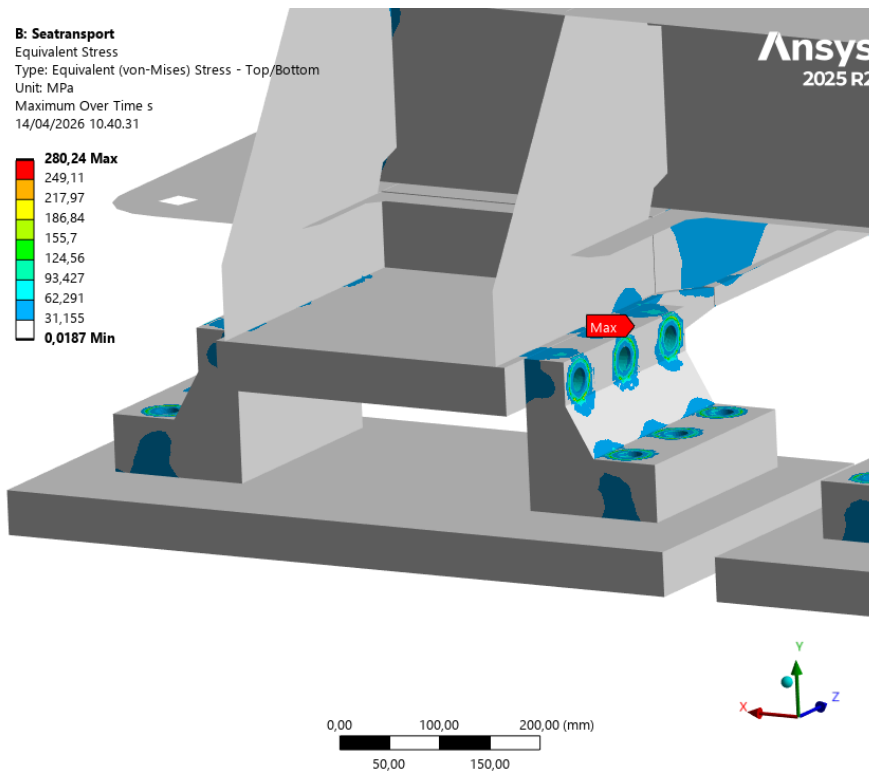


Figure 45 Maximum equivalent von Mises stress over time for light transformer transport support.

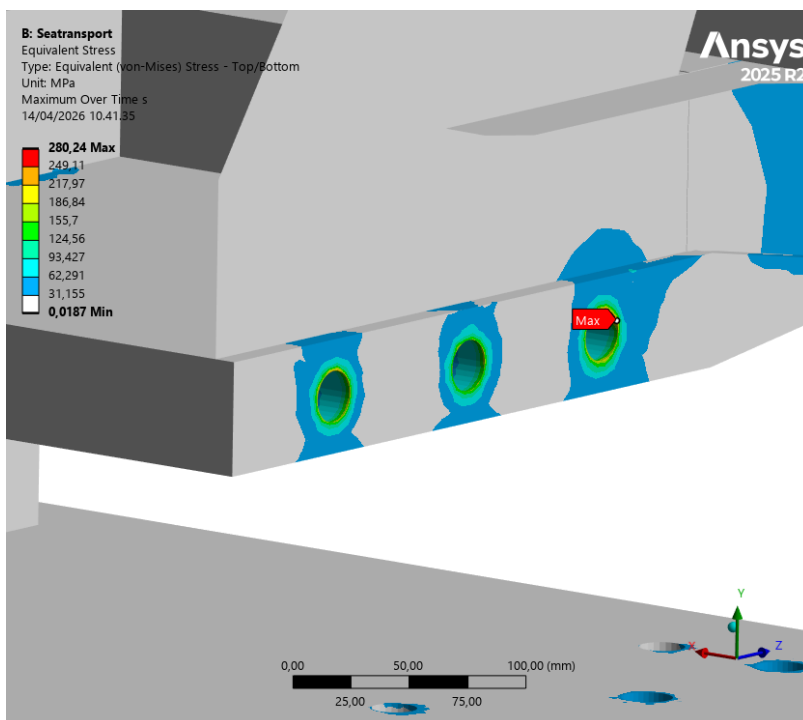


Figure 46 Maximum equivalent von Mises stress over time for light transformer transport supports with point of maximum stress visible.

Horizontal and vertical TS-bolts have large differences in their loads although for both force peaks in the corner most feet assemblies. Figure 47 and Figure 48 indicate that the outer most bolts are most loaded with the vertical bolts while the inner most row of bolts is most loaded with the horizontal bolts. For vertical bolts this difference can be explained by the higher moment force outer bolts experience due to their longer distance to the centre of gravity, while for horizontal bolts the uneven loading is due to small elastic deformations in the tips of the beam structures. Vertical bolts peak at 19 374 N and horizontal bolts at 9666 N for axial forces.

B: Sea transport without preload

Axial Force
Type: Directional Axial Force(X Axis) (Unaveraged)
Unit: N
Solution Coordinate System
Maximum Over Time s
13/05/2026 11:31:06

19374 Max
17222
15069
12916
10764
8610,9
6458,1
4305,4
2152,7
6,7514e-16 Min

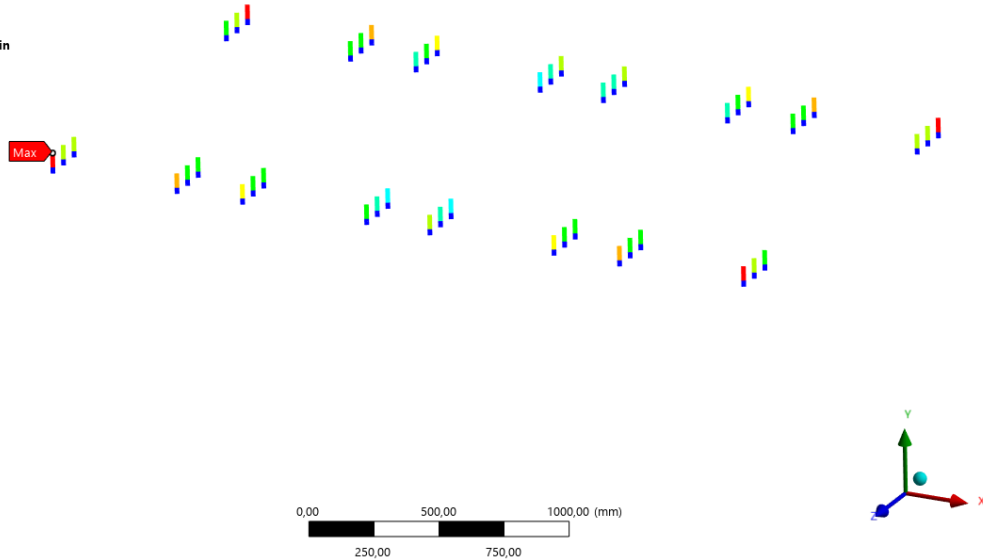


Figure 47 Vertical bolt maximum axial force over time analysed without preload. (Light transformer)

B: Sea transport without preload

Axial Force 2

Type: Directional Axial Force(X Axis) (Unaveraged)

Unit: N

Solution Coordinate System

Maximum Over Time s

13/05/2026 11.32.17

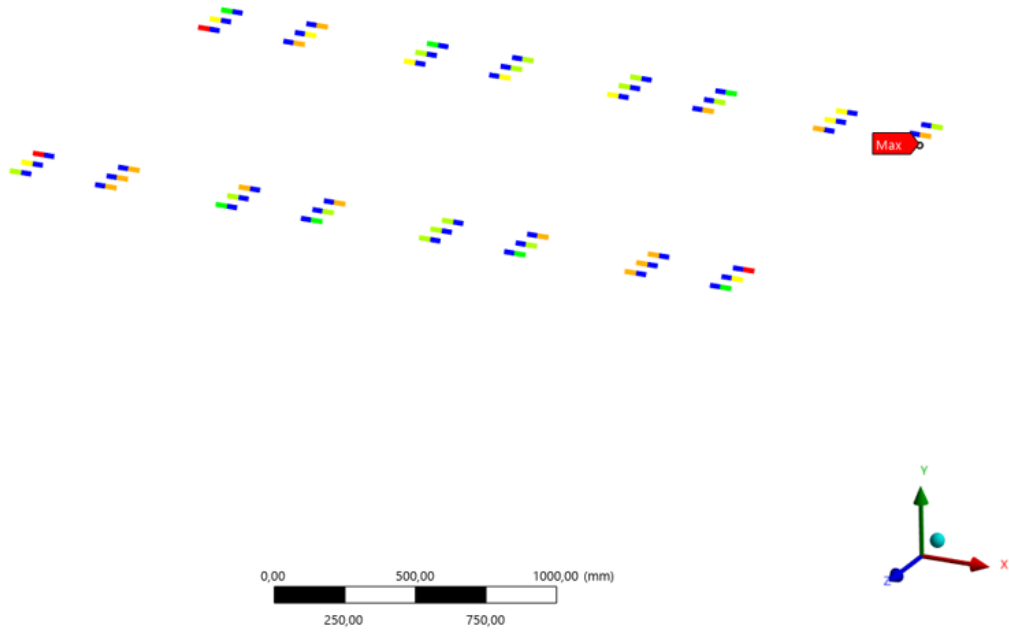
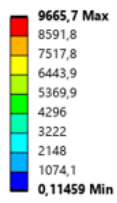


Figure 48 Horizontal bolt maximum axial force over time analysed without preload. (Light transformer)

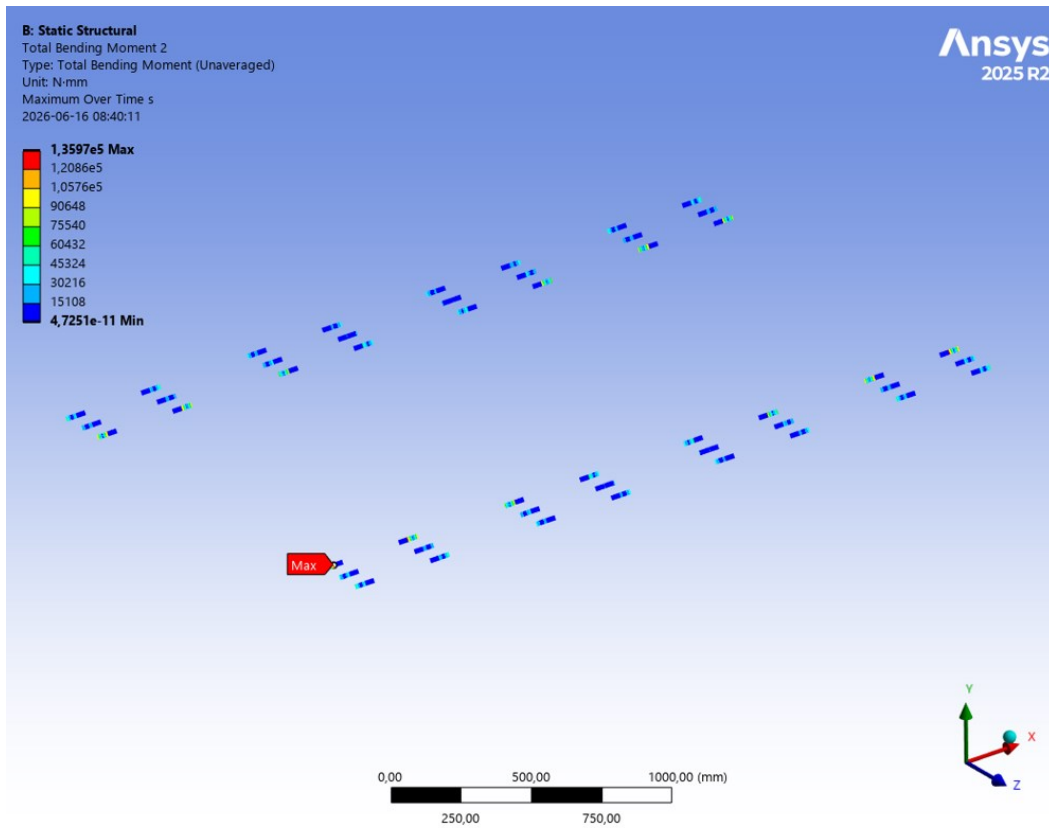


Figure 49 Vertical bolt maximum bending moment over time analysed with preload. (Light transformer)(Minimum preload)

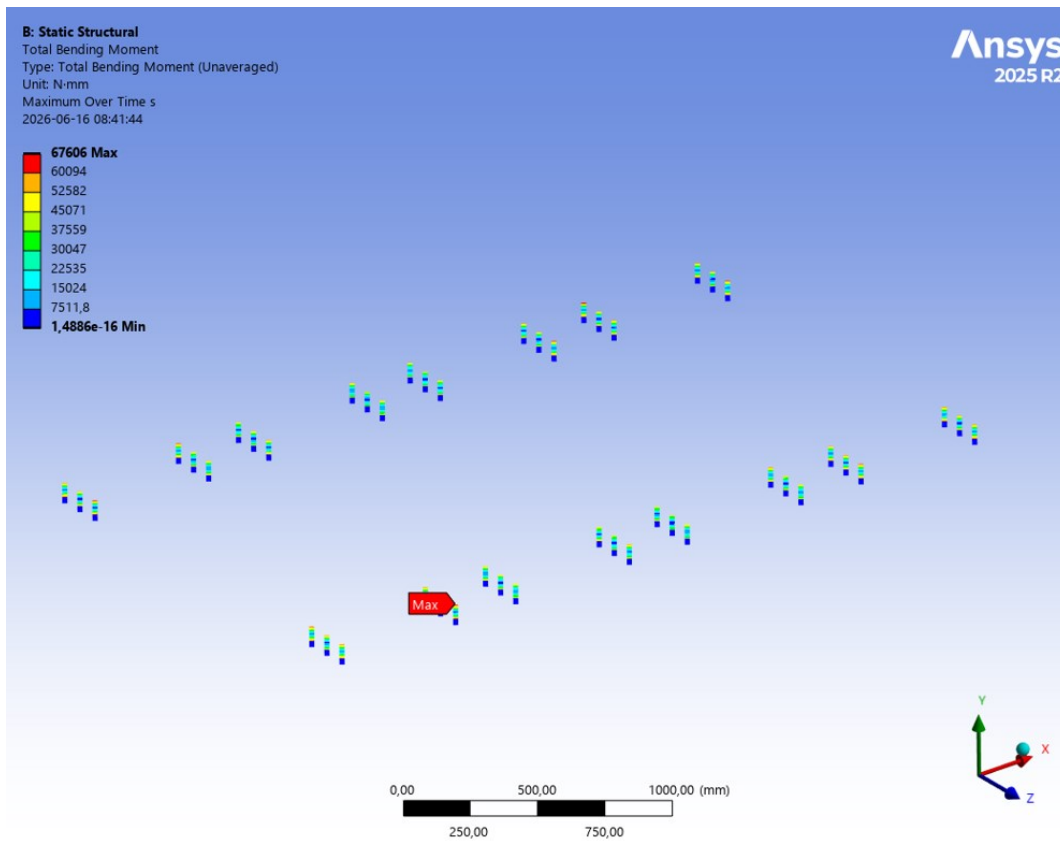


Figure 50 Horizontal bolt maximum bending moment over time analysed with preload. (Light transformer)(Minimum preload)

For bending moment magnitudes are reversed in comparison with axial force results (Figure 49, Figure 50). Vertical bolts have a smaller bending moment while horizontal have a larger moment as can be expected considering the load paths in this design. While horizontal bolts are most loaded in the same points as in the axial study, vertical bolts peak at the corner feet assemblies where load is more equally distributed. Peak bending moments for horizontal bolts are 135 970 Nmm and 108 620 Nmm with low and high preloads respectively. Vertical bolts had peak bending moments of 67 606 Nmm and 70 264 Nmm with low and high preloads respectively. These are node specific peaks (Figure 51, Figure 52), for the stress check an average for the loaded part of the bolt was calculated which are shown in Table 30. Pins under the transport supports took no loads (axial or shear) and were thus excluded from the results.

