Structural Optimisation of Midship Region for Ro-Pax Vessel in Early Design Stage using FEA

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ABSTRACT

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The international shipping industry covers around 90% of the world trade. The bulk transport of raw materials, affordable import and export of food and manufactured goods between inter continents won't be possible without the maritime transport. The requirements from the maritime industry and the classification societies related with safety, energy efficiency, environmental protection, etc. force us to look for more efficient and cost effective technologies. This calls for the development of an integrated multi-disciplinary and -objective design optimisation platform to be used in early design stage of traditional ship design process.

The thesis focuses on demonstrating the application of an automated platform for the structural optimisation of the midship of a typical Ro-Pax vessel in the early stage of ship design process. It is based on the undergoing researches on the framework of European Research Council funded project, HOLISHIP (Holistic Optimisation of Ship Design and Operation for Life Cycle), which focuses on developing innovative holistic design optimisation methods for European maritime industry. The thesis covers the development of a parametric model of the midship using commercial finite element software ANSYS® and which will eventually be used for optimisation using modeFRONTIER® with an aim to reduce the total weight of the structure. A number of in-house tools/ modules are also developed and integrated in the automated platform. The study also extended to implement surrogate models to replace the finite element analysis which allow the customer to reduce the calculation time.

Keywords: Maritime transport, Ro-Pax vessel, Structural Optimisation, HOLISHIP, Surrogate models

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NOMENCLATURE

B	Stiffened panel width
B_1	Breadth of the plate
c	Distance from neutral axis
D	Flexural rigidity of plate
E	Modulus of elasticity
h_{wx}	Web height of longitudinal stiffeners
h_{wy}	Web height of transverse frames
I	Moment of inertia
L	Length of stiffened panel
L_1	Length of the plate
M	Bending moment
M_x	Bending moments per unit length of sections of a plate perpendicular to x axis
M_y	Bending moments per unit length of sections of a plate perpendicular to x axis
N_x	Number of the longitudinal stiffeners
N_y	Number of the transverse stiffeners
p	Pressure
t	Plate thickness
T_1	Draught
t_p	Attached plate thickness
t_{wx}	Thickness of longitudinal stiffeners
t_{wy}	Thickness of transverse frames
W	Stiffened panel weight
w_{Max}	Maximum deflection
γ_M	Partial safety factor for material
γ_R	Partial safety factor for resistance
ν	Poisson's ratio
ρ_p	Density of plate
ρ_{sx}	Longitudinal stiffener density
ρ_{sy}	Transverse frame density
σ	Bending stress
σ_{Max}	Maximum stress
σ_{VM}	Von Mises Stress

σ_{xx}	Stress in X direction
σ_{yy}	Stress in Y direction

1. INTRODUCTION

The advancement in maritime industry for the past few decades are very fast. Each development proposes new and advanced techniques as well as lot of requirements in order to achieve the goals. In earlier days the global trade was dominated by shipping industry while today the concept of shipping industry is entirely changed from cargo trade to pleasure, research, etc. involving lot of complexity. The growth in the maritime industry requires more and more advancement to compete with other transportation modes.

One of the challenges in the shipping industry is that the ships are usually built in very short lead time without any predefined models. Most of the ships are unique in their design and specifications, especially in case of passenger and Ro-Pax Vessels. This forces us to consider special care in the production of ships from initial stages of design to the delivery stage and its entire life cycle. The designs should be flexible to cater various requirements as well as able to adapt the sustainability issues which are crucial in the present maritime scenario.

The added complexity in new ships demands practical and flexible support systems which are quite difficult to implement in the current design strategies. The overlap between various design phases also limits the flexibility of the existing methods. These complexities suggest the requirement of a more system based design approach which can address the inadequacy of the present approaches and allows the designer to integrate various requirements which may arise in future.

The introduction of new regulations by various maritime organisations and the fluctuation of fuel price levels force the shipping industry to explore new and cost efficient designs to achieve economic growth and profits. This also points to the requirements of an optimum design strategy.

A systematic approach wants to be implemented in the design of ships so that the design considers the ship as a complex system integrating a variety of subsystems and their components, all are serving well defined functions. The approach should also capable to address the whole life-cycle, the design stage, operation and disposal.

An optimal ship in this respect is an outcome of multi-objective and multi-disciplinary optimisation process which will integrate the entire ship system for its life-cycle. Mathematically, every element in the life cycle system of a ship itself is a complex non-linear problem. The ship design optimisation always involve conflicting requirements resulting from the design constraints and optimisation criteria arising from various stake holders such as ship owners, builders, classification societies, etc. Usually a ship is need to be optimised for cost effectiveness, higher safety and comfort of passengers and crew, highest operational efficiency or lowest required freight rate, satisfactory protection of cargo and the ship itself and for minimum environmental impact. But many of these requirements are conflicting and it is difficult to take a decision regarding the optimal ship.

So it is necessary to develop a systematic approach which can handle the various design requirements and objectives in a holistic way even though they are complex in the case of ship design.

1.1. HOLISHIP

HOLISHIP (Holistic Optimisation of Ship Design and Operation for Life Cycle) is a European Research Council funded project which focuses on developing innovative holistic design optimisation methods for European maritime industry. The project brings together all stakeholders to improve the design of maritime products by combining the design objectives of the various disciplines involved. It is a system based approach which considers all the main design functional requirements, constraints, and performance indicators [59]. This includes building, transport and operational costs, life cycle cost, environmental impacts, potential loss of lives for passenger vessels, etc. The main technical and regulatory constraints are considered at an early design stage by accounting the various system complexities. The project proposes the development of an integrated design software platform for the entire life cycle of the maritime product under consideration. The software will be capable to run virtual testing and demonstration of the product.

HOLISHIP project utilizes advanced Computer Aided Engineering (CAE) capabilities and integrates techno-economic databases with calculation modules, software tools and optimisation modules. A virtual model, Virtual Vessel Framework (VVF), for the entire ship allows the designer to carry out virtual testing prior to the construction and a better evaluation of design goals at an early design stage; thus, the designer can look for alternate designs if necessary[60].

The HOLISHIP concept results in significant improvements during all life cycle phases of the product and for various industry players such as design and engineering companies, shipyards, equipment manufacturers, ship operators, consultants, etc. The benefits proposed during the different phases of the life cycle for each group are shown in Fig. 1.

