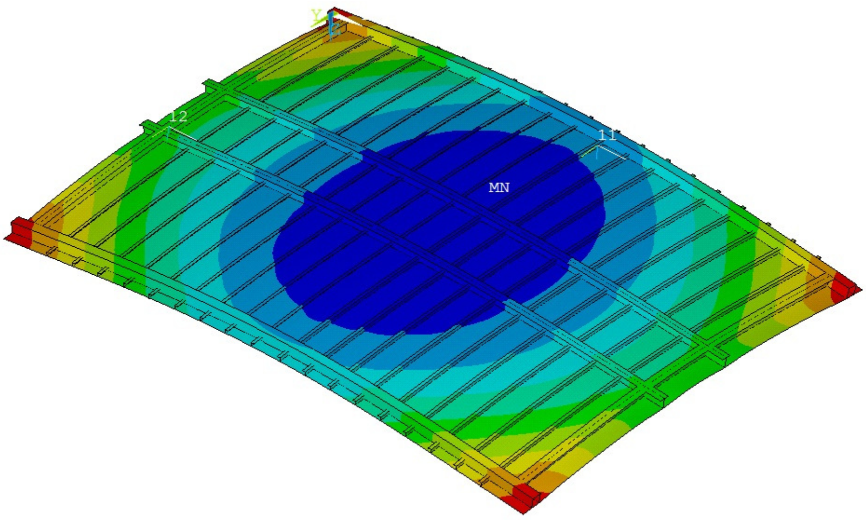

Parametric Design and Optimization of Steel Car Deck Panel Structures

Master of Science Thesis

BARIS ALATAN
HAMED SHAKIB

Department of Shipping and Marine Technology
Division of Marine Design
CHALMERS UNIVERSITY OF TECHNOLOGY
Gothenburg, Sweden, 2012
Report No. X-12/280

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Department of Shipping and Marine Technology
Chalmers University of Technology
SE-412 96 Gothenburg
Sweden
Telephone +46 (0)31-772 1000

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Abstract

Stiffened panels play a significant role in marine industry because of their high strength-weight ratio, they account for a significant amount of a vessel's weight. Hence, weight optimization of these structures can reduce the material costs and to a great extent increase the cargo capacity of a vessel.

This thesis looks into the performance of three steel car deck panels with respect to their weights for a Pure Car and Truck Carrier (PCTC). The focus is on the structural arrangement rather than a comparison of steel and alternative materials, since lightweight materials are still not economically viable for these types of ships. Two of the car deck panels have conventional structural arrangement stiffened with longitudinal and transverse stiffeners while the third one uses diagonally positioned beams. In order to carry out a consistent comparison, the car deck panels are optimized by means of finite element analysis and parametric sensitivity analysis. The panels are modelled with linear elastic materials and a global strength analysis is made with a uniformly distributed load. Results prove the accuracy of the way that an older car deck panel (Concept B) had been developed over time, resulting in the car deck panel currently in use (Concept A). Results also show that the current car deck panel structure could be developed further by utilizing the optimization techniques, reducing their weight by up to 6%.

Keywords: car deck panel, finite element analysis, optimization, parametric sensitivity analysis, stiffened panel structure, strength, weight.

Preface

This thesis is a part of the requirements for the master's degree in Naval Architecture and Ocean Engineering at Chalmers University of Technology, Gothenburg, and has been carried out at the Division of Marine Design, Department of Shipping and Marine Technology, Chalmers University of Technology. The current investigation has been done as a real case study with data from a ship owner (client) who wishes to be anonymous. For this reason, specific data related to the client have been omitted from the report. The project was performed in cooperation with TTS Marine AB.

We would like to acknowledge and thank our examiner and supervisor, Professor Jonas Ringsberg at the Department of Shipping and Marine Technology and our supervisors Thomas Falk and Peter Anderson at TTS Marine AB for their supervision.

We would also like to thank Professor Anders Ulfvarson and Luis Sánchez-Heres for their support and assistance during the project.

Gothenburg, June, 2012
Baris Alatan and Hamed Shakib

Contents

1. Introduction

1.1. Background

Thin-walled structures are widely used in the maritime industry because they make the structure more cost-effective by offering a desirable strength/weight ratio. Reduction in the structural weight of ships will increase their cargo-carrying efficiency. This increase in efficiency is obtained by either carrying more cargo with the same displacement or by increasing the speed of the ship. Moreover, the substantial decrease in material cost supersedes the higher production costs. One can easily predict that both improvements are also important from a sustainability point of view. Less emission of hazardous gases produced by marine diesel engines and reducing the use of natural resources are the examples of these structures' advantages in terms of sustainability.

Different types of materials such as steel, aluminium, composite and plywood are used in car deck structure design. According to Jia and Ulfvarson [1], utilizing alternative materials to produce lightweight decks in marine structures will lead to weight reduction in panels. However, this advantage is overshadowed by the significant manufacturing and material costs. As a result of this, the focus in the marine industry has been shifted toward the structural designs and optimization of panels, either by means of modifying the dimensions or utilizing alternative configurations for the panel structures. In his study on plates subject to shear loading, Alinia [2] has presented the relationship between the increase in critical shear stress for buckling when a plate is stiffened and certain parameters such as the aspect ratio and the type and number of stiffeners. Maiorana et al. [3] have in a similar study presented the dependence of the critical buckling load of a longitudinally stiffened plate on the stiffener position, the load that the panel is subjected to (in-plane bending, compression or shear), the type of cross-section, stiffener flexural rigidity and panel aspect ratio. Likewise, in a study on a longitudinally stiffened panel subjected to bending moment in its own plate, Alinia and Moosavi [4] have shown that by placing the stiffener at its optimal position, an increase in the critical bending stress coefficient can be increased by as much as six times. Furthermore, this optimal position is dependent on the stiffener's flexural rigidity and the panel's aspect ratio. Regarding alternative configurations, Maiorana et al. [3] have presented the fact that stiffeners with closed cross-sections have a better buckling performance than open cross-sections, which are generally used by the marine industry. Nie and Ma [5], on the other hand, have investigated possible improvements that can be achieved by adding thin-walled box beams to the decks of a warship. They have concluded that in this way up to a 20% decrease in deck stresses and more than a 90% increase in deck buckling stress will lead to significant improvements in hull strength and survivability with less than a 10% increase in structural weight.

In this thesis, a methodology is presented to investigate the possibility of obtaining weight reductions in steel car deck panels while fulfilling certain criteria such as deflections and stresses occurring in the panel. This is to be done by means of numerical optimization techniques, as have been tested and proven resourceful in a number of research projects, such as those by Vanderplaats [6], Kumar et al. [7], Brosowski and Ghavami [8] and Vanderplaats and Moses [9].

1.2. Objective

The main objective with this investigation is to evaluate and compare a series of car deck panels by their weights and performance in satisfying design requirements. These are the required free height above the fixed below deck, the deflection along the edges and the stress requirements of the relevant classification society, which in this study is Det Norske Veritas (DNV). The following targets were determined to have a consistent comparison between different concepts:

- A conventional car deck panel concept currently in use was optimized and if possible further developed by means of parametric study and optimization.
- The same procedure was applied to an older concept which had been replaced by the current concept.
- Finally, alternative concepts were created and optimized.

The performances of optimized car deck panel designs were then compared and it was concluded whether or not the current design should be changed. Furthermore, parameters or details that play an important role in the weight optimization of a car deck panel were studied.

1.3. Methodology

A reference car deck panel of a pure car and truck carrier (PCTC) type of vessel was proposed by TTS Marine AB for this study. Following the methodology shown in Fig. 1, performance criteria were first imposed on deflections and stresses that occur in the models for different working positions of the car deck panel. Finite element models of all the geometries were created by parameterizing the variables such as plate thickness, web or flange width, or stiffener spacing. The static analysis was carried out in ANSYS Mechanical APDL 13.0 [10]. Several runs were made to make a mesh convergence analysis. These parametric models were then used in the goal-driven optimization part of the project. Using design of experiments, several combinations of the parameters are created as "design points". By running the analysis for each design point and recording the outputs, a mathematical formula was fitted to the data, which is called a "response surface". The parameters with significant influence on the outputs were determined from this formula, with which new response surfaces were created. Numerical optimization techniques are then utilized to obtain weight-optimized panel structures. Goal-driven optimization is presented in detail in Section 2.

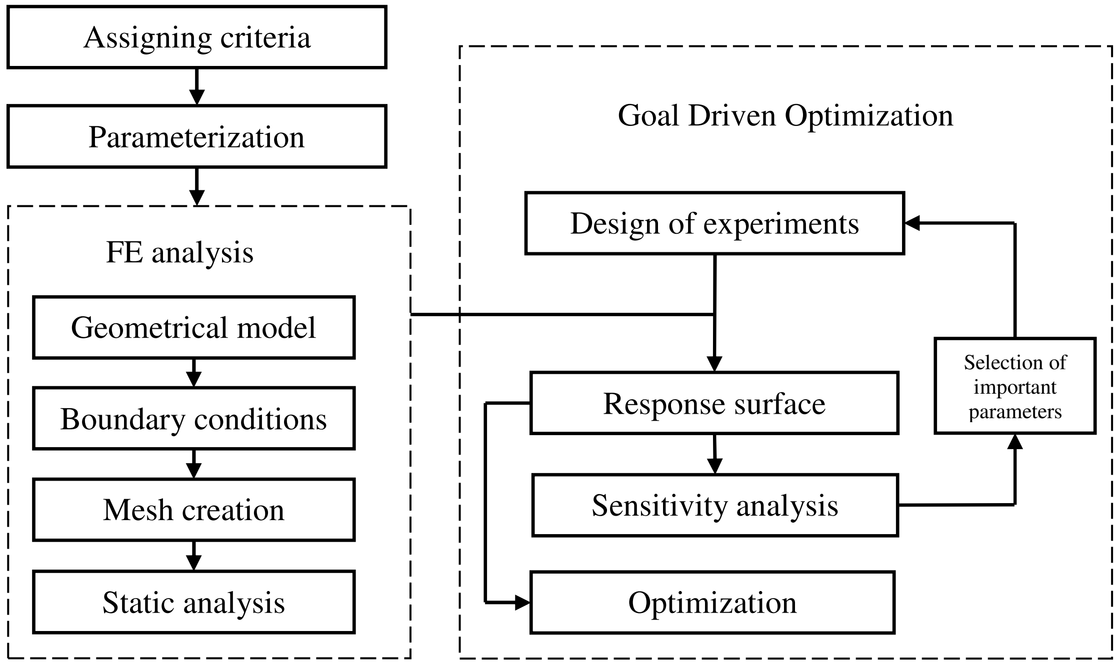

Fig. 1. Flow chart showing the steps and their interactions.

1.4. Limitations

Stress components are to be below the yield stress limit of the material. By considering steel as the only material used, linear elastic material models are assumed to be sufficient. The geometries were created by continuous shell elements without taking welds into account.

Dynamic effects are important limiting factors in marine structures. According to Jia and Ulfvarson [1], vibration and damping problems will arise when ship structures are made lightweight. The fact that ship structures are subject to cyclic loads from waves renders the fatigue life of the structure a significant issue as well. It is important to be aware of structural responses to these factors for reaching a feasible design. A lighter design increases the natural frequency of the car deck, which must be considered in order to avoid causing resonance frequencies in the system. On the other hand, a structural arrangement of the car deck may affect its vibration modes and consequently its stress conditions. Results of this investigation do not include such effects. Weight optimization of a car deck panel with regard to static loads does not necessarily improve its performance towards dynamic loads. Further investigation has to be carried out in order to compare the structural response of the car decks to static and dynamic loads and their correlations.

The design criteria for deflections and stresses were based on global strength of the panels with a uniformly distributed load and self-weight acting on the structures. Axle loads from the cars are not taken into account, which is important as a design criterion from a local strength point of view. It affects the scantling of secondary stiffeners and local plate buckling of the car deck panel but not the criteria in this study.

The investigation focuses on the analysis of a car deck panel in a particular location in a ship. A car deck panel in a different position in the ship might have different dimensions and will be subjected to different accelerations. This results in a completely different loading condition. Therefore, the conclusions of this thesis might not necessarily be directly applicable to panels different than the ones presented here.

Production costs have great impacts on design of such structures and usually lead to a contradiction preventing a lighter design to be achieved. The stress condition varies over the plate, which requires different dimensions to be applied to different parts. Production costs, on the other hand, would decrease if the parts had the same dimensions. For instance, parts with the same thickness could be cut from the same plate and a manufacturer could take advantage of the economy of scale. Therefore, the same dimensions are applied to all parts regarding the maximum stress value. Parameterization of Concepts A and B (Sections 4.2 and 5.1) is an example of such a contradiction where unique parameters were assigned to different parts. Otherwise, no financial analysis was made in this investigation. Consequently, the use of technology that could improve the design by overcoming some limitations is not discussed. For instance, by considering developments in welding technology the minimum plate thickness that is allowed to prevent buckling could decrease. This could result in decreasing to a great degree the weight of the structure.

