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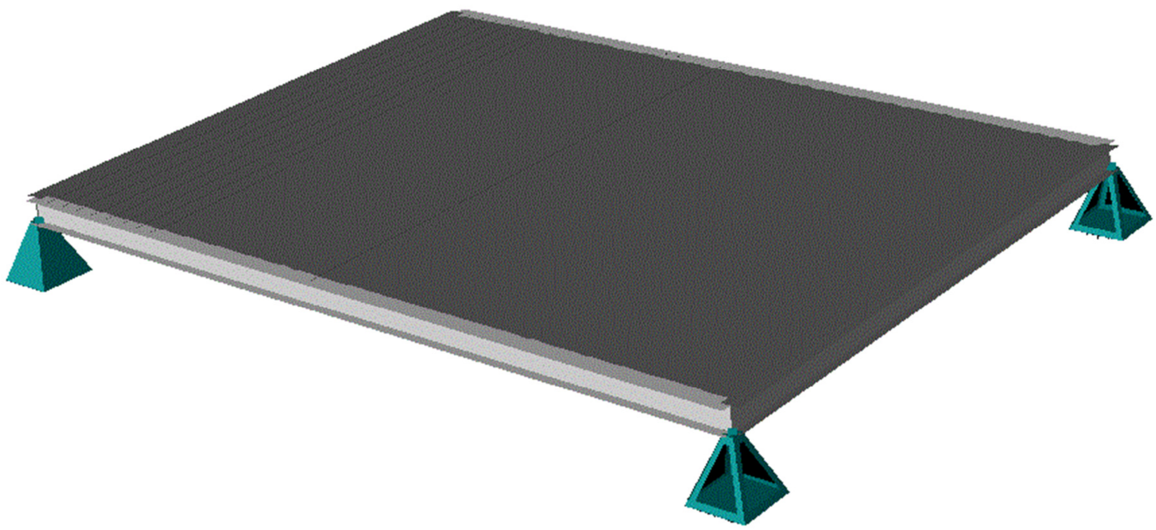

Alternative Design of Steel/Aluminium Car Deck Panels

Master's Thesis in the International Master's Programme Naval Architecture and Ocean Engineering

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Department of Shipping and Marine Technology
Division of Marine Design, Research Group Marine Structures
CHALMERS UNIVERSITY OF TECHNOLOGY
Göteborg, Sweden 2014
Master's thesis 2014: X-14/303

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Cover: Proposed design of an aluminium/steel car deck panel.

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ABSTRACT

This thesis evaluates if parts of the structural elements in a liftable car deck can be replaced with aluminium to obtain a lower weight. In order to be able to compare the final solution to an existing one, two car deck panels with different dimensions (14.37x14.64 and 14.37x10.22) on a Pure Car and Truck Carrier (PCTC) were used as a reference. Different concept designs where the steel top plate was replaced by an aluminium structure were evaluated by utilizing engineering beam theory, goal driven optimization and finite element analysis. The evaluation resulted in a steel/aluminium car deck design with extruded aluminium profiles as a stiffened top plate and a conventional steel beam system structure as support.

The final design of the larger car deck panel resulted in a weight reduction of 7.5% while the smaller car deck panel was weight reduced with 28.7%. Based on this difference, studies of how the free length between supports affects the steel structural weight were carried out.

A cost analysis was performed to evaluate whether the design was economically feasible. It was found that the payback time of the proposed design is 2.4 times the desired payback time, compared to the average life time of a PCTC which is 23 years, it was concluded that with the aluminium cost of 2014 the design is too expensive.

Key words: aluminium, finite element analysis, goal driven optimization, liftable car decks, weight reduction.

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Preface

[The current version of the thesis is a public version of the work. Due to confidentiality, some of the sections/paragraphs/tables/figures/text is deleted; this is marked in the report. Interested readers are kindly asked to contact TTS Marine AB in Gothenburg, Sweden, and ask for permission to read the full report.]

This thesis is a part of the requirements for the Master's degree in Naval Architecture and Ocean Engineering at Chalmers University of Technology, Göteborg. The work has been carried out at TTS Marine AB in Göteborg from January to April of 2014 and at the Division of Marine Design, Department of Shipping and Marine Technology, Chalmers University of Technology between April and June of 2014.

We would like to thank our examiner and supervisor, Senior Lecturer Per Hogström at the Department of Shipping and Marine Technology, for his excellent guidance and support throughout the work with this thesis.

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Furthermore, we would like to express our appreciation to the employees at TTS Marine AB for their valuable feedback. Special thanks are extended to Henrik Westermark for his ingenuity and his enthusiasm for this project.

Göteborg, May, 2014

Erik Andersson

Axel Guhrén

Notations

Roman upper case letters

B	Breadth of car deck panel [m]
E	Young's modulus for steel [Pa]
I	Moment of inertia [m^4]
L	Length of car deck panel [m]
M	Extra cost for manning [SEK/m]
M_y	Bending moment [Nm]
P_a	Price of aluminium (including FSW) [SEK/kg]
P_s	Price of steel [SEK/kg]
Q	Load on a beam [N]
S_z	Static moment [m^3]
V_y	Tyre load on deck plate [N]
Wa_a	Aluminium weight for new design [kg]
Ws_a	Steel weight for new design [kg]
Ws_r	Steel weight for reference solution [kg]

Roman lower case letters

b	Extra weight for brackets [%]
b_o	Cost of one bolt [SEK]
f_1	Material factor
q	Load per length [N/m]
w	Displacement [m]
z	Distance from neutral axis to fibre studied [m]

Greek lower case letters

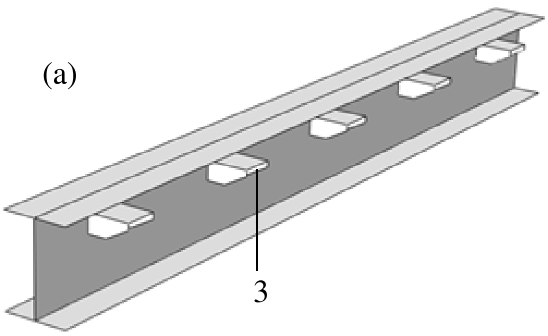
δ	Deflection [m]
ϵ	Error in response surface
κ	Curvature of a line
σ_{corr}	Critical buckling stress after correction [N/m^2]
σ_e	Critical buckling stress [N/m^2]
σ_{vM}	Equivalent stress according to von Mises [N/m^2]
σ_x	Stresses normal to the cross-section [N/m^2]
σ_y	Stress limit for reaching the yield point [N/m^2]
τ_{xz}	Shear stress [N/m^2]

Terminology

This section presents the definitions of commonly used terms in this thesis.

- **Aluminium stiffeners:** The webs and flange of the aluminium profile. Figure III, (1).
- **Attachment points:** The corners of the beam system, which is simply supported. Figure II, (2)
- **Beam system:** The supporting beam system which consists of a frame and longitudinal/transverse stiffeners. The beam system is made of steel
- **Brackets:** Vertical supports for the aluminium panels, adds a minor contribution to the weight. Figure Ia, (3)
- **Frame:** The outer beams of the beam system. Figure II, (4)
- **Friction Stir Welding (FSW):** A method to weld the profiles together.
- **Investment:** The total manufacturing and material cost of a car deck panel.
- **Lashing holes:** The cargo (Vehicles) is fixed in place during voyage by attaching a lashing strap between the hole and the cargo. Figure III, (5)
- **Longitudinal frame:** Beams of the frame mounted in the longitudinal direction of the ship. Figure II, (6)
- **Longitudinal stiffener:** Beams within the frame mounted in the longitudinal direction of the ship. Figure II, (7)
- **Mechanical fastening:** Alternative, mechanical method to attach the aluminium profiles together. Figure Ib, (8)
- **Panel:** The complete structure. The panel consists of the stiffened aluminium top plate, the frame and the longitudinal/transverse stiffeners.
- **Payback:** The time it takes for a more expensive solution to pay off by savings in operation cost.
- **Profile:** An extruded aluminium profile consisting of a top plate, web and flange. Figure III, (9)
- **Stiffened top plate:** The profiles welded together to form a top plate with integrated stiffeners. Figure II, (10)
- **Transversal frame:** Beams of the frame mounted in the transverse direction of the ship. Figure II, (11)
- **Transversal stiffener:** Beams within the frame mounted in the transverse direction of the ship.

(a)



(b)

[Figure deleted due to confidentiality]

Figure I

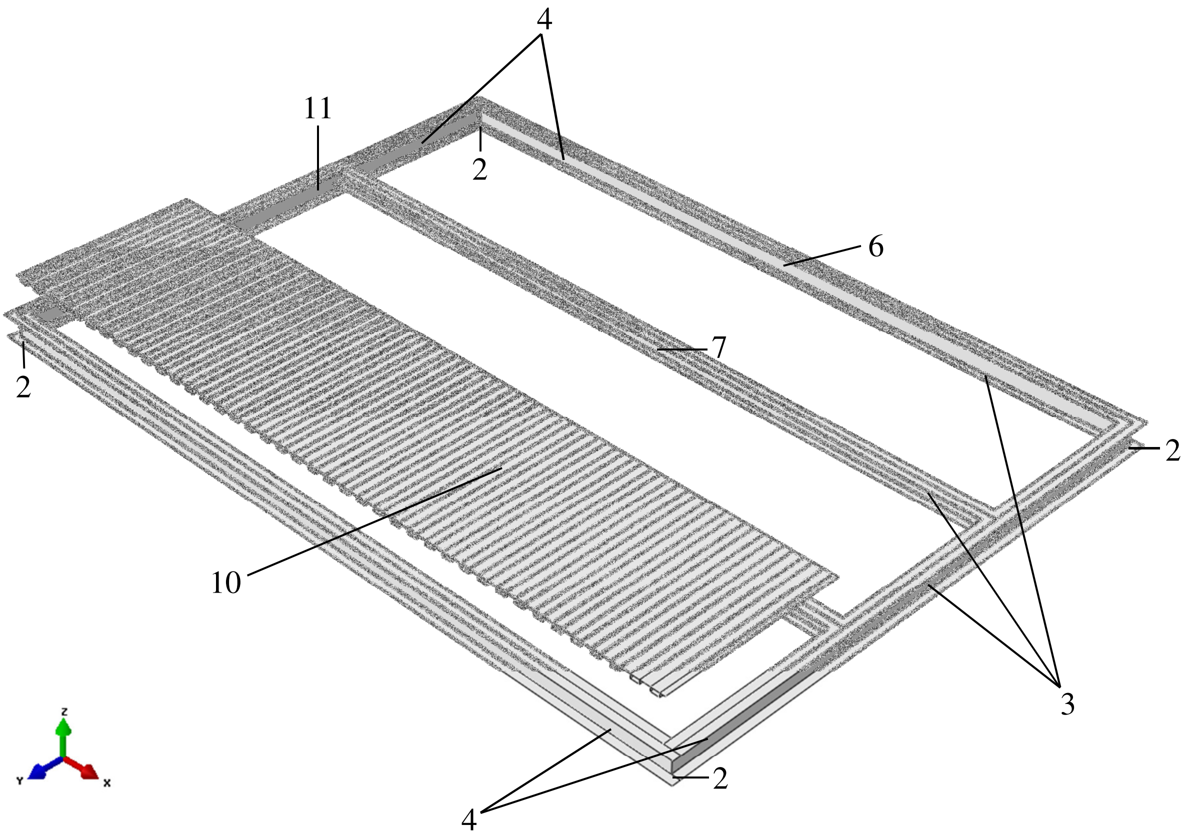


Figure II

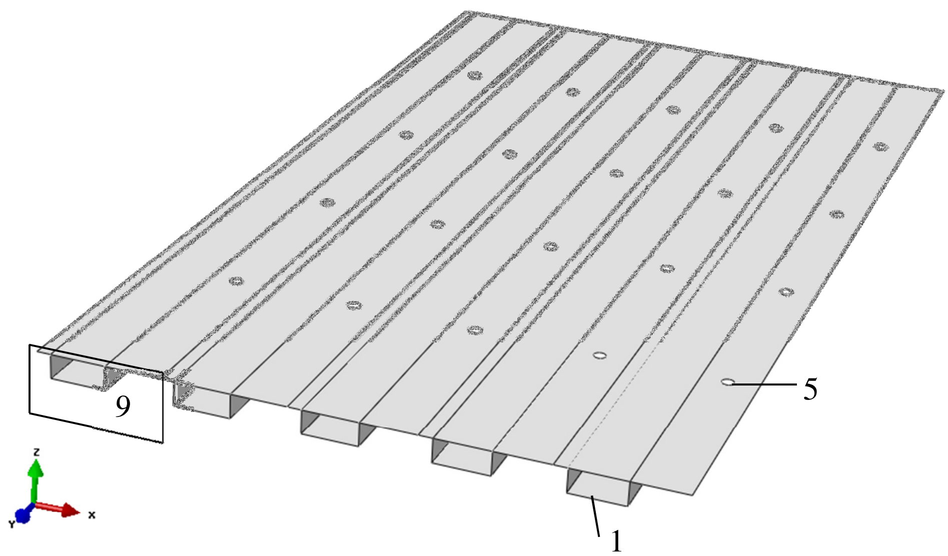


Figure III

Confidentiality

In agreement between the client TTS Marine AB and Chalmers University of Technology, entire Section 7.2 and parts of Section 7.3 is removed due to confidentiality. The full version can be obtained by contacting TTS Marine AB.

1 Introduction

During the last decades environmental issues have gotten a more significant role within society at large but also in the shipping industry. During this period, the fuel oil prices have been increasing, giving the shipping industry smaller marginal profits due to higher expenses (Heydová et al, 2011). Furthermore, the International Maritime Organization (IMO) are imposing more stringent requirement regarding hazardous gases, such as NO_X and SO_X. These changes are making light weight alternatives in ship equipment a viable option as a result of the possible long term savings in fuel expenses. Regardless of the increased production cost often associated with alternative materials, effort is put into finding light weight solutions for the marine industry.

For example, Lauenstein and Sökjer-Petersen, (2001) investigated the possibility to reduce the weight in car deck panels by using extra high tensile steel. Gunnarsson and Hedlund, (1994) concluded that extruded aluminium profiles could be used in car deck structures in order to obtain a lower weight. Forslund, (2002) studied how the use of aluminium in the stiffened top plate affects the effective breadth of a steel beam. Furthermore, a sandwich structure made from extruded aluminium was constructed and tested with good results regarding structural strength by Hanson, (2000). However, this design was too costly to be used.

If the higher investment cost can be motivated by an increased earning capacity and following reduced operation costs, the liftable car deck panels in Pure Car and Truck Carriers can be constructed from an alternative material instead of steel. Lightweight materials can be sustainable both from an economic and environmental point of view since it gives the possibility to reduce the average amount of fuel needed for each unit transported.

1.1 Background

In a PCTC vessel there are usually two different kinds of adjustable car decks; liftable and hoistable. Liftable and hoistable car decks are non-integrated decks, divided into panels that each can be moved in the vertical direction. While not in use they are stowed beneath the deck head. To keep the efficiency high and to optimize the cargo capacity in PCTC vessels these car decks are beneficial. The vertical movement is performed by a mobile deck lifter for liftable car decks and an integrated system for hoistable car decks. This gives the possibility to configure panels and optimize the height of the headroom for certain set of cargo.

Traditionally, these decks are made of steel and have a structural arrangement with a top plate supported by a grid of longitudinal and transverse stiffeners that translates the loads of the cargo to the load carrying pillars through the frame. Furthermore, to optimize the cargo intake, the structure is usually restricted to a specific building depth. Consequently, the elastic properties of the material are not utilized fully. An example of a conventional car deck structure can be seen in Figure 1.1. The holes that can be seen in the structure are used to attach the lashing hooks to ensure the cargo does not move due to wave induced motion.

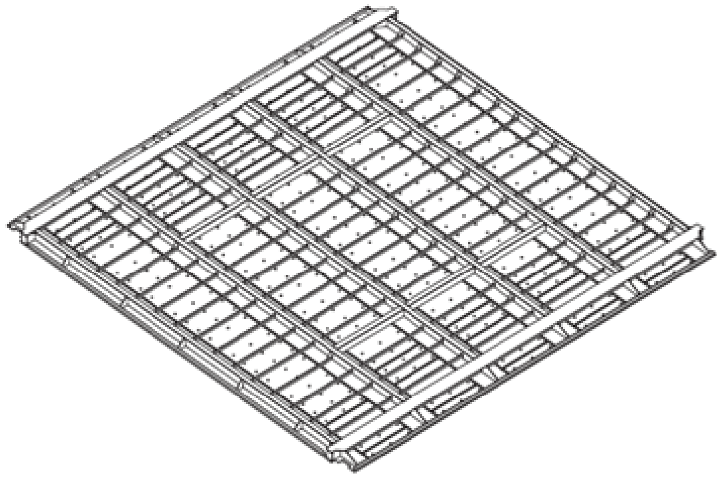


Figure 1.1 *Structural arrangement of a conventional car deck.*

Replacing a certain amount of steel with aluminium might increase the long term profits for the shipowner. While replacing the entire structure with aluminium would be impossible due to the building depth limitations, there is a large weight-reduction potential of replacing the top plate of car decks with an aluminium solution. The reason for this is the fact that the steel top plate of a car deck cannot be made thinner than 6 mm due to manufacturing limitations. Consequently, the top plate corresponds to approximately half of the total weight.

1.2 Problem definition

The objective of this thesis is to evaluate the possibility of replacing today's conventional car deck panels in PCTC vessels with an alternative light weight structure while satisfying the design requirements of building height, deflection, as well as the classification society Det Norske Veritas (DNV) stress requirements for car deck panels. This is done by replacing the top plate and stiffeners with an aluminium solution. Aluminium makes it possible to reach a lower weight for the car deck panel but with the drawback of higher investment cost. The aim is to find a cost effective solution that gives the shipowner a reasonable payback time compared to the expected increased investment cost.

Furthermore, manufacturing and material costs are relevant to evaluate since they affect the investment cost. If the investment cost is too high compared to the benefits, the solution might not be desirable for shipowners to invest in. A more detailed analysis of the wishes and demands of the stakeholders are presented in Section 1.4.

1.3 Methodology

The car deck panel comprises of two main parts; the stiffened top plate and the beam system, which in this thesis are treated separately. The stiffened top plate supports the vehicles and transfers the loads to the beam system, which in turn transfers the loads to the attachment points of the car deck panel. For the stiffened top plate, deflection is not a critical design parameter, thus it can be constructed from aluminium, making for a great potential weight saving. However, aluminium has a larger material cost than steel. Thus, the first step is to focus on the detailed design of the stiffened top plate.

This decides the boundary conditions for the beam system, which is treated in the following part.

The stiffened top plate is designed utilizing the software MATLAB (The MathWorks, 2012) and Abaqus (Dassault Systèmes, 2013). A program written in MATLAB is used to evaluate the global responses while Abaqus is used to evaluate the local stresses.

The design of the beam system is based on the stiffened top plate, since the profiles are designed to be placed freely over a certain distance. This distance decides the minimum number of longitudinal stiffeners in the beam system. The beam system is analysed in ANSYS (ANSYS, (2014)) with a response surface screening in an initial stage for finding a viable solution, and optimized by utilizing the goal driven optimization tool (GDO).

Subsequently, the stiffened top plate and beam system are simulated together using GeniE (DNV Software, 2014) to ensure that no stress exceeds the maximum permissible stresses. Extra attention is given to the stresses in the boundary conditions, which can be different from when simulating the stiffened top plate and beam system separately.

The last step is calculating the cost of the new design as well as comparing it to the reference solution. Furthermore, weight reduction and the corresponding fuel savings are taken into account. These calculations indicate whether the solution is economically sustainable or not.

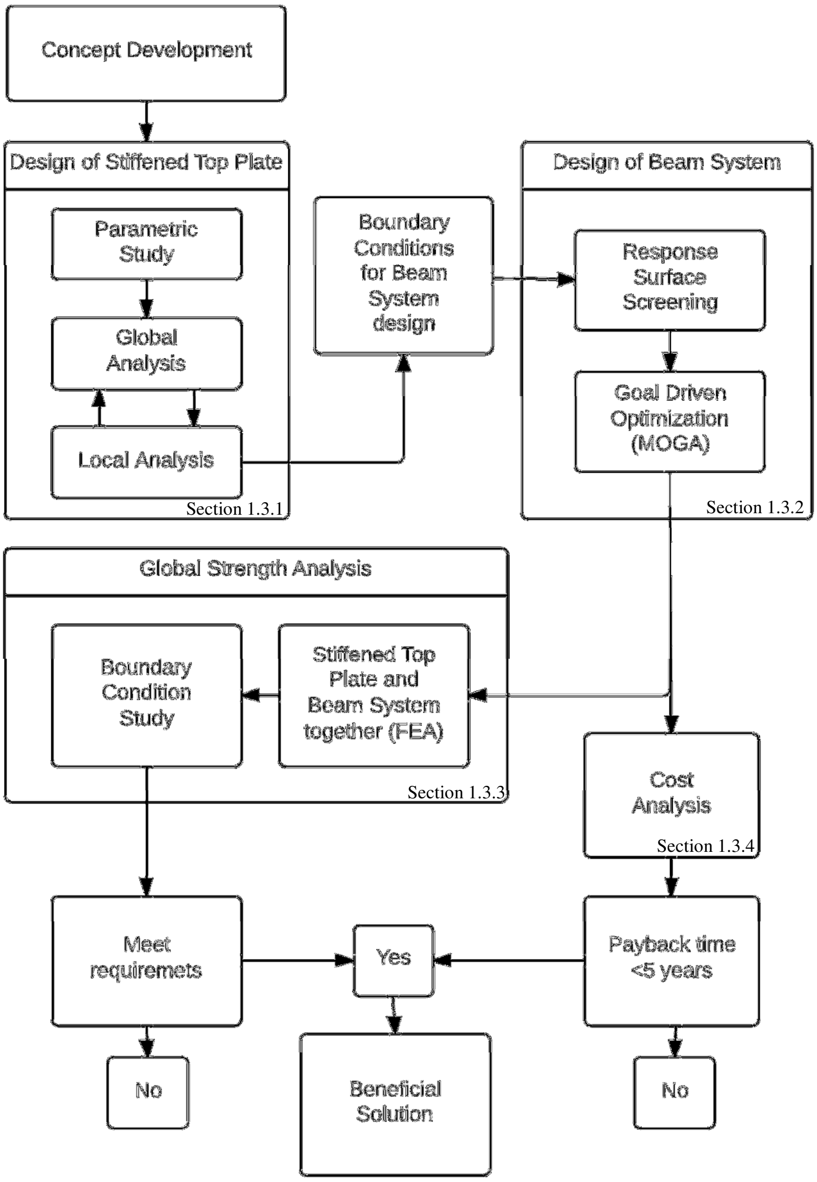


Figure 1.2 *Flowchart describing the work process.*

1.3.1 Design of stiffened top plate

Several concepts are evaluated for the stiffened top plate. One interesting type of stiffened top plate is extruded aluminium profiles which are connected to each other. The profiles can be connected either by welding or by mechanical fastening. The concepts are presented in detail in Section 4.1.

To compare the concepts, an elimination-matrix, see Table 4.2, is utilized. Different aspects such as estimated weight, estimated manufacturing cost, possibility of avoiding exceeding the maximum permissible stresses, buckling and failure due to point loads, insecurity in terms of knowledge and risk of fatigue are weighted. In this comparison, one concept is chosen to be evaluated further in an iteration process. The purpose of the iteration is to minimize the material of the stiffened top plate, since it will account for the majority of the production cost.

1.3.2 Design of beam system

The purpose of the beam system is to transfer the loads from the top plate to the attachment points of the beam system. This should be achieved with a limited deflection of the system as well as low weight and manufacturing cost. Consequently, if this can be utilized, the beam system can be made lighter. The difference between the concepts evaluated is the number of longitudinal and transversal beams in the beam system.

1.3.3 Final simulation and boundary condition study

When the final design is set, everything is simulated together, using the software GeniE. This simulation shows how the stiffened top plate contributes to the deflection of the beam system. Special attention is given to the boundary conditions.

1.3.4 Cost analysis

The purpose of the cost analysis is to calculate a more exact manufacturing cost of the new design. The investment cost is expected to be higher than that of the reference solution. However, if the weight of the panel is lower, the increased investment will get a payback time due to reduced fuel consumption. Furthermore, aspects such as reduced need of ballast tanks due to a lower vertical centre of gravity can be taken into account.

If the investment for the new design is too high, it will be less likely to be used today. However, it can still be interesting for the future since the fuel prices are expected to increase. In addition, the raw material prices fluctuate which can make a solution more or less viable in the future.

1.4 Design criteria

In order to compare the different concepts with the reference solution as well as designing an applicable structure, the design criteria are established together with TTS Marine AB. These criteria are also used as constraint functions in the optimization.

1.4.1 Dynamic factor

When a ship moves in the water it is subjected to wave induced motion. Because of this motion a dynamic factor is introduced to the evaluation. This dynamic factor is taken as the worst case that will occur for a PCTC. Hence, the dynamic factor is 1.5 in accordance with DNV, (2014a) and is added to all loads when evaluating stresses. A ship is pitching and rolling due to waves at sea-going mode, causing different accelerations at different positions of the ship. Pitching of the ship causes the highest accelerations in the most aft and forward part of the ship, while rolling causes the highest accelerations at the sides. The dynamic loads are in general smaller in the middle of the ship.