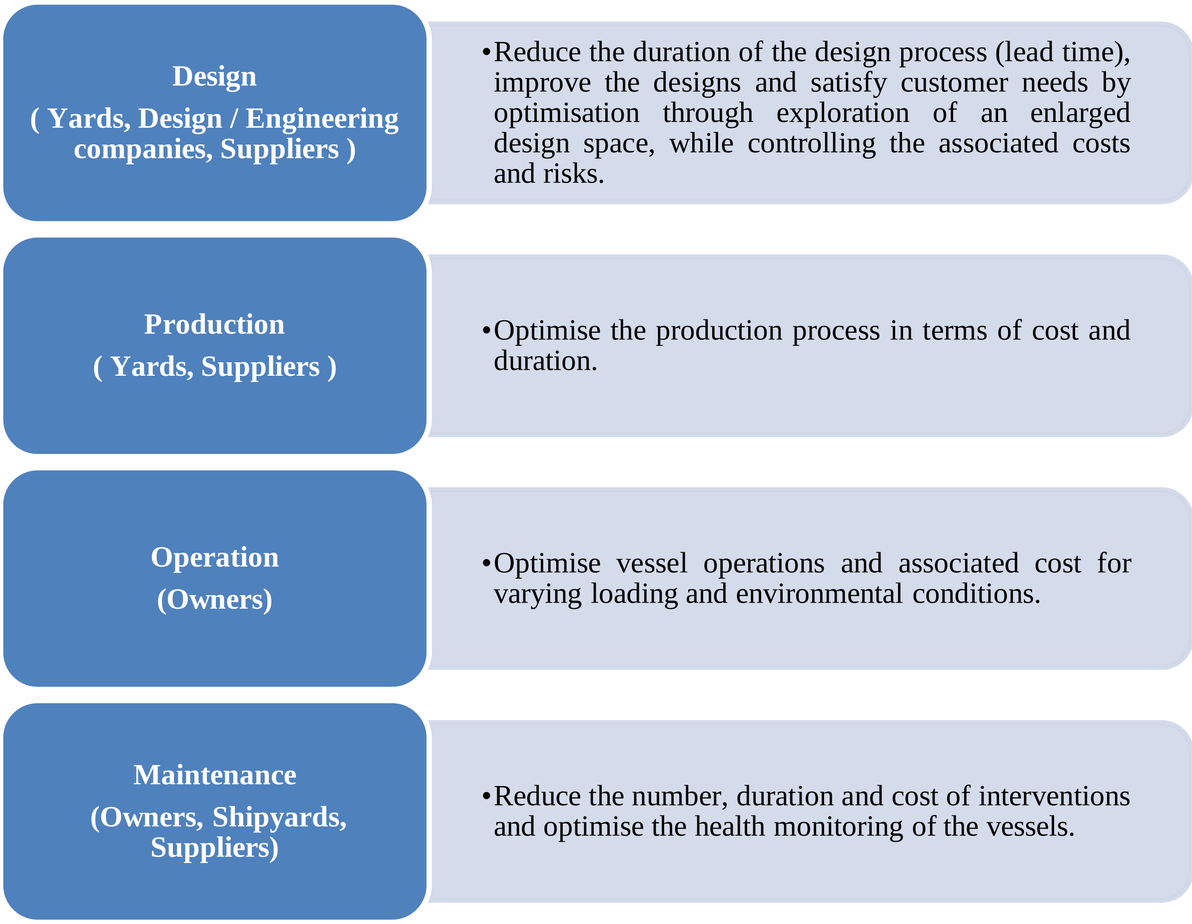

Fig. 1 Benefits for each industry group during different phases of the life cycle [60]

The project is structured as three main work clusters namely, tool development, software integration and application cases or demonstrators.

1. **Tool development:** Methods and software tools will be developed for the individual design aspects. They will be adapted for the automated use in the HOLISHIP integrated design platforms for the intended works.
2. **Software Integration:** Software tools developed in the previous cluster (Cluster 1) will be integrated into HOLISHIP design platforms and the VVF.
3. **Application Cases/ Demonstrators:** This cluster is dedicated for the implementation of integrated software platforms into the design and operation of ship and other maritime assets. The usage and benefit of the developed frameworks will be demonstrated.

The overall HOLISHIP project structure is shown in Fig. 2 and the detailed description of the application case cluster is shown in Fig. 3. There are 8 work packages involved in the project and they are handled by various groups from maritime industry and academic partners from European Union.

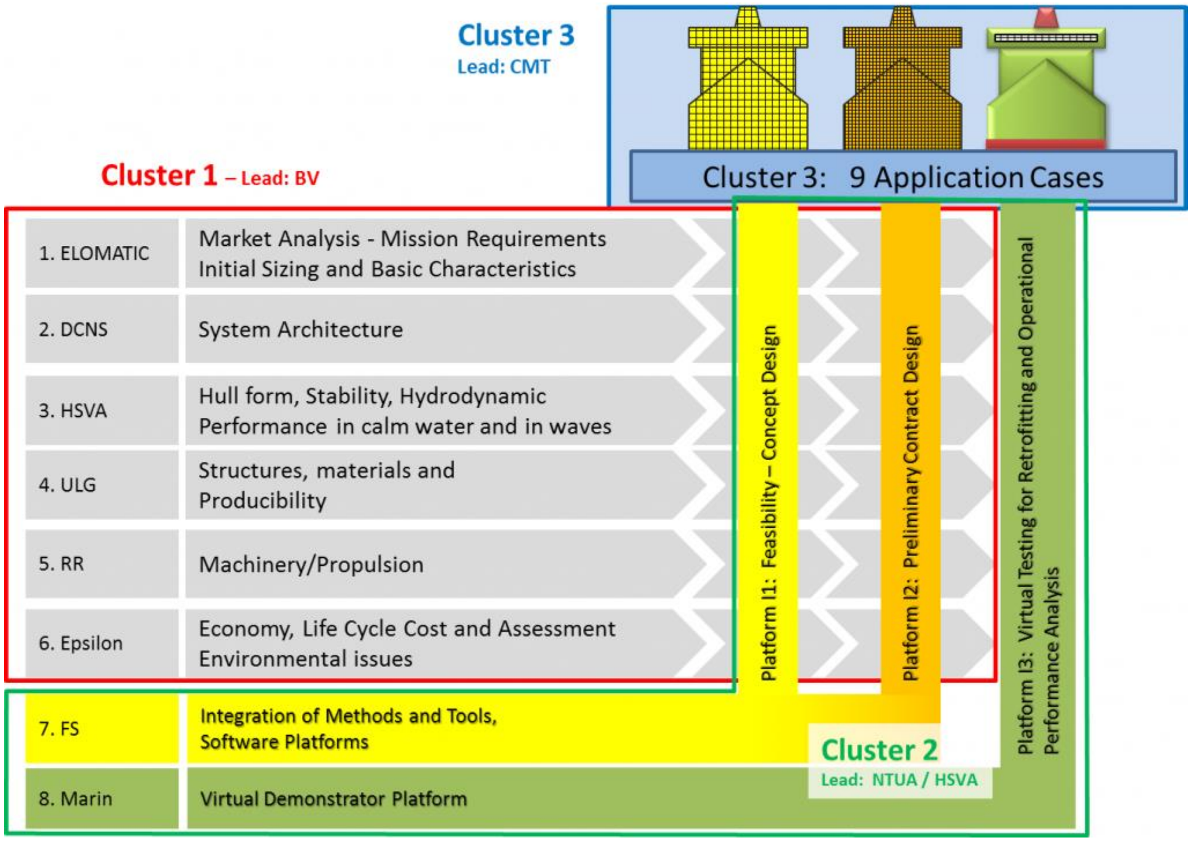

Fig. 2 Global project structure indicating 3 clusters of HOLISHIP [59]