2. Theoretical background of goal-driven optimization

This section presents the theory behind goal-driven optimization, which has been selected as the method for obtaining weight reduction in car deck panels. How the design of experiments is used to obtain a response surface and make a parametric sensitivity is explained. This is followed by a presentation of different numerical optimization methods that are used in this project.

2.1. Design of experiments

The design of experiments specifies changes in input parameters in order to observe the corresponding output response of the system. It allows building the response surface without the need for performing the analysis for all possible combinations of input parameters (fractional factorial design).

A second-order polynomial model is used in this project to be fitted to the response data (response surface). For this purpose, a central composite design according to Montgomery [11] is the most popular and efficient design used. For details, see [11] and [12]. Each combination of input parameters used is called a design point. The number of design points that must be created for the number of input parameters in order to obtain the response surface is shown in Fig. 2.

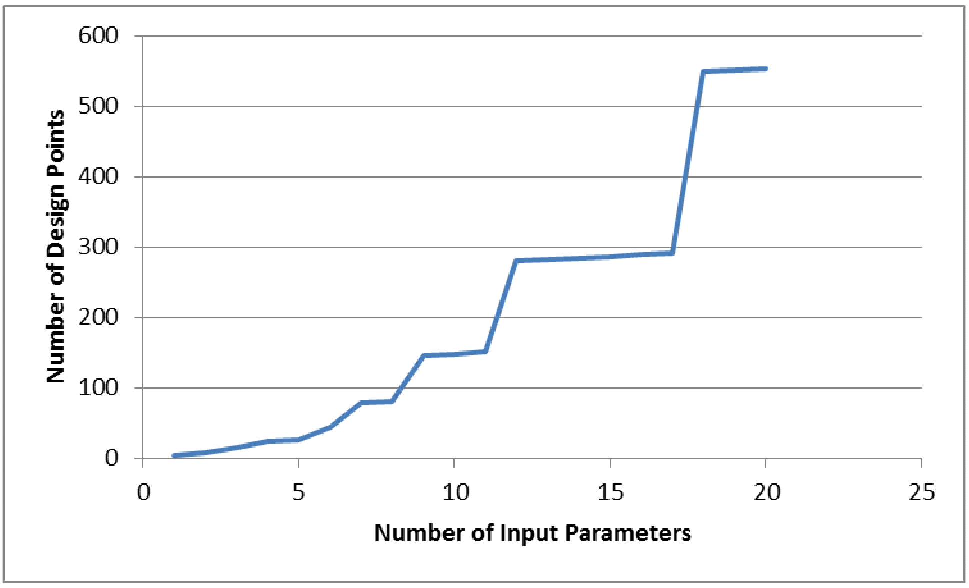

Fig. 2. The number of design points for the number input parameters [10].

2.2. Response surface and parametric sensitivity

Considering the response (output parameters, for example deflection) of a system to be continuous and influenced by several factors (input parameters, for example web height, flange thickness), a response surface is built by plotting the response to possible combinations of the factors (design points). The response surface of *n* factors is plotted in the *n+1* dimension [11], [13]. A response surface as a function of two factors, *A* and *B* is shown in Fig. 3.

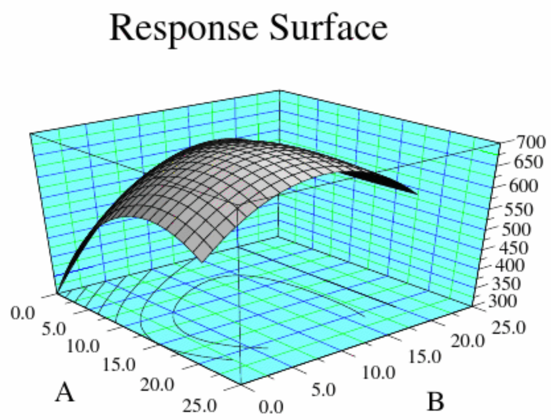

Fig. 3. Hypothetical response surface for two factors A and B, from [13].

A response surface can be approximated by a first-order or second-order mathematical model. If the response is a linear function of independent variables, a first order polynomial model is used:

$$y = \beta_0 + \beta_1 x_1 + \beta_2 x_2 + \ldots + \beta_k x_k + \in \qquad (1)$$

If a non-linear function is needed to model the response, which is mostly the case in real life applications, a polynomial of higher degree is used, such as the second-order model [12]:

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

Dependence on too many parameters can make the optimization process very time-consuming. The number of parameters as shown in Section 2.1 increases the number of design points and consequently the number of analyses that must be run for each design point. This makes the parametric sensitivity analysis an essential step to minimize the initial parameters by establishing those with a greater impact on output values. This can be done by calculating the sensitivity of output parameters to input parameters. The dimensionless sensitivity of each objective y (output) with regard to the variable x (input) ($y = f(x)$) is computed as $Max(y) - Max(y)/Avg(y)$ [14]. A parametric study has been performed utilizing the ANSYS Workbench [10], a framework upon which sensitivity of each individual parameter assigned as input is analysed (goal-driven optimization).

2.3. Optimization

Optimization problems are generally classified as follows [15], [16]:

- Unconstrained problems.
- Linearly constrained problems.
- Non-linear programming problems.

Unconstrained problems are the problems that have an objective with no constraints. Obviously, the objective function must be non-linear since the minimum of the linear unconstrained function is $-\infty$ (neg. infinity). Linear constrained problems have linear constraint functions and the optimization problems, in which one or more constraint functions are non-linear, are called non-linear programming problems. The problem in the current study is an example of non-linear programming problems because of its non-linear constraint functions.

Kumar et al. [7] define the structural design problem as shown in Table 1.

Table 1. A typical structural design problem.

MINIMIZE	SUBJECT TO
Weight or some other design goal	Stress constraints Frequency constraints Manufacturing requirements Reliability, quality and cost considerations Geometry considerations Other miscellaneous design requirements

Vanderplaats [6] provides the mathematical formulation to this problem as follows:

Minimize:

$$F(X) \tag{3}$$

Subject to:

$$g_j(X) \leq 0 \quad j = 1, m \tag{4}$$

$$X_i^l \leq X_i \leq X_i^u \quad i = 1, n \tag{5}$$

where Eq. (3) is the objective function, dependent on the design variables {**X**}, (4) the inequality constraints, and (5) the side constraints. Constraint functions enforce limits on the design variable values. Inequality constraints impose either upper or lower limits, whereas side constraints affect both upper and lower limits.

To solve an optimization problem there is a vast range of analytical and numerical methods available in the literature. A review of all existing methods and their developments is beyond the scope of this thesis. Interested readers are referred to references [9] and [15] - [17]. Instead, random search techniques such as screening and genetic algorithms that are used in this project are briefly described. These methods are considered as direct approaches to optimization problems and because of their simplicity, availability and cost-effectiveness they are preferred over the other methods.

A screening method is a direct sampling method which creates a sample set from the design points and sorts them based on the objectives. It is a powerful method in obtaining the approximate vicinity of global minima and is suitable for use in preliminary design. This forms the basis for advanced methods used for more refined optimization [14].

Genetic search-based optimization methods belong to the category of stochastic search methods. Based on Darwin's theory of the survival of the fittest, these methods represent a set of alternative designs as "generations". The "traits" of individual alternatives are passed on to the next generation through "reproducing" and "crossing". A blending of the best properties of cross-breeding couples leads to the offspring being superior to both parents. Better objective function values are achieved with consecutive generations and the optimum design can be searched by having the degree of superiority of a population as the target of the design process. The fact that gradients of objective and constraint functions are not necessary is very useful since this helps avoid getting stuck in the vicinity of a local minimum [18]. The multi-objective genetic algorithm, (MOGA), a feature of ANSYS Workbench [10], is used in this project.

3. Performance criteria

This section introduces the performance criteria of the reference car deck panel for which this study has been carried out. The same performance conditions were also used for other investigated concepts in the project. Different loads and boundary conditions apply to two different positions where the panel is used. The performance criteria for these two load cases are defined.

3.1. Working positions

The deck plan of the 6th deck of one of a series of 10,190 lane-metres, 55,000 - square metre PCTC ships is shown in Fig. 4. The deck is divided into liftable panels. One of the panels in the middle, marked grey, is selected by TTS as a typical panel to be used as the reference car deck panel in this investigation.

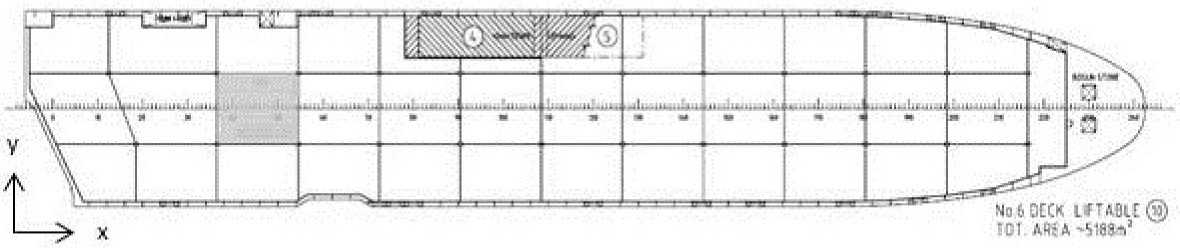

Fig. 4. The position of the reference car deck panel.

The panel has three working positions, as shown in Fig. 5. The first two are the seagoing condition with different requirements for the height above the fixed deck beneath. The third position is the stowed position when the deck is not in use. Different boundary conditions, loads and performance criteria are defined for these two cases.

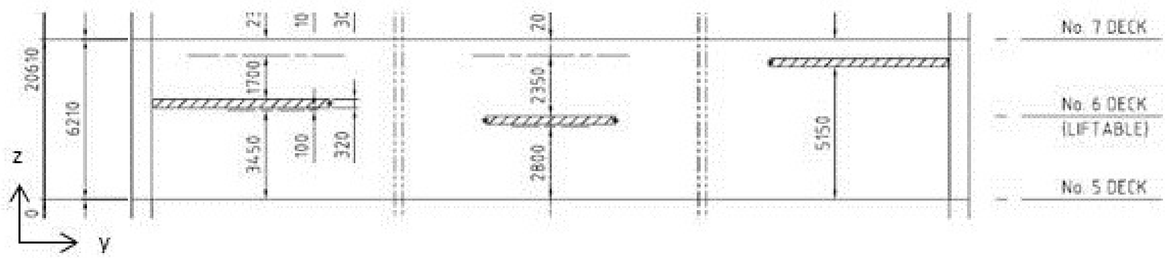

Fig. 5. Working positions of the reference panel.

3.1.1. Seagoing condition (Load Case I)

In this load case, the panel serves as car deck in a seagoing condition. The panel is loaded with a uniformly distributed load (UDL) of 250 kg/m^2 and the self-weight of the panel. The total load is calculated as self-weight (16.2 t) + UDL (250 kg/m^2 * 165.7 m^2) = 57.6 t, where t denotes metric tonne. A dynamic addition of 50%, arising due to the motion of the ship and calculated according to DNV rules [19] increases this load to 86.4 t [20]. In this load condition the panel is simply supported in all four corner areas.

3.1.2. Stowed position (Load Case II)

In this load case, the unloaded panel is lifted to the stowed position. The total load consists of the self-weight and dynamic addition of 20%, which is calculated as 19.4 t [20].

The car deck is usually lifted by a scissor deck lifter which supports the panel in the middle, as show in Fig. 6-b. The deflection of the panel reduces the contact between the flanges of the beams and the platform of the lifter to four points on the edges of platform.

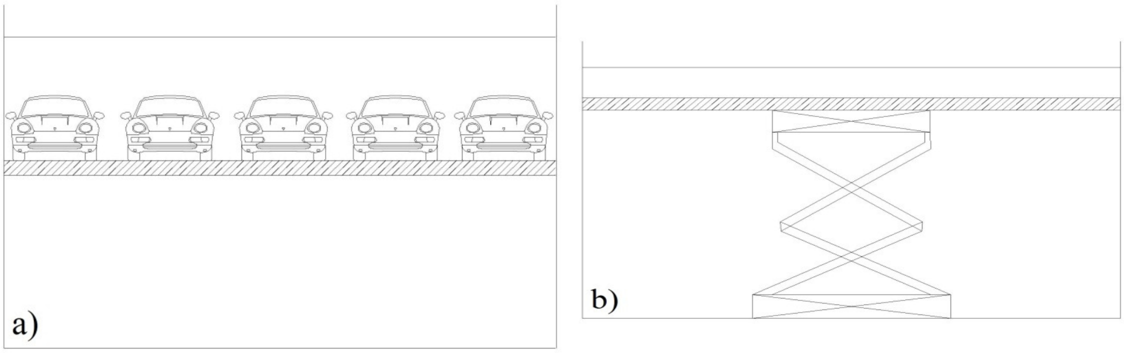

Fig. 6. The panel in (a) seagoing and (b) stowed position lifted by the deck lifter.