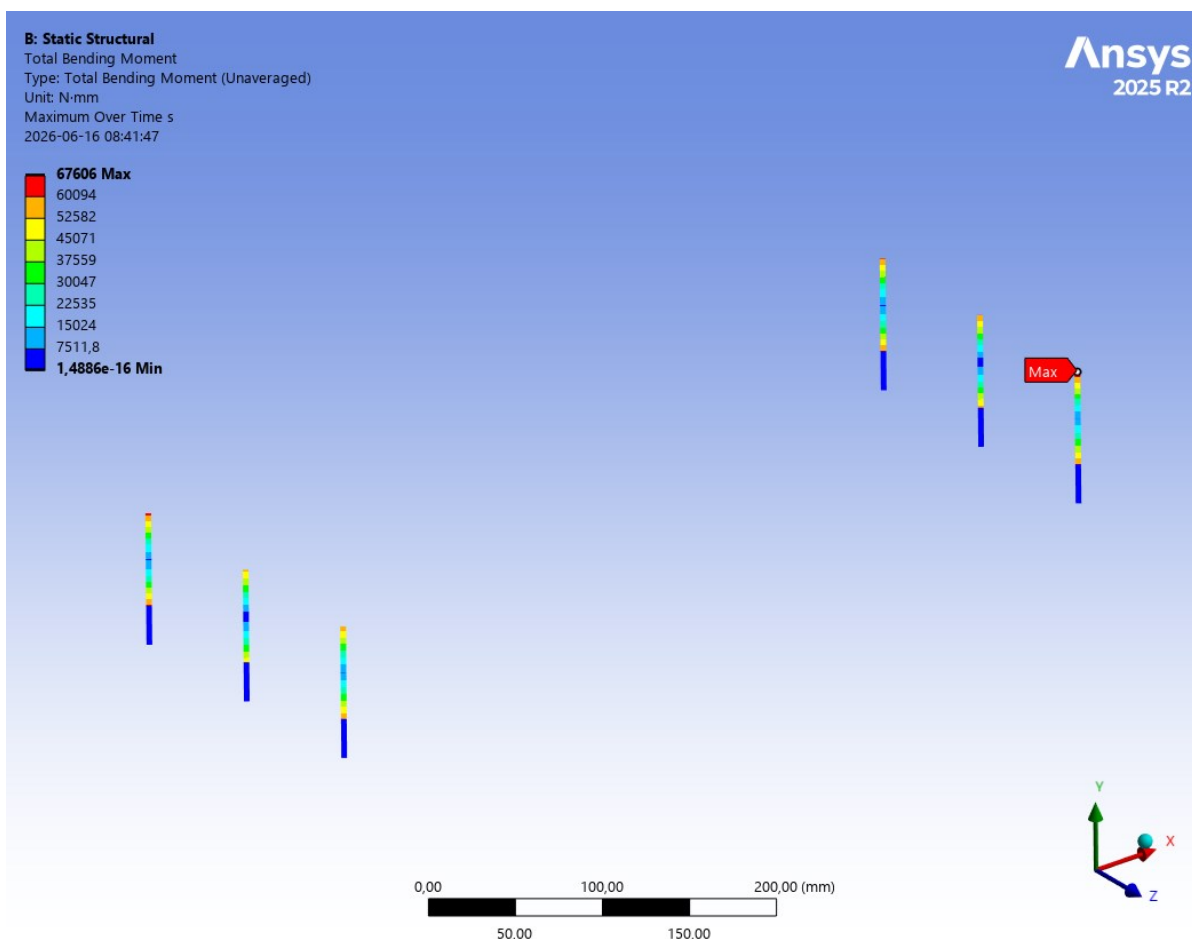


Figure 51 Maximum bending moment over time focused on foot element with highest moment on vertical bolts. (Minimum preload)

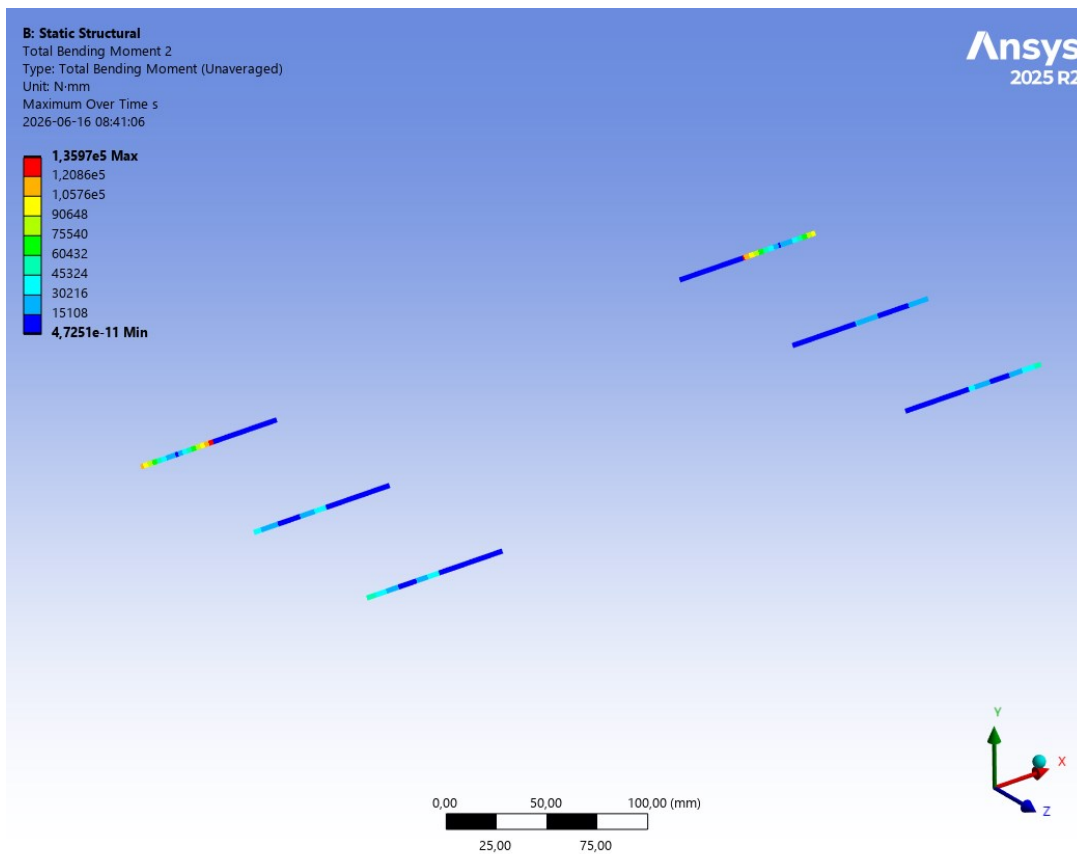


Figure 52 Maximum bending moment over time focused on foot element with highest moment on horizontal bolts. (Minimum preload)

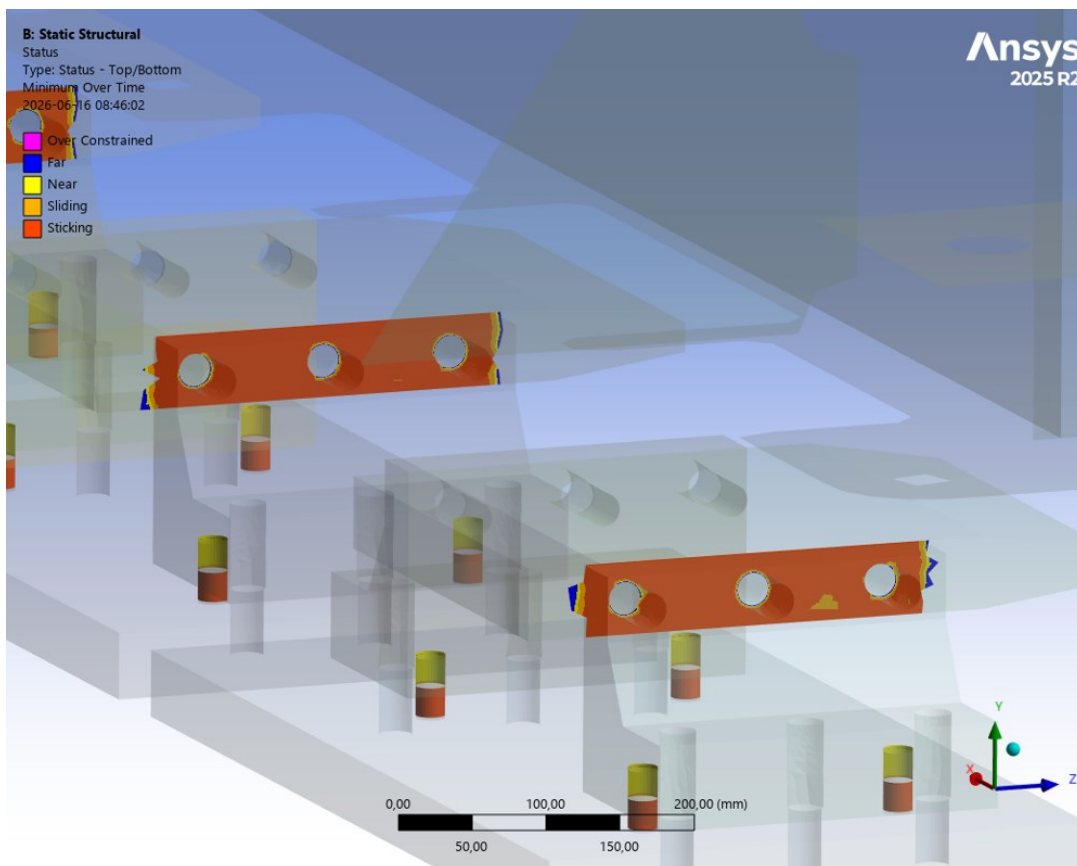


Figure 53 Contact status showing minimum friction over time with lowest preloads active. (Light transformer)

Figure 53 indicates that the contacts status between foot pad and transport support is a frictional-type interface with the friction shim providing adequate friction which greatly helps to alleviate bending moments on the bolts.

Table 30 Bolt properties used for light transformer's transport combined stress check. Bending moments averaged.

Symbol	Horizontal bolts (M24)	Vertical bolts (M24)
$F_{M \min}$	180 000 N	180 000 N
$F_{M \max}$	270 000N	270 000N
d	24 mm	24 mm
d_s	21.19 mm	21.19 mm
d_2	22.05 mm	22.05 mm
P	3 mm	3 mm
$\mu_G \min$	0.07	0.07
α	30°	30°
A_S	353 mm ²	353 mm ²
F_{SA}	9666 N	19 374 N
$M_{b \min}$	63 833 Nmm	35 189 Nmm
$M_{b \max}$	64 854 Nmm	36 378 Nmm

Combined stress check is done by calculating the utilization factor using Equation 17 and values from Table 30.

Horizontal bolts with minimum preload:

$$v \cdot R_{p0.2}$$

$$= \sqrt{\left(\frac{180\,000\text{ N} + 9666\text{ N}}{353\text{ mm}^2} + \frac{63\,833\text{ Nmm}}{\left(\frac{\pi \cdot 21.19\text{ (mm)}^3}{32} \right)} \right)^2 + 3 \left(\frac{180\,000\text{ N} \cdot \left(\left(\frac{3\text{ mm}}{2\pi} \right) + \left(\frac{22.05\text{ mm} \cdot 0.07}{2 \cdot \cos \frac{60^\circ}{2}} \right) \right)}{\frac{\pi \cdot (21.19\text{ mm})^3}{12}} \right)^2}$$

$$v \cdot R_{p0.2} = 601\text{ MPa}$$

$$v = \frac{601}{900} = 0.67$$

Horizontal bolts with maximum preload:

$$v \cdot R_{p0.2}$$

$$= \sqrt{\left(\frac{270\,000\text{ N} + 9666\text{ N}}{353\text{ mm}^2} + \frac{64\,854\text{ Nmm}}{\left(\frac{\pi \cdot 21.19\text{ (mm)}^3}{32} \right)} \right)^2 + 3 \left(\frac{270\,000\text{ N} \cdot \left(\left(\frac{3\text{ mm}}{2\pi} \right) + \left(\frac{22.05\text{ mm} \cdot 0.07}{2 \cdot \cos \frac{60^\circ}{2}} \right) \right)}{\frac{\pi \cdot (21.19\text{ mm})^3}{12}} \right)^2}$$

$$v \cdot R_{p0.2} = 899\text{ MPa}$$

$$v = \frac{899}{900} = 0.999$$

Vertical bolts with minimum preload:

$$v \cdot R_{p0.2}$$

$$= \sqrt{\left(\frac{180\,000\text{ N} + 19\,374\text{ N}}{353\text{ mm}^2} + \frac{35\,189\text{ Nmm}}{\left(\frac{\pi \cdot 21.19\text{ (mm)}^3}{32} \right)} \right)^2 + 3 \left(\frac{180\,000\text{ N} \cdot \left(\left(\frac{3\text{ mm}}{2\pi} \right) + \left(\frac{22.05\text{ mm} \cdot 0.07}{2 \cdot \cos \frac{60^\circ}{2}} \right) \right)}{\frac{\pi \cdot (21.19\text{ mm})^3}{12}} \right)^2}$$

$$v \cdot R_{p0.2} = 629\text{ MPa}$$

$$v = \frac{629}{900} = 0.70$$

Vertical bolts with maximum preload:

$$v \cdot R_{p0.2}$$

$$= \sqrt{\left(\frac{270\,000\text{ N} + 19\,374\text{ N}}{353\text{ mm}^2} + \frac{36\,378\text{ Nmm}}{\left(\frac{\pi \cdot 21.19\text{ (mm)}^3}{32} \right)} \right)^2 + 3 \left(\frac{270\,000\text{ N} \cdot \left(\left(\frac{3\text{ mm}}{2\pi} \right) + \left(\frac{22.05\text{ mm} \cdot 0.07}{2 \cdot \cos \frac{60^\circ}{2}} \right) \right)}{\frac{\pi \cdot (21.19\text{ mm})^3}{12}} \right)^2}$$

$$v \cdot R_{p0.2} = 896\text{ MPa}$$

$$v = \frac{896}{900} = 0.996$$

All bolts have a utilization ratio below 1 and are thus below the yield limit. Ansys has a beam tool that gives maximum combined stress results for all bolts modelled as beams. These results indicate that maximum stress for a bolt during transport analyses is 665 MPa compared to the maximum of 899 MPa indicated by the VDI stress check. Thus, the VDI

combined stress check can be considered to be highly conservative and that even high utilization factor bolts are robust.

5.1.2 Medium Transformer

Stress results

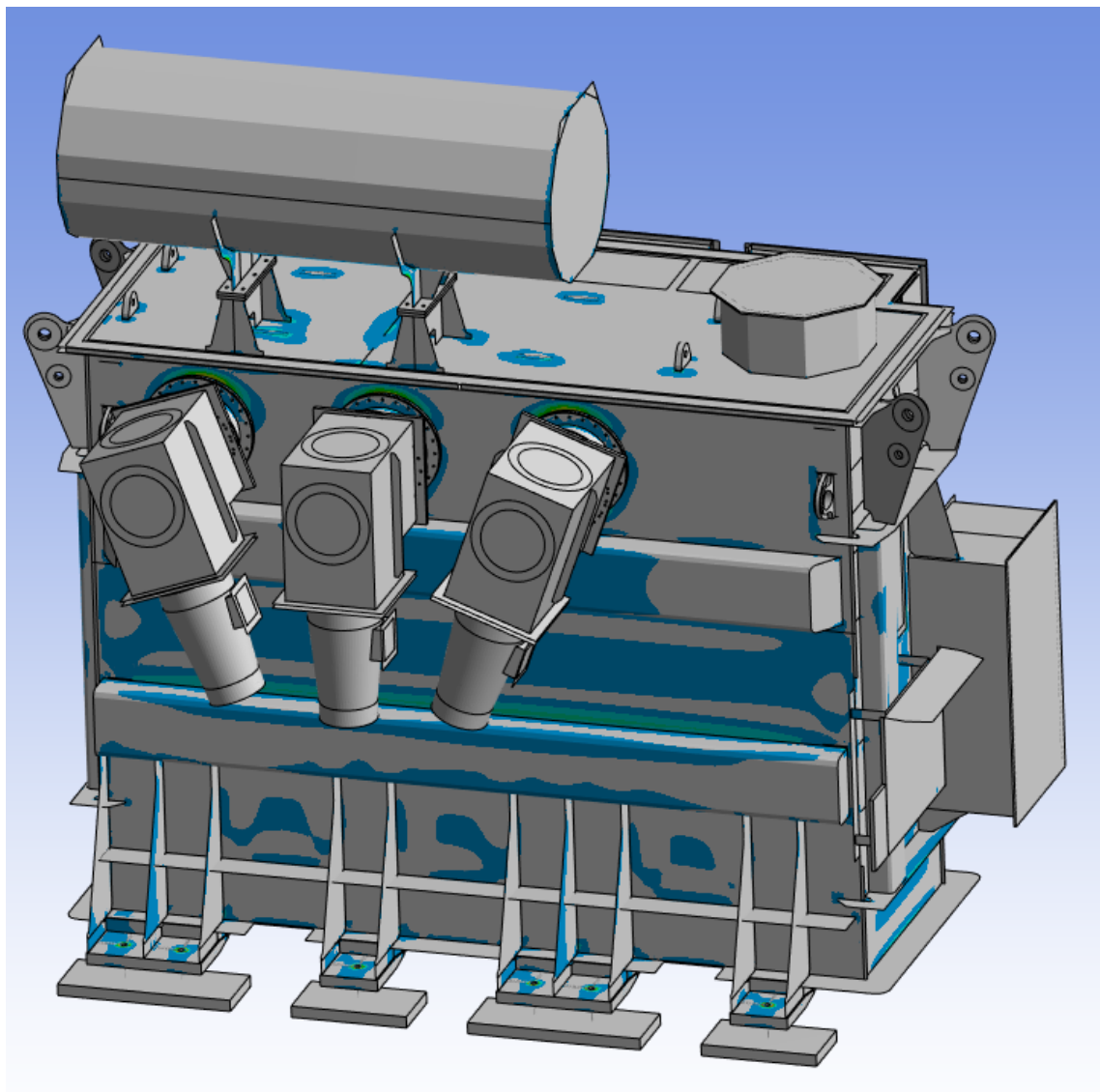


Figure 54 Overview of stress patterns across whole transformer. (Medium Transformer)

Figure 54 indicates that medium transformer exhibits stress patterns comparable to those observed in the light transformer. The highest stress is concentrated around the centre washer area and remains significantly below the maximum allowable limit.