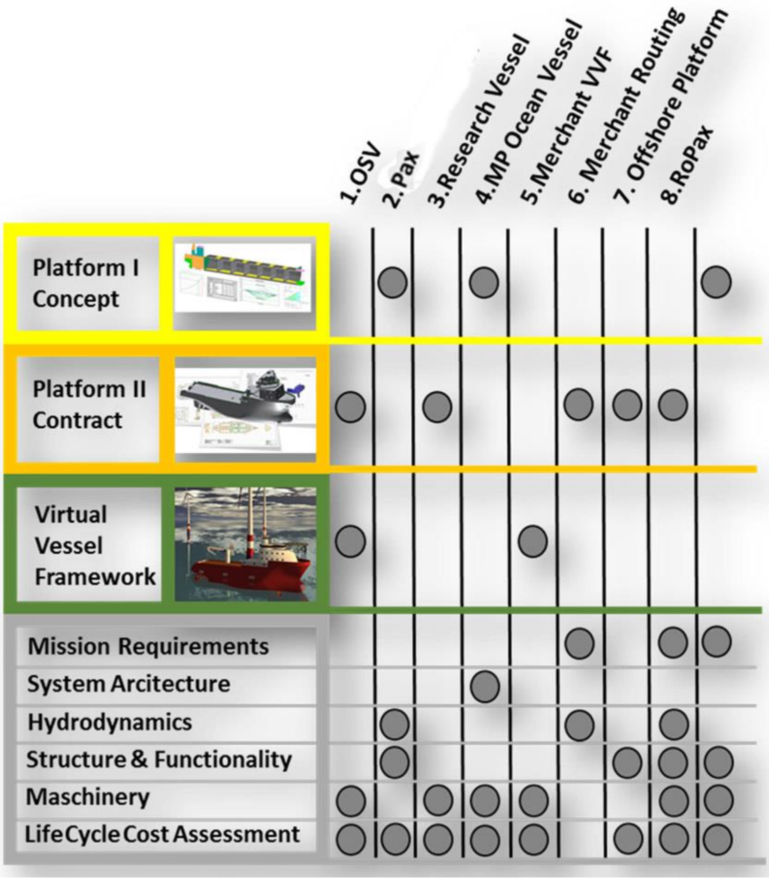

Fig. 3 Details of Cluster 3 – Application cases/ Demonstrators of HOLISHIP [59]

1.2. Objectives

The thesis is carried out within the framework of HOLISHIP project. The study is limited to structural optimisation only which comes under work package 4 of HOLISHIP project. The work package 4 deals with structures, materials and production. The aim of the package is to develop tools which can efficiently reduce the lightship weight of commercial vessels or Ro-Pax vessels in the early stages of design. The requirement from shipyards building Ro-Pax and passenger vessels indicate that one of their prime technical objectives during the design process, especially in the pre-contract stage, is to reduce the lightship weight. This is because at pre-contract stage the use of advanced assessment tools can be applied effectively.

The thesis focuses on the structural optimisation of a Ro-Pax vessel's midship region with an aim to reduce weight. The structural optimisation can be carried out by determining the optimum scantlings for girders, transverse frames, stiffeners and plates. Finite element method has been implemented along with suitable optimisation technique. The optimised scantlings will result in least weight for the midship region, which can be extended to obtain total structural weight.

The main objectives of the thesis are the following;

- Development of a finite element model of the midship region of Ro-Pax vessel in parametric manner using ANSYS® APDL scripts
- Determination of the loading cases and the boundary conditions based on a classification society rules
- Performing the linear structural analyses for the load cases identified earlier along with the computation of structural weight and centre of gravity
- Validation of the applicability of surrogate models to replace the finite element analysis phase in optimisation for large number of design variable problems
- Integration of the finite element model of Ro-Pax midship and optimisation platform, modeFRONTIER®, to create an automated tool for optimisation
- Determination of the optimum scantlings and thereby the minimum weight of the structure.
- Developing a response surface (surrogate model) from the optimised solutions and replacing the finite element package to reduce the calculation time.

Successfully achieving the objectives result a better way to approach the design stage of ships and offshore structure with multiple requirements. The optimised response surface can be extended to other kinds of ships also with slight modifications thereby reducing the lead time and design costs.

1.3. Methodology

The thesis is based on the work carried out during the internship at University of Liege, Belgium. **The aim is to create a fully automatic process which establishes an optimisation workflow with the integration of finite element method**, using ANSYS®, **and optimiser tool**, modeFRONTIER®, **so that there won't be any manual involvement on the graphical user interface**.

The approach is to develop a parametric finite element model of the midship region using ANSYS® APDL. The development will be capable to handle the changes in the structural design parameters such as plate thickness, stiffener spacing, etc. and perform linear structural analysis for load cases like hull girder bending moment and pressure on hull. The load cases and the boundary conditions are identified from a classification society rule, which is the reference for the rule based design. The model should be able to provide the weight of the structure and the position of vertical centre of gravity.

Bureau Veritas (BV) is one of the partner industries for the HOLISHIP project and their rules for the structural assessment of passenger ships and Ro-Ro passenger ships are employed in the finite element analysis.

The modeFRONTIER® will be set up and communicates with the parametric finite element model in a fully automatic way with an aim to evaluate the objective function which is structural weight with constraints like yield strength, ultimate strength, etc. The algorithm employed for optimisation iterates the design variables (scantlings for the midship region such as plate thickness, stiffener spacing and stiffener geometry) and provide the optimum solution regarding to structural weight minimisation (single objective optimisation).

Surrogate models are established using the results from the optimisation. Polynomial response surface, one of the most popular surrogate models, is generated. A large number of iterations have to be carried out in the platform in order to obtain a good response surface. The response surface must be capable to represent the contribution of each quantity accurately so that the process time can be reduced significantly. The idea is to replace the finite element analysis tool with the response surface to reduce simulation time and thereby reducing the total calculation cost.

The various steps involved in the work flow for the internship and thesis are shown in Fig. 4 which is an extension of the work done in the framework of BESST[61] EU Project (Bayatfar et al., 2013)

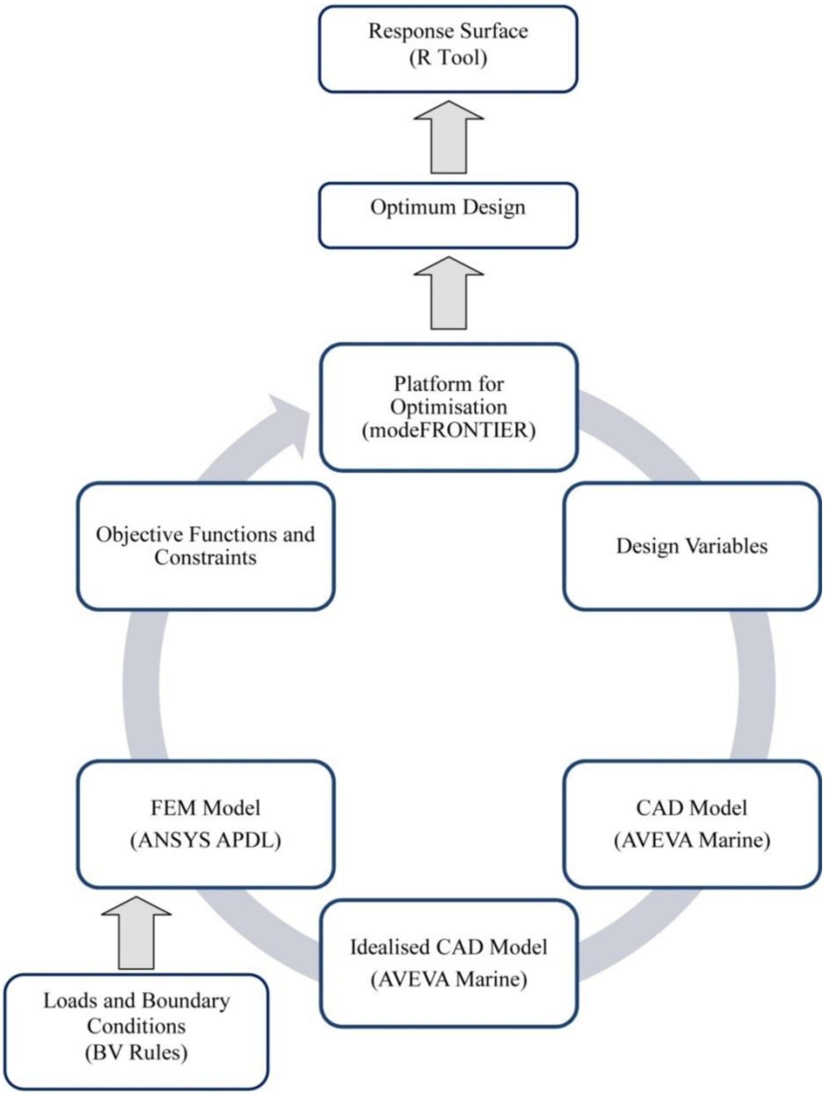

Fig. 4 Methodology Description Diagram

1.4. Software/ Tools

Brief descriptions of software used in the work are given below.