3.2. Performance criteria

Criteria for the optimization of different designs, to assess their performance and to compare them with each other, have to be defined. These criteria are presented by either DNV or the client, and were used as constraint functions in the optimization procedure. The definition of the criteria was made with feedback from TTS, and with reference to Eqs (3) and (4); deflection and stresses are inequality constraints, while weight is the objective function. Other factors such as buckling, fatigue life and natural frequency would usually have to be considered in such a study, but have been left out of the scope of this investigation.

3.2.1. Deflection

The requirement for deflection is that when the panel deflects, a certain free height above the below deck has to remain. Therefore, a certain limiting value cannot be assigned directly for deflection; it is, rather, defined according to Fig. 7 such that the sum of the moulded depth (D) and the maximum deflection of the lowest points of the panel (δ), in addition to an error margin of 20 mm should not exceed a certain limit, which is determined as 423.5 mm.

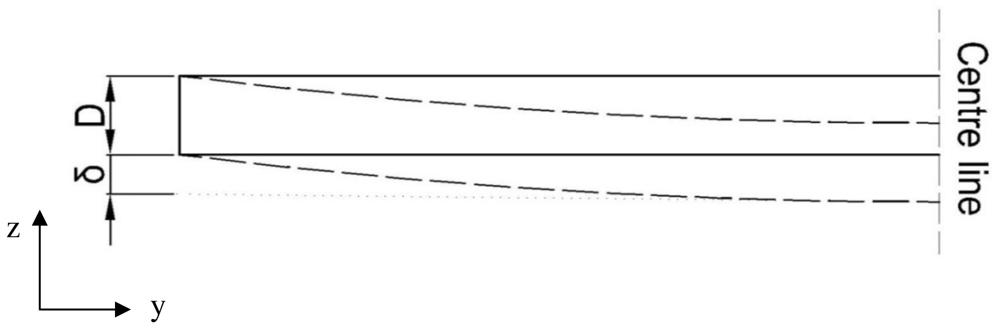

Fig. 7. Moulded depth and deflection of the panel.

There is also a maximum edge deflection criterion which must not exceed 50 mm. It applies to keep the difference in edge heights between two adjacent loaded and unloaded car decks' minimum. This is to ensure the safe passage of cars from one panel to another.

3.2.2. Stress

The maximum stresses that are allowed to occur in the structural elements were calculated according to DNV rules [19] for steel with the properties shown in Table 2. The corresponding values for the two load cases are given in Table 3, where σ_x, τ_{xz} and σ_{vM} denote normal, shear and von Mises stresses, respectively.

Table 2. Material properties of constructional steel used for this study.

Density (kg/m^3)	7850
Young's modulus (MPa)	210000
Poisson's ratio	0.3
Yield stress (MPa)	355

Table 3. Maximum allowable stresses with regards to load conditions.

	Load Case I (Seagoing)	**Load Case II (Harbour)**
σ_x (MPa)	222.4	250.2
τ_{xz} (MPa)	125.1	139
σ_{vM} (MPa)	250.2	278

4. Reference car deck panel – Concept A

In this section the reference car deck panel, its geometrical properties and performance are introduced. This concept is currently provided by TTS to be used for the panels previously shown in Fig. 4. The first objective in this study was the optimization of this concept, to be used as a reference in evaluating alternative solutions. In the following sections, parameterization followed by the FE analysis to define its initial performance is presented. And, finally, the results from the parametric sensitivity analysis that was carried out to define the key parameters are shown.

4.1. Description of the geometry

The geometry of the reference structure, shown in Fig. 8, is rather conventional with 4 longitudinal and 4 transverse beams. It is symmetric with respect to its centre line parallel to the x-axis, which allowed half of the panel to be modelled. In this way, the computational time could be reduced by half. The two transverse beams near the edges (T2 and its "twin" on the opposite edge) extend from one edge to the other. The two in the middle (T8 and its "twin") extend from one longitudinal beam in the middle (L4) to the other (not shown in the figure due to symmetry). In addition to these main beams, the top plate is transversely stiffened with alternating "C" and "L" type profiles. Since two dimensional elements were used, the top flanges of the C type profiles were neglected and they all appear as L type profiles in the figure. The dimensions of these structural elements are presented in Table 4.

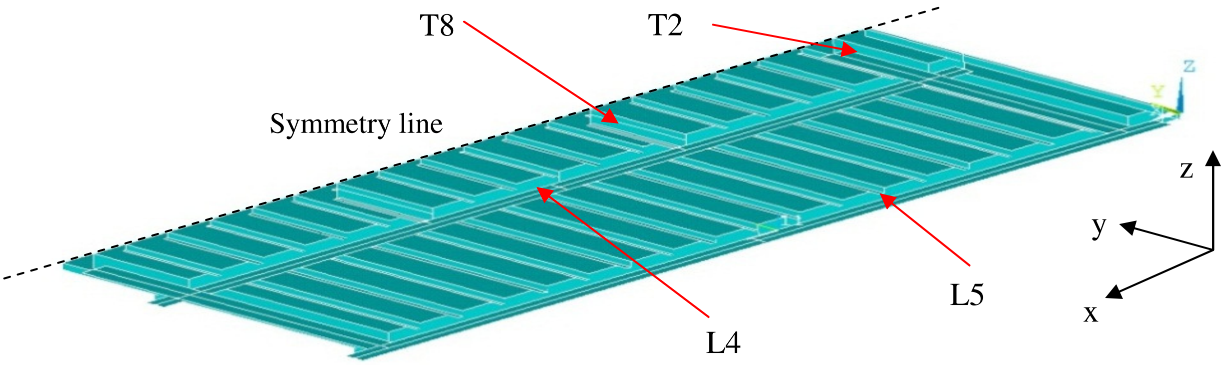

Fig. 8. Half-modelled reference car deck symmetric with respect to the x-axis, which is along the length of the ship in the global coordinate axis.

Table 4. Dimensions of the structural elements of the reference car deck panel.

Name of structural element	Dimensions (mm)
L4 and L5	274 x 6 x 320 x 20
T2	274 x 6 x 425 x 20
T8	274 x 6 x 120 x 20
L type profile	100 x 75 x 7
C type profile	100 x 50 x 7.5

4.2. Parameterization of the geometry

ANSYS Parametric Design Language (APDL) [14] was used to carry out the FE analysis of the panel. With this language, the dimensions of the panel and its structural elements can be defined as parameters in the pre-processing stage, which were used by ANSYS Workbench as input parameters for the goal-driven optimization as described in Fig. 1.

Table 5 shows the list of parameters defined in the modelling stage. These parameters were later used in the parametric sensitivity analysis to determine the ones with higher influence, which were consequently used in the optimization. The parameters defining the geometry and the position of the middle beam are displayed as an example in Fig. 9.

Furthermore, unique parameters were assigned to similar parts of stiffeners. For instance, "TW" is the only parameter for the flange thickness of all beams and they cannot vary independently. This benefits the manufacturing process as product variation is decreased and the quantity of the products to be purchased is increased.

Table 5. Definition and initial values of parameters.

Name	Definition	Original value (mm)
D2	Longitudinal distance from plate edge to T2	615
D4	Transverse distance from plate edge to L4	4035
D5	Transverse distance from plate edge to L5	385
HW	Web height of Beams	274
HWST	Web height of secondary stiffeners	100
TTP	Top plate thickness	6
TW_C	Web thickness of C-profiles	7.5
TW_L	Web thickness of L-profiles	7
TW	Web thickness of beams	6
TF	Flange thickness of beams	20
TF_C	Flange thickness of C-profiles	7.5
TF_L	Flange thickness of L-profiles	7
WF_C	Flange width of C-profiles	50
WF_L	Flange width of L-profiles	75
WF45	Flange width of L4and5	320
WF2	Flange width of L2	425
WF8	Flange width of T8	120

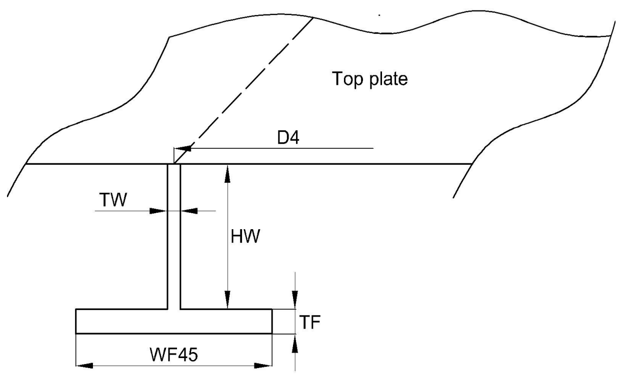

Fig. 9. Parameterization of the middle beam.

4.3. Boundary conditions

A symmetry boundary condition was applied to the symmetry plane nodes of the structure since half of the structure was modelled.

For load case I, the nodes in the corners (intersection line of web of edge beams) are fixed in the z direction. A single node at the centre of the panel (at the symmetry line) is fixed in the x and y directions and no rotation around the z axis to prevent rigid body motion. The boundary conditions applied on the model are displayed in Fig. 10. It should be noted that for the first load case fixing the translation of more than one node would slightly over-constrain the model. The nodes on the intersection line of the edge beams also have a small translation in the z direction as the plate bends. However, this effect is negligible and even necessary for avoiding high-stress estimation at the constraints nodes.

For load case II, deflection of the panel leads to four point contacts where the two short middle transverse beams in the middle touch the edge of the lifter platform. So, the same fixation is applied not to the corners of the panel, but to the nodes in those points.

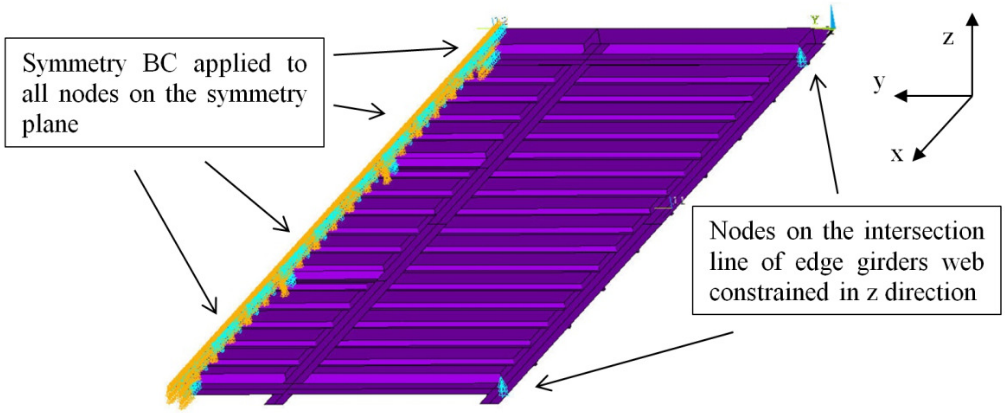

Fig. 10. Boundary conditions of the car deck panel for the load case I.

4.4. Mesh creation

Shell elements are suitable types of elements for the modelling of structures made of thin plates. In this type of elements, the assumption of plane-stress condition simplifies the calculation [10], which is the case for car deck panels. This allows two-dimensional elements to be created where the plate thickness is considered constant.

The geometry of the car deck allows using quadratic 4-node shell elements (no midside nodes) to create a fine mesh. For structural analyses, these corner node elements with extra shape functions will often yield an accurate solution in a reasonable amount of computer time [14]. However, to ensure accuracy in the mesh convergence analysis, 8-node shell elements were selected when using bigger elements. As the element size became smaller they were switched to 4-node elements which are equivalent to an 8-node element twice their size. Both types of elements have six degrees of freedom.

Full integration is used, which allows the shell element to use the method of incompatible modes to improve the accuracy in bending-dominated problems. It only requires one element through the thickness [10].

4.5. Analysis

Static analysis was considered for evaluation and comparison of output parameters where the UDL was applied as a constant load. Edge deflection was extracted from the top plate elements at the edges; see Fig. 11, while the deflection δ is the deflection at the lowest part of the panel (flanges, see Fig. 12). The maximum normal stress component σ_x, shear stress τ_{xz} and von Misses stress σ_{vM} were extracted from the main beams where the maximum values were expected to occur, see Fig. 13. This is obviously the middle area of the beams between supports. The highest values among these were determined to be the maximum stresses occurring in the structure. In this way, the areas with sharp edges with high stress concentration factors arising from coarse mesh were excluded.