1.4.2 Deflection

When the liftable deck is used for cars only, the height from the cargo to the above deck is kept to a minimum in order to optimize the cargo intake. However, a fixed free height needs to be maintained to ensure free passage between decks. Subsequently, in order to maintain this free height, the sum of the moulded depth of the deck and the maximum deflection needs to be restricted; this is established to be 455 mm. When evaluating deflection, no dynamic factor is used since the panels deflect equally when loaded.

1.4.3 Stresses

The maximum load the car deck will be subjected to is an axle load of 1.5 tons. This is decomposed into two criteria which are evaluated separately, a point load of 750 kg/wheel and a uniformed distributed load of 250 kg/m^2. These loads together with the self-weight of the panel will give rise to stresses in the structure, which should not exceed the maximum permissible stresses according to Table 1.1 below. These stresses are dependent on the material constant f_1 which is 0.61 for aluminium (DNV, 2014c) and 1.39 for steel (DNV, 2014b). The friction stir welded material is assumed weaker than the extruded material, consequently the material factor is lower, it is assumed to be 0.45 (TTS Marine AB, 2004).

Different structural members have different stress requirements; Table 1.1 presents the maximum permissible stresses for these. σ_x, τ_{xz} and σ_{vM} denote normal, shear and von Mises stresses. The mechanical properties of the materials used in this study are presented in Table 1.2.

Table 1.1 *Maximum permissible stresses for different structural members in the car deck structure.*

Aluminium	$\boldsymbol{f_1}$	$\boldsymbol{\tau_{xz}}$ (MPa)	$\boldsymbol{\sigma_{vM}}$ (MPa)	$\boldsymbol{\sigma}$ (MPa)	**Reference**
Plate for cars	0.61	55	112.1	97.7	DNV, (2014d)
Stiffeners	0.61	55	122.9	109.9	DNV, (2014e)
FSW	0.45	40.2	82	71.5	TTS Marine AB, (2004)
Steel	1.39	125.1	250.2	222.4	DNV, (2014d)

Table 1.2 *Material properties of the materials used in the study.*

	Constructional steel NV-36	**Aluminium NV-6063-T6**
Density (Kg/m^3)	7850	2700
Yield stress (MPa)	355	170
Poisson's ratio	0.3	0.33
Young's Modulus (GPa)	210	70

1.4.4 Stakeholders' requirements

The above requirements are the most critical in order to design a safe structure that complies with the regulatory requirements. However, other stakeholders are involved in the design and manufacturing of a liftable car deck. Below follows a list of stakeholders associated with the liftable car deck. Table 1.3 shows their respective demands or wishes.

- **TTS Marine AB**: The company that designs and sells the car deck panels.
- **Classification Societies (DNV)**: Approves that the car deck is designed in compliance with standards.
- **SAPA Profiler**: The company that manufacture extruded aluminium profiles.
- **Shipyard**: Installs the car deck during the construction of a ship.
- **Shipowner**: Operates the ship where the car deck is installed.
- **Cargo owner**: Owns the cargo transported on the car decks.
- **Society**: Wants a product that is consistent with a sustainable future.

1.5 Limitations

The limitations in thesis were the following:

- The study focuses on two reference panels with specific dimensions and locations in the ship. One panel in the centre of the ship (14.64 x 14.37 m) and one on the side (10.22 x 14.37 m).
- The maximum vertical extent of the panel, including its deflection, is 455 mm.
- The classification rules used as a base for the design are those of Det Norske Veritas (DNV).
- The analysis is limited to the structure. Hence, any effects of a lower weight, such as vibrations, are omitted from the evaluation.
- A patented solution for lashing in aluminium is used. Consequently, no attempts to modify any parts of the lashing are made.
- The fatigue life of welds and other connections are not evaluated.
- Dynamic loads are evaluated with the use of a dynamic factor.
- The global strength of the beam system is evaluated using a uniformly distributed load while axle loads from the cars are used for the local strength of the panels.
- The behaviour of stresses around bolting holes is not evaluated.

Table 1.3 The stakeholders' requirements.

Stakeholder	Requirement	Value	Demand or wish
TTS Marine AB	Panel Dimension	14.46x14.37 m	Demand
		10.22x14.37	Demand
	Able to withstand local loads	750 kg/tyre	Demand
	Tyre area	150x200 mm	Demand
	Able to withstand global load	250 kg/m2	Demand
	Building depth + deflection	455 mm	Demand
	Competitive product	Able to economically compete with other solutions	Wish
	Weight reduction	25%	Wish
SAPA Profiler	Maximum width	620 mm	Demand
	Minimum thickness	2 mm	Demand
	Maximum height	200 mm	Demand
Shipowner	Easy cargo handling		Demand
	Lightweight		Wish
	Lashing compatibility	Lashing holes minimum 50 mm from webs	Demand
	Payback time	*[Deleted due to confidentiality]*	
Cargo owner	No/low risk of damage of cargo due to the design of the car deck, i.e. falling tools.		Wish
DNV	Maximum permissible stresses (aluminium plate) (MPa)	$\sigma_x = 97.7$ $\tau_{xz} = 55$ $\sigma_{vM} = 112.1$	Demand
	Maximum permissible stresses (aluminium stiffener)	$\sigma_x = 109.9$ $\tau_{xz} = 55$ $\sigma_{vM} = 122.9$	Demand
	Maximum permissible stresses (Steel)	$\sigma_x = 222.4$ $\tau_{xz} = 125.1$ $\sigma_{vM} = 250.2$	Demand
Society	Environmental friendly product		Wish
Shipyard	Easy to assemble		Wish

2 Reference solution

The ship used as a reference in this thesis is a PCTC vessel with approximately 150 liftable car decks, corresponding to an area of 23 680 m^2. The supporting pillars divide the ship into three sections; port and starboard, which are mirrored, and the centre. Consequently, two different car decks are used as reference for this study, hereinafter referred to as side car deck panel and centre car deck panel, the car deck panels selected as reference solutions are marked in grey, see Figure 2.1. The panels have different working positions as presented in Figure 2.2. Depending on the cargo, the decks can be configured vertically to optimize the cargo intake.

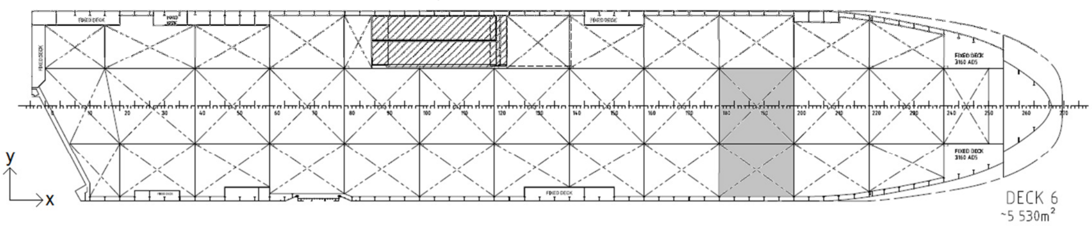


Figure 2.1 *Car decks on deck 6, with the reference car decks in grey.*

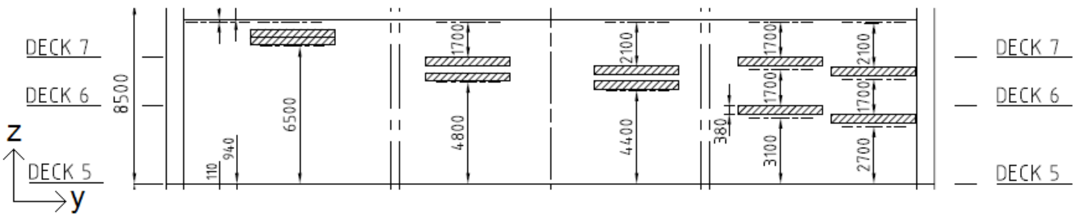


Figure 2.2 *Different configurations of the liftable car deck panels.*

The reference solutions have structural arrangements similar to a conventional car deck; it consists of a steel plate designed to resist a point load of 750 kg/tyre supported by a beam system that can support the weight of the plate and a uniform distributed load of 250 kg/m^2.

The top plate is stiffened with HP 120x6 bulb stiffeners and it is resting on the beam system. The thickness of the top field plate is an important factor for the total weight of the car deck. However, a thinner field plate increases the risk for entire field plate buckling and for deflection, but can be prevented by having a higher number of transversals. In the top plate, there are holes to lash the cargo. The distance between the holes are in the longitudinal direction 480 mm and in the transverse direction 700 mm. Figure 2.3 shows a model of the top and bottom of the centre car deck panel.

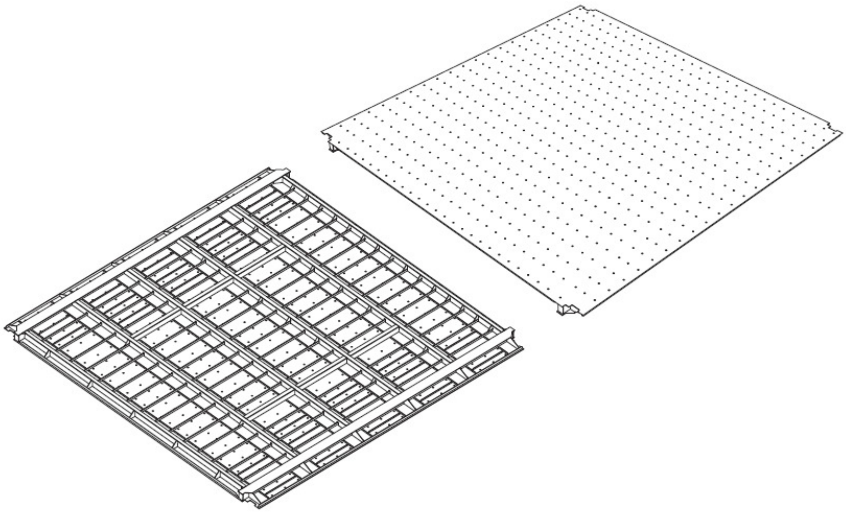


Figure 2.3 *Conventional car deck solution.*

2.1 Centre car deck

The centre car deck panel has a length of 14 370 mm and width of 14 640 mm. In Figure 2.4 the arrangement of the beam system can be seen. It is a conventional structural arrangement with longitudinal and transverse stiffeners. The dimensions of the stiffeners are presented in Table 2.1. The purpose of the two transverse beams in the beam system (T6 and T14) is to support the lifting of the frame while the longitudinal stiffeners translate the forces to the frame. The total weight of the centre car deck is 19 597 kg which corresponds to a structural weight of 93.16 kg/m^2.

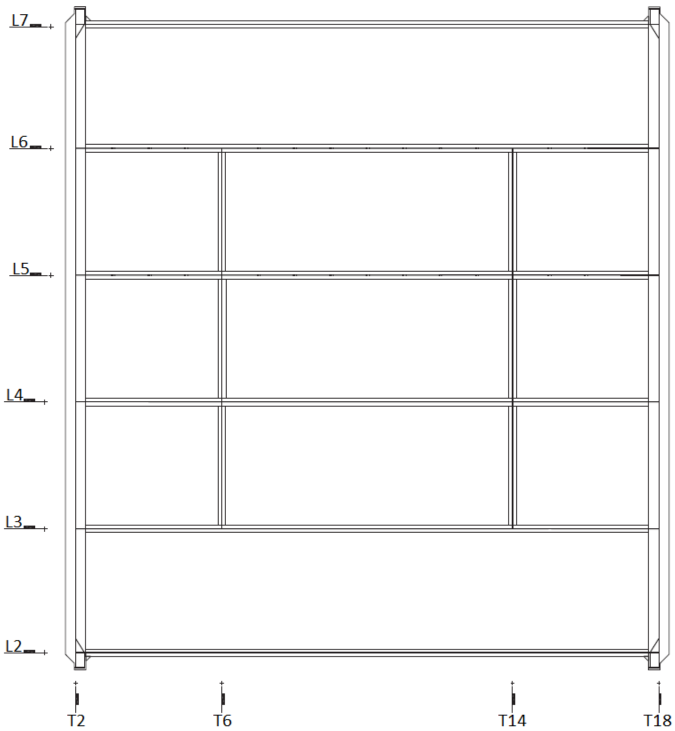


Figure 2.4 *The supporting frame and grid system of the centre reference deck.*

Table 2.1 *The dimensions of the different structural members.*

Name	**Dimensions mm**	**Type**
L3, L4, L5, L6	354x6 + 170x20	T
L2, L7	354x6 + 170x20	T
T2, T18	354x6 + 450x20	T
T6, T14	120x6	HP

2.2 Side car deck

The side car deck panel has a length of 14 370 mm and a width of 10 220 mm. Since the car deck panel covers a smaller area compared to the centre car deck panel, the beam system has a slightly different arrangement. The dimensions of these stiffeners are presented in Table 2.2. Figure 2.5 shows the arrangement of the supporting structure of the side car deck. The total weight of this car deck panel is 12 818 kg, which corresponds to a structural weight of 87.28 kg/m^2.

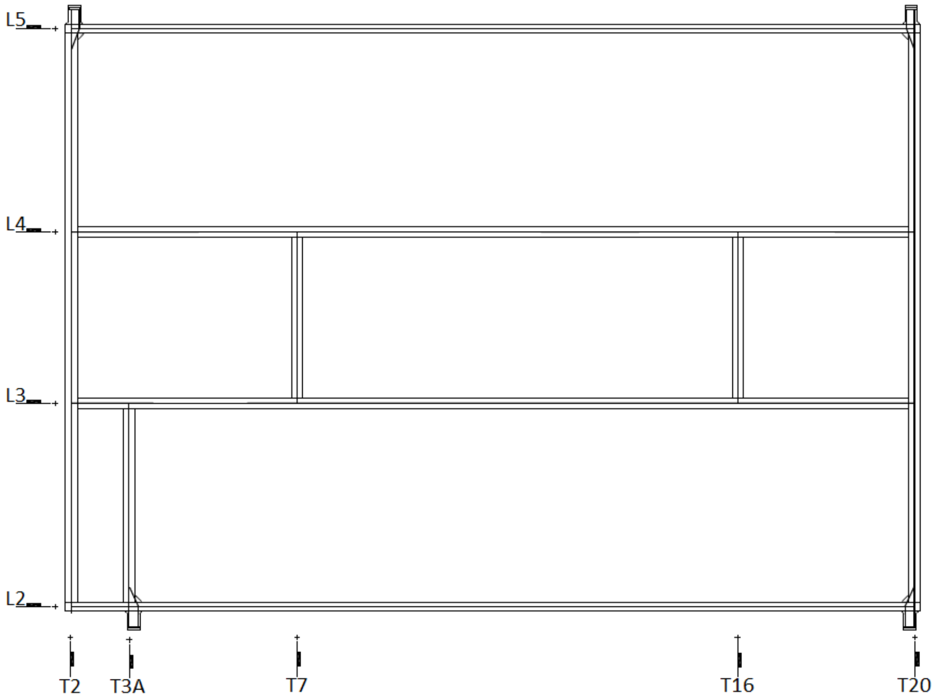


Figure 2.5 *The arrangement of the supporting structure for the side car deck reference solution.*

Table 2.2 *The dimensions of the different structural members.*

Name	Dimensions	Type
L2, L5	354x6 + 150x20	T
L3, L4	354x6 + 170x20	T
T2, T20	354x6 + 200x20	T
T3A	354x6 + 200x20	T
T7, T16	120x6	HP

2.3 Lashing

During voyage, the cargo is attached to the car deck panel by lashing straps with hooks at the ends. The lashing straps are attached to the front and back of the cargo and to the lashing holes located in the top plate, fixating the cargo. In conventional car deck panels the lashing hole is just a plain hole, the steel is strong enough to meet the requirement of the forces the lashing cause. However, if the top plate is made of aluminium, the lashing hooks will cause failure. This thesis is not evaluating alternative concepts for the lashing, but is applying a patented solution from TTS Marine AB.

This solution reinforces the lashing holes in the aluminium panel by introducing a ring with higher stiffness. This solves the problem with potential fatigue cracks, by evening out the load. Furthermore, this solution also handles the stress concentrations that will arise at the edge of the lashing holes.

Naturally, the lashing of cargo will give rise to stresses in the structure. However, the lashing holes used and the tyres will always be separated which results in low interaction between lashing stress and stresses due to cargo. Furthermore, the worst case scenario for lashing is when the ship has an extreme list and cars are almost hanging from their lashings; in this scenario, the tyre load is neglected (Andersson and Öisjöen, 2011). Based on this, it is assumed that the existing lashing solution can be used without further analysis, as long as the thickness is the same or greater.

2.4 Material

Steel is traditionally used in ship building since it is both low cost and strong. However, since the fuel prices increase, in time the benefits of light-weight materials grow more significant. Today's challenge is to develop solutions with lower weight and equal load carrying capacity.

Aluminium is a material that is increasingly used. Aluminium has, compared to steel, a lower Young's modulus, higher price and a lower density. Applying aluminium in some areas might increase the profit over time due to reduced fuel consumption, even if the investment cost increases. Aluminium is also more resistant against corrosion.

High tensile steel (HTS) is a material that can help contribute to weight reduction due to the increased yield strength compared to normal construction steel. However, the Young's modulus for HTS is the same as for conventional steel, which means the stiffness will be unchanged. In this thesis there are limitations in deflection and is most likely the dimensioned factor, thus HTS is not a viable option in this case.

Composites can have very varying properties depending on fibre and matrix materials, fibre directions and ply thickness (Agarwal et al, 2006). Composite materials are however very expensive, and are hard to make beneficial in the problem described in this thesis. Composite materials consist of layers, each with fibres in different direction and matrix as filling, making it possible to have different properties in different direction with low weight. Depending on the material in the fibres and in the matrix, the properties and cost vary. Composites might play a bigger role in the future, if the restrictions regarding emissions increases even more.

3 Theory

In this section the theories used in different parts of the study is introduced. These include engineering beam theory used in the design of all structural members as well as buckling theory, which is used in the stiffened top plate design and goal driven optimization which is used for evaluating the steel structure.

3.1 Engineering beam theory

For the initial design and evaluation of the stiffened top plate, engineering beam theory is used. A few assumptions are made in engineering beam theory. For example, plane sections must remain plane under loading. As long as the shape and size of the cross section is constant in the longitudinal direction as well as the cross section being closed, this assumption should be true for the structures evaluated (Thelandersson, 2002). Furthermore, the load case evaluated is a 2D case, with the loads applied according to Figure 3.1. Consequently, warping or mixed torsion is not evaluated in the initial design of the stiffened top plate. This means that when the load is applied unevenly, the stress response can be slightly higher. However, the local stress concentration from the tyres is assumed to be dimensioning. Furthermore, a conservative approach is made by using a mix between fixed and simple supports in the global strength evaluation.

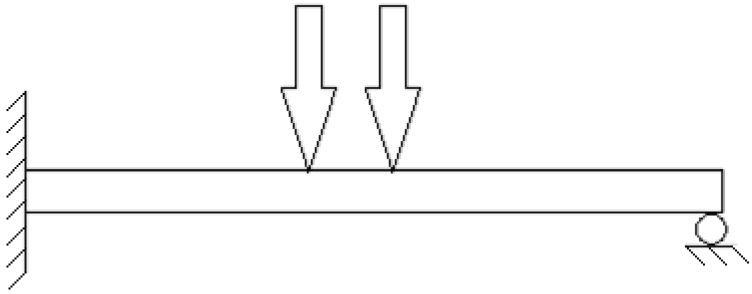


Figure 3.1 *Load case used in the evaluation of the global strength.*

Since the structure is assumed to be subjected to a pure vertical force, the only stress that is evaluated in the initial design is normal bending stress and bending shear stress. The normal bending stress, denoted σ_x, is calculated with Equation 3.1.

$$\sigma_x = \frac{M_y}{I_y} z \tag{3.1}$$

Where M_y is the bending moment, I_y is the moment of inertia for the cross section and z is the distance from the neutral axis to the fibre currently being studied in the cross section (Thelandersson, 2002). Figure 3.2 shows a graphic representation of the normal bending stress, zero stress will occur in the neutral axis and the stress level will increase as the distance from the neutral axis increases.

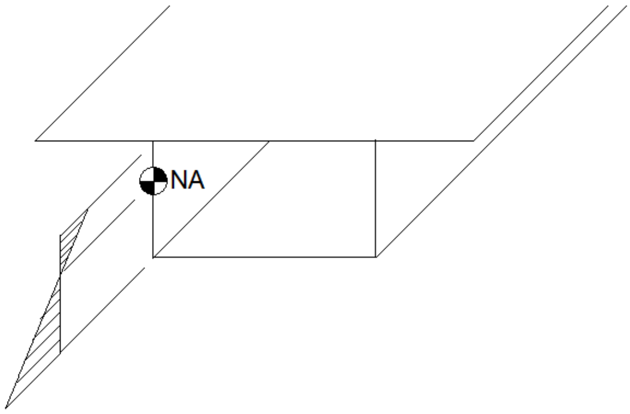


Figure 3.2 *The normal stress distribution due to bending.*

The bending shear stress in the structure due to the load of a tyre is calculated with Equation 3.2.

$$\tau_{xz} = \frac{V_y S_z}{t I_z} \tag{3.2}$$

Where V_y is the load, S_z is the static moment, I_z is the moment of inertia, and t is the thickness where the shear stress is evaluated. It is known that the highest shear stress in the structure will occur in the neutral axis. However, the normal stresses will be zero in the neutral axis which that the highest stresses will occur in the top or bottom of the structure since it is known that the shear stress will be lower than the normal stress for this structure and load case.

Even though both normal stresses, other than bending, and St Venant stresses will occur when the profile is unevenly loaded, this is not evaluated in the initial script. Consequently, the equivalent Von Mises stresses are evaluated with Equation 3.3 (Lundh, 2008).

$$\sigma_{vM} = \sqrt{\sigma_x^2 + \sigma_y^2 + \sigma_z^2 - \sigma_x\sigma_y - \sigma_y\sigma_z - \sigma_z\sigma_x + 3\tau_{xy}^2 + 3\tau_{yz}^2 + 3\tau_{zx}^2} \tag{3.3}$$

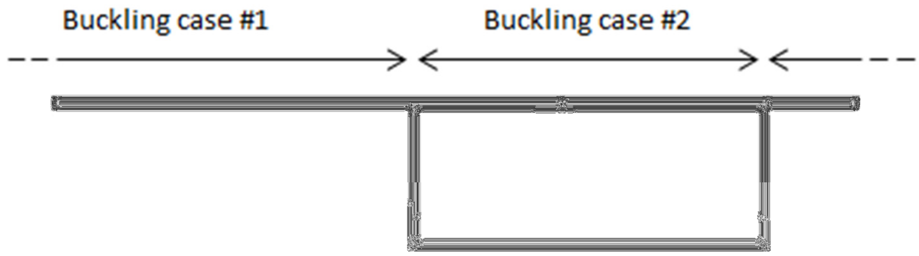
Where $\sigma_y = 0, \sigma_z = 0, \tau_{yz} = 0, \tau_{xy} = 0$. This assumption is made since the profile will be subjected to pure bending in this load case. This means that the approximate von Mises stresses is calculated with Equation 3.4 below

$$\sigma_{vM} = \sqrt{\sigma_x^2 + 3\tau_{xz}^2} \tag{3.4}$$

3.2 Buckling

The parametric study set rough dimensions of the aluminium profile, rejecting all solutions that cannot resist buckling. In the top plate, two cases of buckling can occur. The first case is buckling of the upper flanges; the second is buckling of the upper plate between the webs. See Table 3.1 for a visualization of the cross-section. The reason only the top plate is exposed to buckling is that the whole panel is deflecting downwards, leading to compressive stresses in the top plate and tension stresses in the bottom plate of the stiffened top plate.

Table 3.1 *Cross-section of aluminium profile.*



Buckling case #1: Buckling of the flanges of the top plate.	**Buckling case #2:** Buckling of the top plate between the webs.

For both buckling cases, the concerned area is considered a plate. The profiles are bolted in the ends and can therefore be considered as fixed supports. As long as bolts don't get loose, the profile is fixed. Hence, according to Ringsberg, (2011) the critical buckling stress can be calculated as in Equation 3.5

$$\sigma_e = \frac{\pi^2 E}{12(1-\nu^2)}\left(\frac{t}{b}\right)^2 4 \qquad (3.5)$$

In the equation, t is the thickness of the plate and b is the distance between supports. In the first case, buckling of the flanges, the distance between supports is the length of both flanges combined, since in the actual buckling case the profiles is welded together, making a plate as wide as two flanges. In the second case the distance between supports is the length between the webs of one profile.

However, since this theorem does not take plasticity into account, the stresses have to be checked. If the critical buckling stress σ_e, is higher than the yield point of the material divided by 2, the Johnson's and Ostenfeld's correction needs to be used (Ringsberg, 2011). The Johnson's and Ostenfeld's curve is describing the relation between influences of plasticity and buckling characteristics. The relation can be expressed according to Equation 3.6.

$$\sigma_{corr} = \sigma_y\left(1 - \frac{\sigma_y}{4\sigma_e}\right) \qquad (3.6)$$

In theory, all materials are perfectly elastic up to the yield point. In reality, the materials are starting to deviate from the perfect elastic behaviour at half the yield point. The correction factor (σ_{corr}) corrects the buckling formulas to be closer to reality.