- **ANSYS® APDL**

ANSYS® is a general purpose finite element modeling package developed by Swanson Analysis System, Inc. for solving a wide variety of engineering problems numerically. The use of ANSYS® spread across various disciplines such as structural, fluid dynamics, vibration, heat transfer, acoustic, electro-static, electro-magnetic etc. Finite element solution procedure is usually divided into three phases; Preprocessing, Solution and Post-processing.

APDL is the acronym of ANSYS® Parametric Design Language which is also a part of ANSYS® package. Mechanical APDL allows the user to automate the various processes such as creation of geometry, loads and even post-processing. The software also supports script coding along with the use of graphical user interface. This feature of the software is used in the work.

- **modeFRONTIER®**

The modeFRONTIER® platform is widely used for the process automation and optimization in the engineering design process. The multi-disciplinary design optimization software which allows the workflow creation, design space exploration and integration with third party tools to provide automation of simulation processes [57].

The software uses multi-objective optimisation algorithms to streamline the design processes and reduce the cost and time by improving the results. It allows the user to choose strategy of optimisation based on design space boundaries with the required accuracy and robustness. The platform include advanced algorithms for response surface method based and direct optimisation such as stochastic, deterministic and heuristic methods for single as well as multi objective problems. The algorithms can be run in any modes like manual, self-initialising or fully autonomous. The single objective optimisation feature of the software is used in the present thesis. The software was developed by Italian company ESTECO SpA in 1999.

- **R Tool**

R is a language and environment used for statistical computing and graphics (Venables et al.). R tool provides a wide range of statistical, such as classical statistical tests, linear and nonlinear modelling, response surface, etc. and graphical techniques. The software's capability to develop response surface technique is used for the proposed work.

Response surface technique is basically a mathematical method which correlates several variables with one or more target variable. This is based on a minimum square approximation of lot of pre-calculated data or experiments. The method provides a polynomial which accurately represents the relationship between variables and the target(s). The resulting polynomial can be used in other computations thereby replacing time consuming and complicated processes such as FEM and CFD. The method is much better when replacing experimental data, which is quite complex to process, or replacing FEM or CFD calculation, which take too much time and power to compute.

2. LITERATURE REVIEW

The optimisation of ships and maritime structures are quite well studied in the past decades by many authors. The implementations of response surface methods in various industries are also available along with its application in shipping industry. A detailed literature study has been carried out in order to identify the previous works and their advantages and disadvantages. The study is divided in to two sections; Structural Optimisation and Response Surface Method as surrogate model.

2.1. Structural Optimisation of Ships and Offshore Structures

Zanic et al. (1998, 2001) describe the structural optimisation of ships built in Croatia and Italy. The structural optimisation offers significant savings for ship builders and end users (ship owners) in terms of decreasing the price and weight of steel required for construction, increasing the deadweight, providing special class features with classification societies regarding maintenance, etc. The design methodology applied for one first class passenger ship and two Ro-Pax vessels. SHIPOPT and MAESTRO programs are used for design and structural analysis of ships. The finite element analysis is carried out on 3D full ship model to obtain accurate assessment of structural and dynamic responses such as global deformations, longitudinal stresses at each deck with control of buckling of upper decks, transfer of forces between lower hull and upper superstructure, local deformation around windows, doors, etc. Sensitivity analysis for the hull structural weight with varying web frame spacing is also carried out. There are 264 design variables used and their minimum and maximum limits are specified. The proposed strategy even allows modifying the main structural components quickly without losing the structural integrity of the ships.

Boulougouris et al. (2004) propose an optimisation procedure for the improvement of the existing ship design; the developed method can be used at both the conceptual design stage as well as the later design stage allowing the designer to obtain an overview of the design space. Maximisation of ship's resistance against capsize, expressed by the Attained Subdivision Index, transport capacity in terms of increased deadweight and garage deck space, and building cost reduction in terms of structural steel weight reduction are the objectives considered for the optimisation. Various case studies are carried out by implementing Multi-Objective Genetic Algorithm (MOGA) where as the last case study used MOGA with additive utility functions (UTA). The size of the initial population is a significant parameter in the optimisation procedure in terms of computational cost.

Rigo (2005) presents the theory implemented in the LBR5 software, the analysis and optimisation software for orthotropic structures and sea water. The software is also applied to scantling optimisation of ship structures (Rigo, 2001, 2003). It uses an analytical method based on the differential equations of stiffened plates to compute the overall response of the hull structure.

Klanac and Jelovica (2007) explain a model coupled by vectorization and genetic algorithms for omni-optimisation. The omni-optimisation assumes the capability to perform any kind of problems such as single and multi-objective optimisation using single optimisation algorithm or omni-optimiser. The paper describes the application of omni-optimisation on a fast ferry's mid-ship structure for the minimum weight and vertical centre of gravity. The optimisation focused on multiple constraints based on structural and technical aspects. The objectives are optimised concurrently as well as independently also. There is around 10% reduction in weight and 6.5% in vertical centre of gravity resulted with the implementation omni-optimisation. But the algorithms are not suitable for ship structural design problems involving overall ship structure, which are problems resulting thousands of constraints.

Skoupas and Zaraphonitis (2008) discuss the development of a method for the preliminary design and optimisation of high speed mono hull Ro-Pax vessels. The model is created based on selected design parameters and macros are used to generate hull form, structure and internal layout. The structural design is performed using NAPA software and scripts are utilised for the calculation of plate thickness and the cross sectional characteristics of primary and secondary stiffeners. The final structural scantlings are defined by considering the various load cases and carrying out structural analysis based on DNV rules. The created parametric model linked to modeFRONTIER® for implementing multi objective optimisation using Genetic Algorithms at a later stage. The optimisation of a large and a small vessel are carried out to confirm the applicability of the developed parametric model.

Papanikolaou (2009) points out the importance of holistic concept of optimisation in ship design to simplify the different stages in its life cycle. The method describes the standard ship design optimisation problems and provides solutions with the use of advanced multi objective optimisation methods. The concept proposes improved and modern designs with increased safety and endurance, capability to carry more cargo, reduced engine power requirements, less pollution and thereby improved protection for environment. The developed tool is useful for the design of hull form development for high speed crafts and for the internal subdivision of Ro-Ro passenger vessels for improved safety and efficiency.

Sekulski (2009) proposes a computer code to investigate the possibility of multi objective optimisation for hull structural scantling, which is a key parameter in the weight reduction. The method uses genetic algorithm to optimise the scantlings of the ship hull. Main consideration is given to topology and structural elements in the large spatial section of ship. A high speed vehicle - passenger catamaran's structural weight minimisation is carried out by considering several design variables like plate thickness, stiffener and frame spacing, etc. The numerical results imply that the application of genetic algorithm is an efficient tool for simultaneous optimisation of topology and structural sizing of high speed ships.