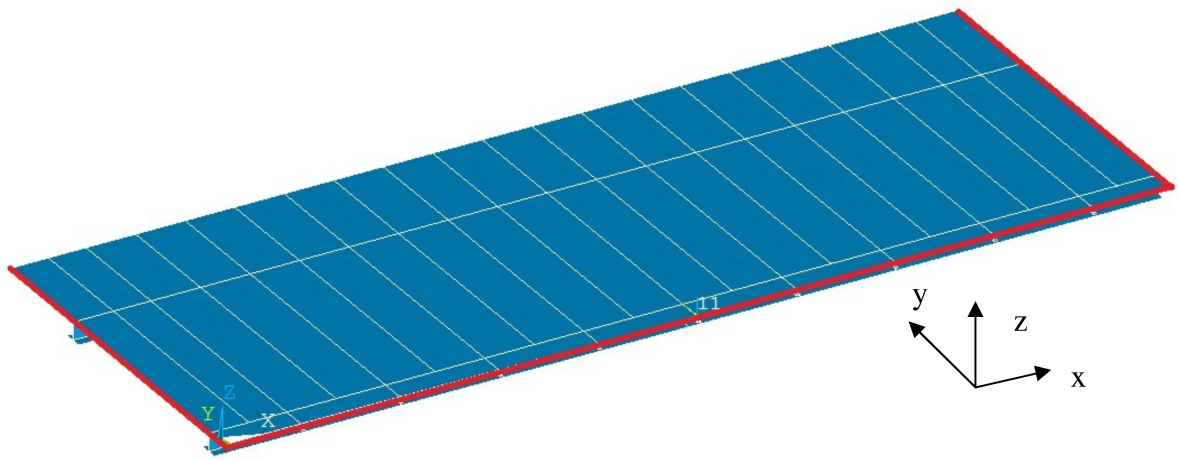

Fig. 11. "Edge Deflection" is measured along the edges marked in red.

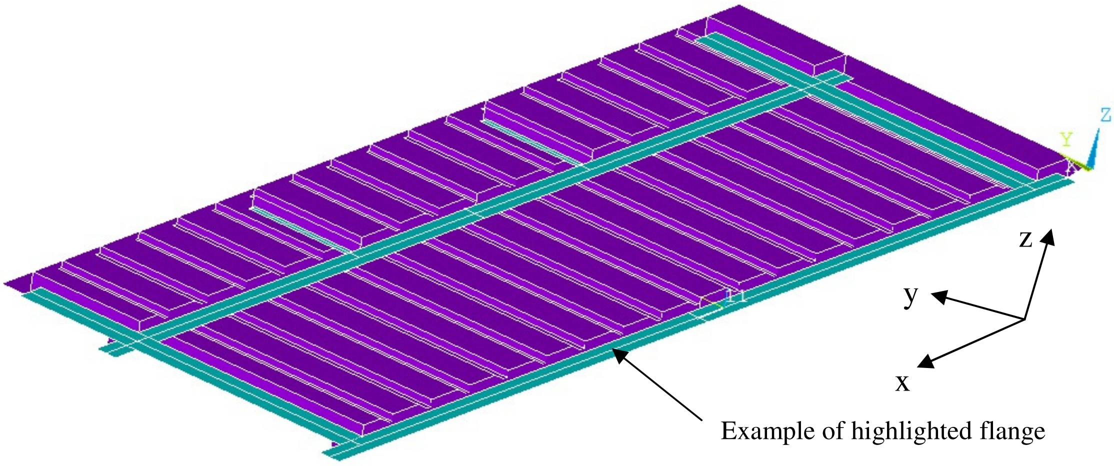

Fig. 12. The value of δ is measured at the highlighted flanges.

Fig. 13. The parts of the panel where the stresses are recorded are circled in red.

4.6. Mesh convergence

Figure 14 shows the mesh convergence analysis obtained by plotting the maximum von Misses stress value as well as the deflection of the lower part of the panel against various element sizes. As can be seen from the figure, the results have a minor change with a 75 mm element size compared to 90, which was the selected element size for the analysis. The model had 11,545 elements with this element size.

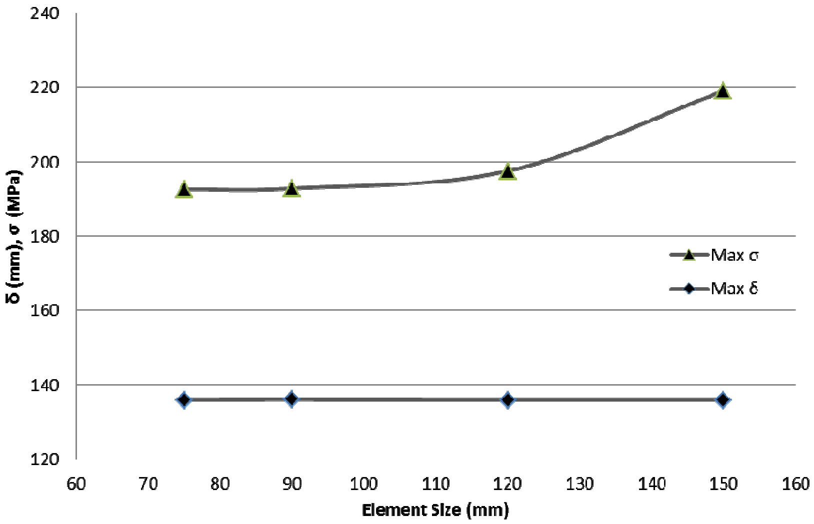

Fig. 14. Mesh convergence analysis for maximum σ_{vM} and δ.

4.7. Performance

The performance of the reference car deck panel and the criteria to be met is given below in Table 6. Note that the 20 mm error margin, which was mentioned in section 3.2.1 for "building depth", has been subtracted from the criterion instead of adding to the result obtained from the analysis. It can be seen that all criteria concerning deflections and stresses are met. The question is now whether a lower weight can be achieved while still satisfying the criteria defined in Section 3.2.

Table 6. Performance of car deck panel A and criteria to be met.

	Load Case I	**Criteria limits**	**Load Case II**	**Criteria limits**
Edge deflection (mm)	49.8	50	-	N/A
Building depth (ttp+hw+tf+δ) (mm)	390	403.5	-	N/A
σ_x (MPa)	184.6	222.4	90	250.2
τ_{xz} (MPa)	61.5	125.1	19.6	139
σ_{vM} (MPa)	218	250.2	124.9	278

4.8. Results from the parametric sensitivity analysis

The optimization of the car deck panels with respect to both of the load cases resulted in two completely different models. This is because of the fact that the two load cases have different boundary conditions, loads and performance criteria. Therefore, a decision had to be made on which load case to choose for the optimization. Looking at the results in Table 6, it was concluded that the structure does not need to be optimized for the second load case, since it satisfies all criteria with a large safety margin. It was noted; however, that should an optimal design be achieved for load case I, the analysis for load case II with the new dimensions would have to be run to make sure that the new structure has a satisfactory performance for this load case as well. The following figures (Fig. 15 - Fig. 18) show the sensitivity of each output parameter plotted versus every input parameter (as was presented in Section 2.2) for load case I.

At a glance at the figures, it can easily be concluded that the secondary stiffeners have a very small impact on the output parameters. The significant impact that the web height "HW" has on building depth, edge deflection and stress, compared to a relatively small increase in weight, is distinctive as the web height proportionally shifts the neutral line of the panel. The opposite trend is shown by the top plate thickness "TTP", which could drastically increase the weight with small changes in panel behaviour.

The important role that the position of the middle beams (D4) plays in changing the stress, but not to the same degree as any other objective's value, is also considerable. The conclusion could be that the maximum stress occurs on the longitudinal edge beam. By positioning the middle beam (L4, see Fig. 8) closer to the edge, the stress in the edge longitudinal is lowered. However, as it moves far towards the centre of the panel the load area that must be carried by the edge longitudinal increases and this results in a higher stress condition. The small influence that occurs in weight is caused by the change in the supported length (T8) for the deck lifter to which the middle beam is connected. Hence, it is reasonable to only select the parameters defining the geometry of the primary stiffeners and their positions to be optimized.

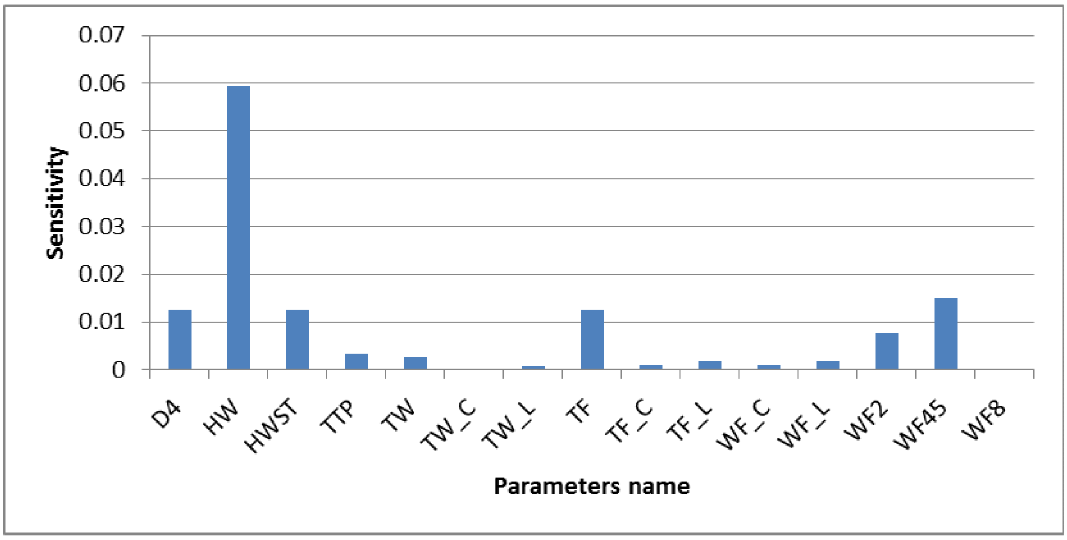

Fig. 15. Sensitivity of building depth to input parameters.

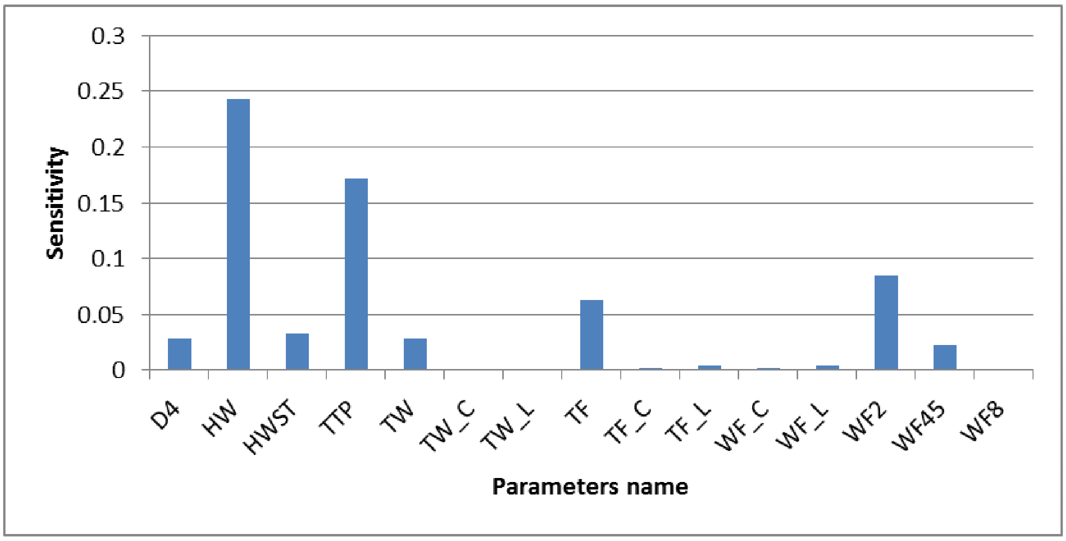

Fig. 16. Sensitivity of edge deflection to input parameters.

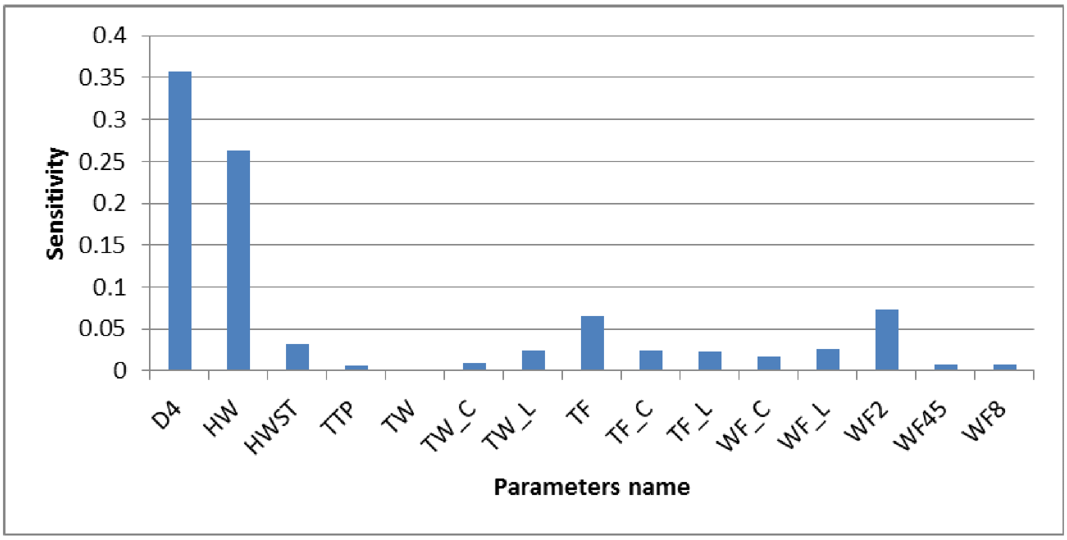

Fig. 17. Sensitivity of maximum von Misses stress to input parameters.