3.3 Deflection

Deflection is a dimensioning factor in this thesis. The stiffness of the profiles and beams is dependent on the cross-section, and the lower the stiffness is the higher will the deflection be. The deflection also depends on the boundary conditions at the attachment points, the deflection of a beam with simple supports in the ends have five times higher deflection compared with a similar case with fixed supports, as can be seen in Equation 3.7 and Equation 3.8 below (Lundh, 2008).

$$\delta = \frac{5QL^4}{384EI} \qquad (3.7)$$

$$\delta = \frac{QL^4}{384EI} \tag{3.8}$$

These equations can be deduced from the differential equation of the elastic line. The curvature of a line can be expressed in accordance with Equation 3.9

$$\kappa = \frac{1}{R} = \pm \frac{|w''(x)|}{(1+(w'(x))^2)^{\frac{3}{2}}} \tag{3.9}$$

Where $w(x)$ is the displacement of the line and R is the radius of the curvature. The following relationship has been made, see Equation 3.10.

$$\kappa = \frac{1}{R} = \frac{M}{EI} \tag{3.10}$$

Furthermore, the deformation of most beams is assumed small, hence $w'(x) \ll 1$ results in the simplification presented in Equation 3.11.

$$-EIw''(x) = M \tag{3.11}$$

This equation shows the relationship between the deflection, bending stiffness and moment.

$$\frac{d^2}{dx^2} w''(x) = \frac{d^4 w(x)}{dx^4} = \frac{q}{EI} \tag{3.12}$$

Depending on the boundary condition, the deflection will be given by the solution to this differential equation. As it can be seen the deflection is highly dependent on the free length between supports (Lundh, 2008).

The beam system in this study is considered being simply supported. However, simply supported in theoretical terms is that one support is pinned and the other one rolled. In reality, the supports are fixed in translational degrees of freedom but free to rotate. In reality, the boundary conditions are something between simply and fixed supports, but closer to simply. To be conservative, all calculations regarding the beam system in this thesis are based on the theoretical term of simply supports.

All deflections in this thesis are obtained by simulations with simple supports. The boundary conditions used in the software is displacement supports, with zero deflection in x-, y- and z-axis in one support, y-axis and x-axis in two supports and z-axis in all supports.

3.4 Goal driven optimization

This section presents the theory used in the Goal Driven Optimization (GDO) tool (ANSYS, (2014)). The GDO tool is a multi-objective optimization technique that finds the design that best fits the user defined objectives based on the input geometry parameters. The method is used to find a weight optimized solution for the supporting steel structure. The section contains a brief presentation of the different parts of the GDO tool which includes design of experiments, response surface, sensitivity analysis and the optimization algorithms.

3.4.1 Design of experiments

In this tool the geometry parameters are given an upper and lower value. Instead of running the simulation for all possible combinations of geometry parameters, the design of experiments determines sampling points to be explored in the most efficient way. This reduces the sampling points needed and consequently it reduces the computation time. The program combines one centre point with points along the axis of the geometry parameters, these points are determined by a fractional factorial design (Dodge, 2008).

In order to predict the shape of the response surface a second-order polynomial model is used. In this case it determines which geometry parameters gives a certain response and to what magnitude. Furthermore, the program uses central composite design to fit the second-order model (Montgomery, 2009).

3.4.2 Response surface

The response surface is used when analysing a problem where the response is influenced by multiple variables and the aim is to optimize this response. In this study it is used to find a design with a limited height and deflection combined with a low weight. This is of course dependent on several different geometry parameters and a range of output parameters such as web height, deflection and geometry mass. The geometry parameters can be seen as a function $y = f(x_1, x_2, \ldots x_k) + \epsilon$ where ϵ is the error in the response. A response surface is then represented by $f(x_1, x_2, \ldots x_k)$. An example of a response surface can be seen in Figure 3.3 (Montgomery, 2009).

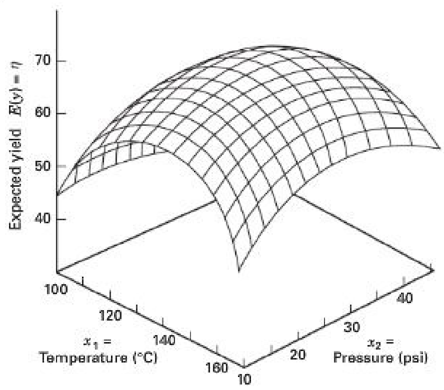


Figure 3.3 *An example of a response surface for an analysis, showing the expected yield for temperature and pressure as the input parameters. (Montgomery, 2009)*

Since the relation between the response and the geometry parameters are unknown, the program approximates these responses with a second-order polynomial model.

$$y = \beta_0 + \sum_{i=1}^{k} \beta_i x_i + \sum_{i=1}^{k} \beta_{ii} x_i^2 + \sum_{i<j}^{k} \sum \beta_{ii} x_i x_j + \epsilon \qquad (3.13)$$

This approximation is usually enough in order to run an analysis of the response surface that corresponds to the actual system (Montgomery, 2009). However, since the software approximates the surface between calculated points, any solution taken from the response surface needs to be rechecked.

3.4.3 Parametric sensitivity analysis

The computational time of the optimization routine is mainly dependent on the number of input parameters. The software runs a parametric sensitivity analysis of the input parameters and based on this the user has the choice to constrain parameters that have a very small effect on the output. Based on the sensitivity analysis, the user can make changes to the range of different parameters which in turn can make the analysis less time consuming.

3.4.4 Optimization

ANSYS offers a range of optimization methods in their GDO tool; in this study screening and Multi Objective Genetic Algorithm (MOGA) has been used. Screening is based on the Hammersley algorithm (Diwekar & Xu, 2005). It uses direct sampling and sorting to find a multiple objective design. This method is well suited for preliminary design since the number of points does not increase exponentially with the number of input parameters. Because of its simplicity, the screening method is preferred as a base for more advanced optimization algorithms.

The MOGA is inspired by natural evolution where crossover and mutation can yield an offspring that is superior to both parents. In this study the MOGA has been used for structural optimization. Interested readers can find more information about genetic algorithms in (Konak et al, 2006) and more general information about the other optimization algorithms in (ANSYS, (2014)).

4 Concept Development

In this thesis, the stiffened top plate is designed before the beam systems. This has both benefits and drawbacks. The most significant benefit is that the stiffened top plate can be weight-reduced at much as possible, and thus reducing the investment cost since the stiffened top plate is made of aluminium. The drawback is it is setting more narrow boundary conditions to the beam systems beneath.

To generate a concept for the stiffened top plate, several proposal designs are weighted against each other in an elimination-matrix. The elimination-matrix is based on a parametric study using, as well as discussions and hypotheses between the authors and experts from TTS Marine AB. The most promising concept of the stiffened top plate is iterated by a global analysis using MATLAB, and a local analysis using Abaqus. Depending on how the stiffened top plate is designed and applied, it might give restrictions to the design of the beam system. The beam system is analysed and optimized using ANSYS workbench.

4.1 Concept generation of stiffened top plate

The first part is to generate concepts for the stiffened top plate. Several concepts are investigated but only one is evaluated further. The top rated concept from the elimination-matrix, see Table 4.2, is extruded aluminium profiles that extends between the longitudinal stiffeners and are fixed with bolts. The benefits of extruded profiles are that they contribute to the stiffness of the whole panel. The profiles are fixed to each other by friction-stir welding (FSW), forming a stiffened top plate structure. FSW allows the profile to contribute to the effective flange of the stiffeners across the panel.

An alternative concept of the stiffened top plate is a honey comb structure. The idea is to have two thin plates with honey comb structures (hexagon) in the vertical direction in between, see Figure 4.1. The honey comb structure is expected to add a certain contribution to the panel stiffness and reduce the deflection. It is also possible to add foam in the honey combs to even out the loads. However, this concept is rejected due to high estimated cost.

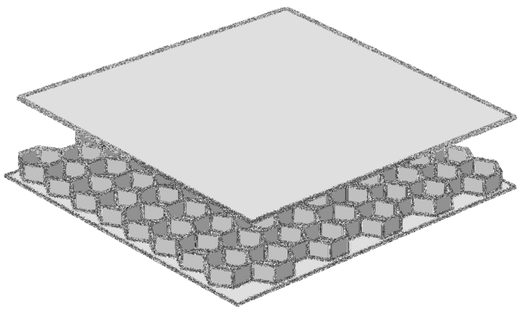


Figure 4.1 *Honeycomb sandwich structure*

Composite materials are a combination of two or more materials. Normally, composite materials consist of fibres of one or more materials in plies upon each other. The filling material (matrix) is one or more materials. Depending on how thick the plies are and in what direction the fibres lie in, different properties can be obtained in different directions. Composite materials can give the possibility of reaching desired properties in certain directions with a low weight (Agarwal et al, 2006). This concept is rejected due to the high cost of composite materials. Composite materials are expected to have higher potential in the future, if the oil price continues to increase. A short summary of the top plate concepts are presented in Table 4.1

Table 4.1 *Brief explanation of various concepts for stiffened top plate.*

Concept	**Explanation**	**Benefits**	**Drawbacks**
Extruded profiles	Various cross-sections are evaluated, see Figure 4.2.	Low manufacturing cost.	Sensitive to buckling.
Honey combs	Two thin plates with a honey comb structure in between. It is also possible to add foam in the honey combs to even out the loads.	Expected to add a certain contribution to the panel stiffness. Reduce deflection.	High manufacturing cost. High estimated weight.
Sandwich structure	Composite materials are a combination of two or more materials that together can reach different properties in different directions.	In theory, it is possible to reach high strength levels in desired directions with a low weight.	High manufacturing cost.

Three different cross-sections of extruded profiles are evaluated to make sure the optimal solution can be found. . All three cross-sections are evaluated in a parametric study to see which cross-section can reach the lowest weight per area. The parametric study does only take global analysis into account, but the idea is to get estimations how big potential each concept has from a weight-reduction point of view. The cross-sections with brief explanations are presented in Figure 4.2.

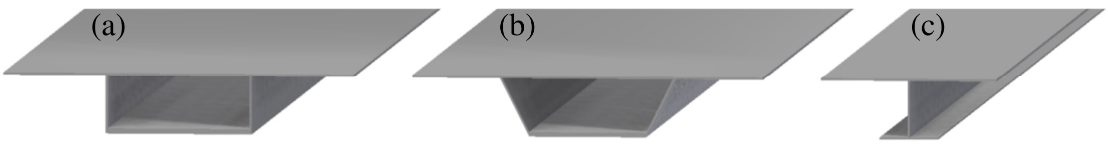


Figure 4.2 *Concepts of extruded aluminium: a) Concept #1: Vertical webs with closed cross-section, making this concept good from a weight-perspective and resilient to buckling. b) Concept #2: Webs inclined with an angle and closed cross-section, making this concepts ideal for resist buckling but of the cost of slightly higher mass. c) Concept #3: Vertical, single-web cross-section, making this concept ideal for reaching low weight solutions, but of the cost if higher risk for buckling.*

In the parametric study the inputs of different dimensions is set as intervals with discrete numbers. All possible combinations are evaluated. The combinations that cause too high stress and/or buckling are rejected. The purpose of the parametric study is to find which cross-section can reach the lowest weight per square meter while meeting the strength requirements.

All combinations of dimensions that do not meet the requirements of buckling and avoiding maximum permissible stresses are rejected, and among the remaining candidates the solution with the lowest mass per square meter is kept. The reason the two first concepts (see Figure 4.2a and Figure 4.2b) have the same number of + in Table 4.2 is that the parametric study finds the lowest weight when the web is vertical. Practically, this means these two concepts are the same.

Table 4.2 *Elimination-matrix for stiffened top plate concepts.*

	Explanation: (+) good (-) bad (?) need more info					
	Extruded #1	Extruded #2	Extruded #3	Honey combs	Honey combs with foam	Sandwich (composite materials)
Estimated weight	+	+	+	+	+	+
Estimated cost	+	+	+	-	-	-
Meet strength requirements	+	+	-	+	+	+
Meet buckling requirements	+	+	-	+	+	+
Resistance for point loads	+	+	+	+	+	+
Resistance for fatigue	+	+	+	?	?	-
Lack of experience	+	+	+	-	-	-
	Concept accepted	Concept rejected	Concept rejected	Concept rejected	Concept rejected	Concept rejected

Composite material (sandwich) is rejected partly due to manufacturing cost, but also due to the fatigue limitations. Fibre breakdown can occur sudden and is considered a risk of safety in this case (Agarwal et al, 2006). The honey comb concepts are rejected due to the lack of experience for this kind of structures in the marine industry. The single-webbed aluminium profile concept (Extruded #3) is rejected due to higher weight compared to the other aluminium profile.

The concept evaluated further is studied using beam theory and plate theory. Engineering beam theory and plate theory is used to perform a global analysis, evaluating how the different concepts behave regarding buckling, deflection and what stresses occur for different free lengths.

4.1.1 Parametric study

The first step of designing the stiffened top plate is to evaluate the aluminium concepts with a parameter study. Each dimension of the cross-sections are divided into a number of points with the lowest respectively highest manufacturable limit as a constraint. All possible combinations of dimensions are then evaluated and the lowest weight solution that meets all requirements is chosen. Hence, all dimensions that cause too high stress and/or buckling are rejected. The target is to obtain dimensions that minimize the weight per square meter of the panel while the stress criterions are fulfilled. See Figure 4.3 how the stress due to bending and weight behaves to the web height, lower plate thickness and upper plate thickness. Figure 4.3 is based on the final design, see Figure 4.2a.

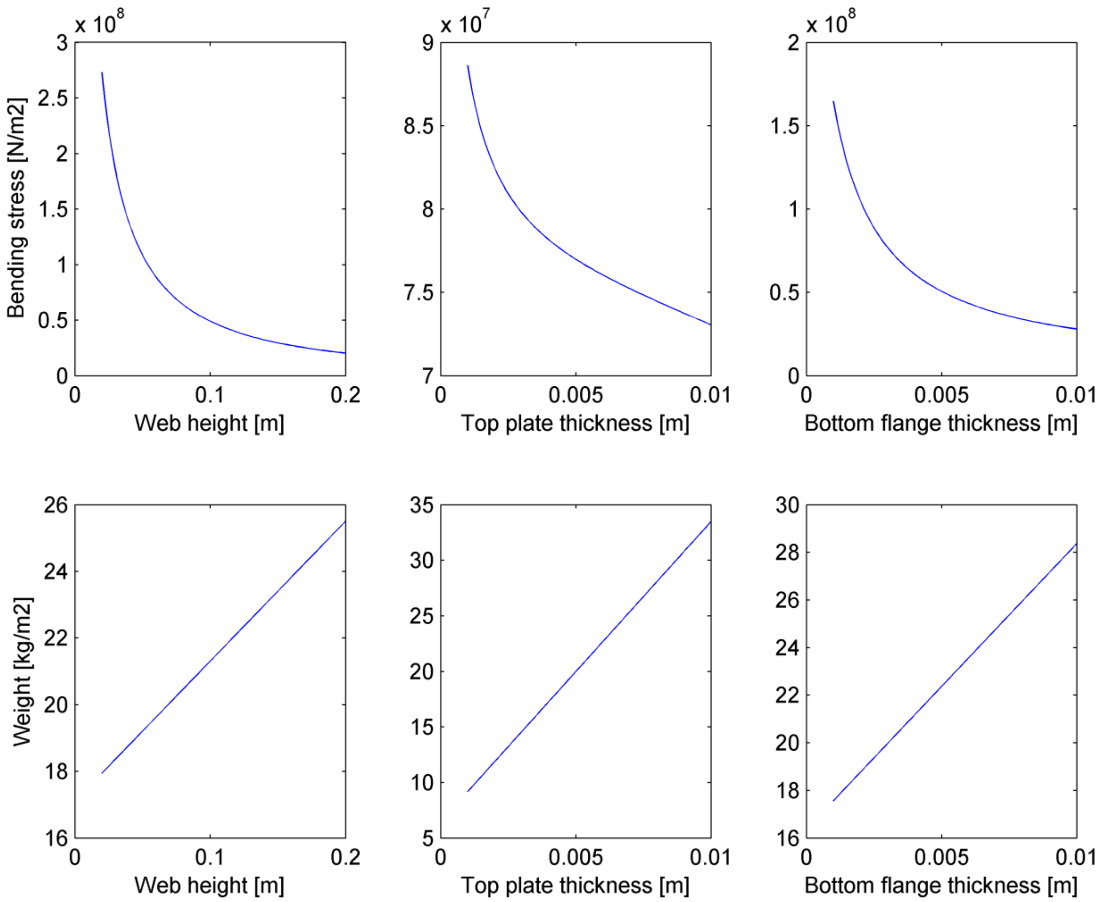


Figure 4.3 *Bending and weight depending on web height, upper and lower plate thickness. In the graphs, all dimensions are set according to the end design, except for the variable under study.*

Figure 4.3 shows that the web height has the highest contribution to the bending stiffness, while it has a low contribution to the weight. The upper and lower plate thicknesses affect the neutral axis, which in turn affects bending stresses. To lower the bending stress the lower plate thickness needs to be increased. It is evident that to obtain a low weight solution, a high web height should be used while the plate thicknesses are kept at a minimum. The parametric study is strongly dependent on the free length between supports of the profile. Several cases are carried out for different number of beams. Since the ratio length-height is big, the case is considered being pure bending. Therefore, no further evaluation is done of the shear force since it is assumed to be low.

4.1.2 Design iteration of the aluminium profile

Since the design from the parametric study is over or under dimensioned, a more detailed analysis is necessary. To reach the final design of the aluminium profile it is iterated with a design spiral using engineering beam theory and FEM. The design spiral iterates the design between a global analysis and a local analysis. The global loads analysis calculates stress due to bending (See Equation 3.1) and buckling (See Equation 3.5 and 3.6), see Section 3 for more details. The local analysis calculates stresses due to local loads near the tyres and boundaries using FEM. The goal of this process is to optimize the stiffened top plate before designing the beam system.

4.2 Concept generation of the beam system

The purpose of the beam system is to transfer the uniformly distributed load, which is set to 250 kg/m^2, to the attachment points. The design is restricted in building depth and deflection according to the design criteria as presented in Section 1.4. Depending on how the aluminium profile for the stiffened top plate is designed, the design of the beam system also has restrictions to the minimum number of longitudinal stiffeners to avoid to high stresses in the aluminium profiles due to bending.

Since the main purpose of this thesis is to keep the weight down it is beneficial to have as few beams in the structure as possible. In general, few large beams contribute more to the stiffness per mass than many small beams do.

$$I = \frac{bh^3}{12} \tag{4.1}$$

As can be observed in Equation 4.1, the height of the beam has a cubic contribution to the moment of inertia, which is the reason few large beams is to prefer to obtain a high stiffness to low weight. Therefore, the lowest allowed number of beams in the longitudinal direction is set by the aluminium profile. Consequently, only having beams in the longitudinal direction translates most of the loads to two out of four sides of the outer frame. By having transverse beams as well even outs the loads to all four sides. Only having beams in one direction must not however be a problem since the frame can be dimensioned differently on the different sides. The different concepts shown in Figure 4.4 will only differ in the number of transverse beams.

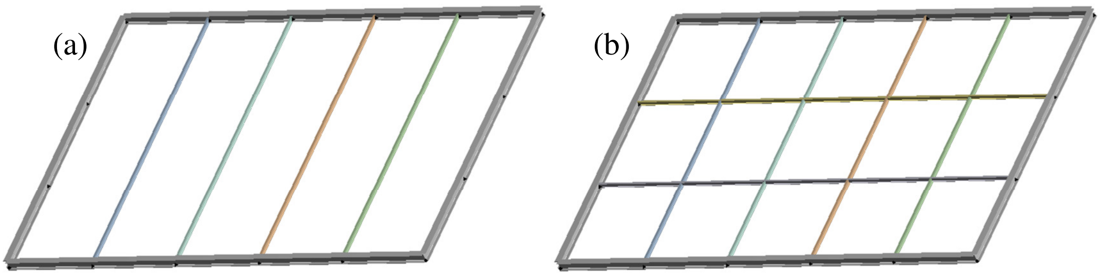


Figure 4.4 *Transverse stiffener: Concept (a) has no transversal beams, making it ideal for keeping manufacturing costs low. This concept also has the potential of reaching the lowest mass. Concept (b) has two transversal beams, which allows the structure to reach a lower deflection, with the cost of higher mass.*

Having beams in a diagonal pattern is possible to transfer the loads directly to the attachment points. However, the length of a diagonal beam is longer than a longitudinal, and therefore the deflection and/or dimensions are larger. The diagonal pattern of the stiffeners is not evaluated since it leads to a higher weight of the panel (Alatan and Shakib, 2012).

To avoid having the neutral axis of the structure in the middle of the stiffeners, they are fixed on the upper flange or on the bottom flange of the frame, see Figure 4.5. Both of these scenarios have pros and cons. If the stiffeners are fixed on the bottom flange, the aluminium profiles can fit on top and thus reducing the manufacturing cost, but with the drawback of even more limited web height of the longitudinal stiffeners. It is however possible to fit the aluminium profiles even if the longitudinal stiffeners are attached on top of the frame by bolting shorter profiles between each longitudinal stiffener. This is a dilemma of investment cost in relation to weight, where the investment cost is increased due to more bolting is needed, and the weight is reduced since the height of the web of the longitudinal stiffener can be used in higher extent to increase the moment of inertia.

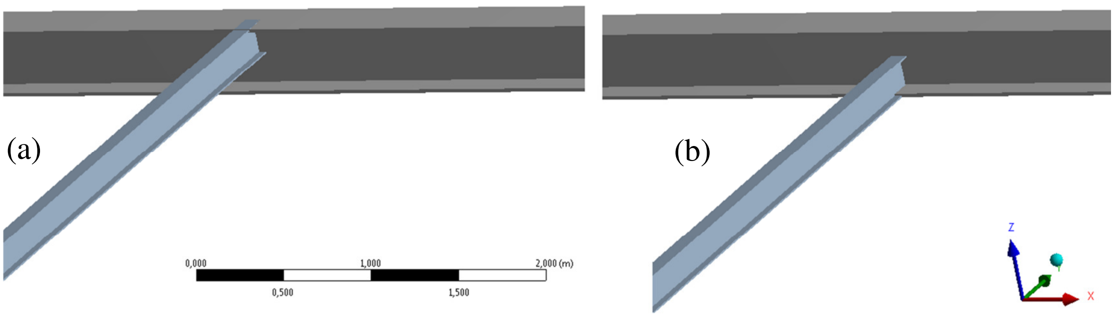


Figure 4.5 *Longitudinal stiffeners attached on upper flange (a) and on lower flange (b) of the frame.*

4.3 Assembly methods

This section presents how the structure is assembled. Depending on which assembly method is used; the investment cost, strength, fatigue life and boundary condition can vary.

4.3.1 Aluminium-aluminium connection

The idea with the aluminium profiles is that they shall be welded together and form a stiffened top plate. Since keeping the material mass for aluminium down is of value, welding is an interesting method since it doesn't require any overlap. Friction-stir welding (FSW) is a commonly used process-method for aluminium alloys that allows the plate to achieve good weld strength without post treatment (Nicholas ED, 1998). This method is applied in this thesis and can be motivated by the low concentration of defects and is in general a good choice when working with low thicknesses. Compared to conventional welding methods, friction stir welding is stronger in the welds which is necessary in this case because of the point loads from the tyres.

Other methods discussed are fusion welding and Tungsten Inert Gas (TIG) welding, and mechanical fastening. According to Ericsson and Sandström, (2011) FSW offer several benefits over TIG and fusion welding. FSW offers stronger welds that are more fatigue resistant, compared to TIG. Simultaneously, FSW is using a lower temperature than fusion welding, resulting in lower thermal stresses. Hence, FSW seems like the superior choice in this case due to the solid-phase weld, low distortion and low cost (Nicholas and Thomas, 1997). Mechanical fastening is the cheapest choice and has the potential of reaching the most economical beneficial solution. However, mechanical fastening is not evaluated in this thesis due to the lack of experience for the distribution of forces in this kind of structure. For example, utilizing this solution would significantly reduce the effective flange contribution from the aluminium on the steel beams.

4.3.2 Aluminium-steel connection

The aluminium profiles need to be mounted to the beam system, and depending on how they are mounted the boundary conditions vary. In this thesis the profiles are bolted into the frames and stiffeners of the beam system. This means that as long as the bolts do not come loose, the profiles can be considered as having fixed supports in the ends.

An alternative and potentially cheaper way is to lower the position of the longitudinal stiffeners and having the aluminium profiles placed on top of the stiffeners. Hence, the aluminium would only be bolted in the frame, but since the deflection is such a dimensioning parameter in this case, this is not beneficial.

4.4 Concept development summary

The final concept of the stiffened top plate is several aluminium profiles with two vertical webs, see Figure 4.2a, welded together using FSW. This cross-section has the potential of keeping down the weight of the stiffened top plate and therefore also the investment cost, while it meets the strength requirements. These profiles are bolted onto the beam system in the frame and in the longitudinal stiffeners.

Under the stiffened top plate, the beam system is designed with two longitudinal stiffeners for the side car deck panel and four for the centre car deck panel. Either of the car deck panels have transversal stiffeners because it only reduced the deflection marginal with the cost of higher weight, according to simulations, see Figure 4.4a for centre car deck panel. The longitudinal stiffeners are attached on the upper flange of the frame due to the magnitude of the web height and deflection correlation, see Figure 4.5a.

5 Structural Analysis and Optimization

In order to investigate the deflection and stresses in the structure, the finite element method (FEM) is used. Since the stiffened top plate and beam system are investigated separately, two different FEM softwares are used. Abaqus (Dassault Systémes, 2013) is used to verify the results obtained from the global analysis as well as evaluating local stress responses while ANSYS workbench (ANSYS, (2014)) is used when evaluating the steel structure. The optimization of the steel structure is also carried out in ANSYS workbench.