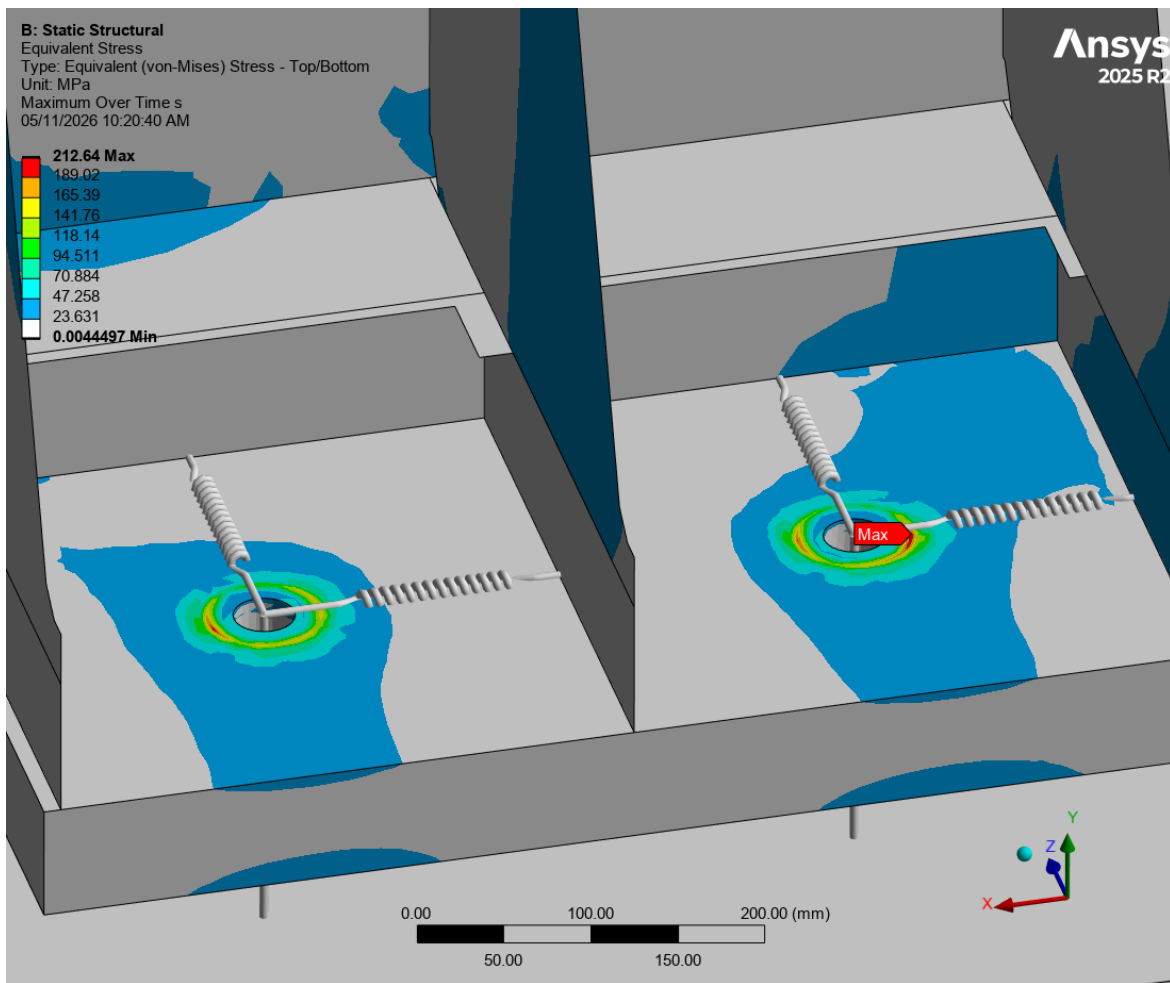


Figure 55 Maximum equivalent von Mises stress over time for Medium Transformer foot plates.

Washer surfaces in Figure 55 indicate a maximum peak stress of 213 MPa. The nominal maximum stress levels are ~160 MPa which is well below the maximum allowable stress of 291 MPa for this type of plate.

Deformation results

Similarly, as in the Light transformer analysis, maximum deformation over time increases the farther away from the feet you move. However, deformation is higher on one side due to the uneven placement of the active part inside the tank as shown in Figure 56.

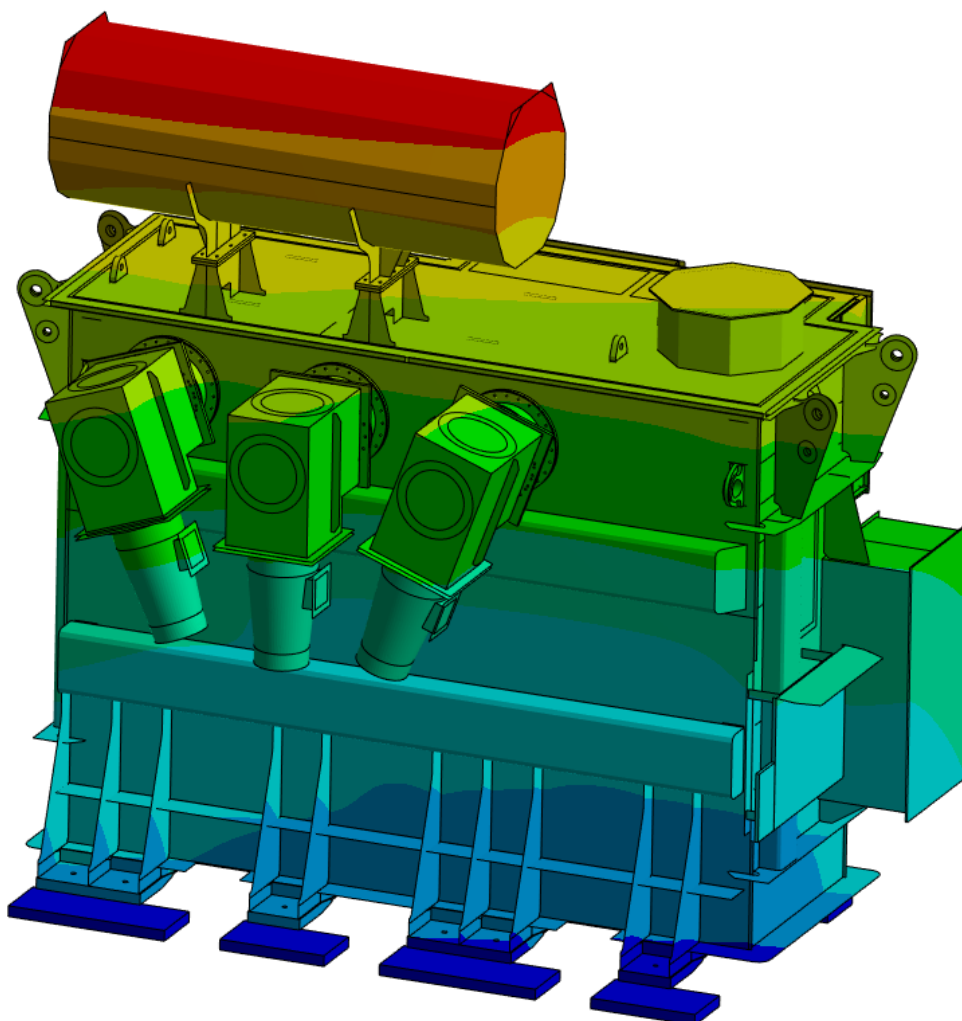


Figure 56 Overview of maximum deformation over time for the whole tank. (Medium Transformer)

The concept rubber elements behave as expected (Figure 57). Deformation is about 50% higher than with the smaller elements, which is on par with size and stiffness difference between them. Static deflection is 6.6 mm, and maximum dynamic deflection is ~17.4 mm. The greatest horizontal deflection occurs in the Z direction (transversal) and is approximately 3.3 mm while highest vertical deflection is downwards ~17.1 mm. Vertical deformation is verified with highest vertical force from

Table 27 and spring coefficient from Table 11 using Hooke's law (Equation 23):

$$\frac{171\,600\text{ N}}{10\,300\text{ N/mm}} = 16.66\text{ mm}$$

The difference between analyzed and calculated deflections is due to the force not dividing precisely equally between the rubber elements in the double beam. However, the difference is acceptable.

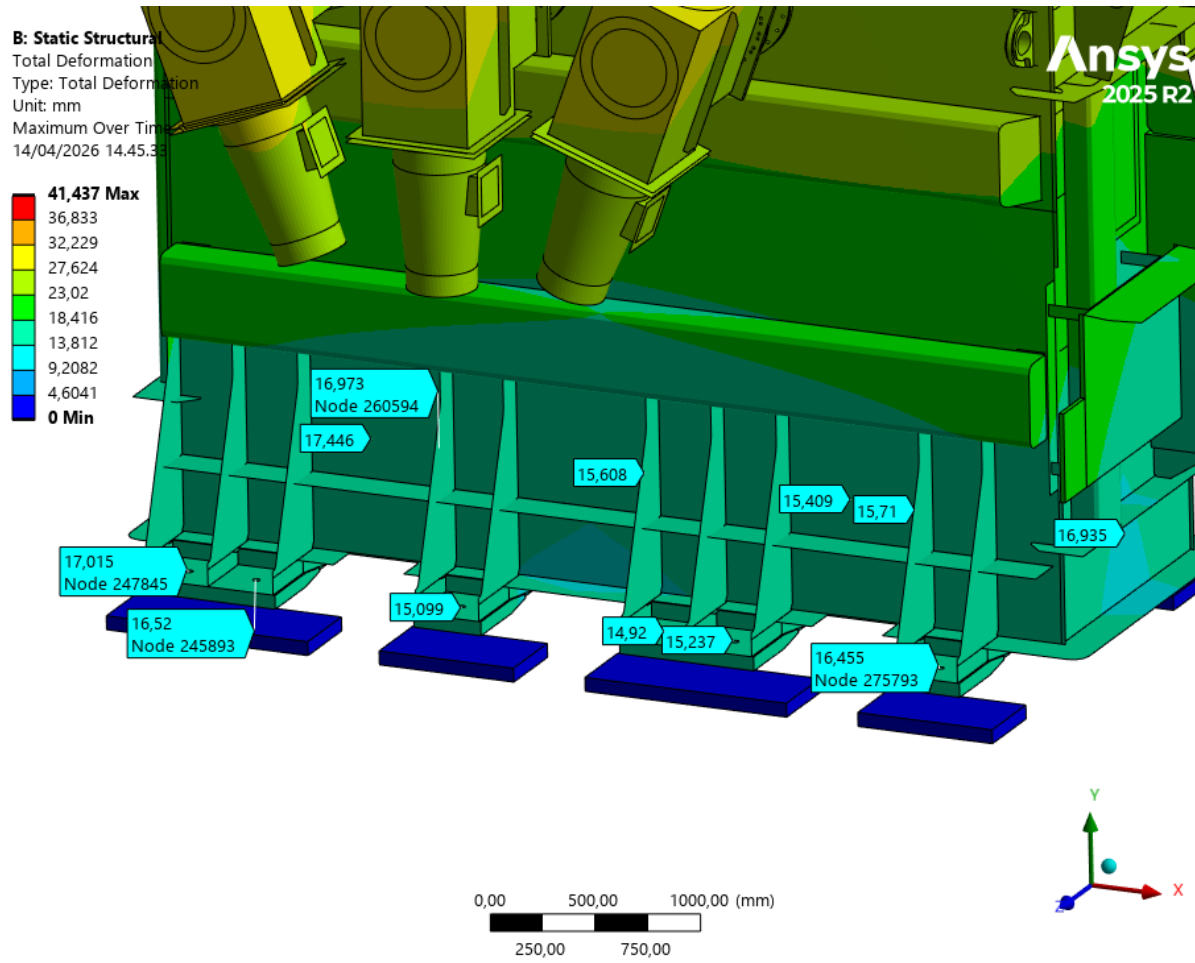


Figure 57 Maximum deformation over time for Medium Transformer.

Reaction force results

B: Static Structural

Solution

Time: 11, s

14/04/2026 14.48.39

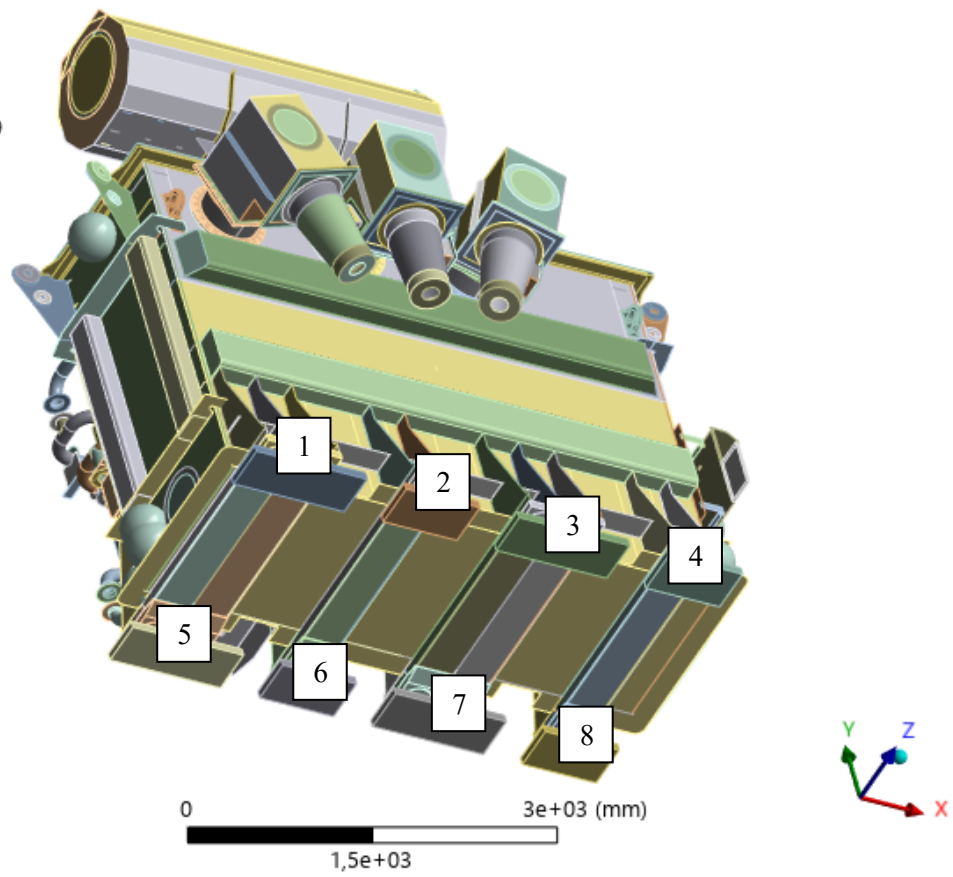


Figure 58 Numbering for reaction force results for Medium Transformer.

Table 31 Reaction forces per foot. Numbering in accordance with Figure 36. (Medium Transformer)

	Maximum horizontal force [kN]	Maximum vertical force per pad [kN]	Maximum vertical force per rubber element [kN]
Pad plate 1	30.5	334.4	167.2
			167.2
Pad plate 2	30.5	148.2	148.2
Pad plate 3	30.4	301.4	150.7
			150.7
Pad plate 4	30.6	168.0	168.0
Pad plate 5	30.6	343.1	171.6
			171.6
Pad plate 6	30.7	152.7	152.7
Pad plate 7	30.5	309.7	154.9
			154.9
Pad plate 8	30.8	171.5	171.5

The corner pads, especially those under the conservator, experience the greatest vertical stresses. In the medium transformer, the active part is also offset towards the -X direction, which results in extra inertial forces on the walls. These forces are mostly transferred to pad plates 1 and 5, which explains the load disparity. As with the previous reaction force analysis horizontal forces are balanced across all pads. A maximum lifting force on a single rubber element was observed as 34 442 N; this is used as maximum axial force for centre bolts.