Papanikolaou et al. (2010) present a methodology for the holistic design of conventional Ro-Ro Passenger ships. Parametric design tools are developed with commercial software, NAPA and coupled with multi objective optimisation software, modeFRONTIER®, to create an integrated ship design platform. The integrated platform is capable to explore the design space and providing optimal designs in an efficient and rational way. The objective function considered was Net Present Value (NPV), which involves many factors like building cost, operational costs and annual revenues. The methodology was implemented for a series of Ro-Pax vessels and found quite useful in conceptual design stage.

An integrated platform for the design and optimisation of ship design process in early stages is developed by Papanikolaou et al. (2011). The methodology applied for an Aframax tanker design for Caribbean trade with an aim to reduce Oil Outflow Index (OOI), lower Energy Efficiency Design Index (EEDI) and lower Required Freight Rates (RFR). The approach is able to handle various aspects of ship design in a holistic way. Even the challenging CFD simulations are implemented by the authors using response surface methods effectively. The developed response surfaces will reduces the complexities in the CFD simulations and thereby reducing the time required for the long computations.

Kitamura et al. (2011b) describe the structural optimisation of double bottom bulk carrier in its initial design stage using finite element method. The design variables include five geometric dimensions and thirty-one plate thicknesses defining shape and size of the bottom structure respectively. The yield stress and the buckling stress are considered as the constraints with minimisation of weight as objective function. The sensitivity analysis and the equation approximating the stress developed (Kitamura et al. 2011a) are used as the finite element analysis requires high computational effort. The optimisation procedure carried out with five combinations of the design variables and the constraints and their results are compared. The simultaneous use of design variables defining shape and size of the bottom structure make the space of the structural optimization larger and improve the quality of structural design. Genetic Algorithm (GA) employed for the optimisation.

Fu et al. (2012) develop a modified collaborative optimisation (CO) model suited for optimisation of ship design phases. The model is implemented for the multi-objective optimisation of a 3100TEU container ship structure to obtain minimum structural mass in the static analysis and to minimise maximum acceleration of structure in the dynamic analysis. The optimisation technique used is a variant of Simulated Annealing (SA) algorithm called Adaptive Simulated Annealing (ASA) in which the parameters that control temperature schedule and random step selection are automatically adjusted as the algorithm progresses allowing the algorithm more efficient and less sensitive to user defined parameters than standard simulated annealing.

Sun and Wang (2012) introduce a hybrid process of modelling and optimisation to reduce the large time cost in structural optimisation of ships. Support Vector Machine (SVM), a statistical learning theory rooted technique specialised in studying the situations with a small number of samples, used in combination with genetic algorithm for the optimisation of structural scantlings from the mid-ship of a very large crude carrier (VLCC) under the direct strength assessment. There are thirty five design variables considered including the plate thicknesses and beam dimensions. The constraints are the yield utilisation factors defined in common structural rules (CSR) and five load cases considered in the analysis with an objective to reduce the mass of the structure. A comparison is also made between the SVM and radial basis function network (RBFN), a typical artificial neural network (ANN). The former one shows strong ability to simulate real analyzing code with high accuracy and outstanding performance of generalization compared with the latter one.

Sekulski (2013) proposes an evolutionary algorithm for multi objective optimisation of structural elements in the large spatial sections of ship hulls based on fitness function comprising few selection criteria. The example presented for the application of developed algorithm is the structure of a fast passenger vehicle ferry. There are 37 design variables considered and a well defined, single valued and having real, positive integer values, objective function is used as the fitness function.

Cui and Wang (2013) present the application of knowledge based engineering on the structural design and optimisation of container ship cargo tank. Ship design is a complicated multi-disciplinary task in which the knowledge based engineering can be implemented to reduce the design errors, mistakes and reduce design cycle significantly. Knowledge based system acquires knowledge from design experts, design standards and rules, successful precedents, etc. Multi-Island Genetic Algorithm (MIGA) employed for the structural optimisation with an aim to reduce the weight of the ship cargo tank. Both design rule method and interpolation method are employed. The use of design rule method provides more economical advantage than interpolation method where as interpolation method provides design solutions with better reliability than former one. The relevant knowledge are automatically distracted from the knowledge base and executed together with the knowledge reasoning technique during the new ship structures design process.

Ji and Wang (2013) present a methodology to carryout multi objective optimisation for ultimate strength of ship hulls under multiple load cases. The stiffness analysis using the linear finite element analysis employed instead of the non linear finite element analysis based on ultimate strength since the latter requires more time. The assumption is that the ultimate strength will be optimised as long as stiffness optimised. Kriging approximation model based Latin Hypercube Sampling and Multi Island Genetic Algorithm employed for the stiffness optimization. The load cases considered includes sagging, hogging, horizontal bending and torsion. The methodology applied on the hull of a chemical tanker with an aim to maximise the vertical, horizontal bending and torsion simultaneously. The weight of the ship hull remains the same as initial design is the constraint for the optimisation. There is a noticeable increase in the ultimate strength after optimisation compared to the initial design.

Priftis et al. (2016) describe a methodology for the parametric design and optimisation of a midsized containership (6,500 TEU) using modern features such as computer aided design and computer aided engineering (CAD /CAE) in the conceptual stage of ship design with short lead time. A parametric model representing all the internal and external structural components of the ship and codes to determine the necessary design constraints and effectiveness indicators are included in the model. The paper explains the advantages of using modern optimisation techniques in shipbuilding industry by developing an efficient model capable to handle multi-objectives, structural design, operation and economy. The non-dominated sorting genetic algorithm II (NSGA-II) utilised to carry out the optimisation. The implementation of the parametric models in the early ship design stage will result in numerous benefits to ship builders and ship owners. The method can also be applied to other containerships sizes and ship types (Koutroukis et al., 2013).

Lee et al. (2017) confirms the applicability of multi objective optimisation techniques in the design of derrick structures considering their structural efficiency. The developed appropriate meta-model of maritime systems compared with the fitted models created with Neuro-Response Surface Method (NRSM). The prediction errors are high for complicated non-linear problems which is the case for ships. The proposed methodology is capable to use as an engineering design optimisation tool in the early stages of design.

2.2. Response Surface Method

Surrogate model or metamodel is an engineering method, which can be used to approximate multivariate input - output behaviour of complex systems based on a limited set of computational simulations. Surrogate models represent the underlying simulation system as closely as possible while being computationally cheaper to evaluate. They can be used for design optimisation, design space exploration, parametric studies, etc. One of the most popular surrogate models is polynomial response surfaces.

Response Surface Methodology (RSM) is a set of mathematical and statistical techniques for building empirical models which can be used for exploring optimum conditions through experiments. Originally the response surface method was developed to model experimental responses (Box and Draper, 1987). Later the method starts to use for the modelling of numerical experiments. The difference between the two is in the type of error generated by the response. Inaccuracy in physical experiments arises due to measuring errors whereas in computer experiments, it is due to numerical errors (as a result of incomplete convergence of iterative processes), round off errors or the discretisation of continuous physical phenomena (Giunta et al., 1996; Toropov et al., 1996; Van Campen et al., 1990). The errors are assumed to be random in RSM.