5.1 Aluminium profile

The aluminium profile is evaluated and weight reduced in MATLAB while Abaqus is used to verify the results obtained when utilizing engineering beam theory and to study the profiles in more detail. S4R shell elements are used since this is a thin structure. For this analysis a profile with the dimensions obtained from the parametric study script is modelled. The material properties used for the analysis are the ones presented in Table 1.2

Since the material changes shape during loading the analysis is run with a geometric nonlinear model, generally nonlinear geometry should be used if the deformations are larger than $1/20^{th}$ of the parts largest dimension (Dassault Systémes, 2008). While the plate deflection is small, it behaves in a non-linear way. Consequently, a non-linear analysis is used. When trying to reduce the weight of a structure it is important that a geometric nonlinear analysis is used in order to avoid over dimensioning. The reason for this is that the results from a nonlinear and linear analysis of the same structure can differ a lot. This is due to the change of shape and consequently changes of stiffness in the material. The load of a linear analysis retains its direction which will give a higher stress response while the load of a nonlinear analysis will follow the deformations. An exaggerated illustration of this can be seen in Figure 5.1 (Dassault Systèmes, 2008).

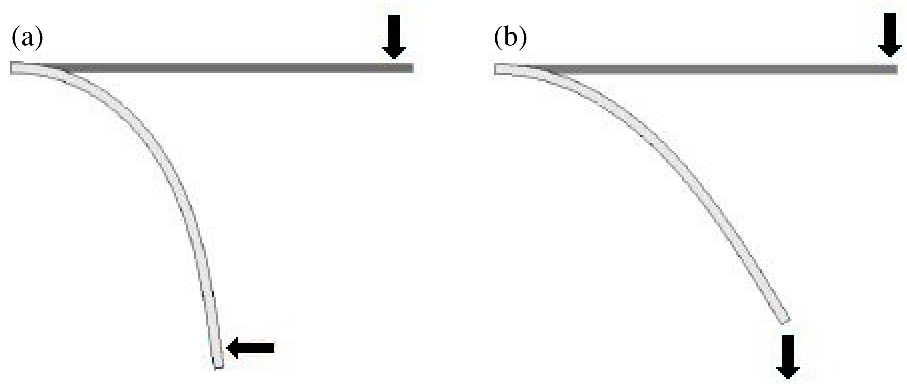


Figure 5.1 *The principal difference between loads of a non-linear analysis (a) and a linear analysis (b). Dassault Systèmes, (2008)*

The load cases for this analysis are when the wheels from two cars are situated exactly in the middle between supports, one case where the wheels are situated between the webs, hereinafter referred to as load case 1, and one case when the wheels are between the stiffeners, hereinafter referred to as load case 2. This is assumed to be the worst case scenarios. Figure 5.2 shows the structure as well as the wheel prints, the boundary conditions for the analysis are fixed at the short edges while the other edges have a symmetry boundary condition.

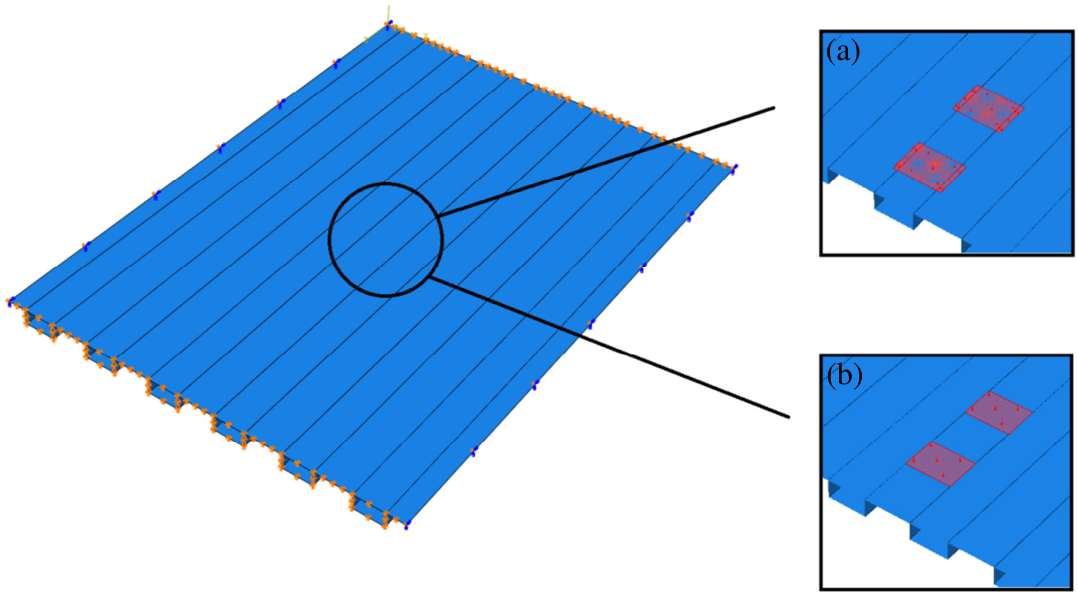


Figure 5.2 *The two different load cases studied in the FE-analysis.*

The stress levels obtained from the beam theory analysis originates from pure bending. However, the wheel prints give rise to local stresses; hence this is evaluated with FEM. When performing a FE-analysis, there are possible sources for error, for example the mesh might be too coarse in order to obtain reliable results. Thus, a mesh convergence study is carried out in order to confirm the accuracy of the results. Furthermore, the global mesh sizing for shell elements should be kept above a minimum of 5 times the biggest thickness (Hogström, 2010). This is due to a limited number of integration points in the thickness of the element, if the distance between these points is too large, errors in the solution will occur.

Figure 5.3 shows the convergence study conducted. The red line represents 5 times the biggest thickness for the model. It is evident that the solution converges for mesh sizes between 0.0375 and 0.0285. At mesh sizes lower than 5 times the thickness there is an increasing numerical error in the solution. For the analysis a mesh size of 0.0325 is chosen.

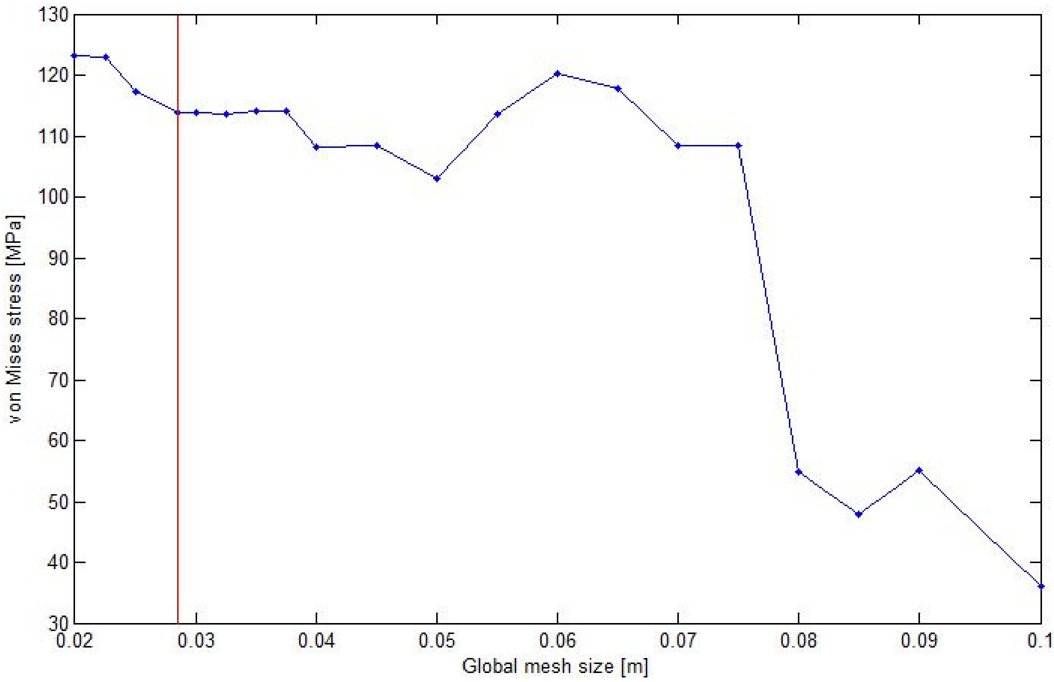


Figure 5.3 *Convergence study for the Abaqus model.*

The above models are assumed to be sufficient to achieve a reliable result for the weight reduction iteration, since the stresses decrease and approach zero over the surface. However, the middle third of the car deck is modelled in order to verify this assumption and to evaluate the how the different point loads affects the global structure. Figure 5.4 shows the structure and the wheel prints from above. The boundary condition is fixed at all edges. This load case is referred to as load case 3.

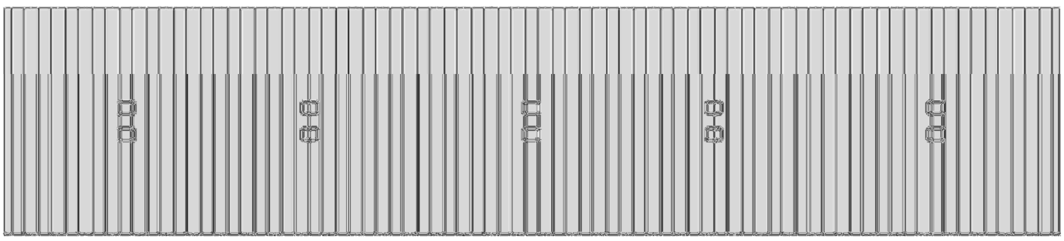


Figure 5.4 *One third of the car deck modelled in Abaqus.*

The distance between wheel prints are defined using a normal car as a reference, while the load is as presented in Section 1.4. It is assumed that the cars will be parked as close as possible to each other, which means that the distance between two tyres is set to 0.2 m. Figure 5.5 shows the dimensions of the car used.

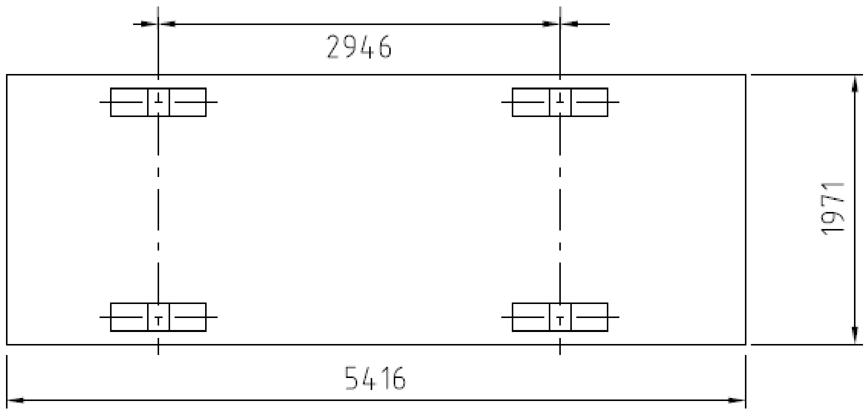


Figure 5.5 *Dimensions of the car used for determining the distance between tyre prints in load case 3.*

5.2 Steel Structure

For the analysis of the steel structure, ANSYS workbench is used. For this analysis the system is considered to have supports as described in Section 3.3. This result in lower stresses in the corners compared to fixed supports. Furthermore, the dimensioning parameter for the supporting steel structure is the deflection, not the stresses. Hence, the choice of elements and modelling system is based on this assumption. Line bodies with assigned cross sections are used for modelling the structure since it is quick and easy to make changes. The element type used in this evaluation is BEAM188 elements, where each node has 6 degrees of freedom (x- y- z-translation and x- y- z-rotation). Furthermore, warping is unrestrained.

The load used for the analysis and optimization is the UDL as presented in Section 1.4, converted to a line load. The self-weight is not a part of this analysis due to pre-tension of the steel structure.

The elements of a FE-model highly affect the accuracy of the results. Because of this a convergence study is carried out where the line segment division of the structure is changed to see if the results differ too much. In this case the solver uses BEAM 188 elements which use cubic interpolation in order to solve the deformation. The maximum deformation occurs in the middle of the longitudinal beam as can be seen in Figure 6.8; hence, both the longitudinal beam and the frame were refined for this convergence study.

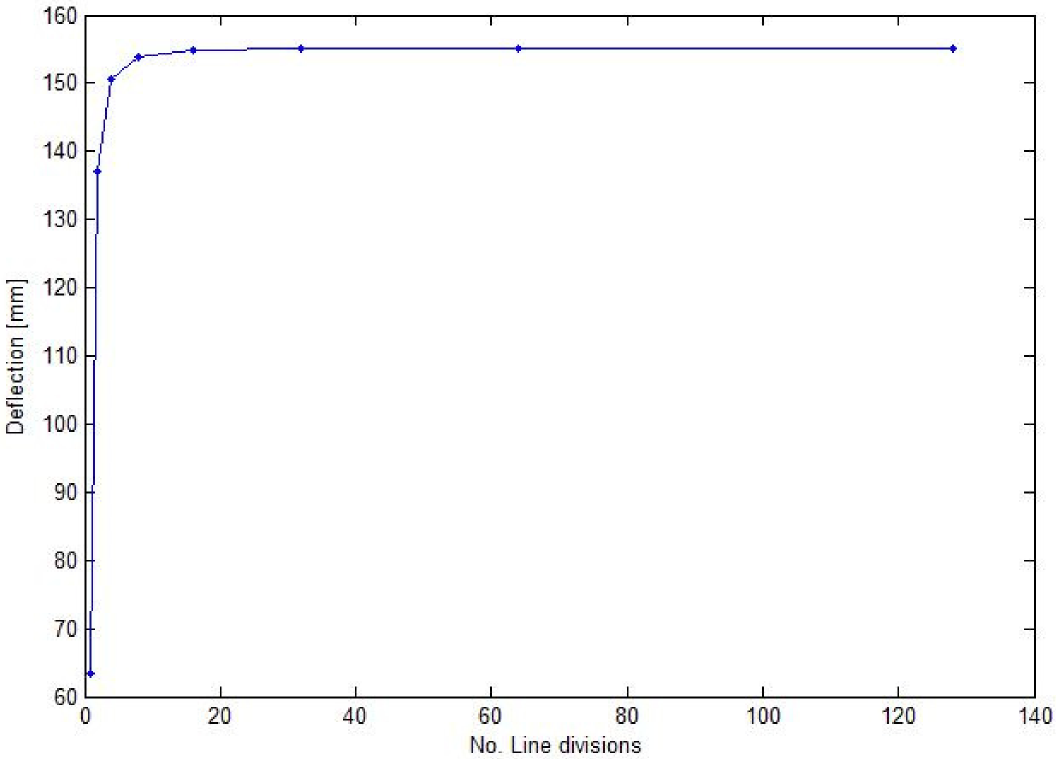


Figure 5.6 *The result from the convergence study, showing deflection (mm) for different number of line divisions*

Figure 5.6 shows that the results converge after 16 elements per line. The solution converges to the fifth decimal place at a line division of 128 elements. Hence, in order to reduce computation time, the simulations are run with 16 line divisions.

5.3 Summary of the structural analysis

For the analysis of the stiffened top plate 3 main load cases are used

- Load Case 1 (LC1): The load from two tyres (22.095 kN) situated in the middle between supports in the transverse direction and on top of the stiffener in the longitudinal direction. See Figure 5.2a.
- Load Case 2 (LC2): The load from two tyres (22.095 kN) situated in the middle between supports in the transverse direction and between stiffeners in the longitudinal direction. See Figure 5.2b.
- Load Case 3 (LC3): One third of the panel loaded with cars in the worst possible configuration which corresponds to a total load of 110.475 kN. See Figure 5.4.

In addition to the above load cases the weld is evaluated by having two tyres (22.095 kN) situated in the middle between supports in the transverse direction and on top of the weld in the longitudinal direction. Furthermore, the structure is evaluated by turning one tyre (11.047 kN) 90 degrees with the same configuration as LC1 and LC2.

The deflection of the beam system is evaluated with a uniform load of 250 kg/m^2 without the addition of a dynamic factor. This corresponds to 516.475 kN for the centre car deck and 360.545 kN for the side car deck. Furthermore, the stresses in the side car deck are evaluated with the same load with a dynamic factor of 1.5 added. This corresponds to a load of 540.817 kN.

6 Results

This section presents the findings for each evaluation performed. The final design of the aluminium profile is presented in Section 6.1 followed by results from the structural analysis. The beam system evaluation is shown in Section 6.2. Section 6.3 presents the analysis in GeniE that confirms the strength of the car deck panel. The section is concluded with a short summary.

6.1 Aluminium profile

Several iterations were made in the aluminium design before a solution satisfying the design requirements were found. The dimensions of the final design are presented in Figure 6.1; the free length of the profile is 3.2 m. It can be seen that the top plate has a quite large thickness; this is a result of the high local stress concentrations in the top plate due to the tyre load. The only method to reduce these stress concentrations is to increase the number of stiffeners or by increasing the top plate thickness. The reason for this limitation is that material only can be added in one direction when using extrusion as a manufacturing process. Figure 6.2 shows the aluminium profile with lashing holes. The bending stress in the profile is affected by the web height and the position of the neutral axis, a combination of the two were found where the lower flange thickness could be kept low, reducing the total weight of the profile. The final design has a weight of 20.673 kg/m^2. While the proposed design has a uniform thickness in the top plate, the mass could be reduced slightly by removing material at low stress locations; this would require a new FE-model without shell elements. This is due to the high uncertainty in the results in areas where sharp edges, due to transitions between different thicknesses, are introduced.

[Figure deleted due to confidentiality]

Figure 6.1 *The dimensions of the proposed aluminium profile in mm where the free length is 3.2 m*

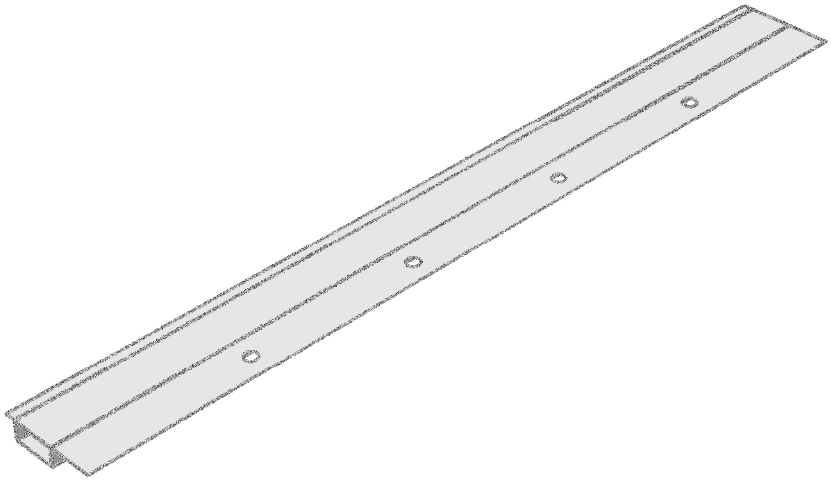


Figure 6.2 *Aluminium profile with lashing holes*

6.1.1 Stresses in the aluminium profiles

The figures in this section show a selection of results from the structural analysis performed in Abaqus. The complete result, showing that all stress requirements are fulfilled can be found in appendix A. Figure 6.3 and Figure 6.4 show the Von Mises stresses in the profile for load case 2. If compared to the requirements presented in Section 1.4 it can be seen that the stresses are at the limit in the top plate while they are slightly below the limit for the stiffener. Furthermore, the stresses approaches zero already two profile lengths from the load. This shows that the assumption that the existing lashing solution can be used, without too much interaction with the local loads, is true.

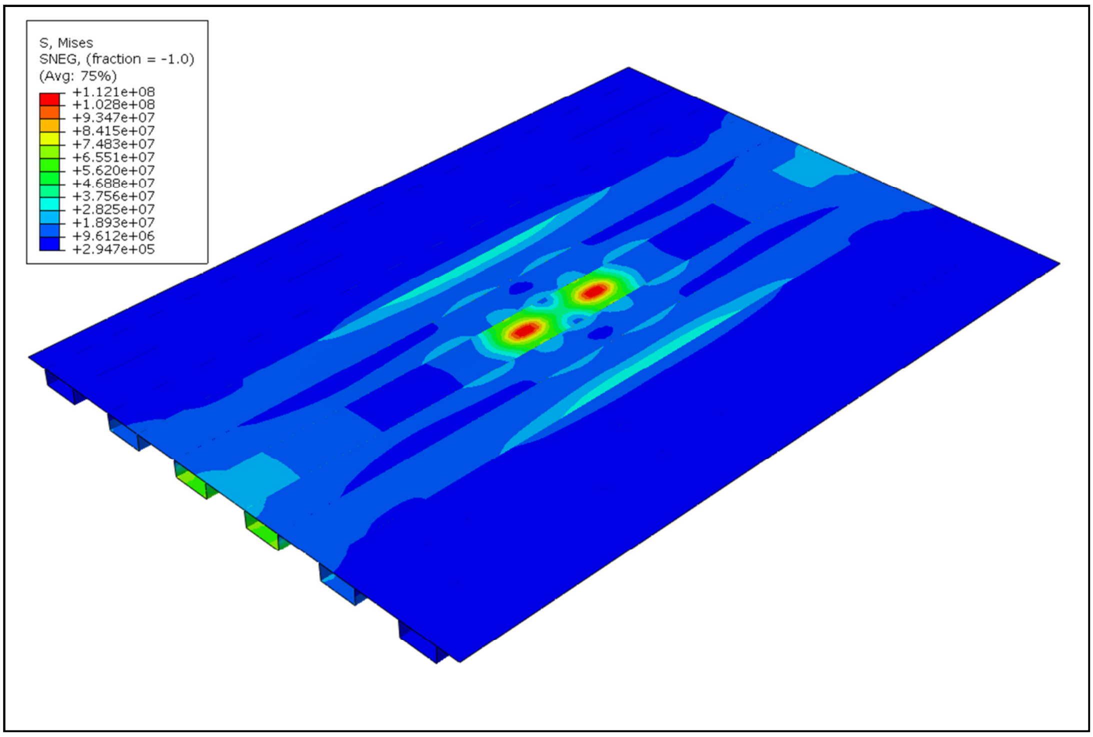


Figure 6.3 *The Von Mises stresses in the stiffened top plate due to the tyre load.*

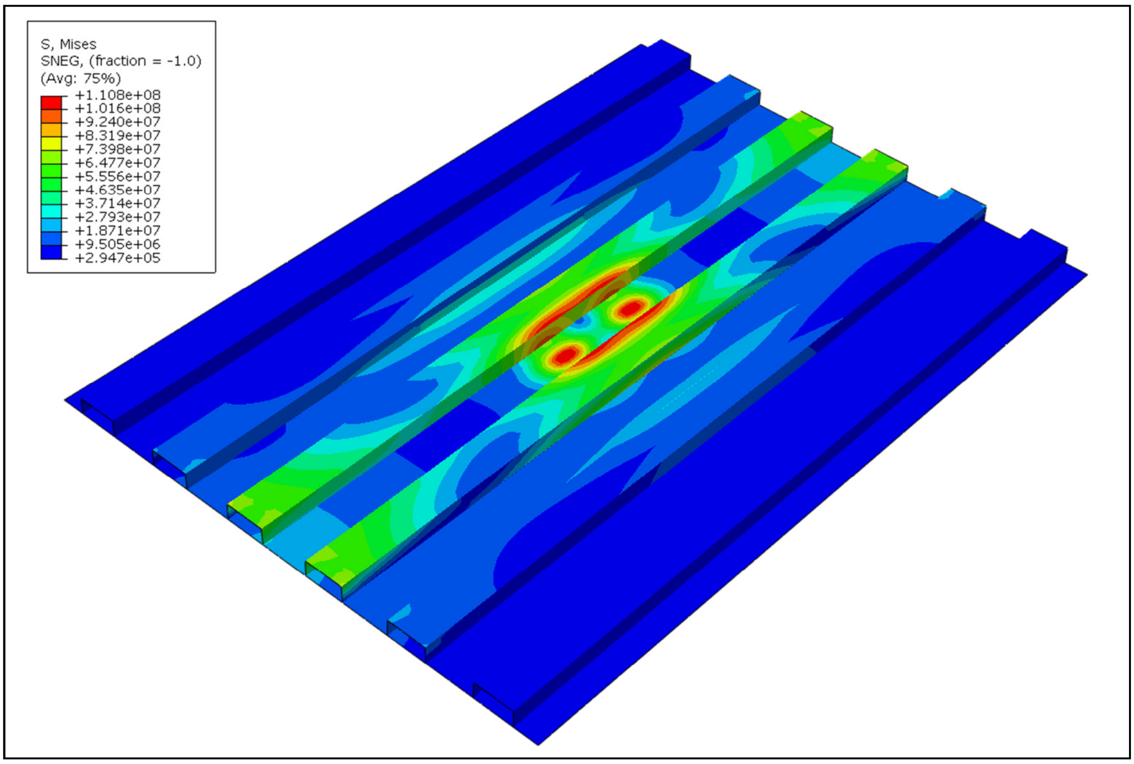


Figure 6.4 *The Von Mises stresses in the stiffeners due to the tyre load.*

Figure 6.5 shows the stress response for the worst load case the welded material will be subjected to i.e. when a tyre is placed directly on the weld. The red line represents the approximate location of the weld and the stress magnitude is at the allowed limit.