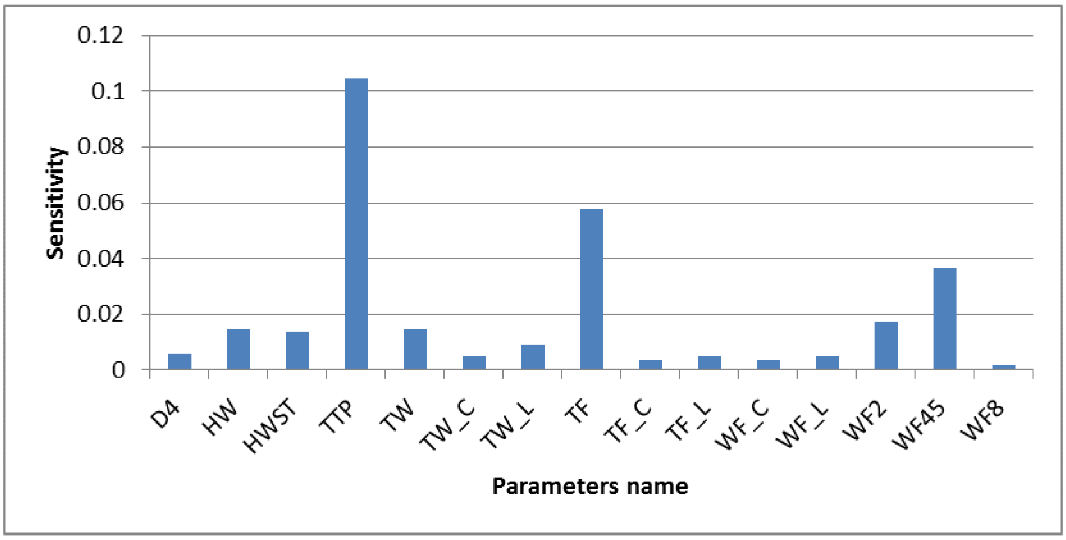

Fig. 18. Sensitivity of weight to input parameters.

Upper and lower limits were given for theses parameters as shown in Table 7, thus defining the side constraints according to Eq. (5). Values were chosen in order to keep the dimensions reasonable from manufacturing and operational points of view. As was mentioned earlier, the top plate thickness has a significant impact on the panel weight. However, it was kept 6 mm in all analyses as required from TTS Marine in order to avoid thermal residual stresses causing plate deflection during welding.

Table 7. Parameters of car deck concept A selected for optimization.

Parameter	Original value (mm)	Lower limit (mm)	Upper limit (mm)
D4	4035	2835	4310
HW	274	150	300
TF	20	10	40
TW	6	4	12
WF45	320	200	500
WF2	425	200	600

5. Concept B

In this section, the same calculations and optimization procedure for Concept A are made to evaluate the performance of another car deck, "Concept B". The results of their optimizations are presented in Section 6. This concept and its design were selected together with TTS. It had previously been supplied by TTS, but has later been replaced by concept A. As can be seen, it is stiffened by 6 longitudinal and 6 transversal beams as shown in the Fig. 19, where they have been reduced to four longitudinal beams and four transverse beams (including the deck lifter supports) in Concept A.

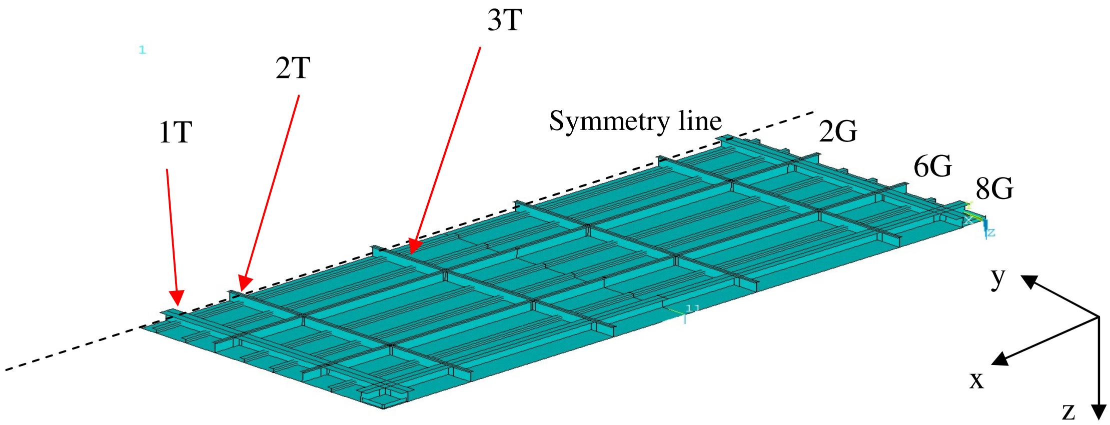

Fig. 19. Half-modelled car deck Concept B symmetric with respect to the x-axis.

Concept A is considered as an improved design of Concept B. The changes made could be explained as follows:

(1) The function of the transverse beams is mostly to provide support to secondary stiffeners. Hence, by positioning the secondary stiffeners transversally, which was the case in Concept A, they are no longer of importance.
(2) Part of the load carried by 2T and 3T is transferred to the constraint points through 8G. So, by taking them away, the load will be taken on by 1T through 2G and 6G instead. This improves the maximum edge deflection that usually occurs at 8G, which is the one with the longest span, making it the most loaded one.

The same load cases, criteria and boundary conditions that were considered for Concept A apply to Concept B as well. Table 8 shows the performance of this panel before optimization is carried out. As can be seen, the edge deflection and maximum normal stress values exceed the criteria limits.

Table 8. Performance of car deck panel Concept B. Values exceeding the criteria are bold.

	Load Case I	Criteria limits	Load Case II	Criteria limits
Edge deflection (mm)	**68**	50	-	N/A
Building depth (ttp+hw+tf+δ) (mm)	382	403	-	N/A
σ_x (MPa)	**228**	222	18	250
τ_{xz} (MPa)	9	125	2	139
σ_{vM} (MPa)	227	250	61	278
Weight (t)	15.4	-	15.4	-

5.1. Parameterization

The following parameters listed in Table 9 define the overall geometry of a car deck model to be optimized. As can be seen, the scantlings of the stiffeners as well as the positioning of the primary beams are considered.

Table 9. Parameterization of the dimensions of Concept B. "Original value" indicates the initial (existing) geometry of the car deck panel.

Parameter	Definition	Original value (mm)
PLL	Half of plate length	7185
PLW	Half of plate width	5785
D_1T	Distance from plate edge to 1 T	585
D_2T	Distance from plate edge to 2 T	2085
D_3T	Distance from plate edge to 3 T	5485
D_2G	Distance from plate edge to 2G	4735
D_6G	Distance from plate edge to 6G	1935
D_7L	Distance from plate edge to first longitudinal stiffener	1235
D_8G	Distance from plate edge to 8 G	535
HW	Web height of beams	274
HW_L	Web height of L-profiles	100
TTP	Top plate thickness	6
TW	Web thickness of beams	6
TW_L	Web thickness of L-profiles	8
TF	Flange thickness of Beams	20
TF_L	Flange thickness of L-profiles	8
WF_1T	Flange width of 1 TR	250
WF_23T	Flange width of 2 and 3 TR	100
WF_26G	Flange width of 2 and 6 G	100
WF_8G	Flange width of 8 G	250
WF_L	Flange width of L-profiles	75

5.2. Mesh creation

4-node shell elements were also used for Concept B. Figure 20 shows the mesh convergence analysis. The maximum von Misses stress and deflection are plotted against the element size. The element size created in the model already converges with 150 mm elements. However, an element size of 100 mm was used in the analysis as the most suitable size in order to obtain at least 2 elements along the web of the secondary stiffeners.

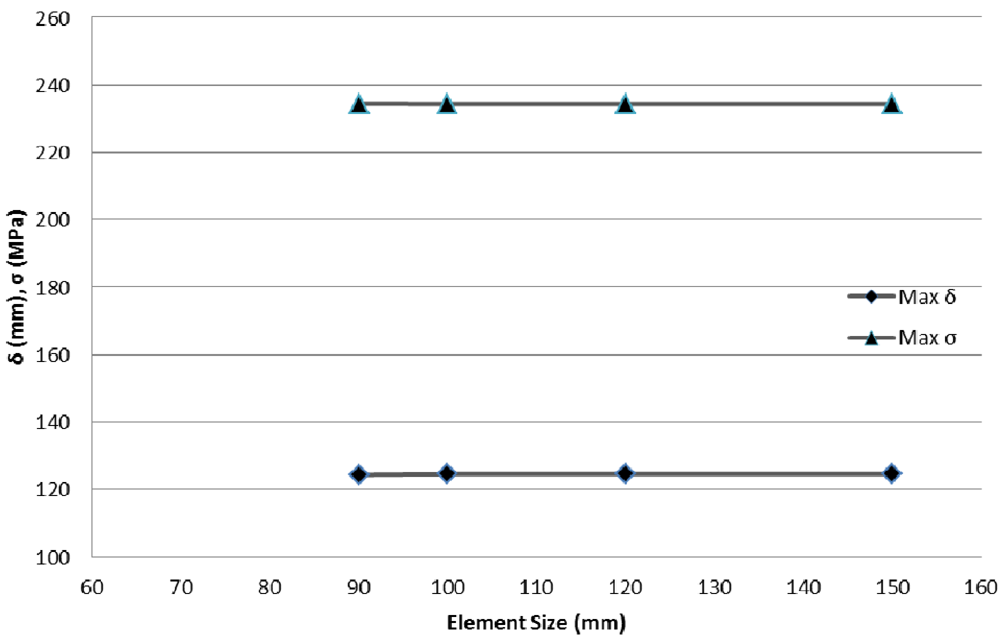

Fig. 20. Mesh convergence analysis for maximum σ_{vM} and δ.

5.3. Results from parametric sensitivity analysis

The parametric sensitivity analyses carried out for the previous concept showed that the dimensions of secondary stiffeners have a negligible impact on the global strength of the panel. They are rather important for the local strength, which is not included in the performance criteria. Hence, parameters could directly be chosen as variables defining the dimensions and positions of the primary beams without a parametric sensitivity analysis.

Parameters to be optimized, their original values and given upper and lower limits are listed in Table 10. These limits were defined to allow the maximum deviation from the original value with the same approach as for the reference car deck panel.

Table 10. Parameterization of the dimension of Concept B.

Parameter	Original value (mm)	Lower limit (mm)	Upper limit (mm)
D_2T	2085	1000	3000
D_3T	5485	4200	7000
HW	274	150	300
TW	6	5	10
TF	20	10	30
WF_26G	100	75	120
WF_8G	250	200	300

6. Optimization

In this section, the optimization results of Concepts A and B are presented. As described in Section 2, a so-called goal-driven optimization [14] has been adopted in order to find the optimum design point. Possible optimum design points are presented as “candidates”. The total number of design points created depends on the number of parameters and the chosen design of experiments, see Fig. 2. It is obvious that with the growing number of design points the calculation time increases. The analysis has to be repeated for each design point in order to obtain the corresponding output parameter value.

ANSYS 13.0 [10] was used as solver using a PC with dual core processor with a frequency of 2.00 GHz. Also, an elapsed time of a maximum of 20 seconds was spent for updating each of the design points. The response surface creation and optimization process could take up to 60 minutes.

6.1. Concept A

The results of the optimization procedure for Concept A are shown in Table 11. It should be noted that the weight in this table (and for all other models) is extracted from the model and therefore different from the real-life weight of the panel, which is 16.2 t as presented in Section 3.1.1. This extracted value is also used as the "self-weight" in all analyses. By looking at the optimum design points, a decreasing trend can be seen for the thickness of the flange and the web, whereas higher values for web height and flange width are reached. A simple comparison between these parameters in a sensitivity analysis (Section 4) explains these changes, i.e. the sensitivity of the building depth is nearly 4 times larger than the sensitivity of the weight to the web height. The performance of the optimized car deck panel for the second load case is checked and the results are given in Table 12. It is shown that the criteria have been successfully fulfilled.