The application of RSM to design optimization is aimed at reducing the cost of expensive analysis methods like finite element method or CFD analysis and their associated numerical errors. The problem can be approximated with smooth functions (polynomials) that improve the convergence of the optimization process. Venter et al. (1996) discuss the advantages of using RSM for design optimization applications. The use of accurate response surfaces in structural optimisation help to overcome the noise and software integration problems associated with traditional multi-disciplinary optimisation approach. The optimisation process carried out with response surface is computationally inexpensive and resulted in an optimum solution. The solution exhibits excellent correlation with finite element results also.

Response Surface Method (RSM) is used as performance prediction method in many engineering optimisation problems (Hong et al., 2000). RSM is an approximate optimisation method which gives the best relationship between design and target variables using polynomial regression techniques. Experimental design and the determination of response function are the key problems in response surface method (Yu et al., 2002). The method is shown in Fig. 5 (Myers and Montgomery, 1995).

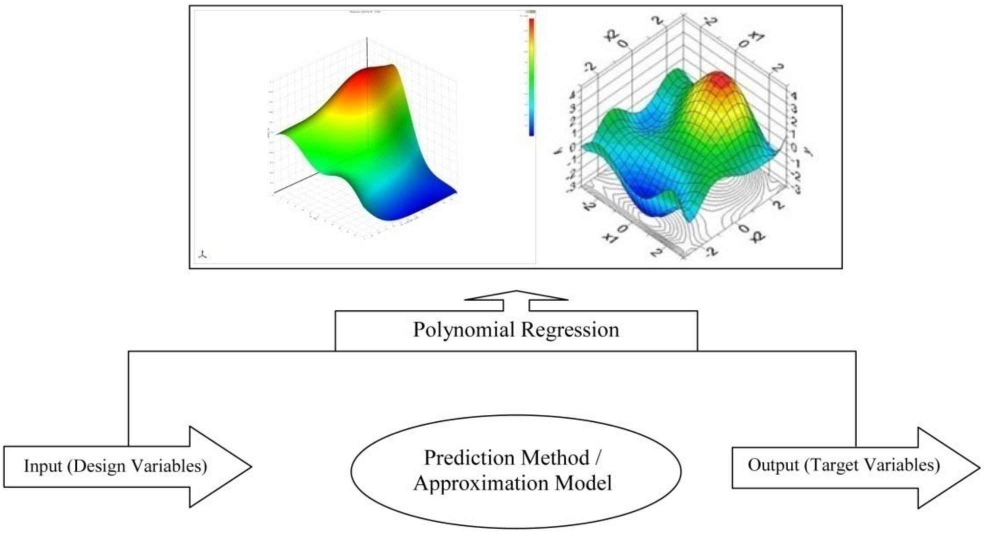

Fig. 5 Response Surface Method (RSM)

The optimisation of transverse bulkhead structure of a crude oil tanker shows that the response surface method is a powerful tool to obtain an optimal design by rationally reducing structural response analyses (Arai and Shimizu, 2001). It is necessary to consider the construction cost in addition to the structural weight for bulkhead optimisation problems to get optimum design similar to the actual design.

Yu et al. (2002) proposes a stepwise response surface approach for the reliability analysis of ship structure. The stepwise regression technique is employed for the development of response surface. The results obtained with sensitivity analysis using RSM are quite acceptable than the ones' with conventional reliability analysis methods such as first order, second order reliability method, etc. A double bottom hull is analysed with the proposed method. The behaviour of large and complex structural systems can be represented by using an appropriate response surface function, which utilises the advantages of conventional reliability methods and overcome their drawbacks.

Yang and Guo (2008) employ response surface method integrated with genetic algorithm for the optimisation of Radar Cross Section (RCS) for naval vessels. The RCS of targets can be reduced by means of various methods such as target shaping, distributed, discrete, active loading, etc. The reduction in RCS improves the stealth performance of vessels. The optimisation is carried out in shaping design in order to reduce the scattering of electromagnetic signals from ship significantly. The electromagnetic signals are scattered from the deck plates, mast and chimney, superstructure corners, armaments, etc.

Characteristic section design method (CSDM) is employed in order to replace the RCS reduction design since it requires intensive electromagnetic numerical analysis and time. The complex and sensitive objective function is approximated using a non gradient and simple polynomial response surface. The results from the computations are verified against the designs obtained with genetic algorithm.

Gorshy et al. (2009) propose a multi-disciplinary optimisation (MDO) technique for ship design considering the structure, cargo loads and propulsive power requirement. The ultimate objective is to reduce the running cost per day of the ship with full load. Latin Hypercube Sampling (LHS) is adopted for the design space exploration and to select sample data from the design variables. It is a variance reduction as well as a screening technique for variables, which highly controls the selection process still allowing them to vary little. Response surface method is employed to reduce the computation expense and achieve desired computational precision in the MDO process. Particle Swarm Optimisation (PSO) is used to obtain the best design scheme from the global design space. The accuracy of the structural optimisation largely depends on the choice of approximating function with its design space. The selection of large number of points is usually not cost effective (Roux et al., 1996). The optimisation is carried out for a 33500 DWT open hatch box shaped bulk carrier ship to minimise the running cost and found that the method is quite promising. Relatively high computational expense is the limitation of the proposed approach.

Lee et al. (2014, 2015) explain the importance of the development of an optimised design tool based on performance of marine systems to use in their initial design stage. Neuro-Response Surface Method is employed to develop the tool. Back-Propagation Neural Network (BPN) is used for generating design space and Non-dominated Sorting Genetic Algorithm II (NSGA-II) is utilised for the optimisation. The developed framework successfully implemented for the case study of 5MW wind turbine and ultimate strength of ship stiffened panels. Commercial software AQWA and ANSYS® APDL are used for analysing the results of developed tool for hydrodynamic and structural performances.

Shi et al. (2015) demonstrate the efficiency and usefulness of reliability analysis using Kriging response surfaces for ship stiffened plates with initial imperfections. The structural reliability analysis requires multiple evaluations of the limit state functions (LSFs) for various random variables. The approach uses Kriging interpolation models as surrogates of nonlinear LSFs and estimates the failure probability with first order reliability method (FORM) using the classical second order polynomial regressions. The method considers 10 design variables for the stiffened plate reliability example and the approach is efficient in the evaluation of ultimate strength by nonlinear FEA in terms of less number of evaluations required compared to the conventional analysis.

Yang et al. (2015) propose a methodology for the structural reliability analysis of the composite structures using RSM in conjunction with nonlinear FEA. The approach uses response surface as the limit state instead of the complicated limit state surface of the model. The in-house program, GLAREL, used for generating the response surface including the linear, square and cross terms and ANSYS® used for FE analysis. The method suggests a reliability based design code for partial safety factors which can be used in design for certain target reliability level.

Parametric structural design optimisation in conceptual design phase is a promising approach which combines the weight reduction, efficiency and safety. Andrade et al. (2017) illustrate the application of design of experiments with RSM and FEA for the parametric optimisation of a platform supply vessel. The results suggest that the method is a reliable one in the early stages of design even in the case of other ship types.

2.3. Comments

Large numbers of literatures are available in the field of ship structural optimisation using finite element method. But the application of the finite element method consumes lot of time. It is important to reduce the computational time especially in the early stages of ship design optimisation. The advantages of optimising the structure won’t be available if the optimisation process consumes lot of time. The application of surrogate models such as response surface method to replace the finite element method is a possible solution to reduce the computation time.