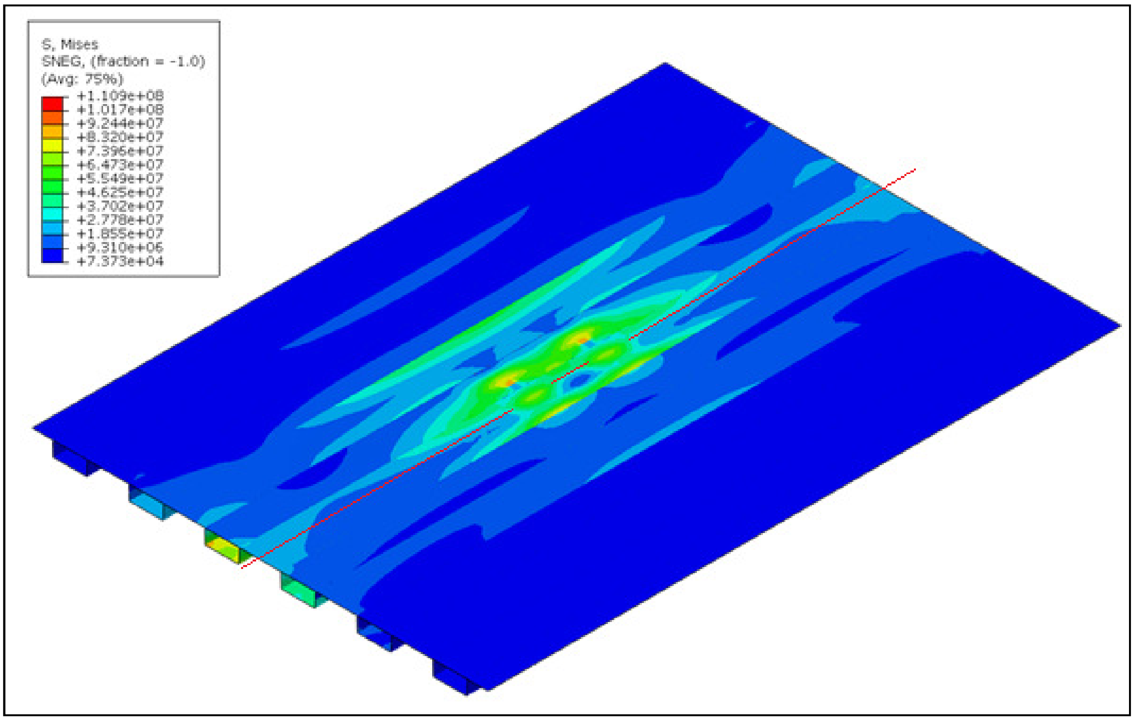


Figure 6.5 *The highest stresses that will occur at the weld due to the tyre load.*

The stresses normal to the cross section for load case 2 are shown in Figure 6.6 and Figure 6.7. This is the load case that yields the highest normal stresses. Here, the normal stress due to bending is dimensioning while for load case 1 the normal stress due to local loads are dimensioning. This is due to the increased thickness in the top plate above the stiffeners. The stresses in the boundaries are slightly lower than the maximum permissible stresses.

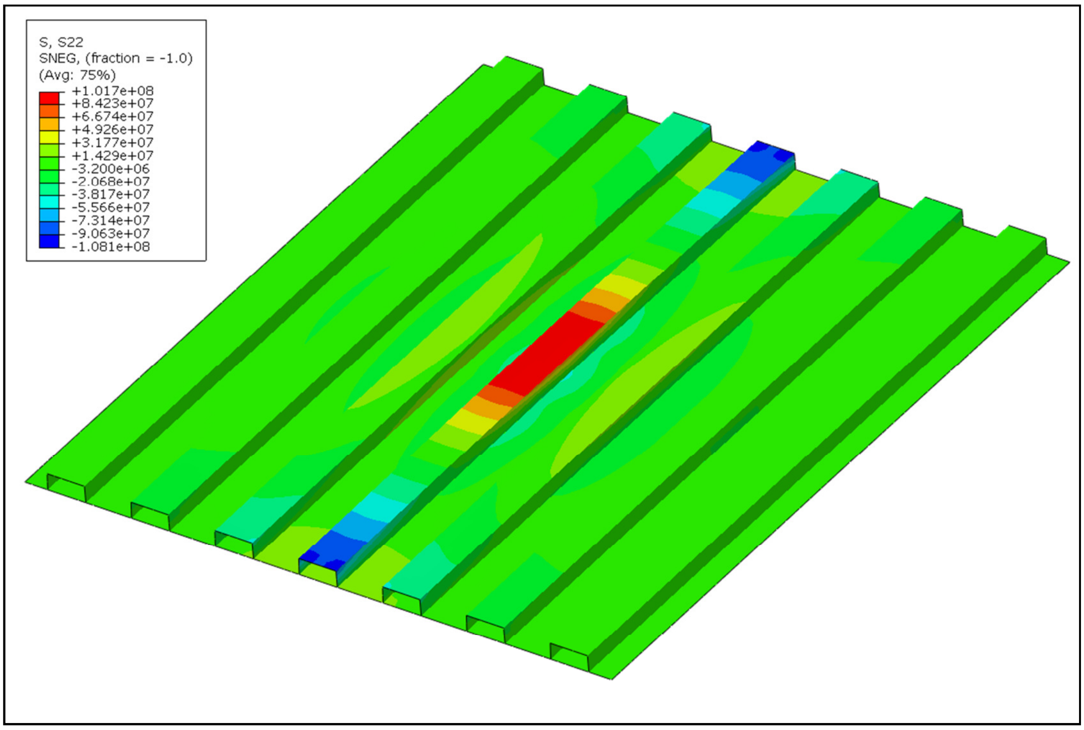


Figure 6.6 *The highest normal stress that occur in the panel.*

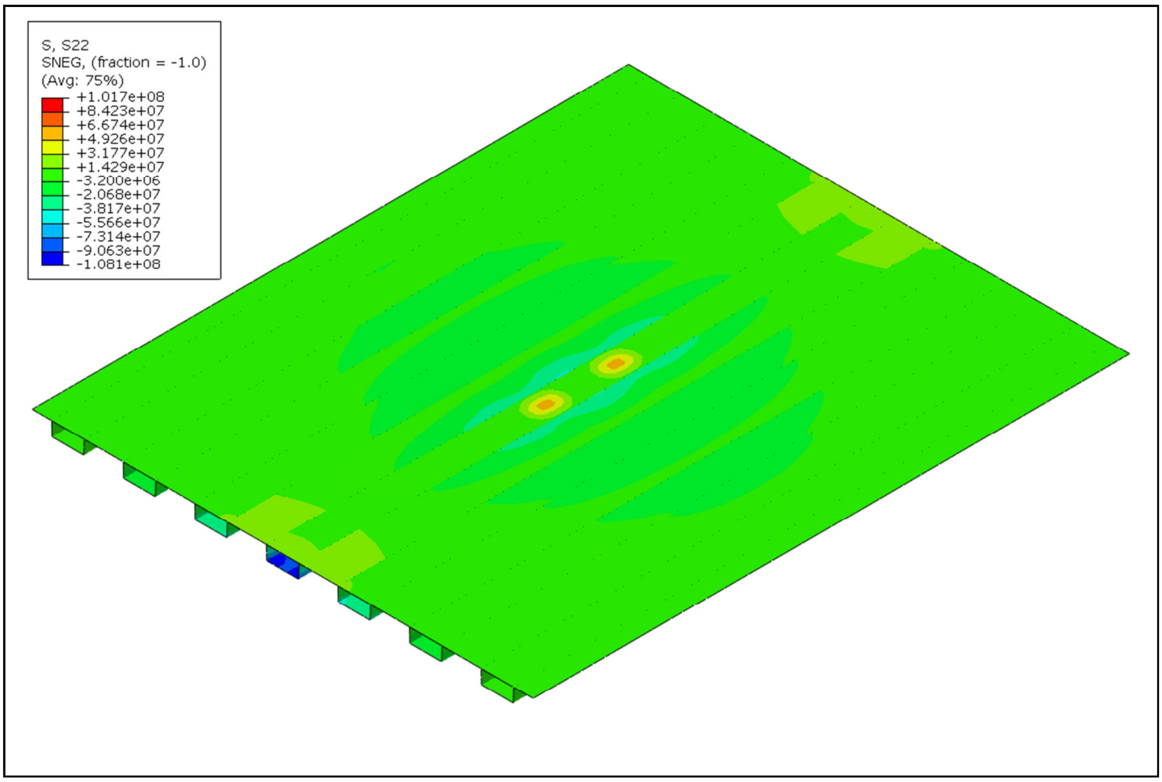


Figure 6.7 *The normal stress in the stiffened top plate for load case 2.*

Contour plots for all load cases are presented in Appendix A. It includes load case 3 as well as a load case where the car is turned 90 degrees. This is to ensure that the structure can withstand the load when cars are moved across the car deck during loading/offloading. In Appendix A, Figure 16, it can be seen that the stress approaches zero already 1-2 profile widths from the tyres.

6.2 Beam system

This section presents the final design of the supporting beam system for the two different dimensions evaluated. As previously mentioned, the deflection is dimensioning. Hence, only the final dimensions of the structural members and the corresponding deflection of the structure will be presented here. The stresses in the structure is however evaluated and presented in Section 6.3.

6.2.1 Centre car deck (14.64x14.37)

The solution found for the centre car deck panel is presented in Figure 6.8. The beam system has a steel frame and four longitudinal steel beams. All the beams in the solution are I-beams. The only solution found where the web height + deflection did not exceed the allowed building depth was by utilizing longitudinal beams, attached in accordance with Figure 4.5a, with a 340 mm web height. Therefore, the longitudinal stiffeners are attached to the upper edge of the frame to avoid exceeding the allowable building depth, which results in additional bolting.

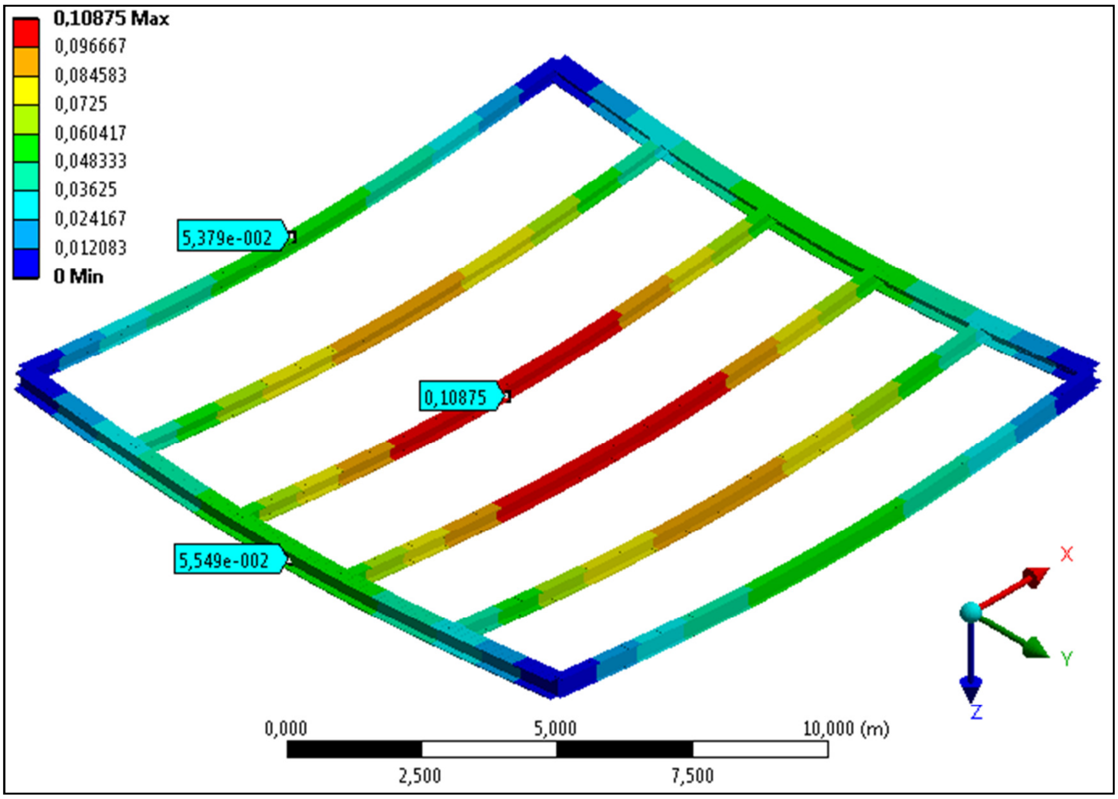


Figure 6.8 *Vertical deflection of the centre car deck panel.*

The dimensions of all the structural members as well as the deflection can be seen in Table 6.1. The corresponding weight for the entire structure as well as a comparison to the reference solution is shown in Table 6.2. The attachments are bolts, brackets, and welds and their approximated weight is 3 % of the total weight.

Table 6.1 *Dimensions and deflections of the beams in the centre car deck panel.*

	Transverse frame	**Longitudinal frame**	**Longitudinal stiffeners**
Bottom flange width (mm)	420	250	250
Bottom flange thickness (mm)	35	15	20
Upper flange width (mm)	350	150	250
Upper flange thickness (mm)	25	6	30
Web height (mm)	400	400	340
Web thickness (mm)	6	6	6
Deflection (mm)	55.5	53.8	108.8
Web height + deflection (design depth) (mm)	455.5	453.8	448.8

Table 6.2 *Weight of the reference solution as well as the alternative design of the centre car deck panel.*

	Reference	**Alternative design**
Aluminium weight (kg/m^2)	-	22.64
Steel weight (kg/m^2)	93.16	63.6
Attachments (kg/m^2)	2,79	2,59
Total weight (kg/m^2)	95.95	88,83
Weight reduction (%)	-	7.42

While these results are from the response surface screening and an optimization would reduce the total weight of the beam system slightly, the weight is too far from the desired reduction which is 25%, see Table 1.3. Consequently, the design is considered not beneficial and not optimized further.

6.2.2 Side Car Deck (10.22x14.37 m)

The solution that yielded the lowest weight for the beam system for the side car deck has a steel frame with two steel longitudinal stiffeners (y-direction in Figure 6.9). All the beams in the solution are I-beams. Due to the magnitude of the span between supports; there was no solution that made it possible to place the aluminium panel on the top of the longitudinal beams. The reason for this is that the stiffeners need to be attached to the upper side of the frame in order to comply with the building height and deflection requirements presented in Section 1.4. Table 6.3 presents the dimensions of the final design while Table 6.4 presents the comparison between the proposed design and the reference solution. As in the previous sub-section, the attachments are bolts, brackets and welds and corresponds to an approximated 3% of the weight of the steel structure.

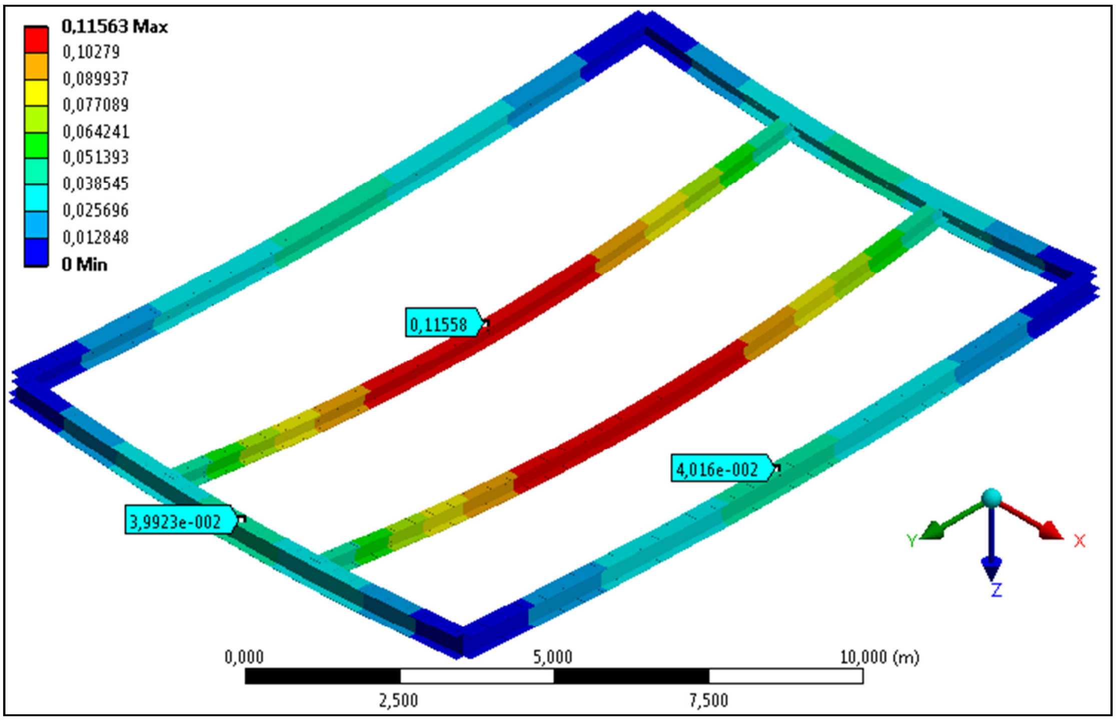


Figure 6.9 *Vertical deflection of the side car deck panel.*

Table 6.3 *Dimensions and deflections of the beams of the side car deck panel.*

	Transverse frame	**Longitudinal frame**	**Longitudinal stiffeners**
Bottom flange width (mm)	309	308	220
Bottom flange thickness (mm)	11	12	20
Upper flange width (mm)	308	315	309
Upper flange thickness (mm)	9	8	20
Web height (mm)	415	415	340
Web thickness (mm)	6	6	6
Deflection (mm)	39.5	40.1	115.0
Web height + deflection (design depth) (mm)	454.5	455.1	455.0

Table 6.4 *Weight for the reference and alternative design.*

	Reference	Alternative Design
Aluminium weight (kg/m^2)	-	20.7
Steel weight (kg/m^2)	87.28	41.54
Attachments (kg/m^2)	2.62	1.87
Total weight (kg/m^2)	89.9	64.11
Weight reduction (kg/m^2)	-	28.69 %

A weight-reduction of 28.69 % is satisfying and will contribute to a significant reduction of fuel consumption, but is only beneficial is the investment cost can be repaid in *[Deleted due to confidentiality]*. See Section 7 for more details regarding the cost analysis.

6.3 Verification analysis of assembled structure

In Section 6.2 it was concluded that the aluminium concept is not beneficial for the centre car deck panel. Consequently, only the side car deck is modelled and evaluated in GeniE (DNV Software, 2014). The result from the FE-analysis can be found in Appendix B, it shows that the design fulfils the stress requirements set up by DNV. Furthermore, the aluminium contributes to the global strength of the car deck panel which results in a lower overall deflection of the beam system. Figure 6.10 presents the deflection when the panel is loaded with 375 kg/m^2. If the deflection in Figure 6.10 is compared to Table 6.3, it is evident that the aluminium reduces the deflection significantly. Without the increased stiffness from the aluminium the deflection is 115 mm while it is 113 mm with aluminium and the dynamic factor added.

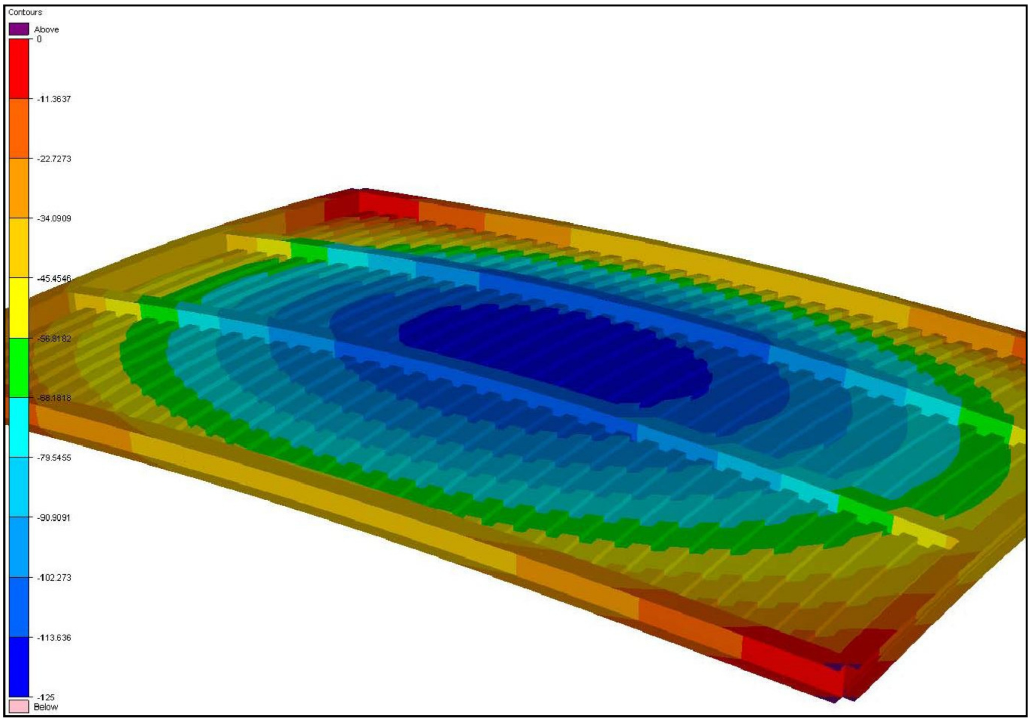


Figure 6.10 *The deflection of the car deck panel.*

6.4 Summary of the structural analysis

A summary of the FE-analysis for the side car deck design is shown in Table 6.5. The maximum stresses from the simulations are shown for each part of the structure as well as the percentage of the maximum allowable stress.

Table 6.5 *Summary of the FE-analysis of the side car deck. Green 0-60%, yellow 60-90% and orange 90-100% of maximum permissible stresses.*

Maximum stress Percentage of allowed	σ_{vM} (MPa)	σ_x (MPa)	τ_{xz} (MPa)
	112.1	-91.85	25.83
deck plate	100%	94%	47%
	79	-71.5	22.1
weld	96.30%	100%	55%
	110.8	-108.1	21.6
stiffener	90%	98.4%	39.30%
	222	221.6	68.15
beam system	88.9%	98%	54.50%

As can be expected for closed cross section as well as I-beams subjected to bending, the shear stresses are significantly lower than the allowable. For the stiffened top plate the highest stress occurs in the top plate due to the local tyre loads In order to reduce this stress the distance between stiffeners need to be lowered or the plate thickness increased. Both methods would result in an increased total weight. The most critical normal stress that arises in the aluminium stiffeners is located in the aluminium-steel boundary for load case 1. There is an uncertainty in this area and further investigation of the boundary condition should be conducted to confirm that the stress concentration does not exceed the allowable.

In the beam system the highest stress concentration of both normal and von Mises stress will occur in the lower flange of the loadbearing frame. This is expected since all the loads from the cargo will be translated through the frame to the pillars.

It should be noted that the load case identified as the most critical is where two cars are parked with the wheels exactly in the middle between supports. This is not a standard cargo configuration and will occur very seldom, if anytime. Hence, the normal stress in the aluminium structure will be lower most of the time. However, regardless of the distance between tyres and supports, the stress concentration in the top plate will always be the same since this is a local phenomenon.

7 Cost Analysis

[Section deleted due to confidentiality]

7.1 Car deck size analysis

The weight-reduction potential of using aluminium as a substitute to steel in the stiffened top plate is higher when smaller car deck panels are considered. Due to this, an analysis is performed where the width of the panel is reduced in steps in order to show how the weight is affected by the lower free length between supports. Figure 7.1 shows how much it is possible to reduce the weight by the use of aluminium, depending on the width of the car deck panel. For this analysis the aluminium profile presented in Section 6.1 is used as it would be too time consuming to design a specific profile for each case. Figure 7.1 also shows the reference solution weight compared to the design with aluminium profiles with two respectively four longitudinal stiffeners.

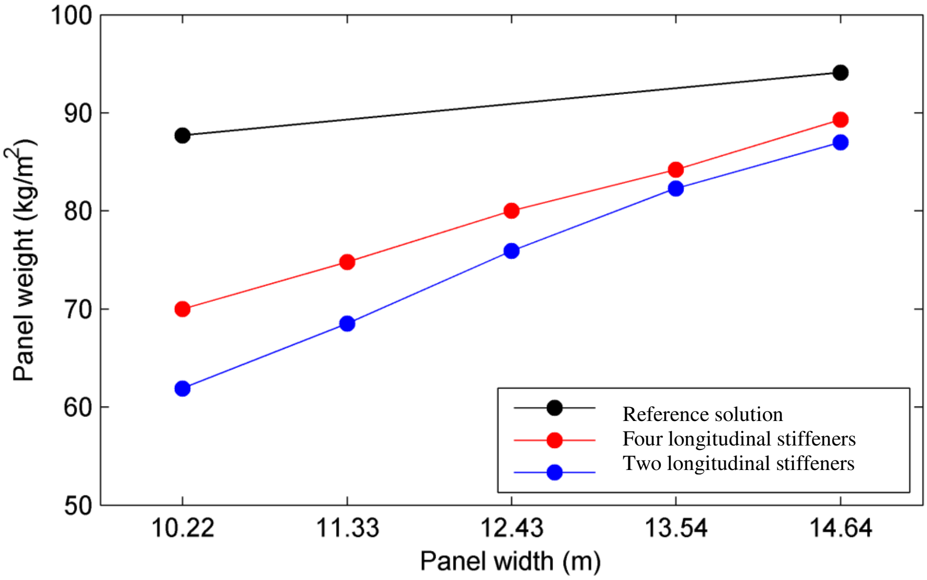


Figure 7.1 *Weight per area for different sizes of car decks.*

The results in Figure 7.1 are based on response surface screenings, hence it has a potential of reaching slightly lower weight than the diagram indicates. All solutions meet the deflection and building height requirements. Figure 7.1 indicates that the weight-reducing potential of aluminium is greater at lower dimensions of car deck panels. This verifies the statement that a 6 mm thick top plate of steel is excessive.