The mass of the modelled transformer is verified using Equation 18. Example with load case 1:

$$\frac{1\,139\,589\text{ N}}{13.19\text{ m/s}^2} = 86\,398\text{ kg}$$

The difference between load cases for this calculation is < 3 kg. The model weighs 86 398 kg which is 4.2% higher than the nominal mass of the transformer provided in the general arrangement drawing (rounded up) and 4.3% higher than the same value taken from a verified and validated analysis using the previous design where due to the measuring point it does not include feet structures. This mass discrepancy is explained by the mass increase due to additional beam structures and the measuring point disparity, thus the discrepancy acceptable.

Bolt analysis results

For bolt loads Hitachi bolt calculator yielded pre-tightening load envelopes of 117.7 to 166kN for the M24 element bolt and 269.9 to 407.3 kN for the M36 centre bolt. Thus 117.7 kN was used for this analysis.

B: Bolt analysis (No preload)
 Axial Force
 Type: Directional Axial Force(X Axis) (Unaveraged)
 Unit: N
 Solution Coordinate System
 Maximum Over Time s
 11/05/2026 9:12:39

21783 Max
 19319
 16855
 14391
 11927
 9462,6
 6998,5
 4534,4
 2070,3
 -393,73 Min

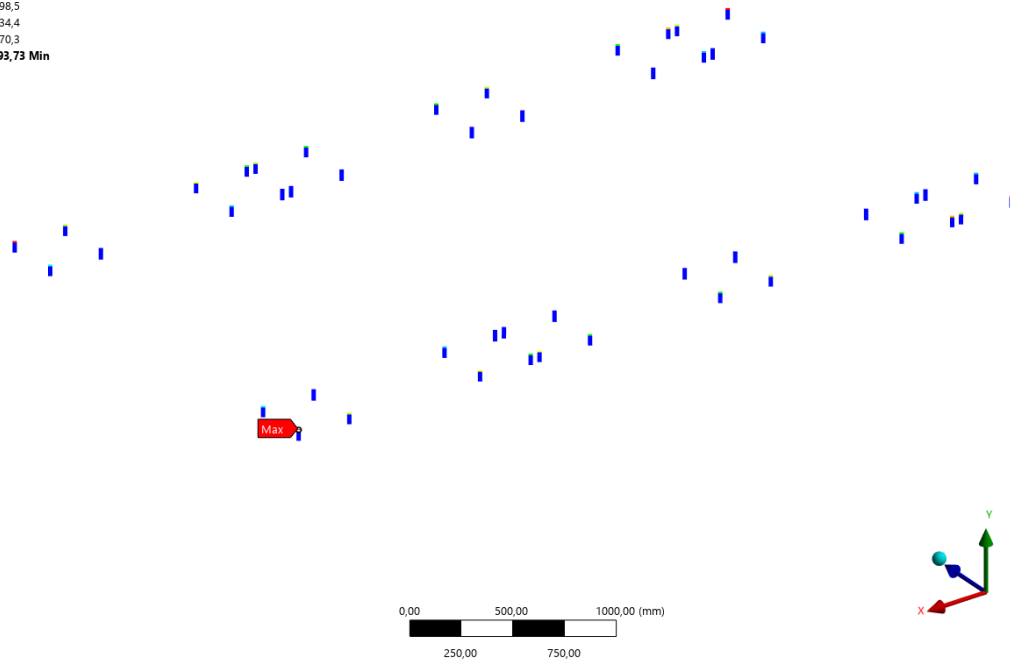


Figure 59 Maximum axial force over time for Medium Transformer element bolts.

Axial force, measured without preload, is most prevalent in the corner most bolts as shown in Figure 59. Similarly, as for the light transformer the inner most bolts of each foot are in compression rather than tension which explains the force difference seen in Figure 60. The maximum axial force over time is 21 783 N.

B: Bolt analysis (No preload)
Axial Force
Type: Directional Axial Force(X Axis) (Unaveraged)
Unit: N
Solution Coordinate System
Maximum Over Time s
11/05/2026 9.12.44

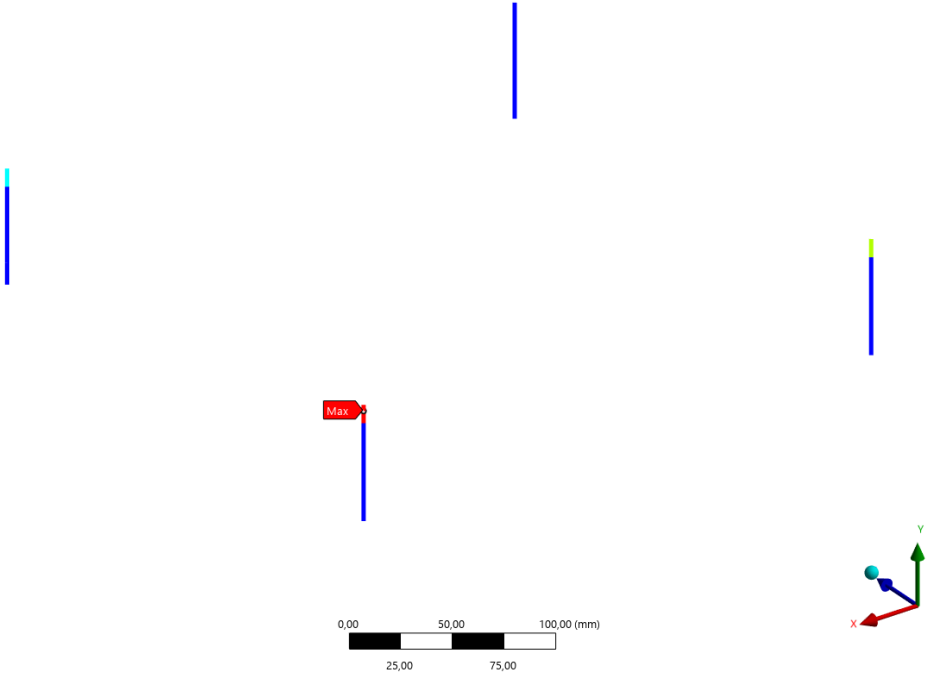
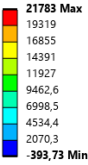


Figure 60 Maximum axial force over time. (Medium Transformer)

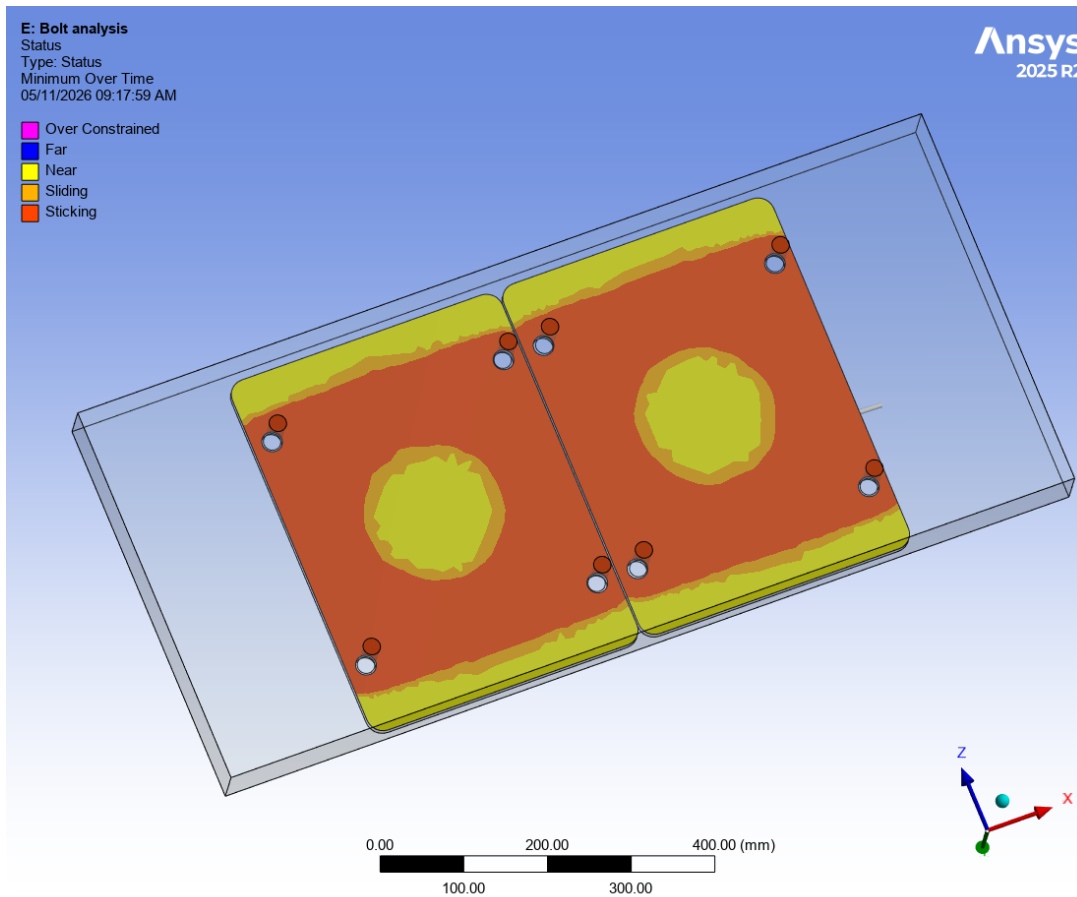


Figure 61 Minimum over time friction between shim and pad plates. (Medium Transformer)

Contact status tool in Figure 61 indicates that clamping force between shim plates and pad plates is enough to induce a frictional bond without sliding.

E: Bolt analysis

Total Bending Moment

Type: Total Bending Moment (Unaveraged)

Unit: N·mm

Maximum Over Time s

11/06/2026 15.02.56

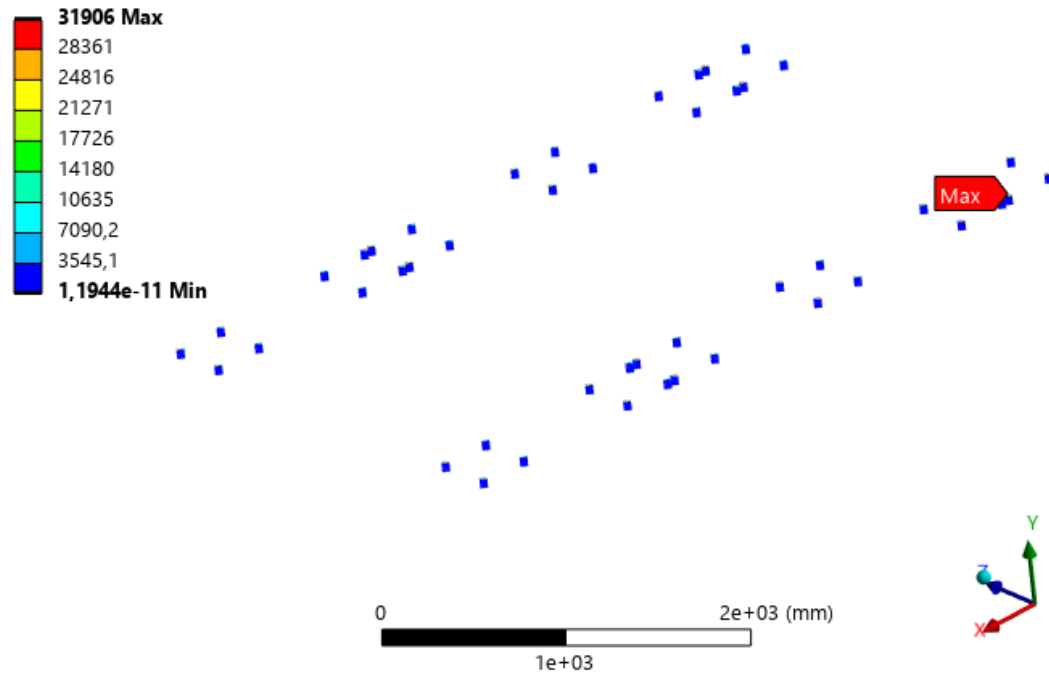


Figure 62 Maximum bending moment over time for Medium Transformer element bolts.

Figure 62 indicates that bending moments are more evenly distributed per bolt in foot than axial forces. Highest bending moments are observed on the active parts side of the tank (Figure 62). Maximum unaveraged bending moment over time is 31 906 Nmm.

E: Bolt analysis

Total Bending Moment

Type: Total Bending Moment (Unaveraged)

Unit: N·mm

Maximum Over Time s

11/06/2026 15.03.13

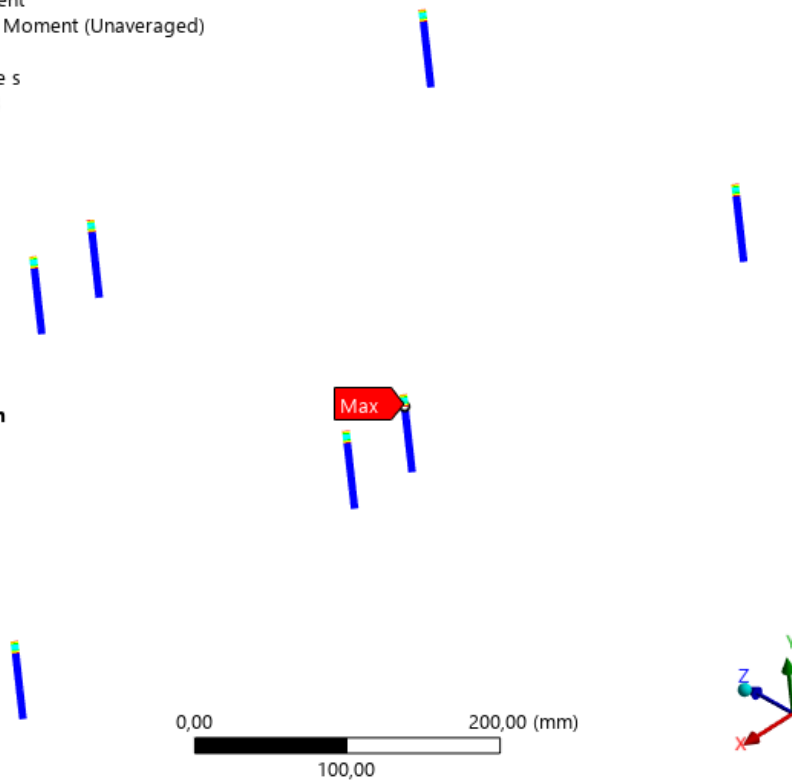
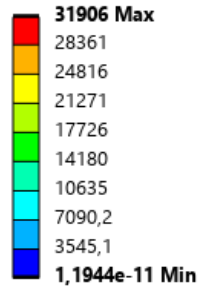


Figure 63 Maximum bending moment over time focused on the point of maximum force. (Medium Transformer)

As for the centre bolt, shear forces acting on the centre bolt are calculated using Equation 24:

$$F_{S\text{ centre}} = 30\,800\text{ N} - 0.15 \cdot 269\,900\text{ N}$$

$$F_{S\text{ centre}} = -9685\text{ N} < 0$$

The result is below 0, thus the bending force is calculated as 10% of the maximum horizontal force:

$$F_{S\text{ centre}} = 0.1 \cdot 30\,800\text{ N} = 3080\text{ N}$$

From this bending moment can be calculated using Equation 25:

$$M_{b\text{ centre}} = 3080\text{ N} \cdot \frac{36\text{ mm}}{2} = 55\,440\text{ Nmm}$$

For the medium transformer a lifting force of 34 442 N was observed during the reaction force analysis, and this will be utilized as the external axial force for the centre bolts.

Combined stress check is done by calculating the utilization factor using Equation 17 and values from Table 32.