Table 11. Optimization of car deck panel “Concept A” by screening and MOGA methods.

	D4 (mm)	HW (mm)	TW (mm)	TF (mm)	WF2 (m)	WF45 (mm)	Building depth (mm)	Edge deflection (mm)	Max. normal stress (MPa)	Max. shear stress (MPa)	Max. von Mises stress (MPa)	Weight (t)
Objectives and constraints							<403.5	<50	<222.4	<125.1	<250.2	Minimize
Initial values	4035	274	6	20	425	320	390	50	185	62	218	15.0
Optimum design points												
Candidate A	4289	299	4	12	549	484	401	46	171	79	194	14.1
Candidate B	4082	282	8	12	573	414	395	50	216	46	242	14.3
Candidate C	3949	299	5	17	486	348	403	44	189	76	225	14.4

Table 12. The performance of the optimized Concept A (candidate A) for the second load case.

	Max. Normal Stress (MPa)	Max. Shear Stress (MPa)	Max. von Mises Stress (MPa)
Criterion	250.2	139	278
Structure response	70.4	33.7	110

6.2. Concept B

The initial spacing of the longitudinal stiffeners in this model is kept constant, which means that the parameters for the positions of 2G and 6G are not continuous. They can be switched by the secondary stiffeners, which means that their positions will have discrete values, chosen as shown in Table 13. Initial values are distinctive in bold. The presence of discrete parameters dramatically increases the computation time. In this case, the number of design points obtained (by design of experiments) from the continuous parameters is multiplied by the number of all possible combinations of discrete parameters. Moreover, a separate set of parametric sensitivity of the continuous parameters is created for each combination of discrete parameters, rather than a single response surface being created for all parameters. Because of this, the optimization for Concept B has been performed in 2 steps. First, the positions of the beams 6G and 2G were optimized by exchanging their positions by the secondary stiffeners, see Fig. 19. Then the optimization process was carried out by having the optimized position of these longitudinal beams fixed for other parameters.

Table 13. Discrete values used for finding the optimum position of longitudinal beams.

d_2G(mm)	d_6G(mm)
4735	**1935**
4035	1235
3335	2635

Response charts for all output parameters have been created for all 9 possible combinations. Figure 21 shows the response chart for the maximum edge deflection.

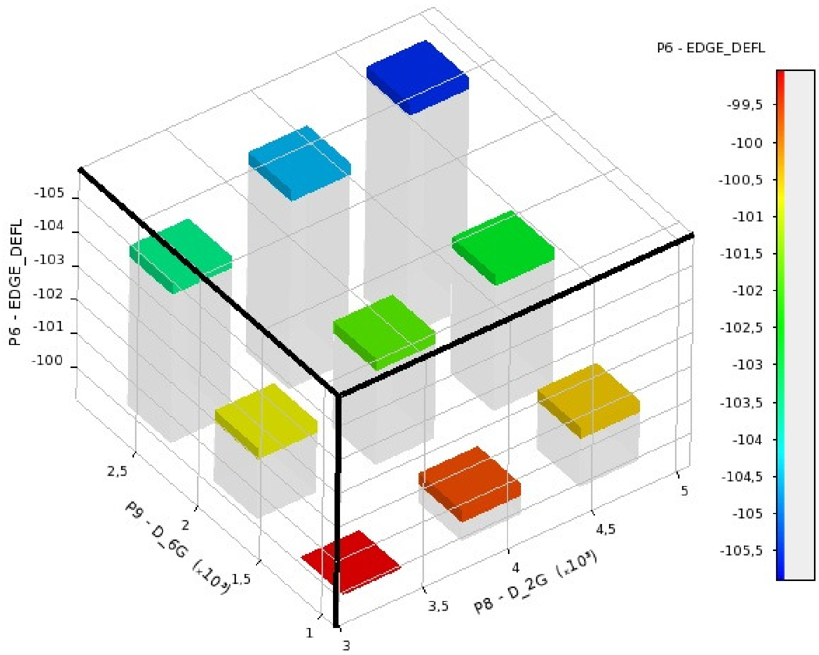

Fig. 21. Response chart for the maximum edge deflection (mm) with a change in d_2G and d_6G discrete values.

Table 14. Optimum positions of longitudinal beams.

	d_2G (mm)	d_6G (mm)	Max. von Mises stress (MPa)	Building depth (MPa)	Edge deflection (MPa)
Constraints			<250.2	<403.5	<50
Initial values	4735	1935	228	382	69
Optimized values	3335	1235	234	383	66

Table 14 shows a better performance with beams closer to the edge of the panel. The chosen design point includes the smallest possible values for each output parameter. The model was updated with the optimized values for the two longitudinal beams and the second step of the optimization was carried out. The maximum allowed edge deflection of the car deck panel is exceeded with the initial parameter values. Table 15 shows the optimization results of an attempt to reach a lighter design fulfilling all the requirements. The optimized design is about 400 kg lighter. The second load case has not been considered due to its smaller impact on the car deck panel compared to the first load case. In other words, criteria will be met for the second load case if fulfilled for the first one. However, beams where the car deck panel is held by the lifter should remain within the lifter platform dimensions.

Optimization of Concept B also shows the same changing trend in parameters as in Concept A where the flange and the web thickness of the beams has been reduced while the web height and the flange width has been increased. On the other hand, a significant change has been made in the positioning of the beams. Fig. 22 shows the optimized car deck panel where the longitudinal beams have been shifted towards the edge of the car deck. The two middle transverse stiffeners have moved more towards the centre and the others towards the edges.

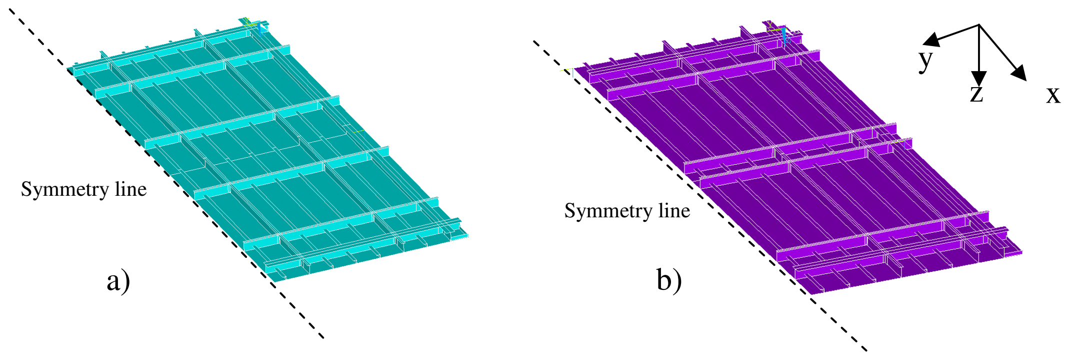

Fig. 22. (a) The original and (b) the optimized car deck panel – Concept B.

Table 15. Optimization results for Concept B with the performance criteria.

	D_2T (mm)	D_3T (mm)	HW (mm)	TF (mm)	TW (mm)	WF_1T (mm)	WF_23T (mm)	WF_26G (mm)	WF_8G (mm)	Building depth (mm)	Edge deflection (mm)	Max. normal stress (MPa)	Max. shear stress (MPa)	Max. von Mises stress (MPa)	Weight (t)
Objectives and constraints										<403.5	<50	<222.4	<125.1	<250.2	Minimize
Initial values	2085	5485	274	20	6	300	100	100	250	383	66	234	9	234	15.4
Optimum design points															
Candidate A	1367	6528	297	13	6	366	84	194	316	399	49	199	17	227	15.0
Candidate B	1448	4234	296	15	5	337	99	171	381	392	49	204	14	239	15.1
Candidate C	2040	5711	290	17	4	382	71	199	352	387	48	167	10	207	15.4
Candidate D	1546	6095	291	29	5	275	54	69	234	402	49	219	13	242	15.7

7. Development of new concepts

This section presents two car deck panel structures that were developed as possible alternative solutions to the first two conventional concepts. First, Concept B that was presented in Section 5 was further improved and a new concept with a much simpler configuration was obtained. The second concept is a panel with diagonally positioned beams. Both new concepts were optimized with the same procedure as described in the previous sections, based on the load cases and performance criteria presented in Section 3. It should be noted that several assumptions and simplifications were made based on the experience gained from the first two concepts.

By assuming that as long as the conventional grillage configuration is not changed, the secondary stiffeners can be kept the same as for the reference panel and do not have to be included in the parametric study. This is because of the same or close values of unsupported length for the stiffeners, which is the governing factor for the required scantlings.

Designing a car deck by assigning unique dimensions to different parts would entail simplicity and productivity. However, this could have a negative impact in restricting the optimization process. This is caused by the fact that the stress and strain condition varies over the plate, and the beams in different positions have different functionalities. By excluding the secondary stiffeners from the analysis, their defining parameters could be replaced by allowing the beams at different parts of the panel to have different dimensions. This is to allow the dimensions of different beams to be defined as different parameters instead of the case with the first two concepts where all the beams had the same web thickness and all the longitudinal and transverse beams had the same flange width, etc. One should still keep in mind that this could slightly increase the manufacturing costs, especially if a small number of car decks are to be manufactured.

Since the overall dimensions of the panel are kept constant, it is reasonable to assume with an element size smaller than or equal to the reference concept, and having two elements along the transverse beams, a mesh convergence analysis is not necessary.

7.1. Development of Concept B (B-II and B-III)

By looking at the optimized design in Concept B, Fig. 22, one can obtain a simpler configuration by combining the three pairs of transverse beams standing close to each other to form single beams. The longitudinal beam in the middle will also have to be carried to its original position, since having just one transverse will not be enough to keep the panel steady when lifted by the deck lifter. The panel in this concept will also be transversely stiffened, with the stiffeners having the same dimensions and spacing as described above. The two adjacent longitudinal beams at the edge are now connected with a flange, forming a box beam and the panel is also supported by a centre transverse beam as seen in Fig. 23. Positioning a box shape (closed cross section) beam at the edge could be beneficial as it resists the torsional moment of the panel. This arrangement is referred to as Concept B-II.

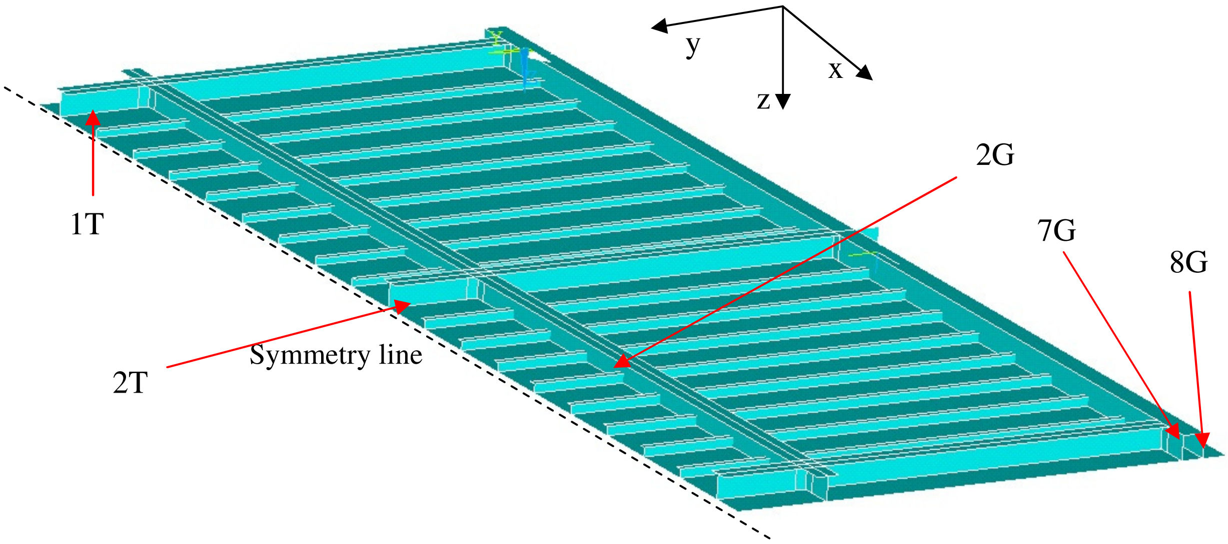

Fig. 23. Geometry of Concept B-II.

The list of dimensions used as input parameters in the optimization is shown in Table 16. The scantlings of transverse stiffeners are the same as for Concept A, and for the sake of simplicity all beams have the same initial dimensions and range.

Table 16. Input parameters for the optimization of Concept B-II.