7.2 Payback time

[Section deleted due to confidentiality]

7.3 Cost allocation

The reason the new design of the car deck panel is more expensive than the conventional design, even if the weight is lower, is that aluminium have a higher raw material price and also requires FSW. As can be seen in Figure 7.2 below, the raw material of aluminium and FSW stands for 55.5% of the total cost. As can be expected, the payback time is varying drastically depending on the aluminium price since it stands for such big portion of the total price.

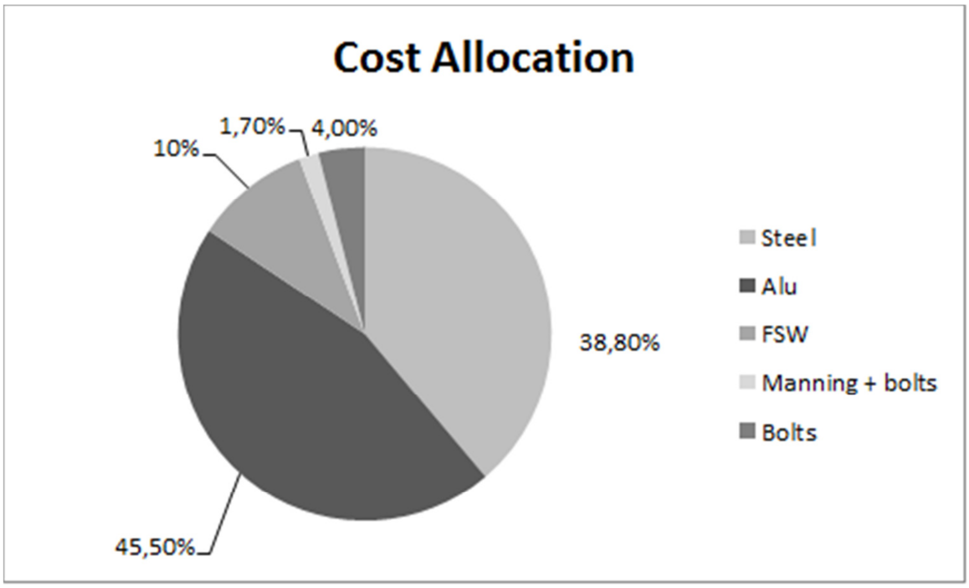


Figure 7.2 *Cost allocation of the side car deck panel*

The manning cost and cost for bolting is dependent of the beam lengths and the number of stiffeners, which makes this cost higher for larger car deck panels. However, to reduce the biggest portion of the total cost, the aluminium, it is necessary to add more steel, which increases the weight. If an alternative to FSW could be found, that would be a possible way to reduce the investment cost and therefore also payback time.

Unfortunately the desired payback time could not be reached. Even if this solution is not economically justified to apply today, it can be beneficial to use in a near future. The investment cost is reduces if the raw material price of aluminium decreases, fuel costs increases or less aluminium is used. Figure 7.3-7.5 shows how the payback time depends on these variables. If the payback time is set to be *[Deleted due to confidentiality]*, then one of the following claims must be true:

- *[Deleted due to confidentiality]*
- *[Deleted due to confidentiality]*
- *[Deleted due to confidentiality]*

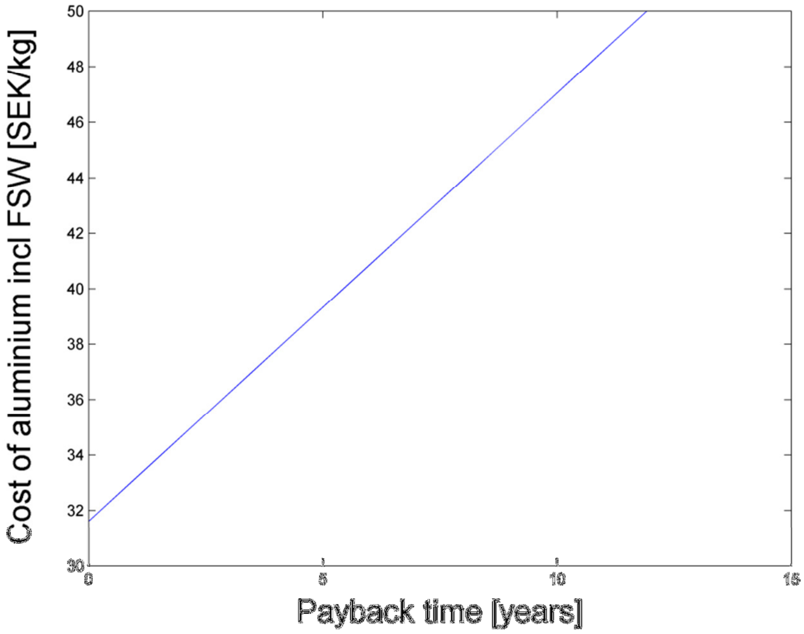


Figure 7.3 *Payback time depending on the change of the aluminium price.*

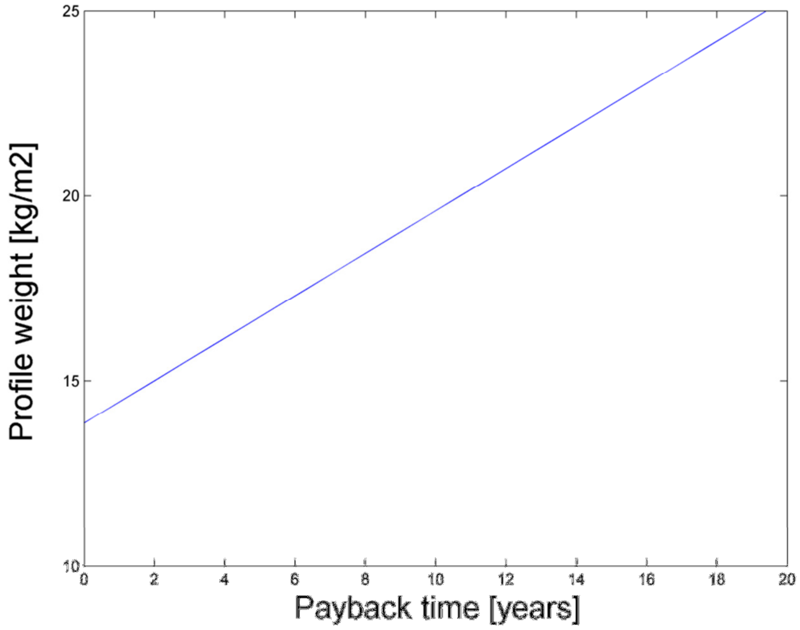


Figure 7.4 *Payback time depending on the change of the aluminium weight.*

[Figure deleted due to confidentiality]

Figure 7.5 *Payback time depending on the change of the fuel oil price.*

8 Discussion

This study shows that the weight reduction potential is highly dependent on the outer dimensions of the car deck panel. If the aluminium concept is to be used on car deck panels with a free length between attachment points surpassing 10-12 meter, the building depth requirement need to be compromised in order to obtain a section stiff enough for possible cost saving. This is due to the web height having a cubic contribution to the moment of inertia as was presented in Equation 4.1. The reason for the loss of stiffness for this concept is also a combination of a lower NA, loss of effective flange and lower young's modulus due to the replacement of the steel top plate. The restriction in building depth is the biggest obstacle for a lower weight for these kinds of structures. Furthermore, if the free length between attachment points can be reduced, for example by utilizing more pillars or by introducing suspension cables, the suggested design can be made more weight- and cost-efficient since the initial assumptions that panels can be placed on-top of the supporting structure would be true, resulting in reduced steel weight, less bolting needed, and the elimination of supporting brackets.

The dynamic factor used for this study is conservative and represents the highest that can occur for the ship. If an adjustable dynamic factor was to be used, depending on the position of the car deck being designed, some deck levels could have a slightly lower aluminium weight. However, the deflection of the steel structure is evaluated without the addition of a dynamic factor, hence the weight saving would be low since the steel structure would be unchanged. Although this would require individually designed decks for each level, the payoff could be high since even a low reduction of aluminium weight would result in a significant reduction in payback time.

In this project the weight-reduction task is divided into two separate parts where the stiffened top plate is weight-reduced first. The design of the aluminium plate is used as a base for the design of the beam system. Hence, the question could be raised whether the results would differ if the beam system would be optimized first. Would this result in a lower or higher free length between stiffeners? While a lower free length would reduce the height of the profiles it has been shown that the local stress concentrations in the top plate is dimensioning and that the stiffeners only account for approximately one third of the profile weight. Furthermore, even though it is possible to optimize the two structures together, creating the FE-model and running the simulations would be time consuming as the number of input variables increase significantly.

As mentioned in the previous paragraph, the beam system is evaluated separately. However, by utilizing bolts as a steel-aluminium connection, the aluminium will contribute to the effective flange of the steel beams. Consequently, the deflection will be lower than the results from the beam system simulation suggests. This can be seen in Figure 6.10. Hence, while fulfilling all the requirements, some of the material in the beam system can be removed.

The FE-analysis of the aluminium profile is conducted with fixed boundary conditions based on the assumptions made in Section 4.3. While it is believed that the assumption is correct and yield reliable results, it would be more conservative to use one side fixed and one side simply supported. However, since this study is about weight-reduction and there are safety factors such as the material factor and the dynamic factor already in place, the boundary conditions are not evaluated further. If it is proven that the assumption is non-conservative, the bending moment would be slightly higher and consequently the bending stresses would increase as well. However, if the permissible stresses for the stiffeners presented in Section 1.4 are compared to the actual stresses in the structure (Appendix A) it can be seen that there is a small margin for increasing the stresses in the area that would be affected by such a change.

9 Conclusions

This thesis presents the design and evaluation of car deck panels of two different dimensions where the steel top plate is substituted with extruded aluminium profiles welded together. Several concepts are discussed and evaluated in a parametric study. The structural strength of the proposed design was evaluated using FE-analysis. Furthermore, the supporting beam system was designed based on the design of the aluminium profiles and optimized using goal driven optimization. Conclusively a cost analysis was carried out.

The major findings of the study are summarized in the points below:

- The proposed design for the centre car deck panel (14.37x14.64 m) yields a weight reduction of approximately 7.5%. This result in a too long payback time compared with the desired *[Deleted due to confidentiality]*. The concept with extruded aluminium profiles is concluded to be unfavourable for panels of this size.
- For the side car deck panel (14.37x10.22 m), a weight reduction of 28.7 % is achieved by using the proposed design. This is within the aim of the study and considered acceptable. However, the payback time is too high (2.4 times the desired) due to the increased material and assembly cost for aluminium.
- If an alternative aluminium-aluminium connection method can be developed, the payback time may be reduced significantly since FSW corresponds to 10% of the production cost.
- It is concluded that the design may become more economically feasible in the future due to increased fuel oil prices, the use of higher quality fuel due to pollution restrictions as well as possible reduction in raw material prices.

From the parametric analysis of aluminium and the FE-analysis it can be seen that local loads are the dimensioning factor for the top plate. The stiffeners only correspond to approximately a third of the panel weight. Hence, the only method to further reduce the weight of the aluminium is to reduce the free length between supports which means adding more steel weight to the structure.

Based on the two first points above and the car deck size analysis described in Section 7.1 it is concluded that the conventional design is more cost-efficient, than the suggested concept, for car deck panels with bigger length between the attachment points. This is due to the increased stiffness gained from the top plate when utilizing the conventional design. The difference in weight per square meter between the two reference car deck panels is 5.9% while the difference when utilizing the aluminium concept is 27.8 %. This shows that the weight-reduction potential for the aluminium concept is higher for smaller panel sizes.

10 Future Work

Since the structure is mounted on a ship, the dynamic factor comes from the waves which mean the loads are cyclic. This thesis has not taken cyclic loads into account and a study concerning fatigue design should be carried out in the future. In that study, also vibrations should be investigated since lower weight cause higher vibrations (Ulfvarson, 2004).

From an economic point of view, the investment cost that is the major problem with aluminium structures, and a more economic oriented study could be carried out in the future regarding manufacturing and attachment methods. The aluminium raw material is approximately 40% of the panel cost, and the FSW is 10%. Finding a substitute to FSW and/or keeping down the aluminium weight even more would reduce the payback time drastically. An alternative method that has been discussed is mechanical fastening; a study of the structural strength of said method could be carried out. If it is shown that it can be used, the only cost associated with the method is a new extrusion tool. Consequently the total cost could be reduced with 10%.

A change in weight affects the stability of the ship. A weight-reduction above the centre of gravity (CoG) of the ship causes higher stability, and below a lower stability; this is due to the change of distance between the CoG and metacentre. However, a more detailed study should investigate more exactly how much the stability is change, since this may be interesting for designing other parts of the ship where the stability may be a limiting factor.

Another area that needs further study is the bolting holes in the flanges of the steel I-beams. While it is believed that the stress concentrations will not be a problem due to the washers on both sides of the flange, this should be studied further.

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APPENDIX A: Abaqus contour plots