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3. OPTIMISATION OF STIFFENED PANEL

Stiffened panels can be considered as the basic structural elements or the building blocks in ships and many other marine structures. Basically such a panel consists of a plate reinforced with orthogonal stiffeners and capable to resist and transmit the forces and stresses throughout the structure. The reinforced plates are also found its application in box girder bridges, air frames, nuclear power plants, etc. because of their simplicity in fabrication and excellent strength to weight ratio. The structural properties used for the stiffened panels depend on their area of applications. For example, plate structures have a higher thickness to length ratio when they used in ship where as thin walled structures are found in aeronautical applications. This is achieved by varying the structural parameters of the plate itself or the geometry and quantity of the stiffeners or the material used. A longitudinally stiffened panel usually seen in ships is shown in Fig. 6.

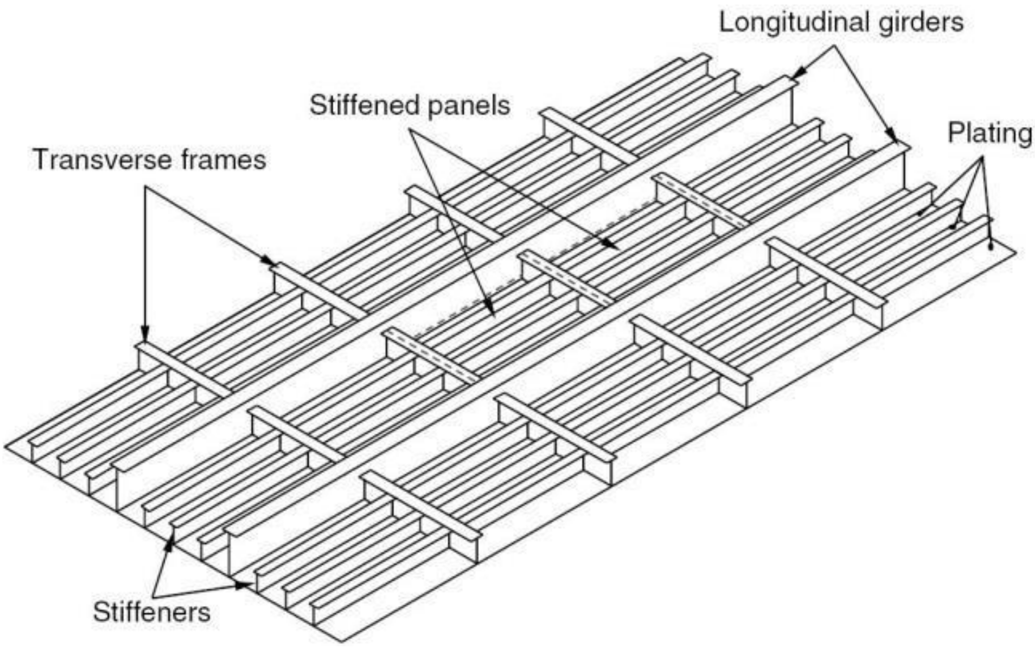


Fig. 6 Flat section composed of stiffened panels [53]

The stiffened panels in ships and marine structures are usually subjected to loads normal to its own plane due to deck loads, cargo loads and water pressure. Due to the overall bending and twisting of the ship hull girder induce in-plane forces at the boundaries of stiffened gross panels. The nature of in-plane forces can be tensile, compressive or shear and some of them can occur simultaneously on all the four boundaries of gross panel (Mansour, 1977). An estimation of ultimate strength of the stiffened panel plays an important role in safety, reliability and economics of structural design (Hughes and Paik, 2010).

One of the important design considerations in ship design is to reduce the dead weight of the ship. Since the stiffened panels are building blocks for the ship, reduction in the weight of the stiffened panel, even if it is really small, without losing their structural integrity yields weight reduction for the entire ship. Any decreases in structural weight will results in cost reduction and improvement in performance also.

An optimisation process is carried out to get a robust, automatic and efficient design for the stiffened panels by integrating the commercial software ANSYS® and modeFRONTIER®. The objective of the stiffened panel optimisation process is to test the codes developed in ANSYS® APDL and understand the working of ANSYS® and modeFRONTIER® coupled loop.

3.1. Description of the Stiffened Panel

A stiffened panel from the deck of a car carrier vessel is considered for the analysis (Jia and Ulfavarson, 2005). The hat shaped stiffeners are replaced with bar stiffeners for simplifying the problem. The structure is made of AH32 steel. The yield stress of the material is 315 MPa and modulus of elasticity is 200 GPa. The material has a density of 7800 kg/m^3 and Poisson's ratio of 0.3 (Harvey, 1982). The geometry and dimensions of stiffened panel are shown in Fig. 7. There are three longitudinal stiffeners and four transverse frames in the stiffened panel.

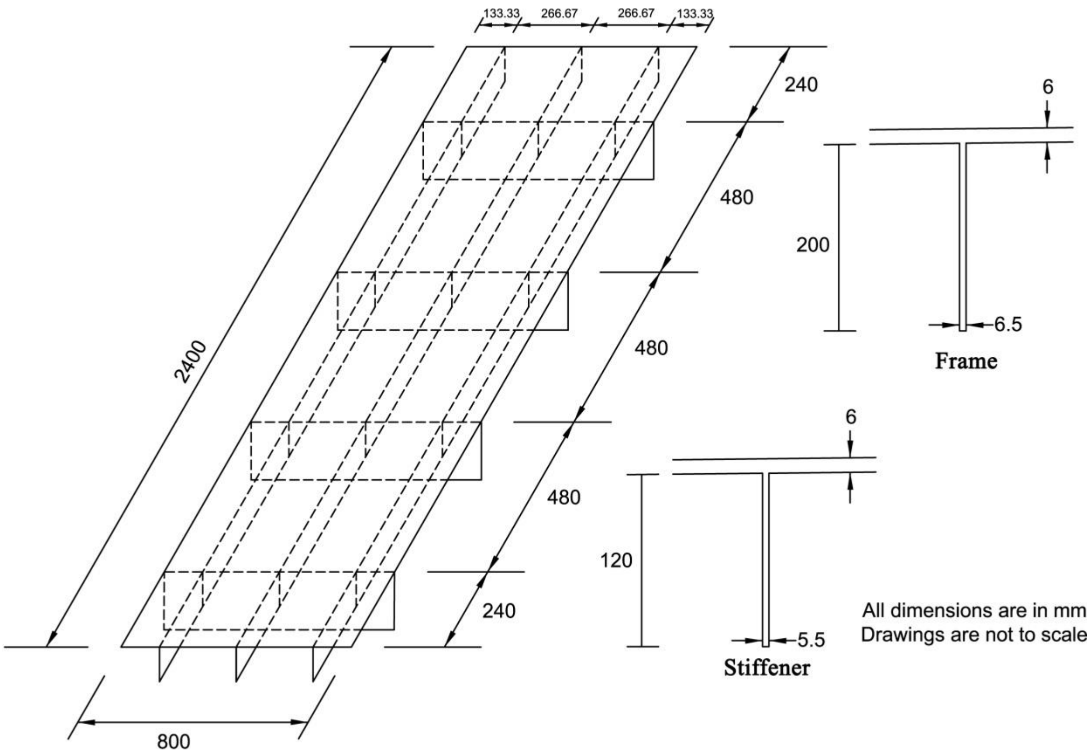


Fig. 7 Geometry and dimensions of stiffened panel assumed for investigation

3.2. Validation of Finite Element Software

Finite element analysis is a powerful technique in modern engineering practice which provides an approximate solution. The results obtained with finite element analysis are depend on various factors such as the correct representation of the problem and simulation, experience and judgement of the engineers involved in the analysis, etc. Small errors in geometry modelling, applying initial and boundary conditions can lead to significant errors in the results. The scenario will become much worse if the errors are small, difficult to identify and they influence the performance significantly. Finite element models are always associated with errors and complexity of the structure increases the probability level of such errors. Model validation should be part of every modern engineering analysis quality assurance procedure (ISO 9001). Finite element validation verifies that the idealisation of the problem and the analysis conclusions reflect real world results.