Table 32 Bolt properties used for medium transformer combined stress check. (Medium transformer)

Symbol	Element bolts (M24)	Centre bolt (M36)
F_m	166 000 N	407 300 N
d	24 mm	36 mm
d_s	21.19 mm	32.25 mm
d_2	22.05 mm	33.40 mm
P	3 mm	4 mm
$\mu_{G \min}$	0.07	0.07
α	60°	60°
A_s	353 mm ²	817 mm ²
F_{SA}	21 783 N	34 442 N
M_b	31 906 Nmm	55 440 Nmm

Combined stress check for M24 element bolts:

$$v \cdot R_{p0.2}$$

$$= \sqrt{\left(\frac{166\,000\text{ N} + 21\,783\text{ N}}{353\text{ mm}^2} + \frac{31\,906\text{ Nmm}}{\left(\frac{\pi \cdot 21.19\text{ (mm)}^3}{32} \right)} \right)^2 + 3 \left(\frac{166\,000\text{ N} \cdot \left(\left(\frac{3\text{ mm}}{2\pi} \right) + \left(\frac{22.05\text{ mm} \cdot 0.07}{2 \cdot \cos \frac{60^\circ}{2}} \right) \right)}{\frac{\pi \cdot (21.19\text{ mm})^3}{12}} \right)^2}$$

$$v \cdot R_{p0.2} = 587.7\text{ MPa}$$

$$v = \frac{587.7\text{ MPa}}{600\text{ MPa}} = 0.98 < 1$$

Combined stress check for M36 centre bolts:

$$v \cdot R_{p0.2}$$

$$= \sqrt{\left(\frac{407\,300\text{ N} + 34\,442\text{ N}}{817\text{ mm}^2} + \frac{55\,440\text{ Nmm}}{\left(\frac{\pi \cdot 32.25\text{ (mm)}^3}{32} \right)} \right)^2 + 3 \left(\frac{407\,300\text{ N} \cdot \left(\left(\frac{4\text{ mm}}{2\pi} \right) + \left(\frac{33.40\text{ mm} \cdot 0.07}{2 \cdot \cos \frac{60^\circ}{2}} \right) \right)}{\frac{\pi \cdot (32.25\text{ mm})^3}{12}} \right)^2}$$

$$v \cdot R_{p0.2} = 579.9\text{ MPa}$$

$$v = \frac{579.9\text{ MPa}}{600\text{ MPa}} = 0.97 < 1$$

The utilization factors for both bolts are less than 1, demonstrating their suitability for forming an effective friction-type bolted joint. Taking bending moment from low preload analysis and using the highest preload in the calculations makes this stress check conservative and indicates a secure joint.

Modal analysis results

Table 33 Modal analysis results for Medium Transformer.

Mode	Direction	Frequency (Hz)	Effective mass (%)		
			X-direction	Y-direction	Z-direction
1	Z	2.91	0.11	0.09	74.59
2	X	3.47	69.96	0.19	0.09
3	Y	6.71	0.22	96.77	0.11
4	Rotate Y	8.59	0.11	0.06	0.21
5	Z	10.91	0.27	0.05	6.93
6	X	10.99	26.47	0.03	0.35
Sum			96.87	97.2	82.28

At the lowest the first six modes capture 82% of the effective mass in all direction, thus higher modes contribute negligible to the resonance response. All first 6 modes are rigid-body modes, which is common for a stiff structure.

Similarly, as with the light transformer results are validated using Equation 28:

$$f_y = \frac{1}{2\pi} \sqrt{\frac{(10\,300 \text{ N/mm} \cdot 1000 \cdot 12)}{83\,000 \text{ kg}}} = 6.1417 \text{ Hz}$$

The FEA has its result frequency characteristically higher (Mode 3) than the analytical calculation with an error of 8.5% which is low.

Transport analysis

Bolt preload for the M30 10.9 TS-bolts was calculated to be from 240 to 360 kN. Both preload cases were analysed separately due to the harsh loads they are subjected to. Slippage was checked for during low preload analysis. No preload was used when analysing external axial forces on bolts.

Sum of downward reaction forces were calculated using results from load steps that had downward acceleration and Equation 18. With these the mass of the transformer is around 85 540 kg which is ~3% higher than the transformer as per general arrangement drawing. Deviation between highest and lowest masses are 5 kg. Reaction forces are load-case-sensitive as expected and stress results indicate that loads are transmitted through expected paths (active parts → tank → beams → transport supports) with the heavier side receiving more stress because of the active part.

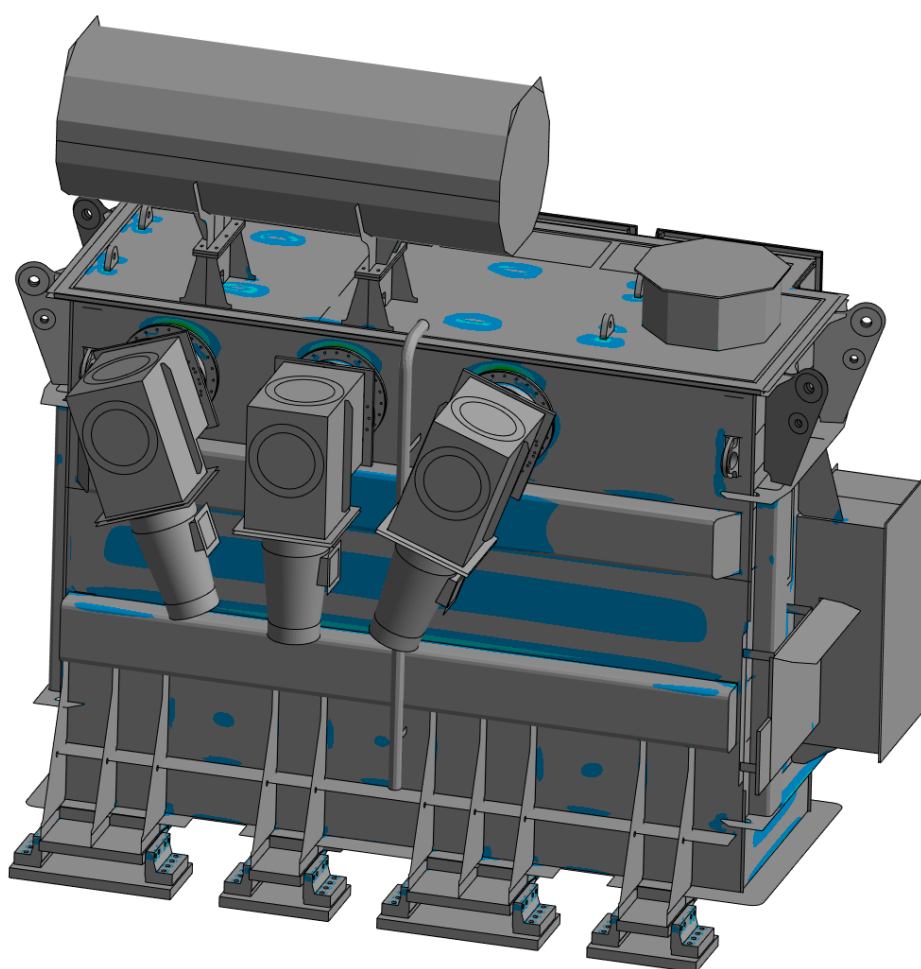


Figure 64 Global stress analysis for light transformer transport analysis. (Medium transformer)

Yielding of the transportation structure was evaluated using equivalent von-Mises stress. Sea transport loads caused higher stresses in the beam structure than the in-place load cases, as shown in Figure 65, but the maximum nominal stress in beams remained well below the allowable limit of 285 MPa.

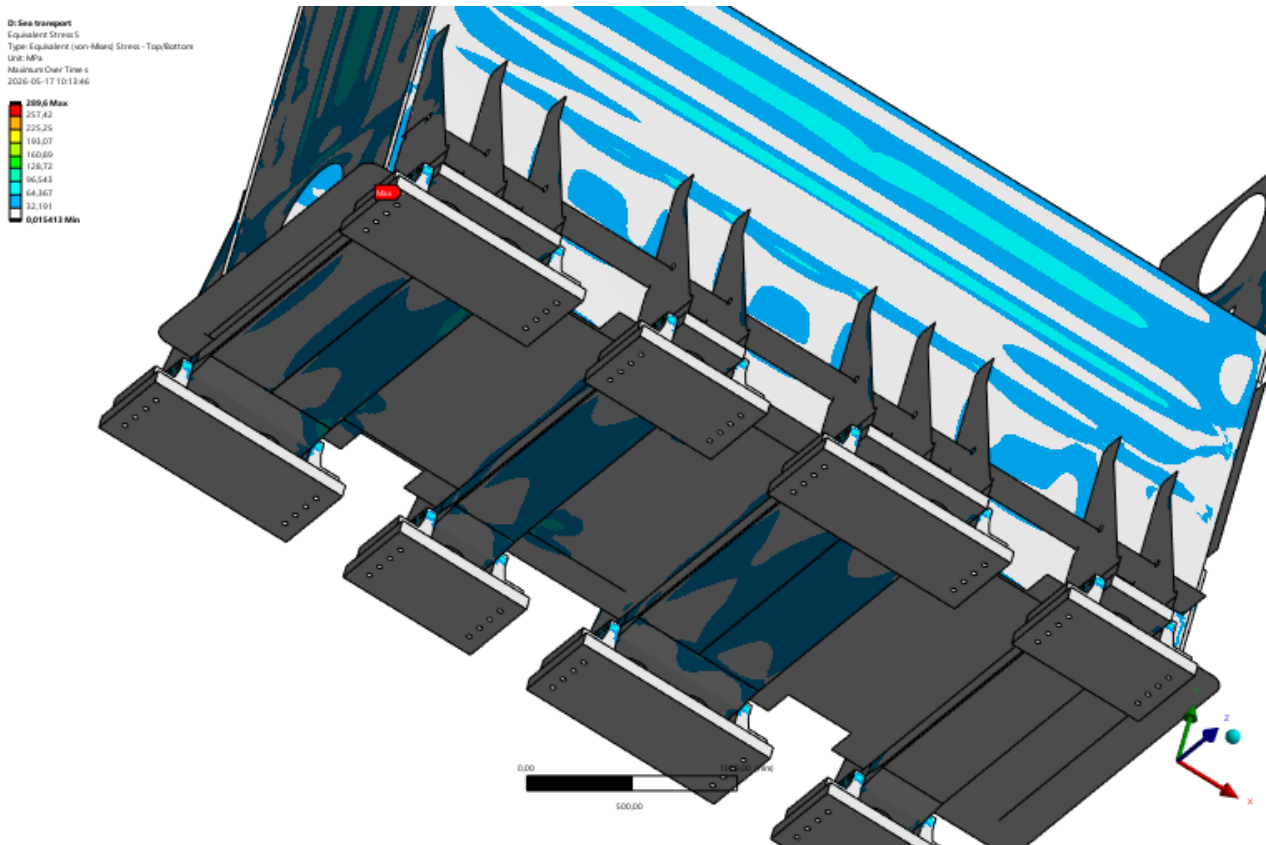


Figure 65 Stress in bottom beams structure during transportation. (Medium transformer)

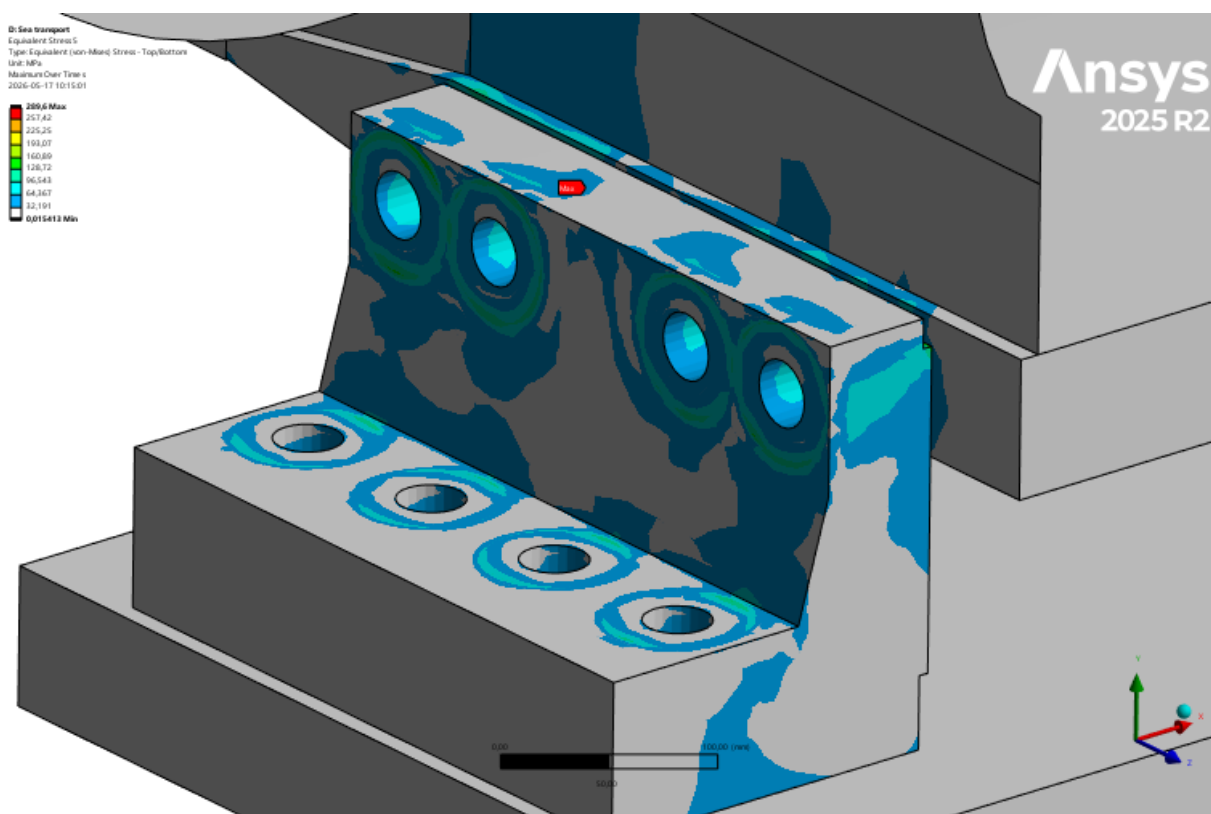


Figure 66 Stress shapes on TS-support during transportation. (Medium transformer)

The maximum peak computed stress of 289.6 MPa slightly exceeds the allowable limit of 285 MPa defined by DNV-ST-E273. However, this exceedance occurs at the edge of a bolt hole as shown in Figure 67, where geometric discontinuities inherently introduce localized stress concentrations. According to established DNV design practice, such peak stresses are typically highly localized and do not represent the structural stress governing failure. Therefore, pointwise maximum stresses obtained from FEA should not be directly used for design verification without considering their physical relevance. When evaluated over a representative region adjacent to the hole edge, the stress level is reduced and approaches the allowable limit. The bolt joint is also simplified to be more computationally effective, and in reality, this edge would be inside the bolt hole where even more material is present to disperse the stress. Furthermore, the observed exceedance of approximately 1.5% falls within the expected uncertainty associated with finite element discretization, mesh dependency, and stress interpolation near discontinuities. Based on these considerations, the result is assessed to be acceptable and in compliance with the intent of the DNV-ST-E273 requirements. A realistic nominal maximum stress in the in the structure is evaluated as 180-190 MPa.

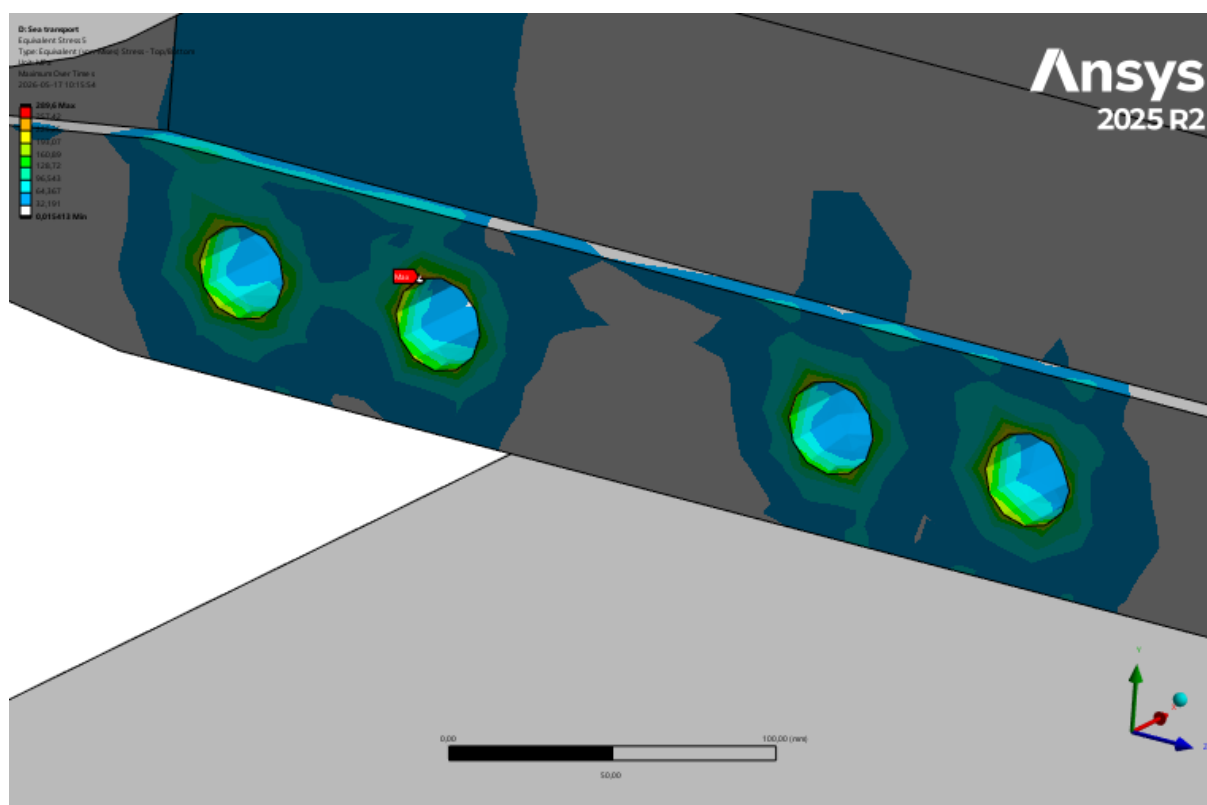


Figure 67 Maximum stress over time showing the local stress peaks during transportation. (Medium transformer)

As with the light transformer horizontal and vertical TS-bolts have large differences in their loads. Figure 68 and Figure 69 indicate that vertical bolts are most loaded in the corners of the transformer and horizontal bolts generally on the inner most row, similar to light transformers. Double beams load horizontal bolts more likely due to increased foot pad bending potential. Vertical bolts have a maximum stress over time of 22 492 N and horizontal bolt 13 782 N.