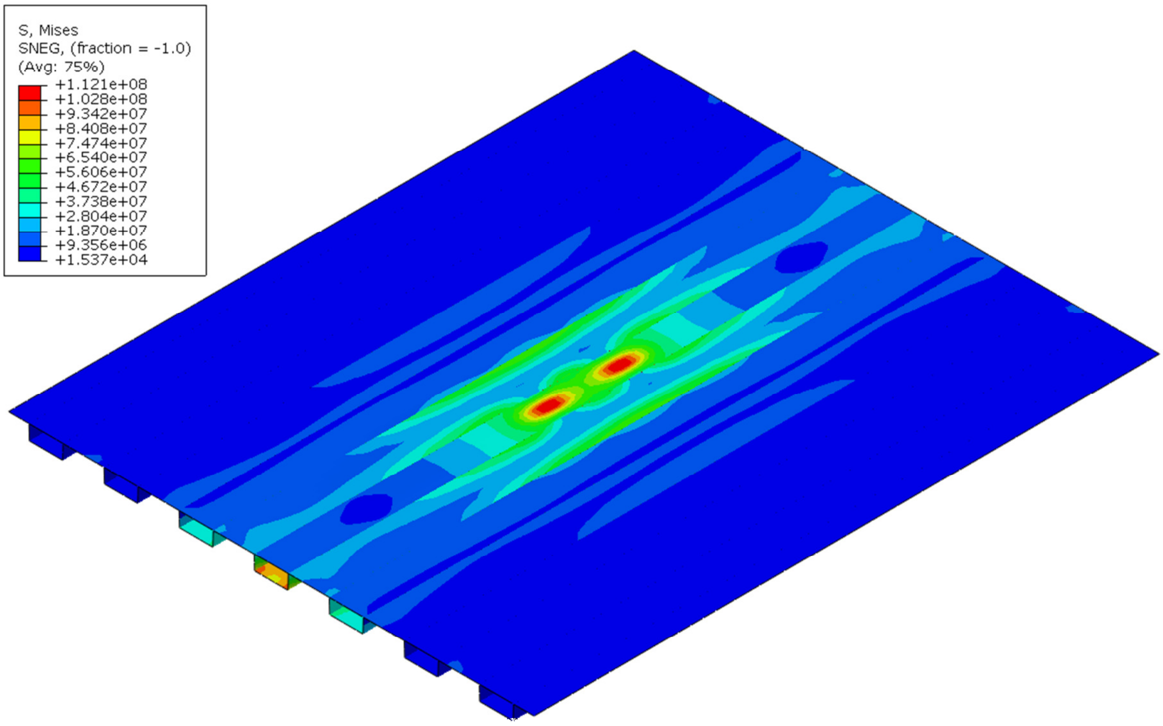

Figure 1 *Von Mises stress, LC1.*

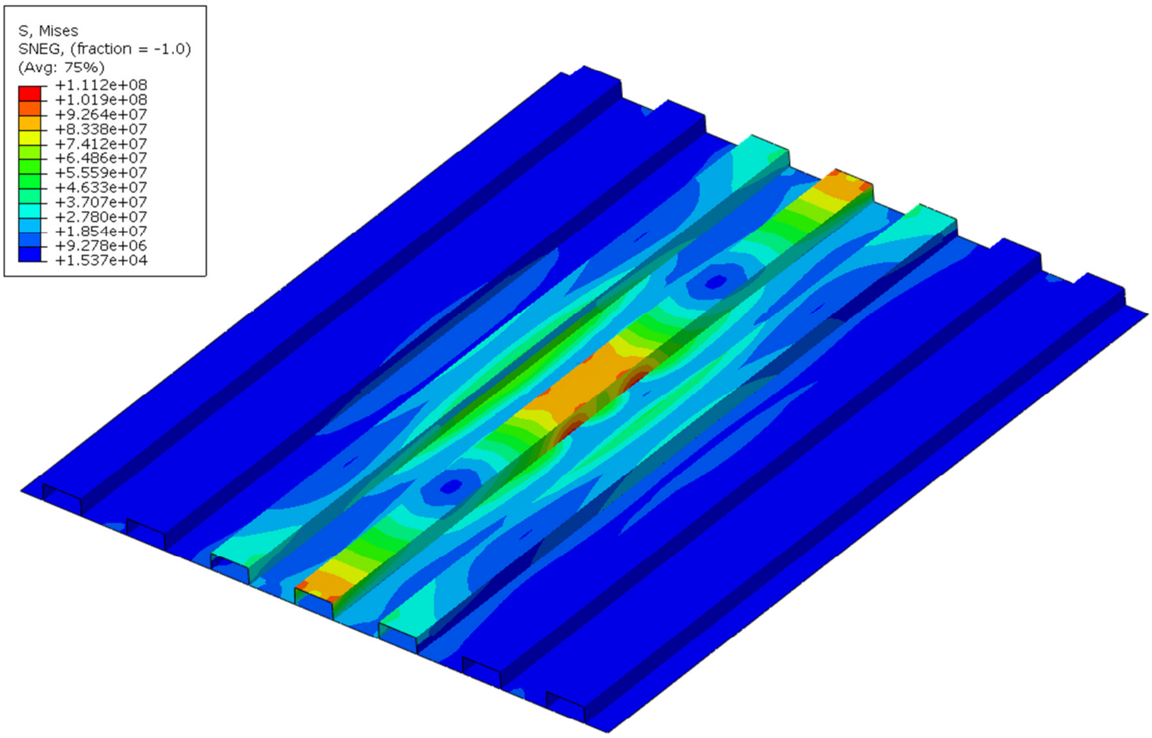

Figure 2 *Von Mises stress, LC1*

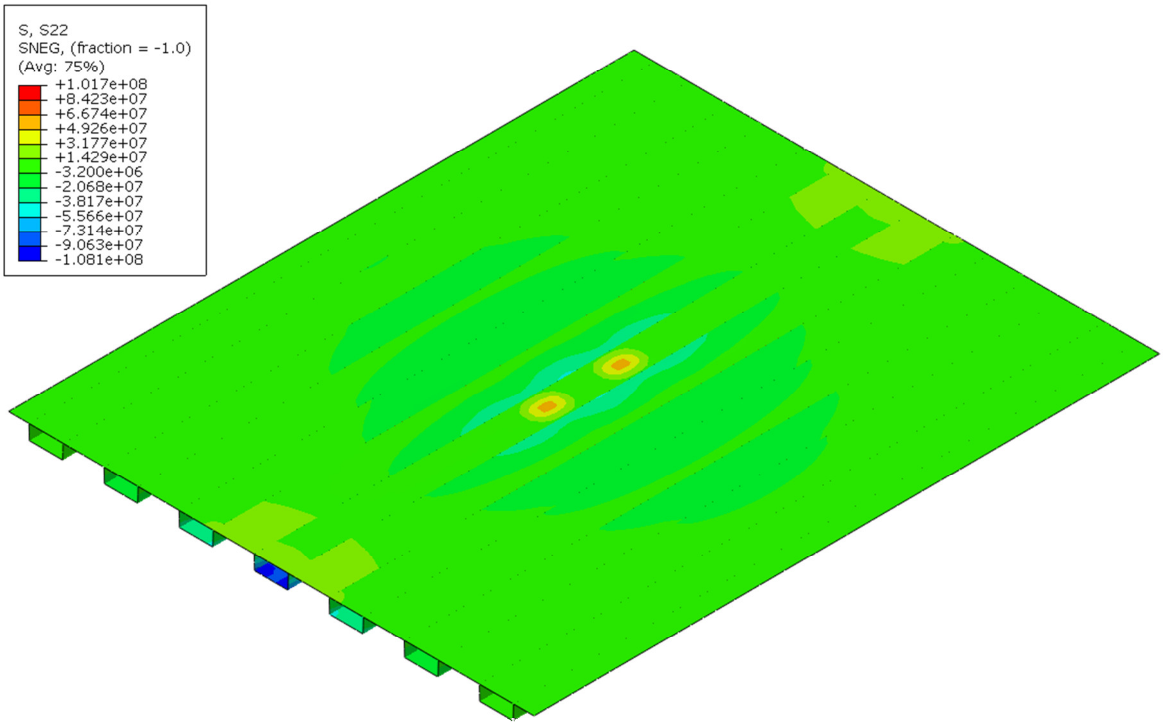

Figure 3 *Normal stress, LC1.*

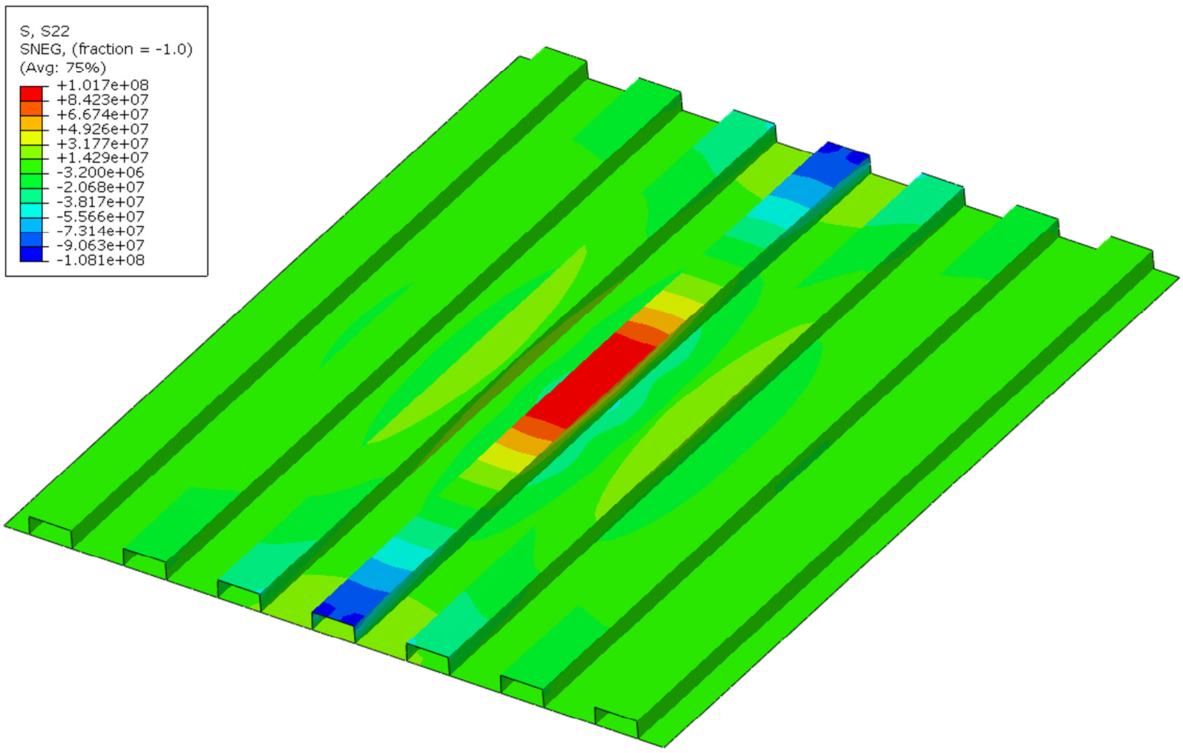

Figure 4 *Normal stress, LC1.*

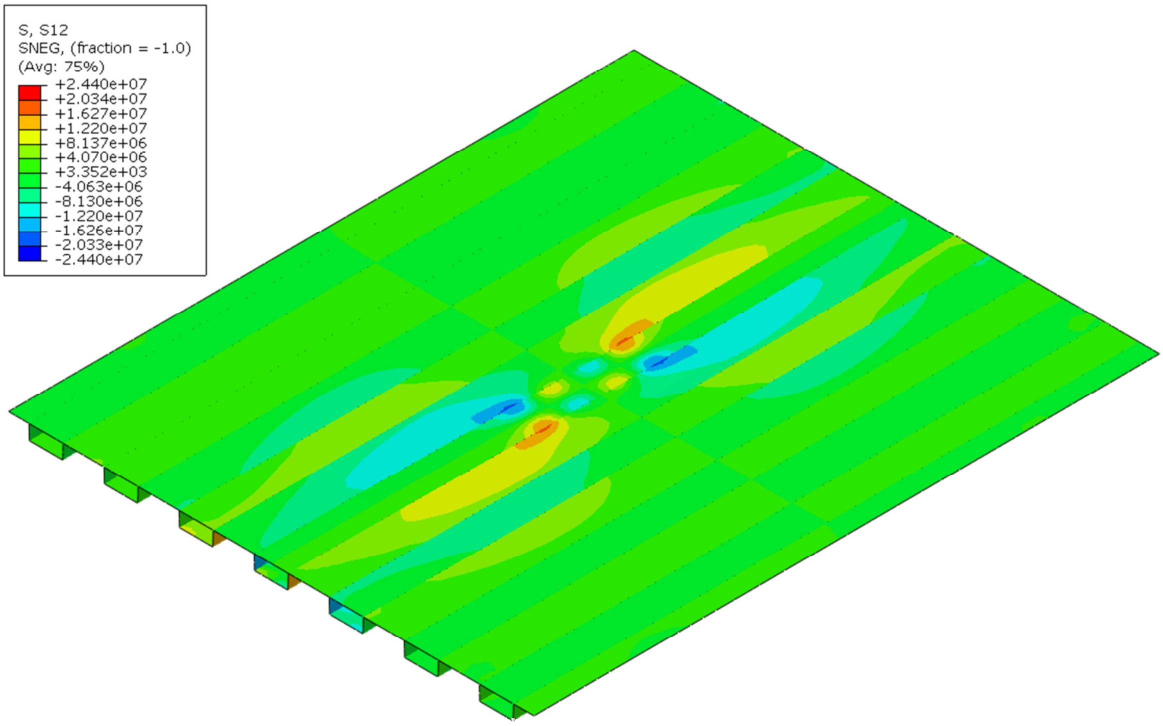

Figure 5 *Shear stress, LC1.*

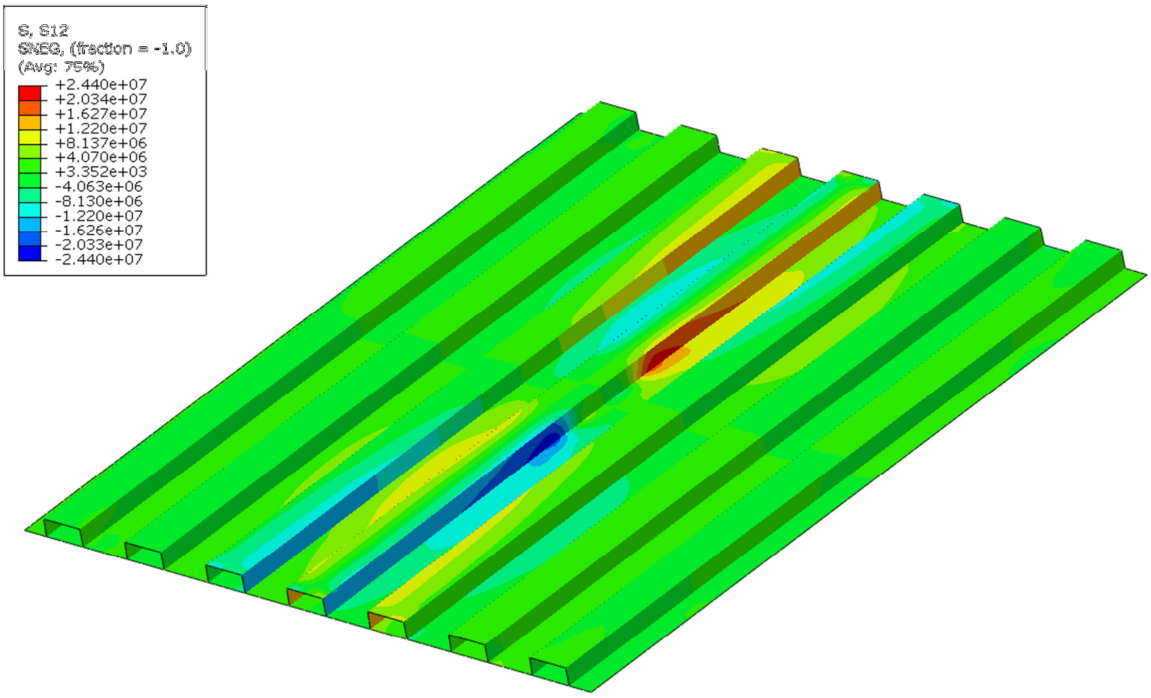

Figure 6 *Shear stress, LC1.*

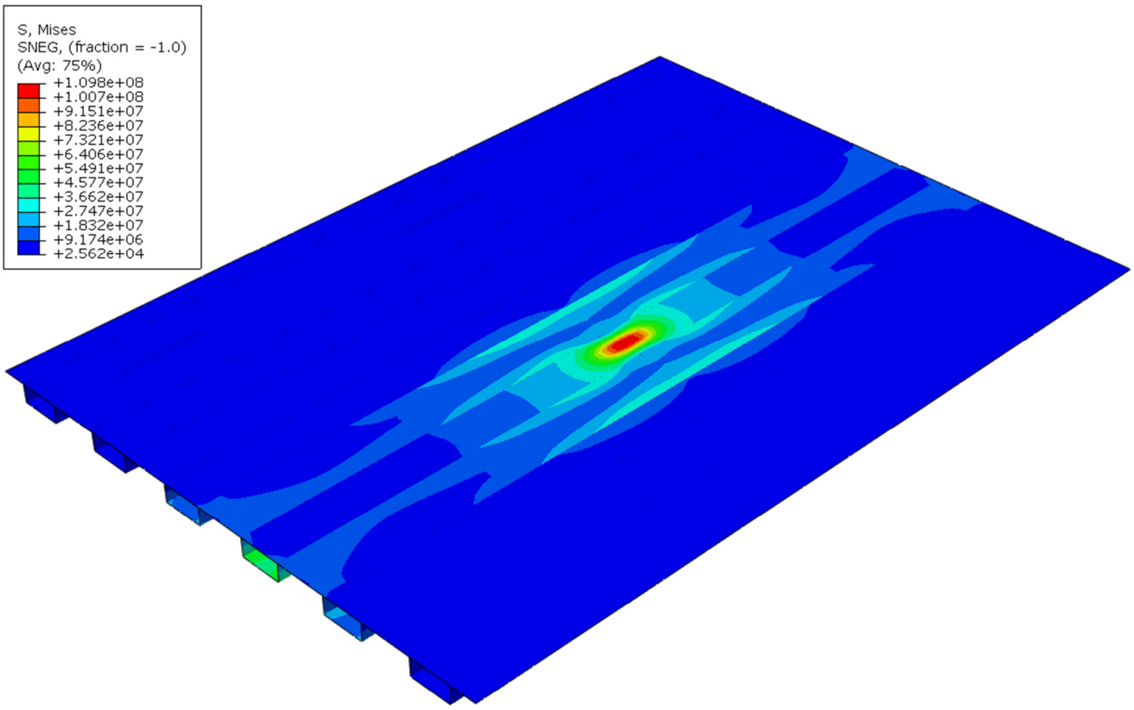

Figure 7 *Von Mises stress, in the top plate between webs, from one tyre turned 90 degrees, LC1.*

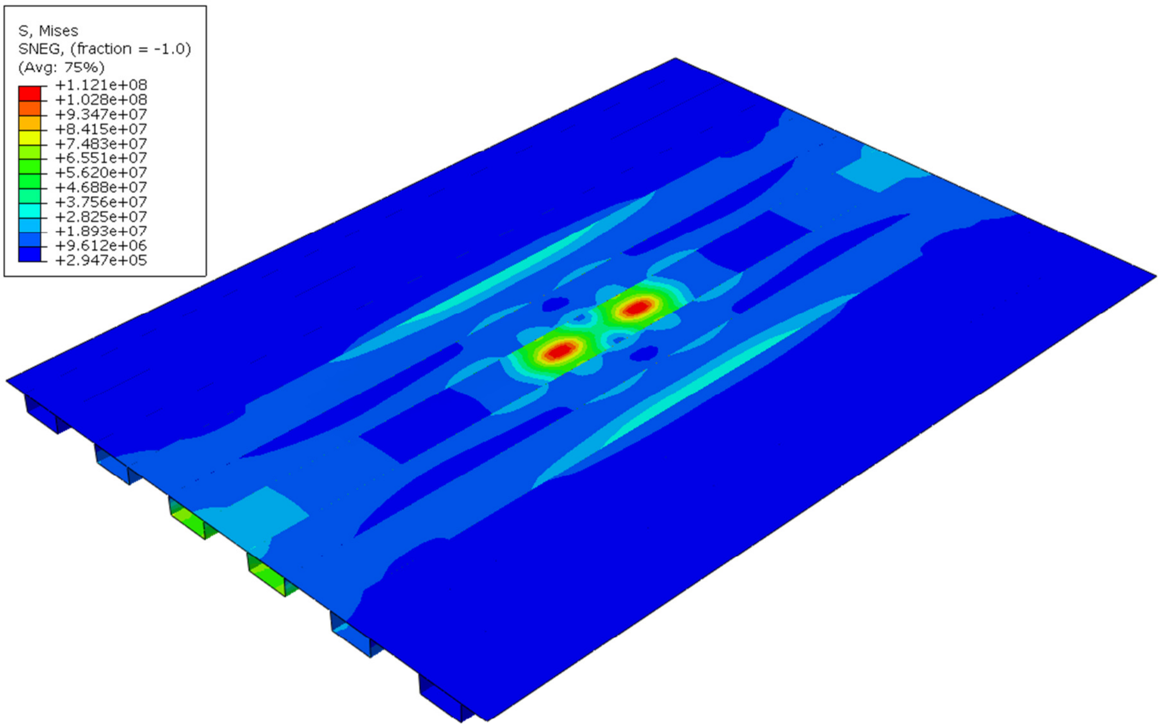

Figure 8 *Von Mises stresses, LC2.*

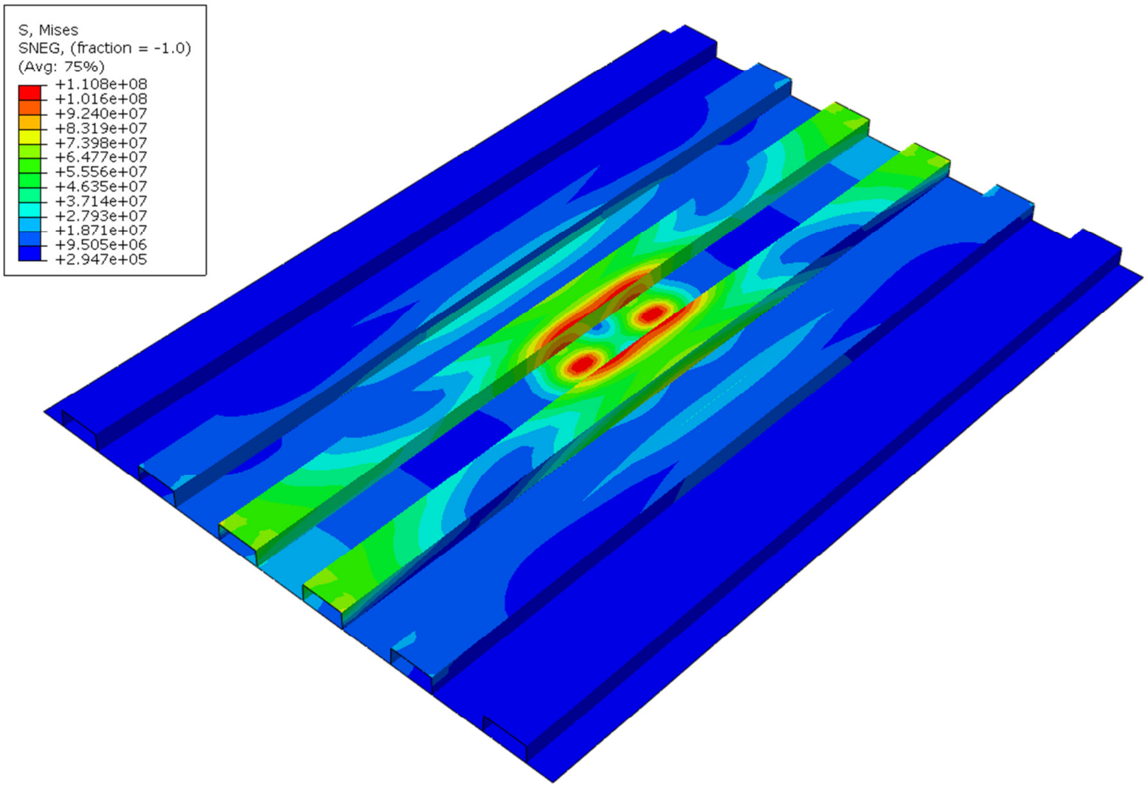

Figure 9 *Von Mises stresses, LC2.*

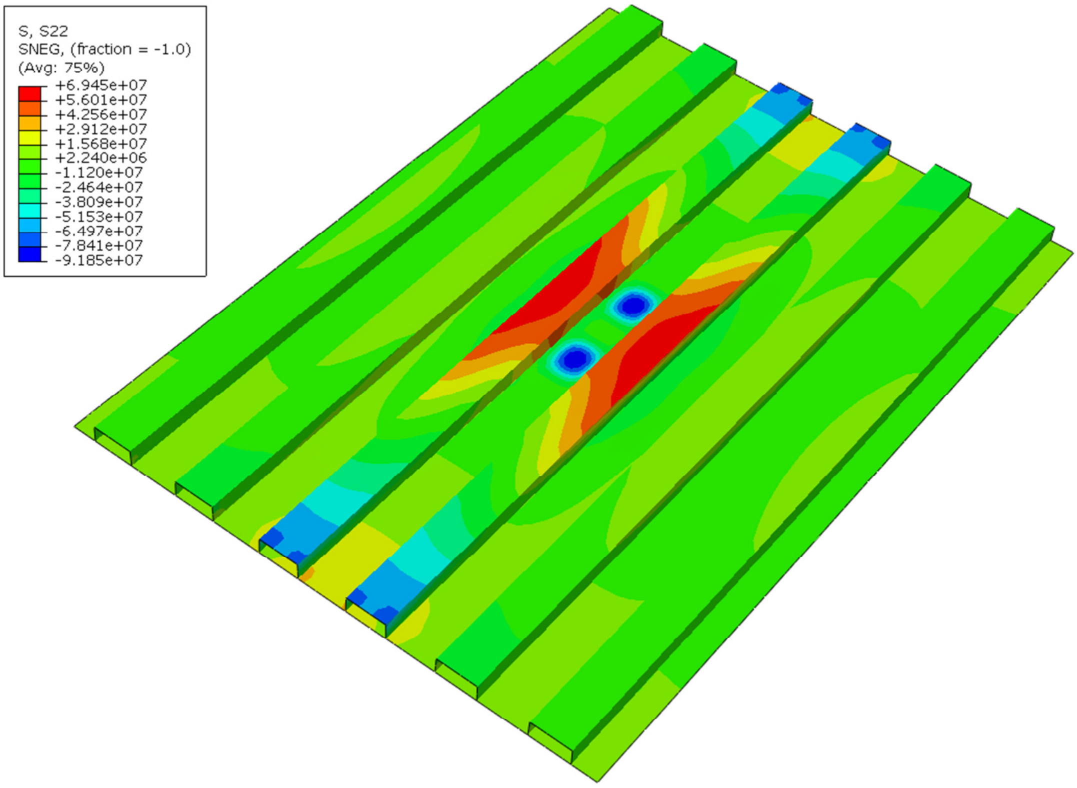

Figure 10 *Normal stress, LC2.*

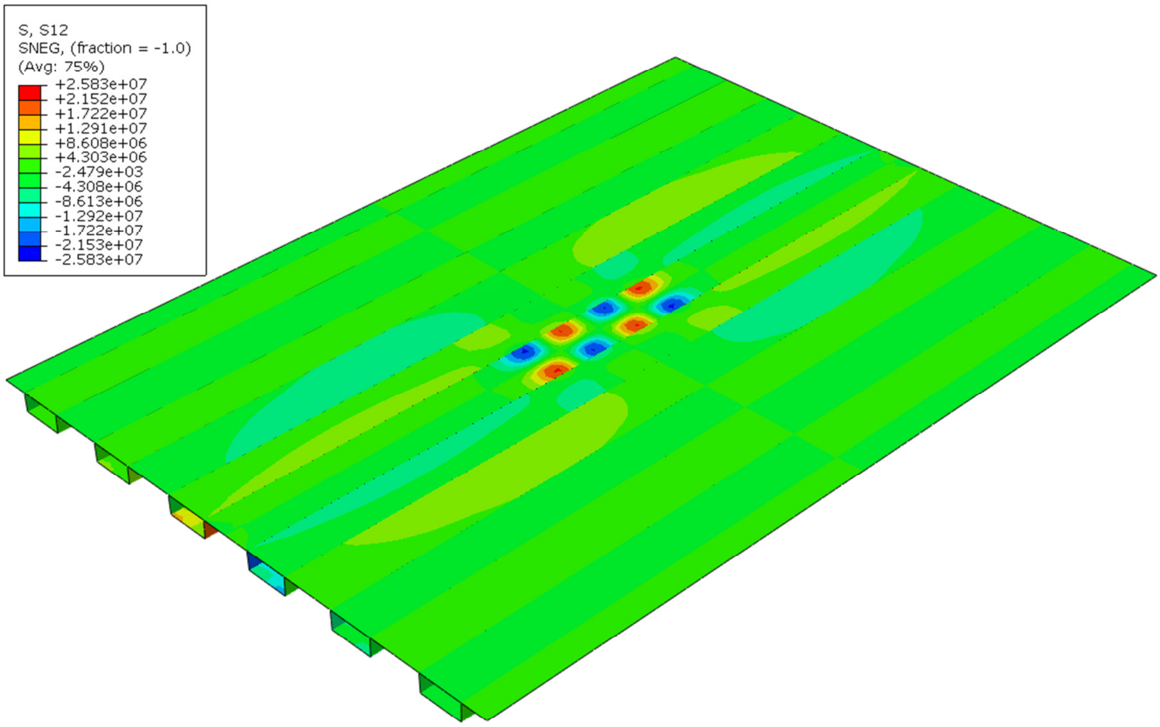

Figure 11 *Shear stress, LC2.*

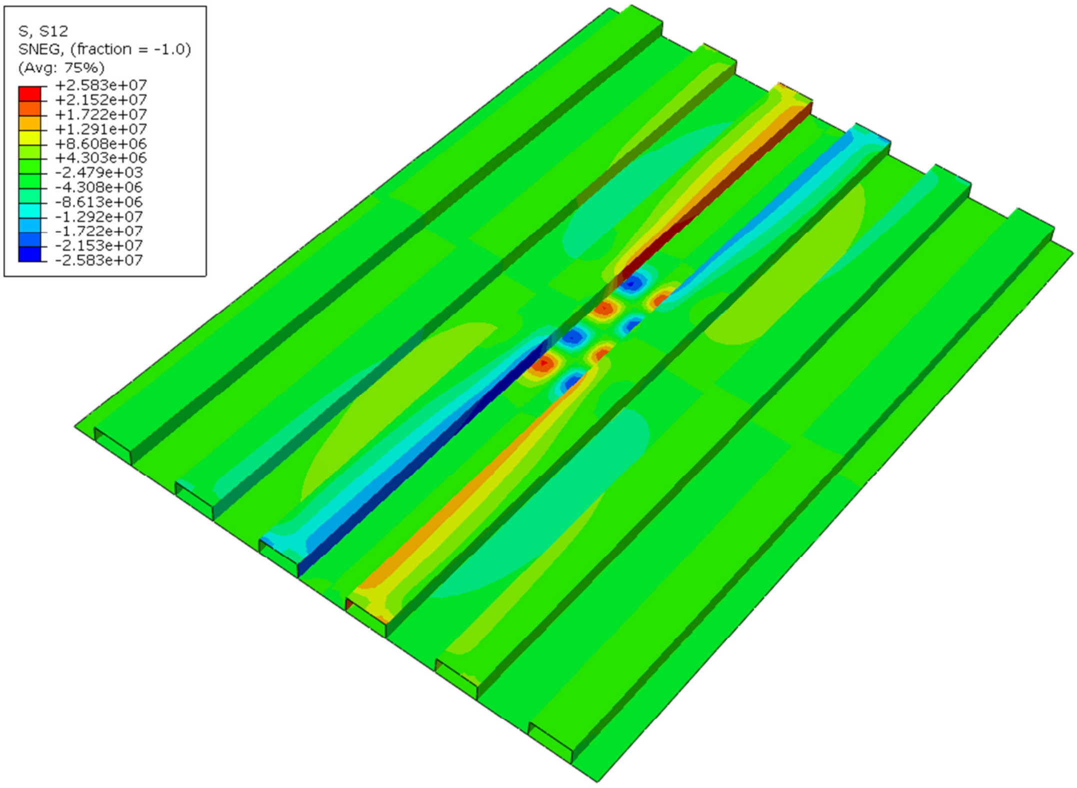

Figure 12 *Shear stress, LC2.*

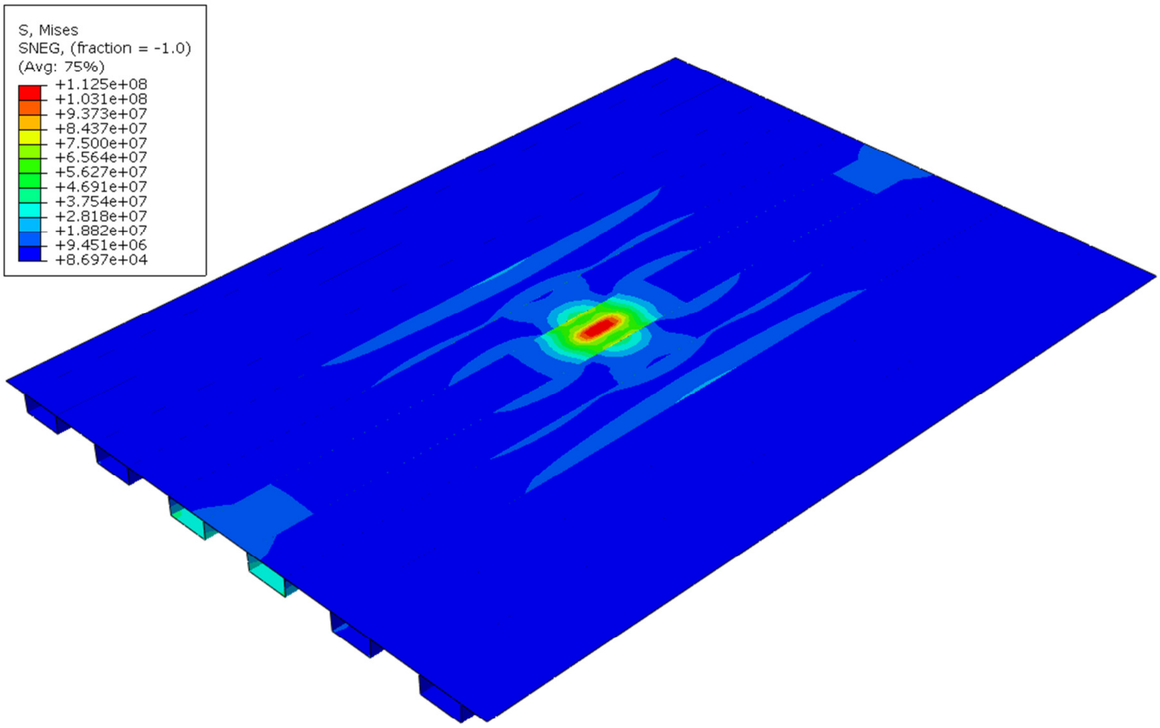

Figure 13 *Von Mises stress, in the top plate between stiffeners, from one tyre turned 90 degrees, LC2.*

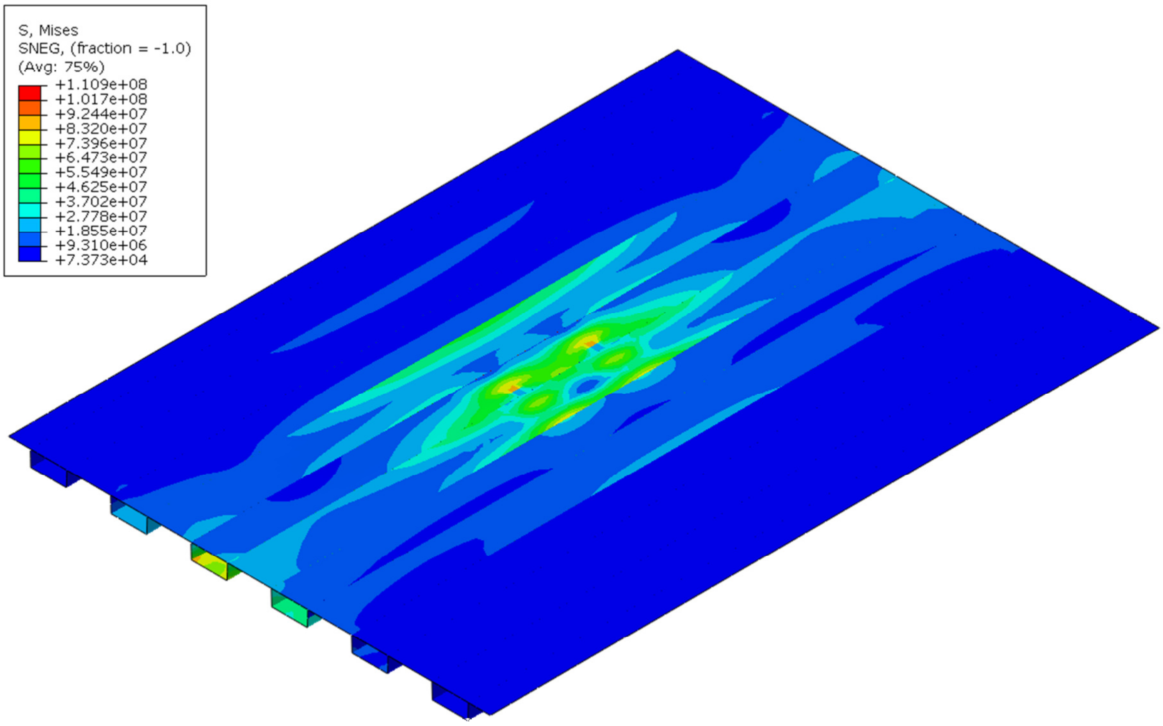

Figure 14 *Von Mises stresses due to the load from two tyres placed on the weld.*