Parameter	Definition	Initial value (mm)	Lower limit (mm)	Upper limit (mm)
D_8G	Distance from plate edge to box beam (8G)	285	200	400
D_7G	Distance from plate edge to box beam (7G)	535	500	900
HW	Web height of beams	274	204	400
T_B	Web and flange thickness of box beam	10	4	20
TW_1T	Web thickness of 1 TR	6	4	20
TW_2T	Web thickness of 2 TR	6	4	20
TW_2G	Web thickness of 2 G	6	4	20
TF_1T	Flange thickness of 1 TR	20	10	40
TF_2T	Flange thickness of 2 TR	20	10	40
TF_2G	Flange thickness of 2G	20	10	40
WF_1T	Flange width of 1 TR and its symmetric	250	200	600
WF_2T	Flange width of 2 TR	250	200	600
WF_2G	Flange width of 2G	250	200	600
WF_B	Flange width of box beam	250	200	600

Optimization results have shown no better solution than the initial design. The reason could be the existence of the middle transverse beam. As can be seen from Table 17, the building depth of the initial design is close to its limit, which restricts the panel from becoming more deflected. So by taking away the middle beam, the point of maximum deflection in the flanges is carried away from the centre point, allowing a bigger margin for global deflection with the same building depth. In this way, Concept B-II could be developed further (Concept B-III), which looks the same as for Concept A except for using a closed cross section box-shaped beam at the edges and without the short beams in the middle for the deck lifter, instead of which the longitudinal beam has been moved to its original position.

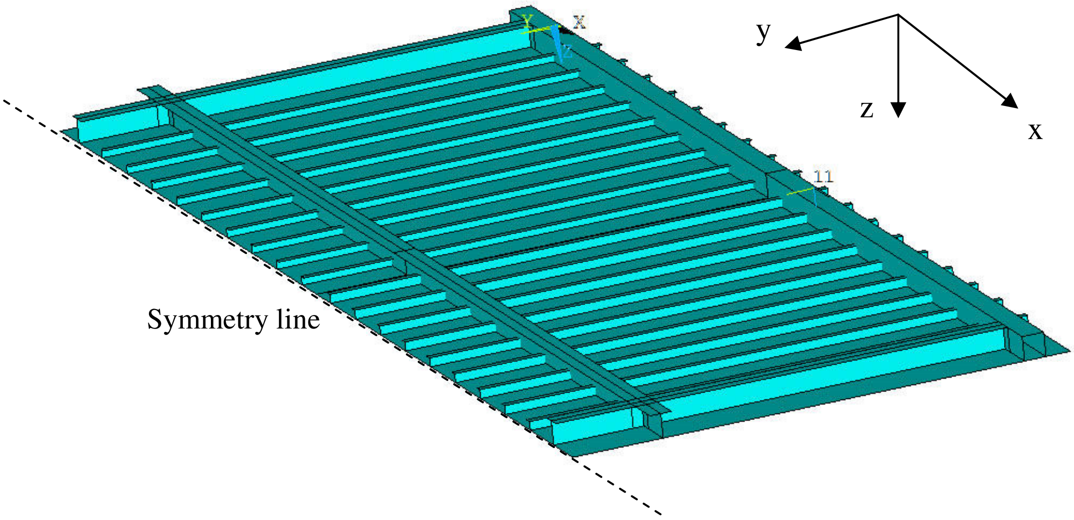

Fig. 24. Geometry of Concept B-III

Results from the optimization of Concept B-III in Table 18 refer to the expected improvement compared to Concept B-II. It has approximately the same weight and performance as the optimized Concept B while obtaining a simpler geometry.

Table 17. Performance of Concept B-II with initial values.

	Building depth (mm)	Edge deflection (mm)	Max. normal stress (MPa)	Max. shear stress (MPa)	Max. von Mises stress (Mpa)	Weight (t)
Objectives and constraints	<403.5	<50	<222.4	<125.1	<250.2	Minimize
Initial values	375	41	187	54	184	16.63

Table 18. Optimization results for Concept B-III with the performance criteria.

	WF_B (mm)	WF_2G (mm)	WF_1T (mm)	TF_1T (mm)	TF_2G (mm)	T_B (mm)	TW_1T (mm)	TW_2G (mm)	HW (mm)	Building depth (mm)	Edge deflection (mm)	Max. normal stress (MPa)	Max. shear stress (MPa)	Max. von Mises stress (MPa)	Weight (t)
Objectives and constraints										<403.5	<50	<222.4	<125.1	<250.2	Minimize
Initial values	250	250	250	20	20	6	6	6	274	418	89	424	59	419	13.8
Optimum design points															
Candidate A	397	260	207	29	13	14	4	6	258	386	44	206	61	227	15.0
Candidate B	397	377	316	16	12	11	8	14	300	402	41	199	47	220	15.1
Candidate C	368	305	445	19	11	12	6	9	279	392	45	219	56	207	15.2

7.2. Concept C

This concept has been developed with two diagonal beams and rectangular frames creating a shape of spider web, as seen in Fig. 25. Utilizing diagonally positioned beams has never been common in the marine industry. By positioning beams along the diagonal, the span is increased, which leads to larger dimensions and consequently to heavier structures. However, an unusual boundary condition is considered in this study, in which the panels are supported at their corners only. Thus, transferring the loads directly to the support points by crossed beams would decrease the load on the edge beams and help fulfil the edge deflection criterion. Furthermore, it seems it would strengthen the support points which are significant to reducing the deflections. This is also beneficial for the second load case since the support points are moved towards the centre. Equally spaced frames were introduced to the geometry; see Fig. 25. The quantity of frames is optimized at first by setting the number of frames as the only variable n. Secondary stiffeners have been placed between the transverse beams depending on the number of frames.

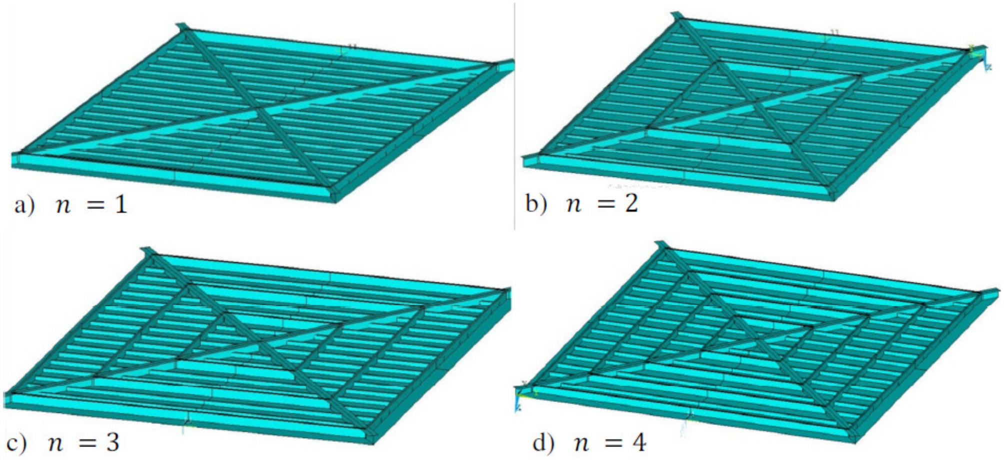

Fig. 25. Panels with crossed beams with a varying number of transverse and longitudinal beams (n=1, 2, 3 and 4).

The car deck panels have been designed according to DNV rules in order to be used as the starting design point. The beams are designed to satisfy global strength, using the UDL as the design load according to DNV [19]. The scantling requirements for the beam at the edge were the largest, since it has the longest unsupported length and the scantlings for all other beams were determined as being the same as this one. The required cross-sectional properties are shown in Table 19. Keeping the thicknesses the same as Concept A, the selected web height and flange width of the beams are shown in Table 20, along with the resulting values for cross-sectional properties for a comparison with required section properties of the beams.

Table 19. Required sectional modulus (Z), cross-sectional area (A) and moment of inertia (I) of the edge longitudinal beam for three different cases.

n	Z (cm3)	A (cm2)	I (cm4)
2	663	3.8	13551
3	442	2.5	9034
4	332	1.9	6776

Table 20. The selected web height and flange.

n	HW (mm)	WF (mm)	Z (cm3)	A (cm2)	I (cm4)
2	334	100	679	22	20479
3	294	90	567	19	14592
4	264	80	450	17	10169

The scantlings for secondary stiffeners were also calculated according to DNV Rules [19]. The outermost stiffener with the largest unsupported length is the worst case, and the required section modulus for all the cases was calculated as 117 cm^3 (this value is the same for all cases, since they all have the same unsupported length). An L-section of 130 x 65 mm with a thickness of 10 mm, which has a section modulus of 125 cm^3, is selected.

The analysis results obtaining the maximum von Mises stress, the total and the edge deflection and the weight with the number of frame beams n = 1, 2, 3 and 4 equally spaced along the plate are listed in Table 21. n = 1 indicates that only the edge beams are used (Fig. 25a). It is obvious that n = 2 (Fig. 25b) is the optimum number of beam frames to be used.

Table 21. Comparison of crossed-beam car decks with a different numbers of frames.

n	Max. Von Misses (MPa)	Total deflection (mm)	Edge deflection (mm)	Weight (t)
1	266	95	72	13.9
2	**234**	**85**	**67**	**14.8**
3	268	179	87	15.9
4	291	124	106	15.7

Figure 26 shows the selected configuration for Concept C. The frame mainly helps to transfer the loads to the main beams. The transverse beams of the frame are parallel to the secondary stiffeners, while acquiring a much shorter unsupported length. So, the stronger contribution of the secondary stiffeners renders the transversal beams of the frame unimportant and they could be excluded.

Table 22 shows the related parameters and their initial values. To avoid difficulties in automatic mesh creation, the web height of all beams is kept as a unique parameter.

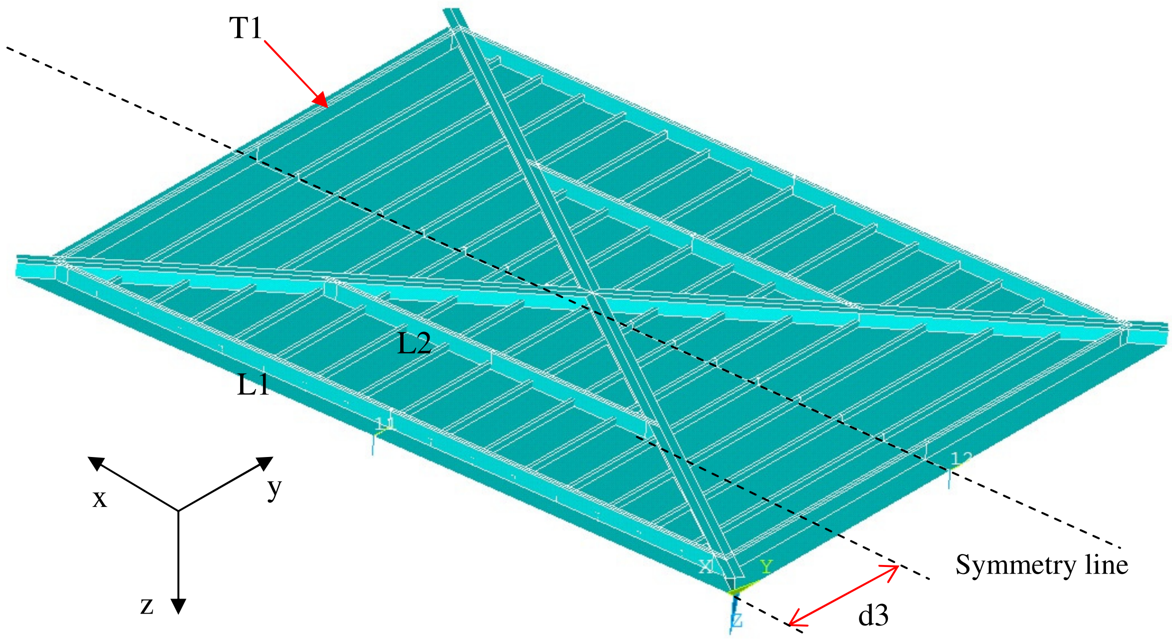

Fig. 26. Car deck panel Concept C with 2 longitudinal beams between the cross-beams.

Table 22. Parameterization of the dimension of car deck with crossed beams.

Parameter	Definition	Initial value (mm)	Lower limit (mm)	Upper limit (mm)
WF_L1	Flange width of edge longitudinal beams	100	50	250
WF_L2	Flange width of middle beams	100	50	250
WF_T1	Flange width of edge transverse beams	100	50	250
WF_C	Flange width of the crossed beams	300	200	500
TW_L1	Web thickness of edge longitudinal beams	6	4	15
TW_L2	Web thickness of edge middle beams	6	4	15
TW_T1	Web thickness of edge transverse beams	6	4	15
TW_C	Web thickness of the crossed beams	6	4	15
TF_L1	Flange thickness of edge longitudinal beams	20	10	30
TF_L2	Flange thickness of edge middle beams	20	10	30
TF_T1	Flange thickness of edge transverse beams	20	10	30
TF_C	Flange thickness of the crossed beams	30	20	40
HW	Web height of beams	334	265	400
D3	Longitudinal distance from the plate edge to middle beams	2700	1700	3700

The optimization of Concept C has entailed no solution with the applied constraints and objectives within the given limits. This means that no combination of parameters could satisfy the criteria. In other words, the entire response surface of the car deck weight is covered by applied constraint values. Table 23 shows an attempt to find the closest points to constraint borders by setting constraint boundaries as "goals" rather than as "hard constraints", obtained by both screening and MOGA methods.