E: Sea transport without preload
 Axial Force
 Type: Directional Axial Force(X Axis) (Unaveraged)
 Unit: N
 Solution Coordinate System
 Maximum Over Time s
 2026-05-15 10:44:40

22492 Max
 20254
 18016
 15777
 13539
 11301
 9063.1
 6824.9
 4586.8
 2348.7 Min

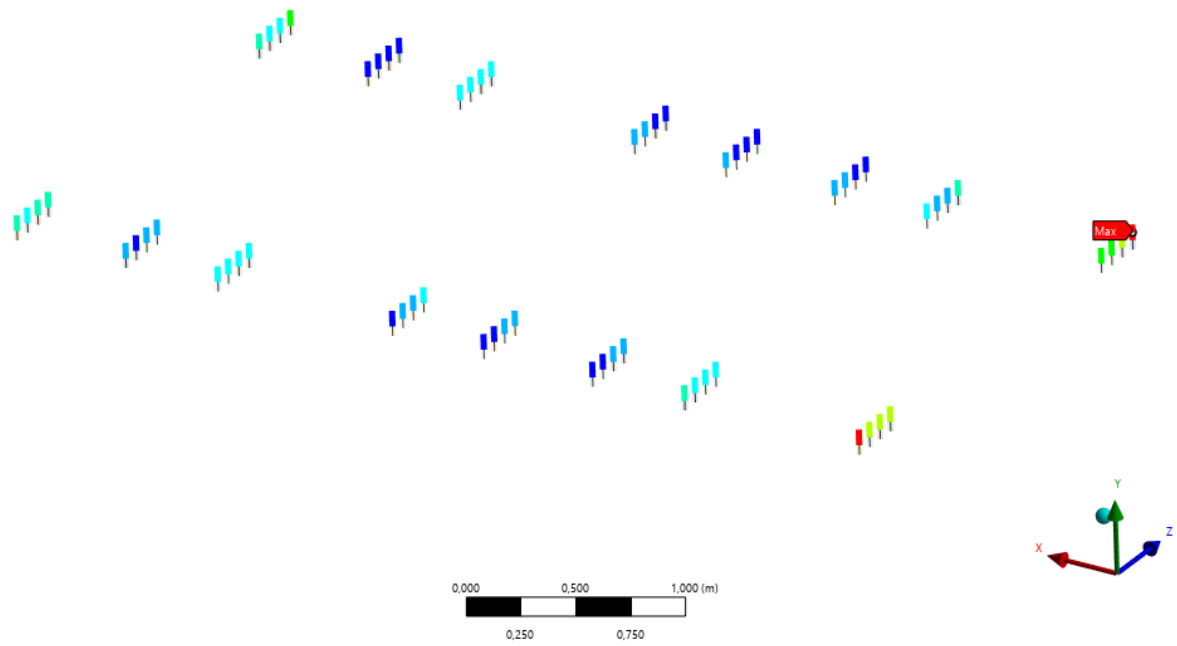


Figure 68 Maximum axial forces over time for vertical bolts during with transportation loads. Analysed without preload.
 (Medium transformer)

E: Sea transport without preload
 Axial Force 2
 Type: Directional Axial Force(X Axis) (Unaveraged)
 Unit: N
 Solution Coordinate System
 Maximum Over Time s
 2026-05-15 10:45:04

13782 Max
 12478
 11174
 9869.4
 8565.1
 7260.8
 5956.5
 4652.1
 3347.8
 2043.5 Min

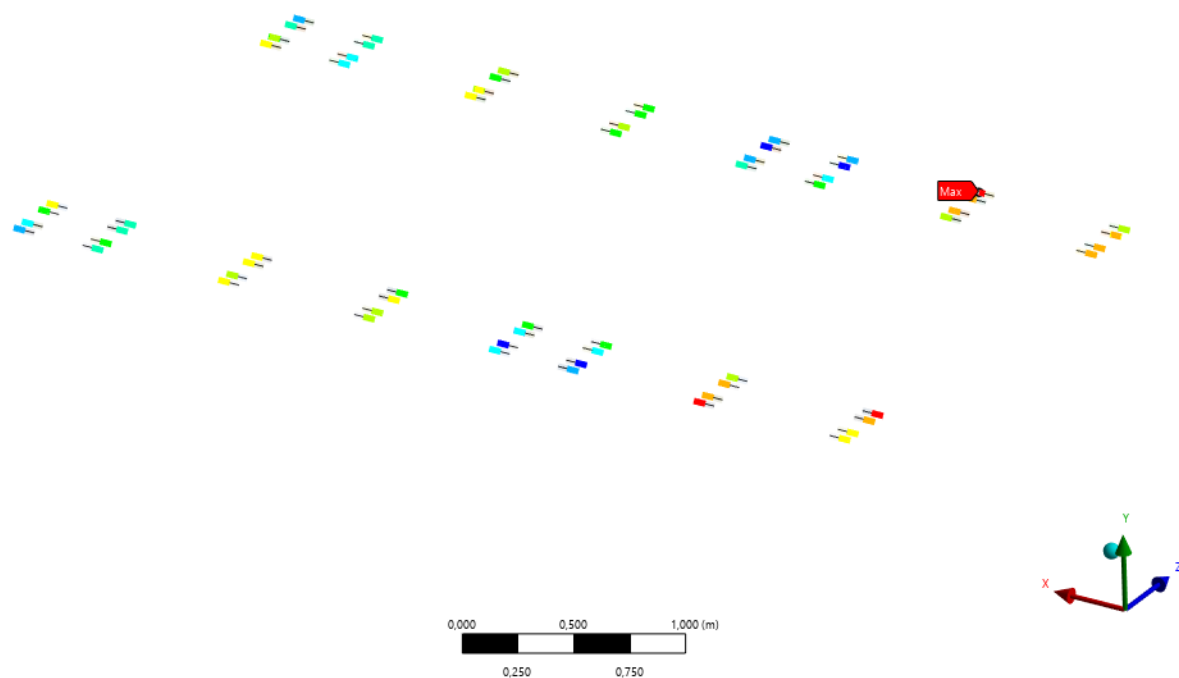


Figure 69 Maximum axial forces over time for horizontal bolts during with transportation loads. Analysed without preload. (Medium transformer)

Magnitudes reverse in bending moments as indicated in Figure 70 and Figure 71. Peak bending moments over time for horizontal bolts are 647 000 Nmm and 632 000 Nmm for low and high preloads respectively. For vertical bolts the peak bending moments are 72 832 Nmm and 84 214 Nmm for low and high preloads respectively. As these are node specific values they were averaged for the load bearing part of the bolts', averaged values are used for stress checks as they are more representative (Table 34). Averaged values for bending moment are in Table 34. Pins under the transport supports took no loads (axial or shear) and were thus excluded from the results.

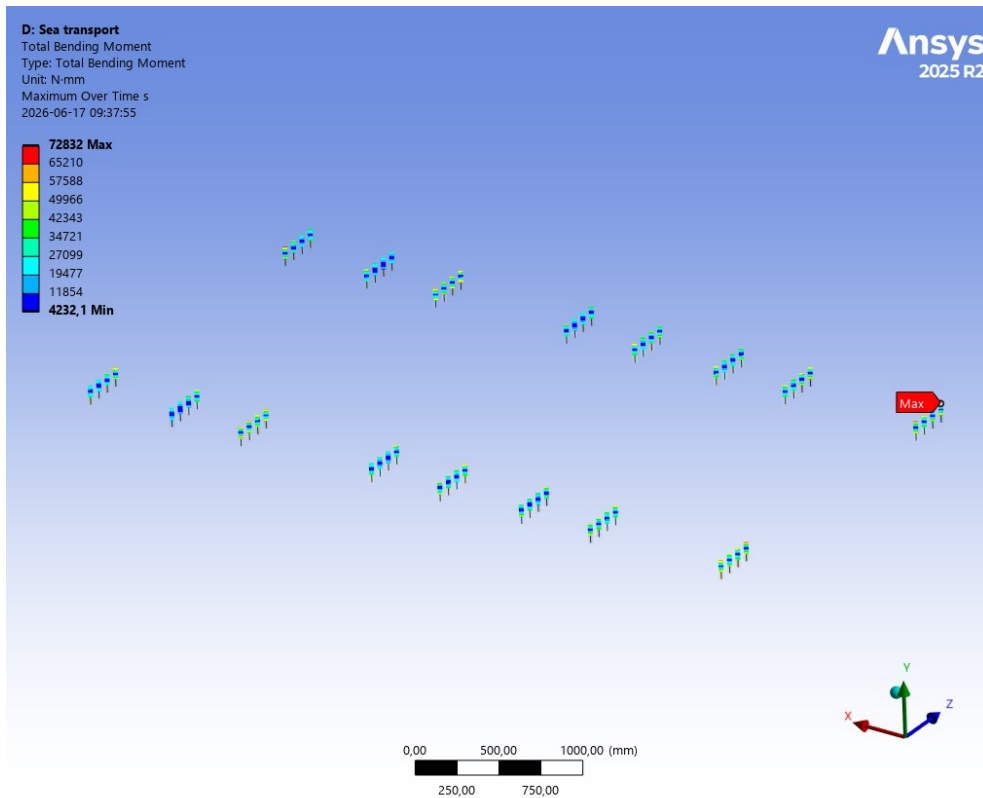


Figure 70 Maximum shear forces over time for vertical bolts during with transportation loads. (Medium transformer) (Low preload)

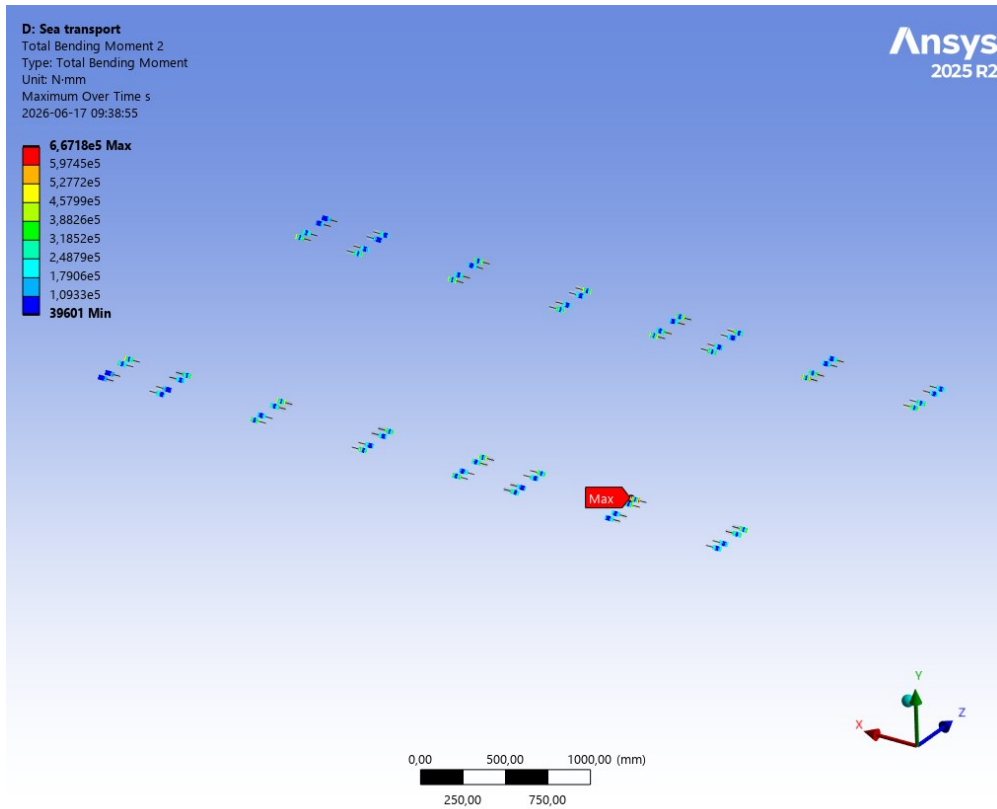


Figure 71 Maximum shear forces over time for horizontal bolts during with transportation loads (Medium transformer) (Low preload)

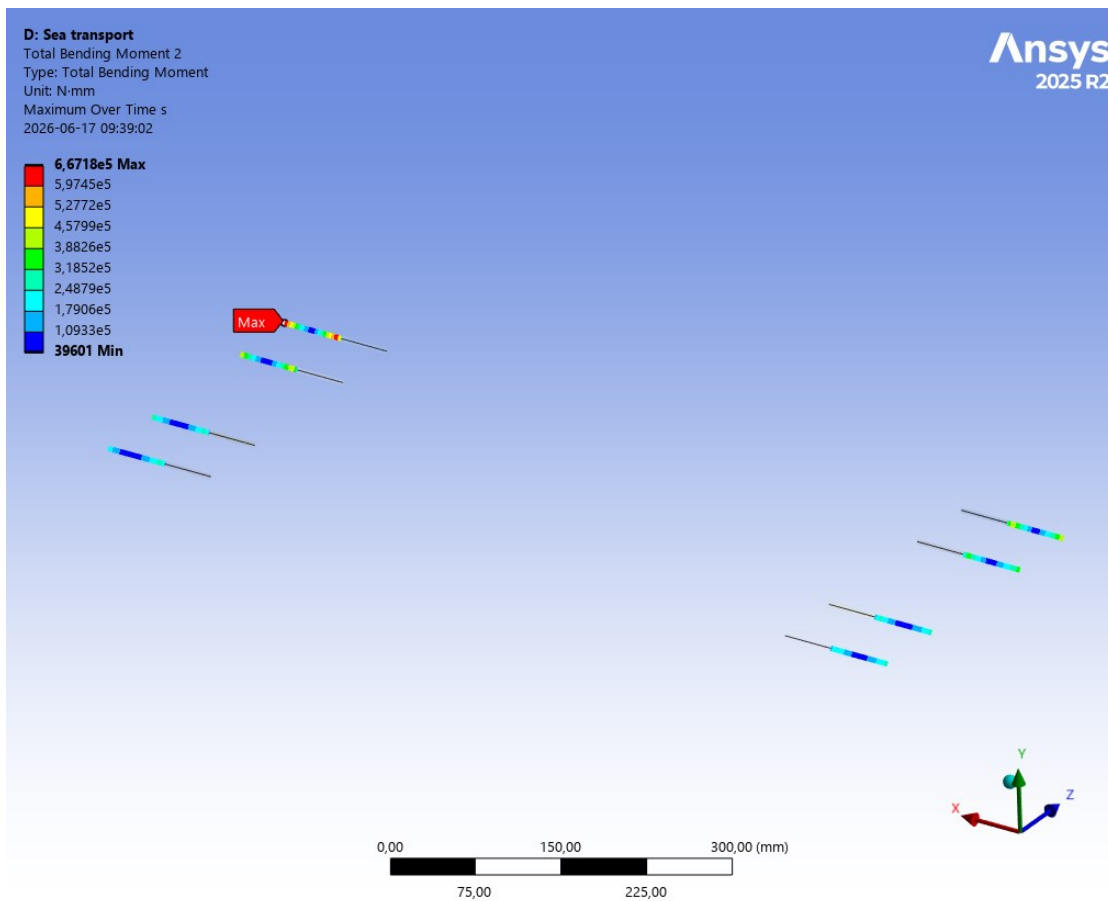


Figure 72 Peak bending stress on the highest loaded foot with low preload. (Medium transformer)

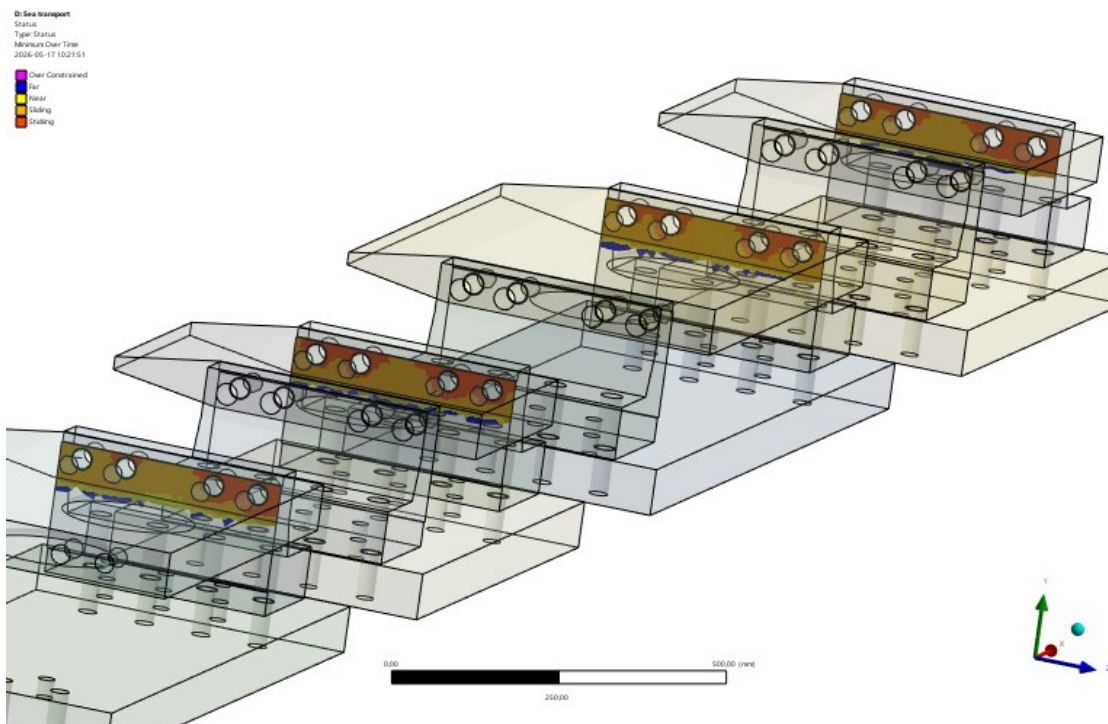


Figure 73 Contact status showing minimum friction over time with lowest preload active. (Medium transformer)

Figure 73 indicates the contact status between foot pad and transport support. The use of friction shims resulted in sticking contact at the interface, which indicates that the shear load was transferred predominantly by friction. Under these conditions, bending caused by relative slip in the joint is expected to be significantly reduced. The remaining bolt bending is therefore governed mainly by local deformation of the surrounding structural components and the joint geometry.