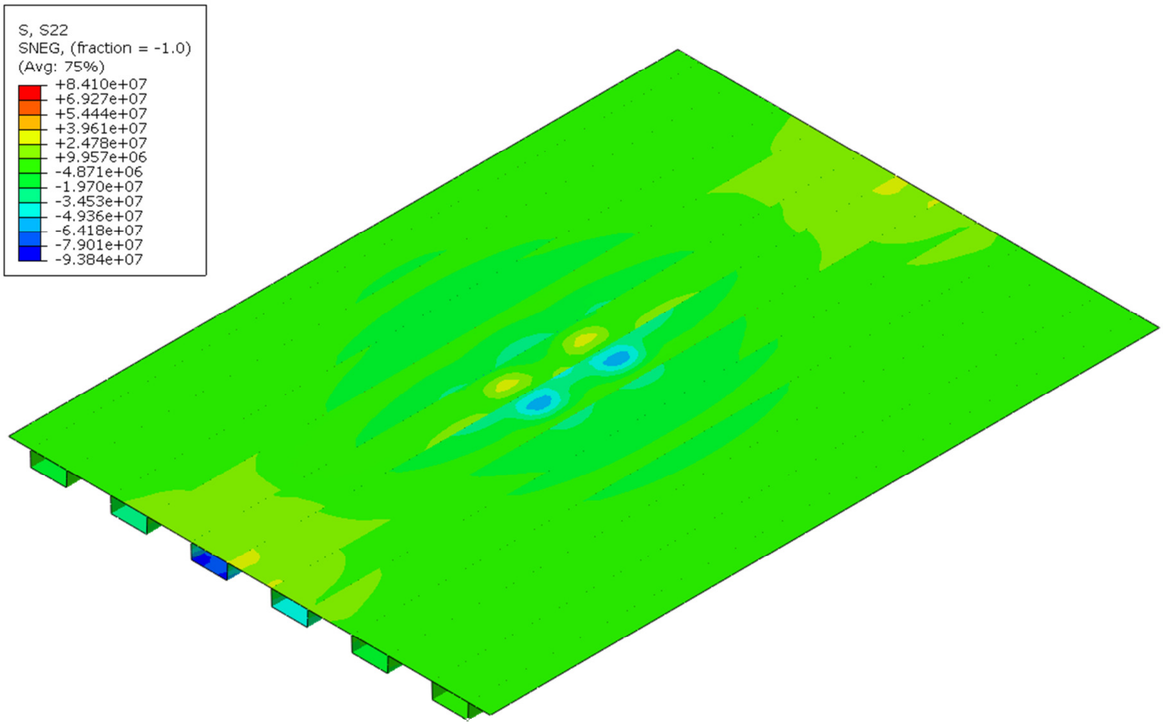

Figure 15 *Normal stresses due to the load from two tyres placed on the weld.*

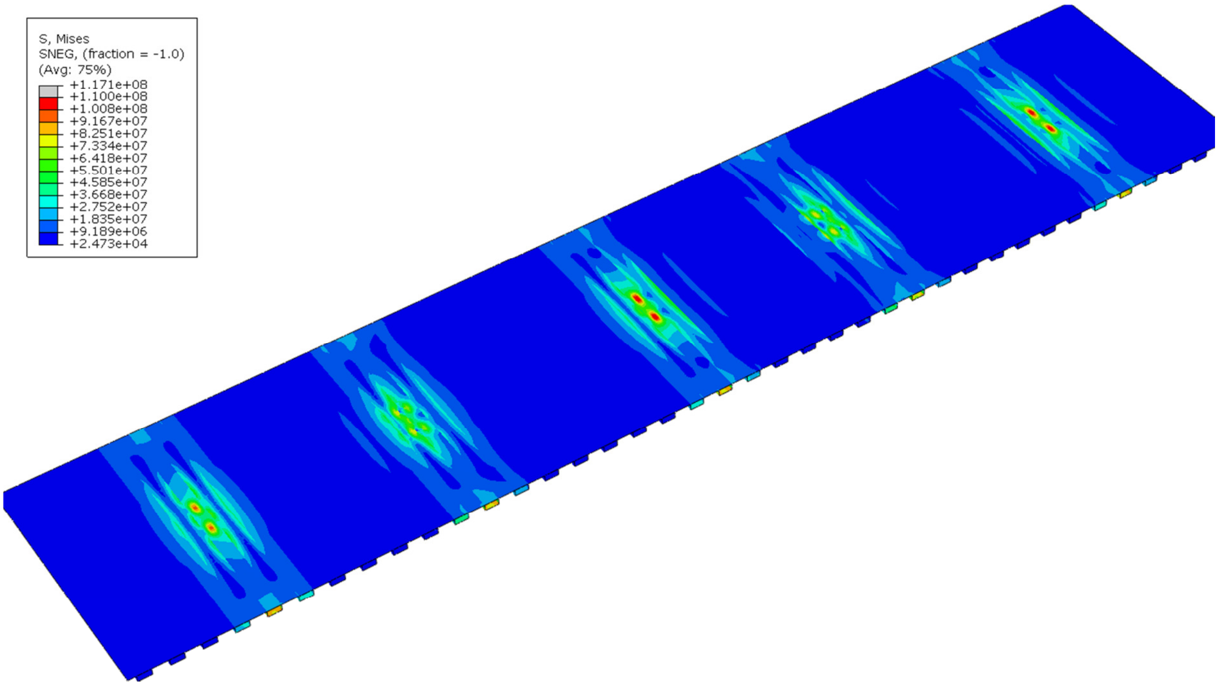

Figure 16 *Von mises stress in the stiffened top plate for load case 3.*

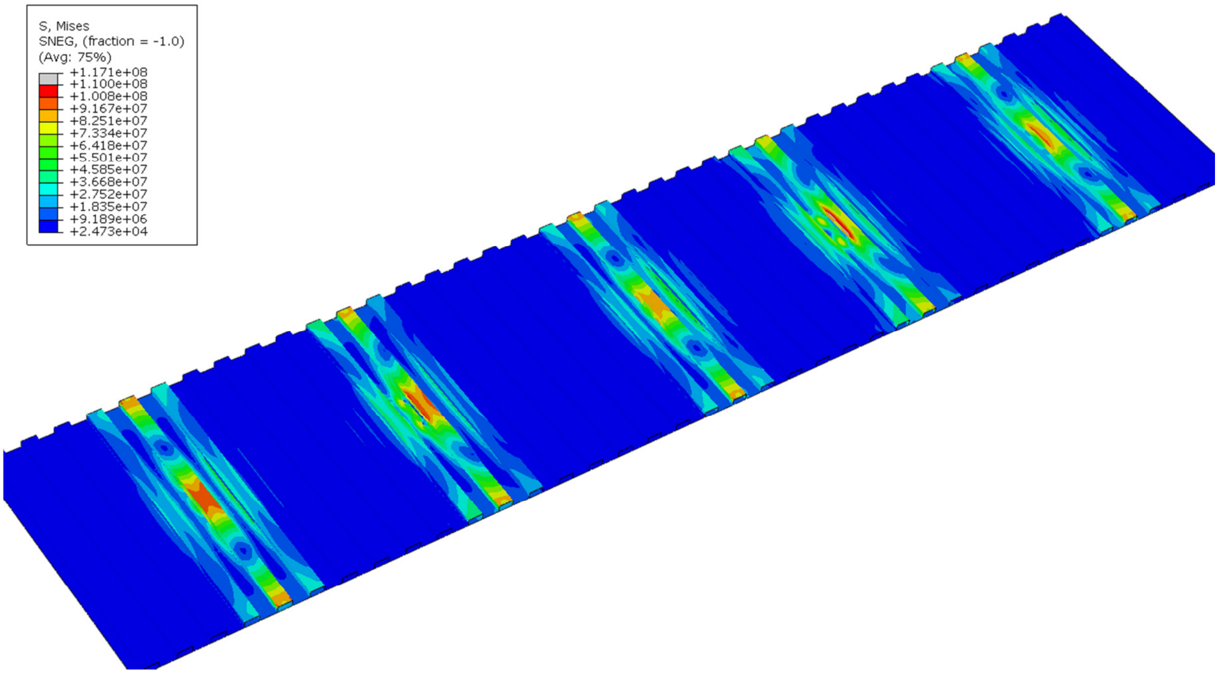

Figure 17 *Von Mises stress in the stiffeners for load case 3.*

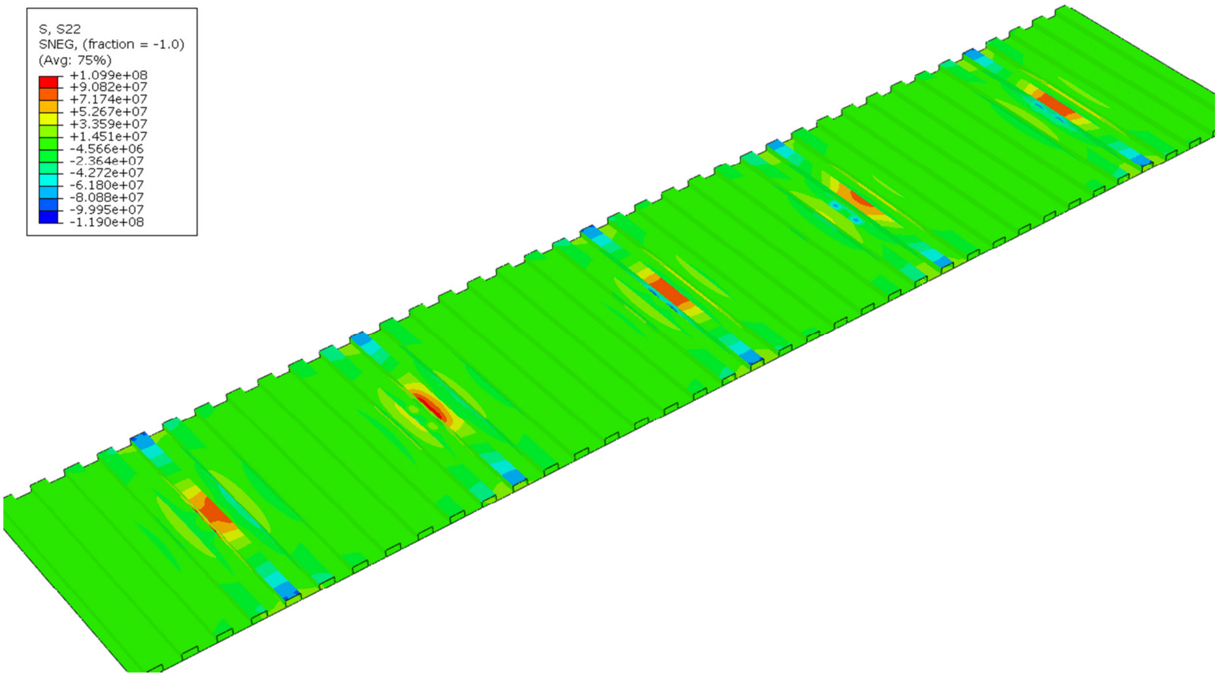

Figure 18 *Normal stress in the stiffeners for load case 3. The negative stress of 119 MPa occurs in the web, in this model this is not the normal stress since the S22 direction of that structural member is not normal to the cross-section, this can be seen in Figure 19. Hence, this is disregarded.*

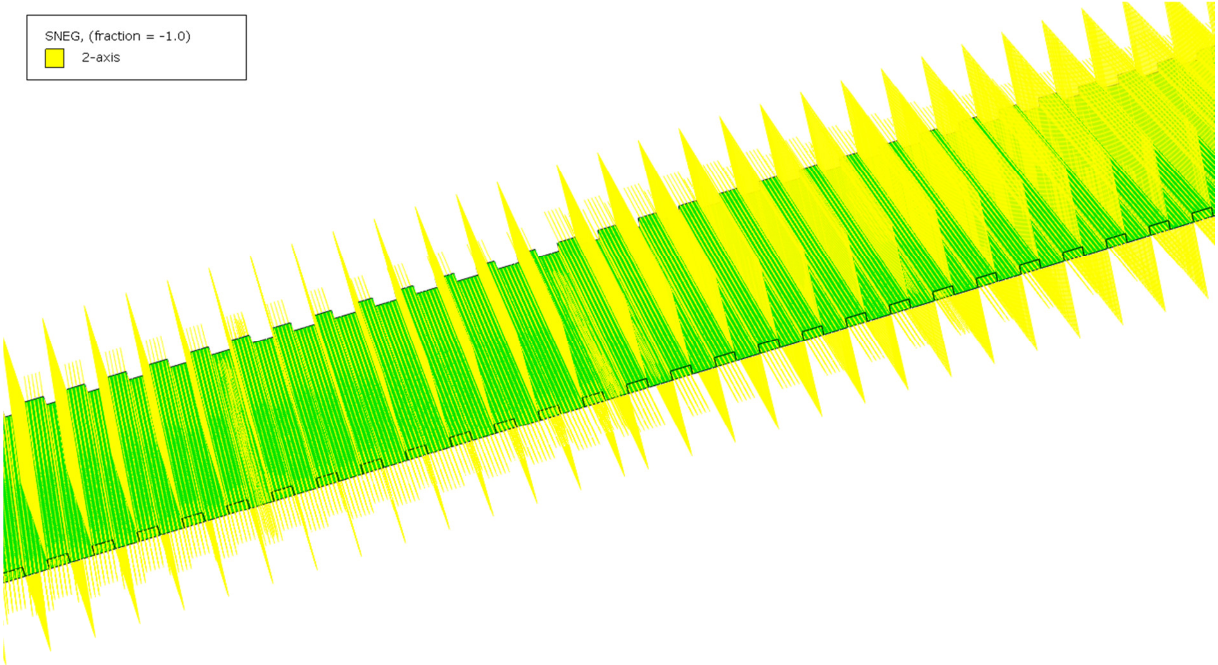

Figure 19 *S22 direction for the structural members, here the S22 of the flange and top plate is normal to the cross-section while the S22 of the webs are not.*

APPENDIX B: GeniE contour plots

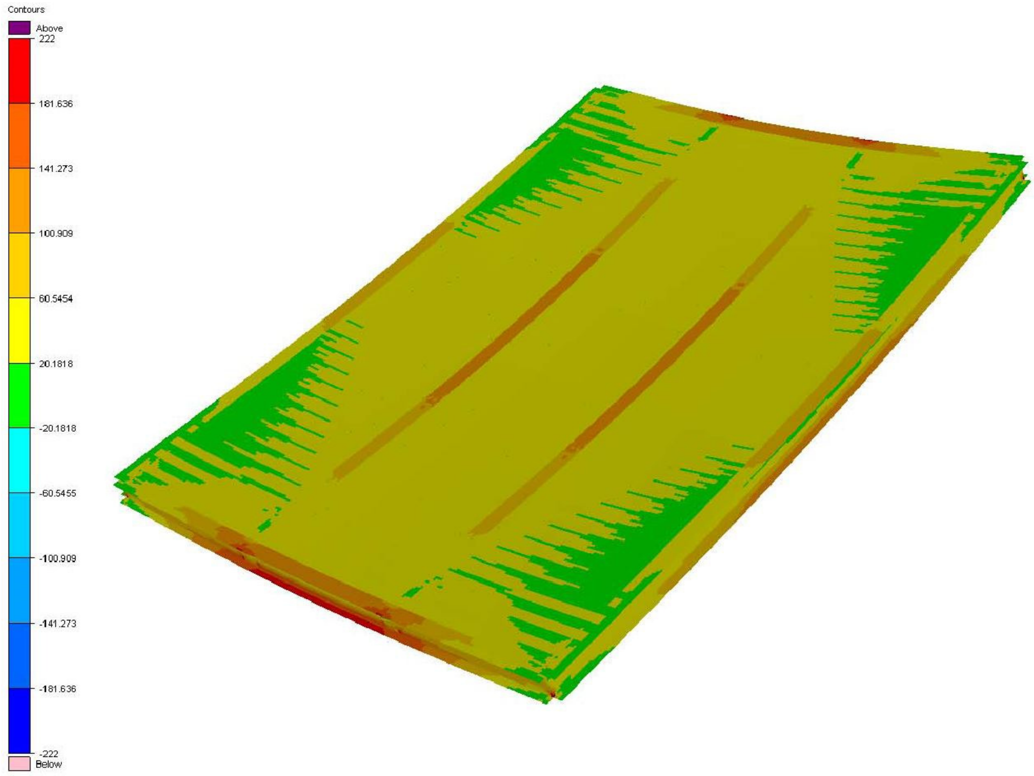

Figure 1 *Von Mises stresses in the car deck structure, top view.*

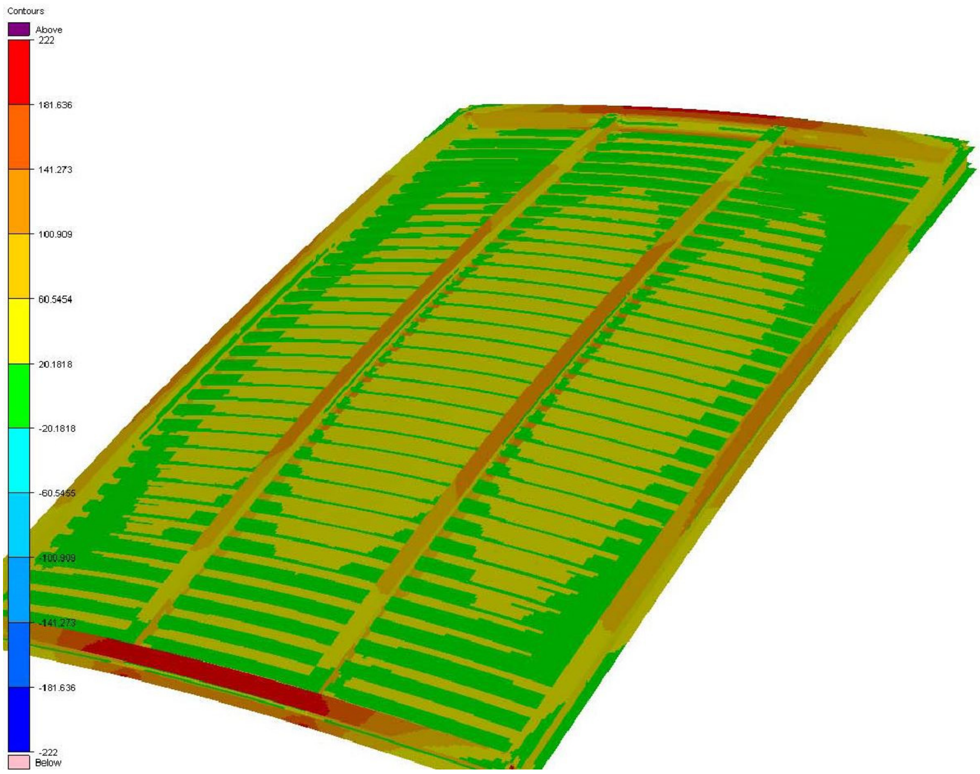

Figure 2 *Von Mises stresses in the car deck structure, bottom view.*

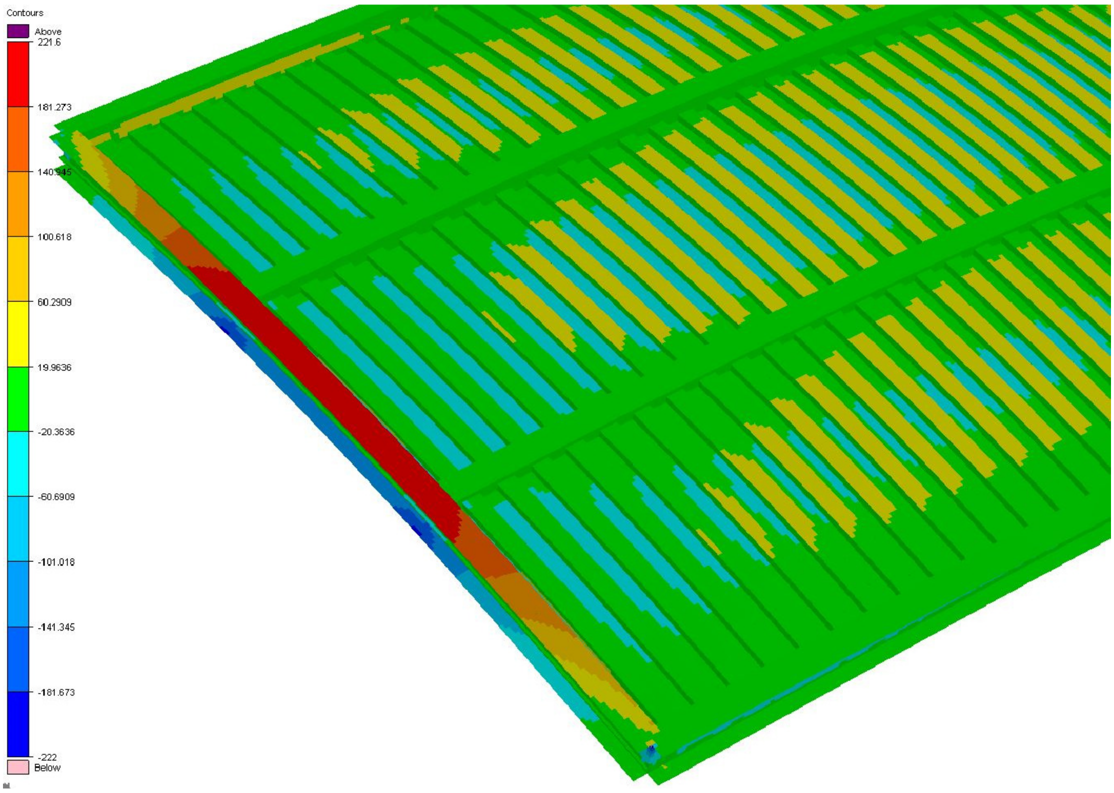

Figure 3 *Normal stresses in the load bearing frame.*

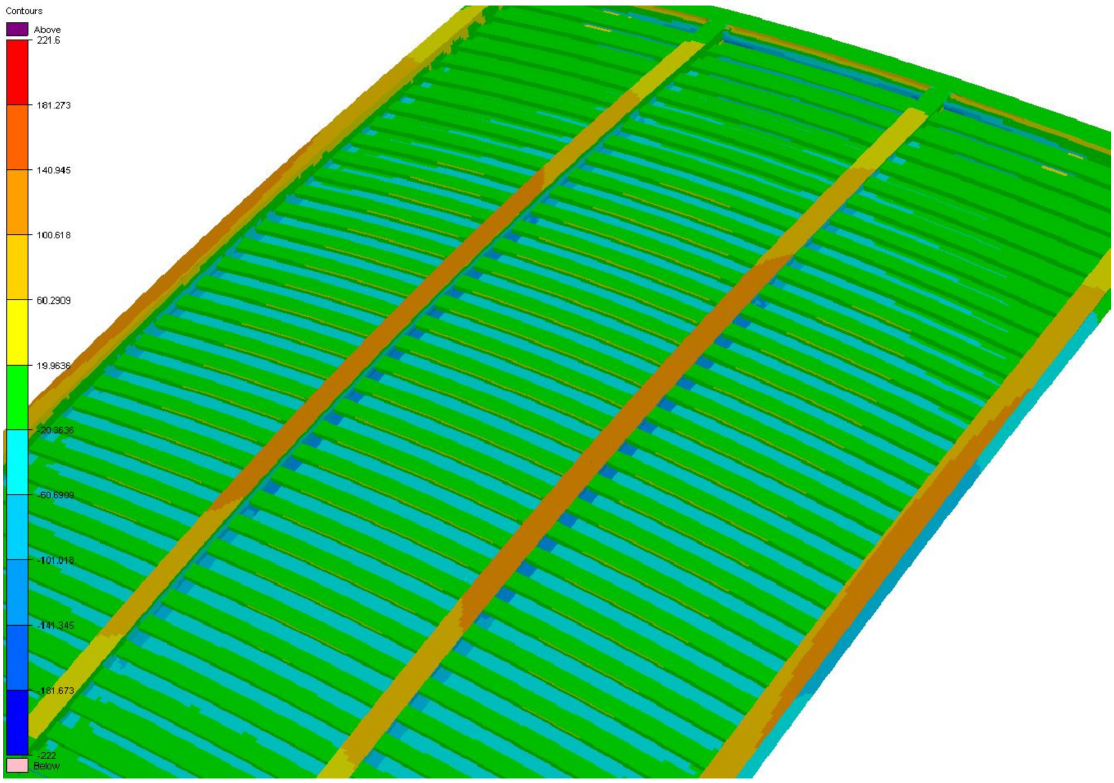

Figure 4 *Normal stresses in the longitudinal stiffeners.*