Table 23. Optimization of car deck with crossed beams.

	D3 (mm)	HW (mm)	TF_C (mm)	TF_L1 (mm)	TF_L2 (mm)	TF_T1 (mm)	TW_C (mm)	TW_L1 (mm)	TW_L2 (mm)	TW_T1 (mm)	WF_C (mm)	WF_L1 (mm)	WF_L2 (mm)	WF_T1 (mm)	Building depth (mm)	Edge deflection (mm)	Max.normal stress (MPa)	Max. shear stress (MPa)	Max. von Mises stress (MPa)	Weight (t)
Objectives and constraints															<403.5	<50	<222.4	<125.1	<250.2	Minimize
Initial values	2700	334	30	20	20	20	6	6	6	6	300	100	100	100	462	67	233	59	233	15.4
Optimum design points																				
Candidate A	2598	274	25	24	15	10	7	4	7	6	431	209	111	146	420	74	185	52	212	16.0
Candidate B	2041	276	22	27	21	28	5	7	12	5	428	248	62	162	413	73	176	81	207	16.7
Candidate C	1729	270	21	28	24	12	6	9	11	7	491	180	229	153	412	70	159	73	205	17.0
Candidate D	2013	276	23	27	21	14	5	7	9	5	377	248	62	162	422	73	174	73	229	16.0

8. Discussion

Optimization results highly depend on the geometry, the applied loads and the boundary conditions of a car deck panel. The number of parameters and their varying impact makes the structure's adaptation to change unpredictable. Therefore, a new set of analyses is required if the working condition of the car deck or its geometry is changed.

The quality of geometry codes allowing for more flexibility in parametric study plays a significant role in ensuring the progress towards an optimum solution. It is a challenging task to create a code with parameters, such as the number and the spacing of stiffeners as well as the scantlings while controlling an appropriate mesh to be created for all design points.

Limited time in this project resulted in a step-wise optimization procedure (as in Concepts B and C). Hence, either the number or the spacing of stiffeners was optimized first. The optimum solution was then used as a constant parameter further on in the investigation. This, however, could raise the question of whether the first obtained solution would remain the optimum one in the second-run optimization.

There is a balance that has to be maintained in the parametric sensitivity analysis regarding the number of input parameters. Since different parameters have conflicting influences on the objective and constraint functions, using too many of them is not likely to end in obtaining satisfactory results. Too few input parameters, on the other hand, will constrain the optimization process and limit the reduction in the objective function.

Table 24 shows the comparison of the performance of different concepts. It should be noted that the weight reduction of Concept B-III is calculated with respect to the original weight of Concept B.

Table 24. Comparison of different concepts after optimization.

	Max. von Mises Stress (MPa)	**Building depth (MPa)**	**Edge deflection (mm)**	**Weight (t)**	**% Reduction in weight**
Objectives	<250.2	<403.5	<50	Minimize	
Concept					
A	193	400	46	14.1	6
B	226	399	49	15	2.5
B-III	227	386	44	15.1	2
C	211	419	74	16	-4.4

It is clear that Concept A is the best concept. This concept provides the lightest solution while satisfying the criteria. The greatest reduction in weight has also been achieved for this concept. Even a further reduction can be achieved if manufacturing concerns were set aside and different beams were allowed to have different scantlings.

The top plate of the panel covers a large area. A small reduction in plate thickness leads to a noticeable reduction in the weight of the panel. However, it was kept constant in this project in order to avoid thermal residual stresses. It is nonetheless possible that with developing welding technology and skilled welders this problem could be overcome.

Optimization of Concept B led to the structural arrangement in B-III that is very similar to Concept A. B-III is only marginally heavier than Concept B, but has a much simpler geometry, making it much easier and cheaper to produce. Furthermore, when produced in reality, B-III is likely to end up lighter than B, since it requires less welding and paint. Since Concept A had in fact been obtained by developing Concept B, there is good agreement between the methodology used in this project and the logic behind the improvements made to Concept B in the industry.

The criterion "Building Depth" is obtained by adding the moulded depth of the panel to the maximum deflection that occurs at the flanges in the middle, where the global deflection is also at a maximum. Thus, a heavier structure is needed to lower the deflection in the middle point of the panel to reduce the "building depth" as was the case for Concepts B-II and C. Hence, positioning beams crossing the centre of the panel is not to be recommended. Diagonally positioning the main beams involves a further disadvantage as it increases the load area that they need to support.

9. Conclusions

The FE method has been utilized to model and assess a reference car deck with respect to design criteria such as the normal, shear and von Mises stresses occurring in the structure; maintaining the required free height above the fixed deck below, and the deflection occurring along the panel's edge. The structure was weight-optimized by means of numerical optimization with certain dimensions defined as parameters, whose upper and lower limits are influenced by operational and manufacturing issues. The lightest possible solution was later compared to alternative solutions, which were either recommended by TTS or designed based on the results achieved from previously considered concepts.

- A weight reduction of 6% was achieved with the reference concept (A).
- The concept (B) that had previously been provided by TTS was optimized in the same way yielding a weight reduction of 2.5%.
- A reduction of 2% was, in turn, achieved from the starting point of Concept B, with a new concept (B-III). The two-edge longitudinal beams were connected with a flange, forming a box beam; the transverse beams near the edges standing very close to each other were replaced with one beam each, and the two in the middle were removed. This final generation is actually regarded as more optimal by having a much simpler geometry.
- A final concept (C), having the main beams placed diagonally, was developed without satisfactory results. It had a poor performance in satisfying design criteria and the final model was the heaviest among the concepts.

From the parametric sensitivity analysis it can be seen that the secondary stiffeners offer a relatively small contribution to the global stiffness of the panel compared to the primary ones. So, placing the main beams within an optimum distance would improve the performance of the panel. Unlike most conventional deck structures, it is fixed only at its corners while having a strict criterion on edge deflection. In designing such structures, the analysis showed that denser beam spacing towards the edges would improve the design rather than spacing them equally. Meanwhile, one should avoid placing the beams passing through the centre of the panel to minimize its building depth.

The conventional design still proves to be more weight-efficient than possible alternative designs. In the same way, looking at Concept B and its "children", it can be concluded that just by simplifying the structural configuration, a design with a similar weight and performance can be achieved. Moreover, it is more convenient in terms of manufacturing and maintenance owing to the simplicity of its geometry.

Parametric study, optimization and use of FE analysis appear to create a consistent methodology in designing a car deck if the parameterization is handled with care. However, existing design rules and engineering logic are essential in obtaining starting design points. It can be concluded that by utilizing optimization techniques, a relatively better solution could be reached as the optimum dimensions for scantlings are found. Otherwise, no major change could be made to the structural arrangement in order to achieve a lighter car deck panel. It is reasonable to expect that improvements in the structural design of car deck panels owing to developments in material and manufacturing technologies will occur in the future. This will most certainly allow numerical optimization methods to be more successful in this area.

10. Future work

Reducing computational time to allow for a sufficient number of different concepts to be examined meant that certain limitations had to be accepted and some assumptions had to be made. Having this in mind, the following points are suggested for consideration in future work:

- Conventional T and L-shaped stiffeners in a grillage system was adopted in this project, while the plate stiffening can be done by other various open or closed shaped beams, i.e. with the use of trapezoidal stiffeners.

- The attention in this project has been directed towards the global strength of car decks under uniformly distributed loads. However, other factors such as local deformations due to vehicle axle loads and buckling of the plate as a constraint are highly recommended for further study.

- As far as the ultimate strength capacity of the structures is concerned, collapse due to buckling is suggested as an important factor to be incorporated in future studies.

- Only a static stress analysis has been in the scope of the project presented in this thesis. Investigating dynamic effects will without doubt lead to more feasible solutions.

- It is suggested that a more sensitive optimization method be adopted for this task, perhaps with a systems engineering approach and assigning different weights to different parameters, constraints and objectives.

- The current investigation was carried out on a particular reference panel on a particular ship. Expanding the analysis by defining several parameters such as ship size, speed, position and the size of the panel, type and weight of vehicles, area of operation, etc., are necessary items if one wants to find out whether different concepts can be optimal for different cases.

- It is seen that not much can be gained by utilizing unorthodox configurations for this reference case. However, combined with a better focus on manufacturing techniques and exploiting alternative materials, it is highly possible that much lighter car deck solutions can be obtained.

- The aim of this master's thesis project has been achieved by linear analysis, assuming that the yield limit is not exceeded. However, material non-linear analysis could be necessary to reach results closer to reality, especially in regions of high stresses, such as sharp corners where the material might start behaving plastically.

- All FE analyses in this project were made with constant loads. In real life, however, marine structures are subjected to cyclic loads arising from ship motions and encounter with waves. It is recommended for further study to incorporate fatigue design into design criteria for such structures.

11. References

[1] J. B. Jia and A. Ulfvarson, "A Parametric Study for the Structural Behaviour of a Lightweight Deck", *Engineering Strctures,* vol. 26, no. 7, pp. 963-977, 2004.

[2] M. M. Alinia, "A Study into Optimization of Stiffeners in Plates Subjected to Shear Loading", *Thin-Walled Structures,* vol. 43, no. 5, pp. 845-860, 2005.

[3] E. Maiorana, C. Pellegrino and C. Modena, "Influence of Constructional Stiffeners on Elastic Stability of Girder Webs", *Journal of Constructional Steel Research,* vol. 67, no. 1, pp. 51-64, 2011.

[4] M. M. Alinia and S. H. Moosavi, "A Parametric Study on the Longitudinal Stiffeners of Web Panels", *Thin-Walled Structures,* vol. 46, no. 11, pp. 1213-1223, 2008.

[5] W. Nie and C.-Y. Ma, "A Comparative Study on the Hull Strength of a Ship with Thin-Walled Box Girders", *Journal of Marine Science and Application,* vol. 5, no. 3, pp. 11-16, 2006.

[6] G. N. Vanderplaats, "Effective use of Numerical Optimization in Structural Design", *Finite Elements in Analysis and Design,* vol. 97, no. 112, pp. 97-112, 1989.

[7] V. Kumar,.. J. Lee and M. D. German, "Finite Element Design Sensitivity Analysis and its Integration with Numerical Optimization Tecniques for Structural Design", *Computers and Structures,* vol. 32, no. 3/4, pp. 883-897, 1989.

[8] B. Brosowski and K. Ghavami, "Multi-Criteria Optimal Design of Stiffened Plates - II. Mathematical Modelling of the Optimal Design of Longitudinally Stiffened Plates", *Thin-Walled Structures,* vol. 28, no. 2, pp. 179-198, 1997.

[9] G. N. Vanderplaats and F. Moses, "Structural Optimization by Methods of Feasible Directions", *Computers and Structures,* vol. 3, no. 4, pp. 739-755, 1973.

[10] ANSYS, Inc., *Ansys 13.0 help.*

[11] D. C. Montgomery, Design and Analysis of Experiments, Hoboken, NJ: John Wiley and Sons, 2009.

[12] J. Antony, Design of Experiments for Engineers and Scientists, Oxford; Burlinton, MA: Butterworth-Heinemann, 2003.

[13] A. Dean and D. Voss, Design and Analysis of Experiments, New York: Springer Verlag, 1999.

[14] ANSYS, Inc., "ANSYS Parametric Design Language Guide", April 2009. [Online]. Available: http://www.ansys.com. [Accessed in April, 2012].

[15] J. Bonnans, J. Gilbert, C. Lemaréchal and C. Sagastizábal, Numerical Optimization, Berlin; Heidelberg; New York: Springer Verlag, 1997.

[16] A. Bhatti, Practical Optimization Methods, New York: Springer Verlag, 2000.

[17] J. Nocedal and S. J. Wright, Numerical Optimization, New York: Springer Verlag, 2000.

[18] C. Y. Lin and P. Hajela, "Design Optimization with Advanced Genetic Search Strategies", *Advances in Engineering Software,* vol. 21, no. 3, pp. 179-189, 1994.

[19] DNV, Rules for Classification of Ships, Det Norske Veritas, Hövik, Norway, 2011.

[20] TTS Ships Equipment AB, "TTS Ships Equipment AB Strength Calculation - Typical CL Panel for Car Decks", Gothenburg, Sweden, 2009.