Table 34 Bolt properties used for medium transformer's transport combined stress check.

Symbol	Horizontal bolts (M30)	Vertical bolts (M30)
$F_{m \min}$	240 000 N	240 000N
$F_{m \max}$	360 000 N	360 000 N
d	30 mm	30 mm
d_0	25.71 mm	25.71 mm
d_s	26.72 mm	26.72 mm
P	3.5 mm	3.5 mm
$\mu_{G \min}$	0.07	0.07
α	60°	60°
A_S	561 mm ²	561 mm ²
F_{SA}	13 782 N	22 492 N
$M_{b \min}$	367 000 Nmm	35 537 Nmm
$M_{b \max}$	340 000 Nmm	41 179 Nmm

Combined stress check is done by calculating the utilization factor using Equation 17 and values from Table 34.

Horizontal bolt with low preload:

$$v \cdot R_{p0.2}$$

$$= \sqrt{\left(\frac{240\,000\text{ N} + 13\,782\text{ N}}{561\text{ mm}^2} + \frac{367\,000\text{ Nmm}}{\left(\frac{\pi \cdot 25.71\text{ (mm)}^3}{32} \right)} \right)^2 + 3 \left(\frac{240\,000\text{ N} \cdot \left(\left(\frac{3.5\text{ mm}}{2\pi} \right) + \left(\frac{26.72\text{ mm} \cdot 0.07}{2 \cdot \cos \frac{60^\circ}{2}} \right) \right)}{\frac{\pi \cdot (25.71\text{ mm})^3}{12}} \right)^2}$$

$$v \cdot R_{p0.2} = 690\text{ MPa}$$

$$v = \frac{690}{900} = 0.77$$

Horizontal bolts with high preload:

$$v \cdot R_{p0.2}$$

$$= \sqrt{\left(\frac{360\,000\text{ N} + 13\,782\text{ N}}{561\text{ mm}^2} + \frac{340\,000\text{ Nmm}}{\left(\frac{\pi \cdot 25.71\text{ (mm)}^3}{32} \right)} \right)^2 + 3 \left(\frac{360\,000\text{ N} \cdot \left(\left(\frac{3.5\text{ mm}}{2\pi} \right) + \left(\frac{26.72\text{ mm} \cdot 0.07}{2 \cdot \cos \frac{60^\circ}{2}} \right) \right)}{\frac{\pi \cdot (25.71\text{ mm})^3}{12}} \right)^2}$$

$$v \cdot R_{p0.2} = 899.6\text{ MPa}$$

$$v = \frac{899.6}{900} = 0.99$$

Vertical bolts with low preload:

$$v \cdot R_{p0.2}$$

$$= \sqrt{\left(\frac{240\,000\text{ N} + 22\,492\text{ N}}{561\text{ mm}^2} + \frac{35\,537\text{ Nmm}}{\left(\frac{\pi \cdot 25.71\text{ (mm)}^3}{32} \right)} \right)^2 + 3 \left(\frac{240\,000\text{ N} \cdot \left(\left(\frac{3.5\text{ mm}}{2\pi} \right) + \left(\frac{26.72\text{ mm} \cdot 0.07}{2 \cdot \cos \frac{60^\circ}{2}} \right) \right)}{\frac{\pi \cdot (25.71\text{ mm})^3}{12}} \right)^2}$$

$$v \cdot R_{p0.2} = 513\text{ MPa}$$

$$v = \frac{513}{900} = 0.57$$

Vertical bolts with high preload:

$$v \cdot R_{p0.2}$$

$$= \sqrt{\left(\frac{360\,000\text{ N} + 22\,492\text{ N}}{561\text{ mm}^2} + \frac{41\,179\text{ Nmm}}{\left(\frac{\pi \cdot 25.71\text{ (mm)}^3}{32} \right)} \right)^2 + 3 \left(\frac{360\,000\text{ N} \cdot \left(\left(\frac{3.5\text{ mm}}{2\pi} \right) + \left(\frac{26.72\text{ mm} \cdot 0.07}{2 \cdot \cos \frac{60^\circ}{2}} \right) \right)}{\frac{\pi \cdot (25.71\text{ mm})^3}{12}} \right)^2}$$

$$v \cdot R_{p0.2} = 742\text{ MPa}$$

$$v = \frac{742}{900} = 0.83$$

All utilization factors are below 1 and thus the joints are adequate. For all analyses Ansys beam tool's maximum combined stress results gave significantly lower results compared to the analytical check with the highest stress being 732 MPa compared to the analytically calculated highest stress of 899 MPa for the same bolt. This indicates that the VDI stress check is highly conservative and that even joints with high utilization factors are robust.

5.1.3 Compliance with standards and goals

The obtained numerical results were evaluated against the relevant design standards to assess the structural adequacy of the transformer feet under the investigated load cases. The applied load scenarios and verification criteria are consistent with DNV-ST-E273, DNV-OS-C101, DNV-RP-C205, and VDI 2230, which together form the basis for offshore structural and mechanical integrity assessment. The governing load cases, including combined static loads and platform-induced accelerations, are aligned with the intent of DNV-ST-E273 and DNV-RP-C205, which emphasize inertia-dominated loading for offshore and transport-exposed equipment. The resulting stresses and reactions in the transformer feet were found to be within acceptable limits, indicating compliance with the structural strength requirements defined in DNV-OS-C101. Bolted joint behaviour was assessed in accordance with the force-based verification philosophy of VDI 2230. The calculated axial bolt forces and utilizations remained below allowable limits for all investigated cases.

Overall, the results demonstrate that the transformer feet design satisfies the relevant structural and bolted-joint requirements. Within the defined scope of the study, the design can therefore be considered compliant with the applied standards for offshore and transport-related loading conditions.

5.2 Discussion

This chapter discusses the numerical results with respect to the defined design objectives, focusing on the structural performance of the transformer feet under dynamic, and transport-related loading conditions.

The applied verification and validation activities indicate that the numerical results are physically plausible and realistic for the intended design scope. While local effects such as detailed thread stresses are not resolved, the model is considered adequate for global structural assessment of the transformer feet and its support system.

5.2.1 Stress and deformation

Stresses in the structure behave as expected, highest stress is located by the centre bolts and does not exceed allowable limits meaning the structures are robust and fulfil the

mandated safety factors from DNV (Table 6). This stress would be on the bottom of the foot pad and dispersed on a larger area (area of the rubber element's hat) thus being lower than analysed. The top surface is in turn stressed by the preloaded bolt.

This stress can be calculated using pressure formula $\left(P = \frac{F}{A}\right)$ with the areas of the washers and maximum preloads of the centre bolts:

Light transformer	Medium transformer
$F_{M \max} = 173.9 \text{ kN}$ $A = 1428.3 \text{ mm}^2$	$F_{M \max} = 407.3 \text{ kN}$ $A = 3258.7 \text{ mm}^2$
$\frac{173\,900 \text{ N}}{1428.3 \text{ mm}^2} = 121.7 \text{ MPa}$	$\frac{407\,300 \text{ N}}{3258.7 \text{ mm}^2} = 125 \text{ MPa}$

These surface pressures are within allowable limits.

VCE Techniek confirmed GM 6000 functions under the specified conditions, and although GM 12 000 is not yet produced, it is expected to perform similarly well. The rubber elements have a high load capacity but are most constrained by the deflection of the rubber inside them as excessive deflection can cause the element's "hat" to touch element screws. The deformations observed in the computational analyses were calculated using linear spring coefficients for the rubber elements as provided by the manufacturer. However, it is well established that rubber exhibits a non-linear stress-strain relationship, with its stiffness increasing as compressive load is applied. Consequently, the actual deformation of the rubber components under operational conditions would be less than that predicted by the linear model, as the material's resistance to deformation intensifies with increasing pressure. Additionally, the springs resist deformation to each direction according to the spring coefficient while in reality especially upward movements are resisted to a different degree compared to downward movements. Springs being used in this manner is an effective but simplified way of modelling that neglects nonlinear and viscoelastic behaviour of the rubber elements.

When looking at the horizontal force distribution the design demonstrates satisfactory performance on balancing the force per foot pad, however some foot pads may have two

rubber elements instead of one which means some elements take half the force when using double and single beams on one transformer. While the rubber element is effective at taking these loads it should be considered to try using either single or double beams exclusively per transformer to avoid fatigue load over time. From a modelling point of view the results are adequate for studying global feasibility, but more accurate material modelling for rubber parts is required for a detailed and accurate analysis. However, accurate material properties are difficult to acquire when dealing with 3rd party firms.

The difference between the maximum forces against most loaded versus least loaded rubber element is 10.8% and 12.2% for light and medium transformers respectively. The rise in this discrepancy is due to uneven active part placement and increase in number of feet.

For both transformers the governing load cases are cases that induce the highest maximum downward reaction forces a foot (corner foot element). These forces appear when downward and longitudinal accelerations are highest, this suggests that longitudinal forces are hardest to dissipate. The calculation of rubber element utilization is performed for the most critical load case (LC: 1 for both transformers), applying Equation 22 in conjunction with the maximum reaction forces obtained from FEA. Utilized feet can be calculated by solving n in the longitudinal direction assuming horizontal and transversal loads are spread equally.

For the light transformer:

$$\frac{14.715 \text{ m/s}^2 \cdot 24000 \text{ kg}}{8} + \frac{\frac{4.9 \text{ m/s}^2 \cdot 24000 \text{ kg} \cdot 1.395 \text{ m}}{1.54 \text{ m}}}{n} + 0 = 70.2 \text{ kN}$$

$$n = 4.09$$

For the medium transformer:

$$\frac{\frac{3.757 \text{ m/s}^2 \cdot 83000 \text{ kg} \cdot 2.16 \text{ m}}{1 \text{ m}}}{12} + \frac{\frac{1.924 \text{ m/s}^2 \cdot 83000 \text{ kg} \cdot 2.16 \text{ m}}{2.14 \text{ m}}}{n} + \frac{13.19 \text{ m/s}^2 \cdot 83000 \text{ kg}}{12}$$

$$= 171.6 \text{ kN}$$

$$n = 6.6$$

This indicates that about 51- 55% of feet elements participate in handling longitudinal loads, which results in more load for the rubber element in the corners. This should be taken into

consideration when conducting preliminary calculations for number of feet and positioning of the transformer on site to minimize longitudinal accelerations by orientation in vessels. This calculation can be used to approximate the number of feet needed to keep the maximum force on a foot element at a certain level provided by the manufacturer of said element. The calculation needs the rough lengths from centre of gravity to farthest foot, maximum force on foot and ULS accelerations. Here's an example with medium transformers parameters where wanted maximum force on a foot is 150 kN:

$$\frac{\frac{3.757 \text{ m/s}^2 \cdot 83000 \text{ kg} \cdot 2.16 \text{ m}}{1 \text{ m}}}{n} + \frac{\frac{1.924 \text{ m/s}^2 \cdot 83000 \text{ kg} \cdot 2.16 \text{ m}}{2.14 \text{ m}}}{n \cdot 0.5} + \frac{13.19 \text{ m/s}^2 \cdot 83000 \text{ kg}}{n} = 150\,000 \text{ N}$$

$$n = 13.94$$

This indicates that ~14 foot elements are needed to achieve a maximum of 150 kN on feet elements. This calculation is only an estimate as it does not take into account the direction-specific and nonlinear properties of the rubber element.

5.2.2 Bolt performance and optimization

Modelling bolts as beam elements provides an efficient means of capturing the global load transfer and stiffness characteristics of a bolted joint, but it inevitably simplifies the local mechanical behaviour. While axial forces, bending moments and overall connection response can usually be represented with reasonable accuracy, local effects such as stress concentrations, thread interaction, hole deformation, and detailed contact pressure distribution are not resolved. Consequently, beam-based bolt models are well suited for system-level structural assessment, but the resulting bolt stresses, particularly bending-related stresses, should be interpreted with caution. For the scope of this thesis beams are an adequate compromise between efficiency and accuracy.

The use of a constant friction coefficient $\mu = 0.15$ for steel-to-steel interfaces in FEA is a practical but simplified assumption. Friction depends strongly on surface condition (roughness, coatings, lubrication) and may vary significantly from below 0.1 to above 0.3. The

CX coating used on most offshore steel structures does not have a defined frictional coefficient, but it can range from 0.08 to 0.25. In numerical models, this parameter directly controls load transfer: higher μ suppresses slip and reduces predicted bolt shear and thus bending, while lower μ increases slip tendency and associated bolt loading. Therefore, $\mu = 0.15$ should be treated as an assumed intermediate value rather than a verified property. While bolt in-place connections proved to be frictional and capable of taking needed loads in FEA with $\mu = 0.15$, they could be enhanced by using friction shims between critical surfaces such as pad-shim, shim-rubber element and foot pad-rubber element. 3M's diamond embedded friction shims accompanied with adequate surface pressure achieve a friction coefficient of 0.45. (3M, 2023) They have the potential for higher friction, but 0.45 is a conservative estimate provided by 3M for critical joint estimation. Table 35 and Table 36 show safety factors against sliding with and without friction shims.

Table 35 Safety factors against sliding with and without friction shims. (Light transformer)

Light transformer							
Maximum horizontal force (kN)	14.8	Preload		Friction coefficient		Safety factors	
				Steel- steel	3M shims	Steel- steel	3M shims
Centre bolt (M24)		Max	173.9	0.15	0.45	1.8	5.3
		Nom	143.4			1.5	4.4
		Min	112.9			1.1	3.4
Element bolts (4 M16 bolts)		Max	78.5	0.15	0.45	3.2	9.5
		Nom	65.8			2.7	8.0
		Min	53.3			2.2	6.5

Table 36 Safety factors against sliding with and without friction shims. (Medium transformer)

Medium transformer							
Maximum horizontal force (kN)	30.8	Preload		Friction coefficient		Safety factors	
				Steel-steel	3M shims	Steel-steel	3M shims
Centre bolt (M36)		Max	407.3	0.15	0.45	2.0	6.0
		Nom	336.9			1.6	4.9
		Min	269.9			1.3	3.9
Element bolts (4 M24 bolts)		Max	169.8	0.15	0.45	3.3	9.9
		Nom	143.5			2.8	8.4
		Min	117.7			2.3	6.9

Safety factors for centre bolts are low with minimum preloads unless friction shims are used, which increase them to safe levels. Without friction shims, raising preload to boost safety factors would require bigger bolts, since existing bolts are already heavily loaded.

The bolt utilization factors for all in-place bolts are 0,97-0,99 which indicates that the bolts operate close to their allowable limit under the governing load case, while remaining within acceptable limits according to VDI 2230. Preload exerts the greatest force in friction-type joints, leading to higher utilization factors when calculated with maximum preloads. Accurate preload is critical in fatigue-loaded bolted joints, which is why a preload envelope was used instead of a single nominal value. As shown by the calculated utilization factors, higher preloads may increase bolt stresses beyond the allowable limit. However, because the calculation method still implies some residual strength between yield and ultimate tensile strength, such conditions do not necessarily cause immediate failure. The conservativeness of the combined stress check can be observed when comparing analytic stress to combined stress from FEA. Nevertheless, excessively high preloads reduce the remaining load-carrying capacity of the bolts and may lead to failure when ultimate loads are applied. Although bolts

may sometimes be tightened to these extremes, they typically remain near the nominal preload, keeping utilization closer to the optimal range of ~ 0.75 - 0.85 .

As all in-place bolt connections are friction-type joints, the recess in the bottom of the foot plate could be removed and remain flat, reducing manufacturing and assembling complexity while increasing beam structure modularity for other uses. The same foot plate could be used with simple bolt anchoring when damping is not necessary or other foot elements.

5.2.3 Modal performance

The modal analysis was conducted to evaluate the global dynamic characteristics of the transformer-support system and to assess potential sensitivity to sea-induced motions and transformer-induced vibrations. As with other analyses, the rubber element was modelled using three springs. This is an effective means of representing flexible support for predicting global natural frequencies. However, it does not take into account the slightly increased stiffness of the rubber when under load, which affects the natural frequencies making them slightly higher. For a global natural frequency, the method is acceptable but resulting frequencies and mode shapes need to be assessed critically.

Analyses indicate no outstanding frequencies or effective mass results and is deemed reliable as confirmed by the single degree of freedom comparison, that shows an expected difference between simplified analytical calculation and complex FEA. Effective mass percentages corroborate that physical mass is most present in the first 6 modes. The heavier transformer exhibits lower natural frequencies than the lighter unit despite being supported by a greater number of stiffer rubber feet. This behaviour is explained by the dominant influence of mass in the mass-stiffness relationship governing natural frequencies as shown in Equation 3. Although the total support stiffness increases due to additional and stiffer feet, the proportional increase in mass is larger than the increase in effective system stiffness. Furthermore, the elastomeric supports exhibit nonlinear stiffness characteristics, meaning that total stiffness does not scale linearly with the number of feet. As a result, the ratio of effective stiffness to mass is reduced, leading to lower rigid-body and isolation-controlled modal frequencies in the heavier transformer. (Den Hartog, 1985)

The mode shapes captured indicate that the system behaves predominantly as a rigid body supported by compliant feet, with deformation concentrated in the elastomeric elements. This confirms that the global dynamic behaviour is governed by support stiffness rather than structural flexibility, and that no critical local deformation modes are present in the investigated frequency range.

Highest modes captured from both transformers the minimum frequency ratios against transformer induced vibration (100 Hz) are $r_{int} = 7.1$ for light and $r_{int} = 9.1$ for medium transformer. These are well into isolation ($r > \sqrt{2}$), this confirms that the highest frequencies of the tank assembly are sufficiently separated from the dominant transformer excitation frequencies, reducing the risk of resonance from high frequency vibrations.

The identified low natural frequencies (4.27 Hz and 2.90 Hz) lie above the dominant frequency range of platform motions at 0.04-0.5 Hz, suggesting limited risk of dynamic amplification due to wave-induced excitation. For wave-induced frequencies the frequency ratios for light and medium transformers are $r_{sea} = 0.12$ and $r_{sea} = 0.17$ respectively. Having better damping capabilities than rubber at these frequency ratios is redundant, as shown in Figure 2, where high and low damping materials show little to no difference at $r < 0.5$. Rubbers viscoelastic nature should dissipate energy through hysteresis and manage motion, stress and fatigue.

A fundamental design challenge in the transformer feet system is balancing structural stability and vibration isolation, as these requirements inherently oppose each other. Stability requires sufficient stiffness to limit displacements and ensure reliable load transfer under combined static and dynamic loads, whereas effective vibration isolation requires low stiffness to reduce natural frequencies and achieve favourable frequency ratios relative to higher excitation sources.

The analysis results confirm that the stiffness of the feet elements governs both static and dynamic behaviour. Higher stiffness improves load distribution and reduces deformation under governing acceleration load cases, particularly in the longitudinal direction, thereby enhancing structural reliability. However, increased stiffness raises the system's natural frequencies and reduces isolation performance. Conversely, more flexible supports improve isolation of high-frequency transformer-induced vibrations by increasing the frequency ratio,

as reflected in the modal analysis where excitation at approximately 100 Hz lies well within the isolation region.

At the same time, excessive compliance leads to increased displacements, which may affect alignment, load distribution, and reduced component clearances. This defines a practical lower bound for support stiffness. Additionally, low-frequency offshore excitations, such as platform motions, cannot be effectively isolated regardless of support properties and are instead governed by global stiffness.

The selected elastomeric feet represent a compromise between these competing requirements. The stiffness is sufficient to maintain structural stability under ULS conditions, while still enabling effective isolation of transformer-induced vibrations. The system therefore operates in a stiffness-controlled regime at low frequencies and an isolation-controlled regime at higher frequencies, which is consistent with expected behaviour for heavy offshore equipment.

5.2.4 Transport performance

Transport analysis indicates that transport supports work as intended to shield the rubber elements from excessive transportation loads. The analysis utilized similar bolt modelling as previous analyses and thus suffer from the same inaccuracies as them. From these the most critical is the limited accuracy of bending moment for beam bolts. Figure 53 and Figure 73 show that the contact between the foot pad and TS-support is sticking when utilizing friction shims. If friction shims are not used the interface slips and induces an excessive bending/shearing force on the bolts. Utilization factors for the bolts are highly dependent on the preloads of the bolts as having a high preload does not have a large effect on mitigating bending when joints are sticking. TS-bolts are well optimized, when high preload results show a utilization factor close to 1 as that means the utilization factor of the low preload can be kept as high as possible. High utilization factor means longer elongation of the bolt and thus better fatigue performance. However, the due to the conservativeness of the combined stress calculation bring down the elongation length by requiring using lower preloads. The peak bending moments were found to be high, requiring stiffer bolts compared to element and centre bolts. To mitigate this shear pins or additional structures under the foot pad could be designed.

Highest bolt loads during transportation were observed during load steps simulating quartering as these concentrate forces on the corner feet (LC: 9-16). Of these load cases most critical were load cases with high transversal and downward accelerations, more specifically LC: 9, 11, 13 and 15. These load cases also have the highest sum of accelerations of all analysed cases.

Pins were found to bear no shear loads when preload was present as the transport support-pad plate interface was always a friction-type joint. In future use of this design, pins and their holes can be removed for easier manufacturing and assembly.

6 Conclusion & Future Work

This study addressed the need for a more robust and standardized interface between offshore transformers and their supporting structures. Rather than evaluating individual performance metrics in isolation, the work focused on whether a unified design approach could reconcile the competing demands of structural integrity, vibration behaviour and practical implementation. The outcome demonstrates that such integration is achievable within a modular framework, provided that load transfer, support compliance and assembly logic are considered as a coupled design problem rather than independent tasks.

RQ1 – Load Cases & Motions

The governing loads acting on the transformer support feet are observed to vary significantly depending on whether the load path is relatively compliant or rigid. For in-place operational conditions incorporating rubber elements, the load path becomes softer, which alters the load distribution among the support points. In this case, longitudinal acceleration was identified as the most critical load component, primarily because the rubber elements do not participate equally in load transfer. This non-uniform load sharing introduces uncertainty in the actual load distribution and may lead to localized overloading of individual elements.

A key limitation of the present analysis is the absence of detailed material data describing the nonlinear deformation behaviour of the rubber elements. As a result, potential effects such as stiffness variation under load and unequal element participation were not explicitly captured in the modelling. Consequently, the predicted load distribution should be interpreted as an approximation, and the results are subject to uncertainty associated with these assumptions.

In contrast, for transport conditions where the structure behaves more rigidly, the load distribution is primarily governed by global acceleration fields. In this case, the most critical stress conditions are associated with high combined acceleration load cases and load concentration at corner supports, reflecting typical rigid-body behaviour.

With respect to dynamic excitation, transformer-induced vibrations intended to be isolated are predominantly associated with frequencies of approximately 100 Hz or 120 Hz, depending on the electrical line frequency. These frequencies fall within a range where vibration isolation using compliant support elements is effective. Conversely, sea-induced motions, typically occurring in the range of 0.04–0.5 Hz, are well below the achievable isolation range of practical passive systems. As such, these low-frequency excitations cannot be effectively isolated and are instead addressed through design strategies that avoid resonance and ensure adequate structural robustness.

RQ2 - Vibration Control

The minimization of vibration transmissibility from an offshore transformer is primarily achieved by selecting support feet with sufficiently low dynamic stiffness, such that the system natural frequency is placed well below the dominant excitation frequencies of the transformer. However, this approach inherently introduces a trade-off: while lower stiffness improves high-frequency isolation, it simultaneously increases susceptibility to resonance excitation and excessive displacement under low-frequency environmental loading.

To mitigate this effect, a moderate level of damping must be incorporated. This serves to reduce the amplitude of the resonance peak and limit amplification during transient conditions, such as wave-induced motions. Nevertheless, excessive damping is not desirable, as it degrades isolation performance at higher frequency ratios. The design therefore requires a balanced compromise between stiffness and damping, rather than the independent optimization of either parameter.

Within the scope of the present study, elastomeric support elements with natural frequencies in the range of approximately 5–10 Hz were identified as a practical solution. This range provides effective isolation of transformer-induced vibrations, particularly those associated with electromagnetic excitation, while maintaining sufficient stiffness to ensure acceptable structural behaviour under offshore operating conditions. However, this selection is strongly dependent on the assumed excitation spectrum and boundary conditions.

A key limitation of the proposed approach is that it primarily considers transformer-induced vibration and does not explicitly account for low-frequency excitation originating from other

onboard machinery or global structural dynamics of the installation. In such cases, the selected natural frequency range may coincide with dominant excitation components, potentially resulting in resonance amplification rather than isolation. This highlights the importance of evaluating the full system-level vibration environment when defining final design parameters.

Despite these limitations, analytical assessment of the selected configuration indicates that for excitation frequencies above approximately 30 Hz, the systems achieve a frequency ratio exceeding $r > 2$, corresponding to a regime where transmissibility is significantly reduced. While the modal analysis confirms the effectiveness of the design for higher-frequency excitation, it should be emphasized that the model is based on simplified assumptions, including linear stiffness behaviour. Consequently, the results should be interpreted as indicative rather than definitive, and further validation through detailed experimental testing is recommended.

RQ3 - Bolt/Shear integrity

The proposed design aims to eliminate shear loading in the bolts by introducing a foot geometry that redirects horizontal forces into the compliant rubber element. This enables the use of high preload levels, allowing the joint to function as a friction-type connection in which shear forces are resisted through interface friction rather than bolt shear capacity. While this approach is theoretically well-established, its effectiveness in practice is strongly dependent on maintaining sufficient and stable preload throughout the service life. Factors such as preload relaxation, surface wear, and environmental influences may reduce the available friction capacity, thereby reintroducing unintended shear load transfer to the bolts. The use of friction shims can improve the robustness of the connection by increasing the friction coefficient and stabilizing surface conditions.

From an assembly standpoint, the removal of the recess in the foot pad improves accessibility and simplifies installation. This represents a clear practical advantage compared to more complex existing solutions. Nevertheless, the simplified assembly assumes adequate alignment and fit-up conditions, which may not always be achievable in large-scale installations without additional adjustment provisions.

In terms of manufacturing, the design introduces stricter requirements for welding accuracy and dimensional control. The use of dedicated jigs and adherence to a defined welding sequence are necessary to achieve the intended geometry and load distribution. While this increases process complexity compared to previous solutions, it also reduces geometric variability and can improve repeatability if properly executed. However, this benefit is contingent on manufacturing discipline, and deviations from the intended process may negate the expected improvements in accuracy.

Overall, the proposed concept demonstrates clear advantages in terms of load transfer mechanisms and assembly simplification, but its performance is sensitive to preload stability and surface conditions. These dependencies should be explicitly considered when evaluating its robustness for offshore applications.

RQ4 – Selection & Standardization

Consistent module selection is achieved by first evaluating the maximum allowable load capacity of an individual foot element and subsequently determining the required number of elements such that the load per element remains within this permissible limit. This ensures that all support components operate within their intended mechanical range.

Preliminary estimates of the system natural frequency are obtained through analytical calculations based on simplified single-degree-of-freedom assumptions. To account for idealizations inherent in such models, a conservative adjustment is applied, typically in the range of 10–20% higher than the analytically derived value.

When multiple rubber element sizes satisfy both load-bearing and geometric requirements, the final selection is made based on the specific installation conditions and functional priorities. In general, lower stiffness configurations are preferred when vibration isolation performance is the governing criterion, whereas higher stiffness configurations are favoured when structural stability, load distribution, or displacement control are of greater importance.

Overall Findings

From a structural standpoint, the concept succeeds by enabling controlled load paths that remain valid across both operational and transport scenarios. A key implication of the work is that bolted joints in offshore transformer supports should be treated as functional components of the load-transfer system, not merely fastening elements. The findings illustrate that joint behaviour is dominated by interface mechanics and preload stability rather than bolt strength alone. This reinforces the importance of treating preload control, surface condition, and contact configuration as primary design variables. The concept demonstrates that when these factors are properly managed, bolt behaviour becomes predictable and decoupled from the dominant structural loading mechanisms.

The dynamic response of the system highlights that transformer support structures operate in fundamentally different regimes depending on excitation source. Instead of attempting full-spectrum vibration control, an effective design must selectively address relevant frequency bands. The results indicate that practical solutions emerge when high-frequency excitation is treated through isolation, while low-frequency effects are addressed through stiffness and system-level tuning. This dual-regime behaviour is a defining characteristic of offshore-mounted equipment and should form the basis for future design approaches.

The study also emphasizes that transport conditions represent a distinct design problem rather than an extension of operational loading. The introduction of dedicated transport load paths demonstrates that separating these phases allows each to be optimized independently without compromising overall system performance. This distinction is particularly important in offshore applications, where transport-induced loads may exceed operational loads but occur over fundamentally different timescales and structural configurations.

Beyond structural behaviour, the results demonstrate that design standardization is achievable without sacrificing flexibility. The modular approach enables scalability while maintaining consistent design logic, which reduces dependency on project-specific engineering decisions. This represents a shift toward design systems rather than individual solutions, where repeatability and predictability become central performance criteria. In practice, this has implications not only for engineering efficiency but also for manufacturing quality and installation reliability.

The study also highlights that the predictability of the concept is contingent on modelling assumptions that simplify real-world behaviour. Uncertainties related to material nonlinearity, long-term behaviour, and contact mechanics introduce limitations that cannot be resolved through numerical analysis alone. These factors do not invalidate the findings but indicate that the current results should be interpreted as a controlled approximation of reality rather than a direct representation of in-service conditions.

In summary, the work demonstrates that offshore transformer support systems can be systematically engineered using a modular and physics-based approach that aligns structural, dynamic, and practical requirements. The proposed concept establishes a framework in which load transfer, vibration response, and constructability are resolved simultaneously, rather than through iterative compromise. While further validation is required, the results indicate that this approach provides a credible pathway toward more reliable and standardized offshore transformer installations.

6.1 Future work

While this thesis demonstrated engineering feasibility of the design further investigation is required before full industrial implementation. Additional analyses should include jacking condition assessments, in which the structural response during handling and installation is evaluated. These load cases may introduce localized stress concentrations and non-uniform load distributions that are not captured in the present study. In addition, the functionality and structural integrity of jacking bolt systems should be verified, as their use introduces localized load paths and potential stress concentrations that require detailed assessment prior to product deployment.

Further experimental testing has already been scheduled regarding the fatigue performance, deflection and vibration control of the GM 6000 foot elements. The elements will be tested with a reactor that induces a wider range of frequencies compared to a transformer, which will give valuable insight into the accuracy of the numerical models utilized in this thesis. Furthermore, the element will be compared to existing onshore and offshore anti-vibration solutions. Overall testing improves confidence in the predicted natural frequencies and mode shapes and support validation of the applied modelling assumptions. Additional future studies could consider the effects of:

- Potential micro-slip at support interfaces
- Material nonlinearities, particularly in elastomeric components
- Manufacturing tolerances and assembly conditions, affecting load distribution between support points

Finally, further optimization of the transformer foot design may be carried out to improve load distribution, reduce stress concentrations, and enhance dynamic performance. This could include alternative beam placements or refined stiffness characteristics of isolator materials. Overall, the identified areas for future work highlight the steps required to transition from numerical feasibility assessment to a fully validated and deployable transformer support solution.

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