Analytical solutions for the deflection and stresses on the stiffened panel are not readily available and formulations of them are based on lot of assumptions which may give solutions far from reality. Theoretical solutions for a plate without any stiffeners are available in many literatures (Timoshenko and Krieger, 1959, Hughes and Paik, 2010). A plate clamped at all edges is used for the mesh convergence study and finite element validation hereafter.

3.2.1. Mesh Convergence

Mesh convergence refers to the smallness of the elements required in the model to ensure that the results of a simulation are not affected by changing the mesh size. A sufficiently refined mesh ensures that the analysis results are adequate. The numerical solution will tend to a unique value as the mesh density increases. On the other hand this increases the computer resources required to carry out the simulation. So it is important to find out the best mesh size for the analysis which provides unchangeable results even with more refinement and efficiently using the computer resources available.

The linear structural analysis has been carried out on a plate using FEA software ANSYS®. Thin shell element SHELL181 (4 nodes) with six degrees of freedom are employed for the modelling and analysis. The structure is loaded by a lateral pressure of 50 kPa and clamped boundary conditions are applied. The dimensions of the plate are 750x250x6 mm. The mesh size for further investigations is selected based on the convergence study carried out on the model. The maximum deflections for fifteen different mesh densities are found out. The aspect ratio always kept at one for best results. The number of elements and the corresponding maximum deflection obtained for the plate are plotted in Fig. 8.

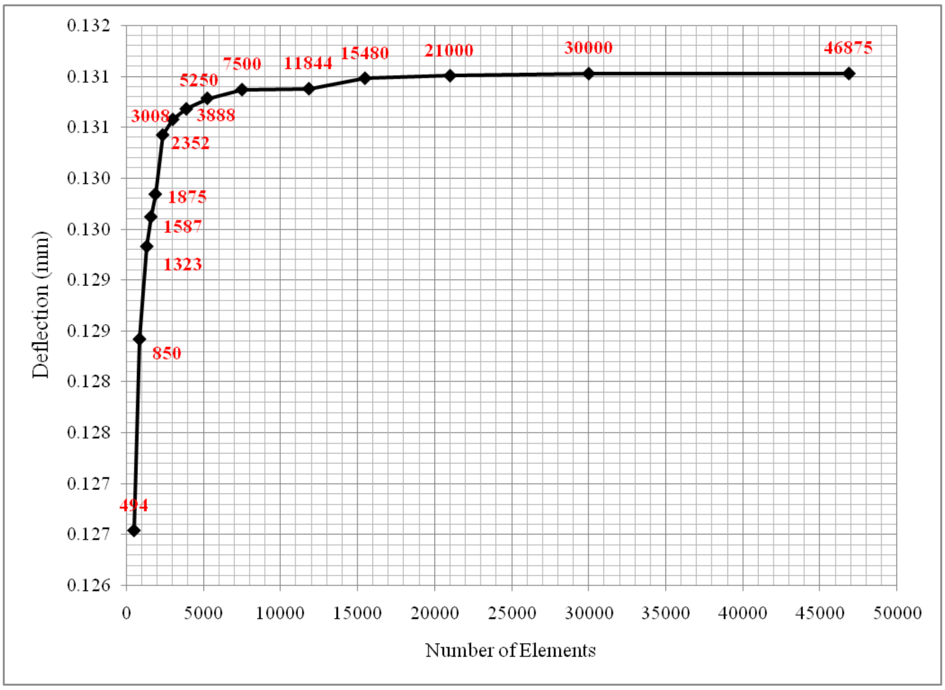


Fig. 8 Plot of Maximum Deflection versus Number of Elements

The number of elements exceeds approximately 21000, the results are converging and further refinement is not affecting the results. The mesh size corresponds to the converged solution is 3x3 mm. The deflection remains the same when the mesh size decreased to 2.5x2.5 mm and 2x2 mm. Therefore a mesh size of 3x3 mm is selected for further analysis.

3.2.2. Analytical Method

Timoshenko and Krieger (1959) developed solutions for rectangular plates with different edge conditions. Their solution for the deflection of a plate clamped at all edges under uniform distributed loading is given by;

$$w_{Max} = 0.00256\, p\, \frac{{B_1}^4}{D} \tag{1}$$

$$D = E\, \frac{t^3}{12(1-\nu^2)} \tag{2}$$

Similarly the moments acting on the edges and centre of the plate are given by the following expressions.

$$M_x = -0.0831\, p\, {B_1}^2 \text{ on clamped edge } x{=}B_1 \tag{3}$$

$$M_y = -0.0571\, p\, {B_1}^2 \text{ on clamped edge } y{=}L_1 \tag{4}$$

$$M_x = 0.0414\, p\, {B_1}^2 \text{ at centre} \tag{5}$$

$$M_x = 0.0150\, p\, {B_1}^2 \text{ at centre} \tag{6}$$

The bending stresses at various points on the plate can be calculated using the flexure formula;

$$\sigma = \frac{M}{I} c \tag{7}$$

The deflection and stress intensity on a plate clamped at all edges can also be estimated using the following relations.

$$w_{max} = \frac{0.0284\, pB_1^{\ 4}}{E\, t^3 \left[1.056 \left(\frac{B_1}{L_1} \right)^5 + 1 \right]} \tag{8}$$

$$\sigma_{max} = \frac{pB_1^{\ 2}}{2\, t^2 \left[0.623 \left(\frac{B_1}{L_1} \right)^6 + 1 \right]} \tag{9}$$

Hughes and Paik (2010) proposed a solution for the deflection and stresses on a plate exerted by lateral pressure and clamped at the edges. The maximum deflection and stress are given by;

$$w_{Max} = 0.98\, p\, B_1^{\ 4}/384D \tag{10}$$

$$\sigma_{max} = 0.5\, p \left(\frac{B_1}{t}\right)^2 \tag{11}$$

3.2.3. Finite Element Method

The stiffened panels in ship structures are mainly subjected to lateral or transverse loads. Cargo load on the decks, hydrostatic pressure acting on the bottom and side shells of the ship, etc. are few examples of lateral loads acting on ship.

Linear static finite element analysis has been carried out on a plate without stiffeners for the verification of finite element tool, which is used for further analysis of stiffened panel, box girder and midship region of Ro-Pax vessel. Finite element software ANSYS® is used to carry out the finite element analysis of the plate. A uniform lateral pressure of 5 kPa applied on the plate with all of its edges in clamped condition. Elastic shell elements (SHELL181) are employed to discretize the geometrical model of rectangular plate and each node has six degrees of freedom (Three translations along *x*, *y* and *z* directions and three rotations about the nodal *x*-, *y*- and *z*- axes). 3x3 mm mesh size is selected as described in mesh convergence study. The clamped condition in software is achieved by restricting all six degrees of freedom of the nodes in the edges of the plate.

3.2.4. Comparison of Results

The maximum deflection and the stresses on the plate obtained using finite element analysis and analytical solutions are compared. The values obtained at the centre and edges of the plate are given in Table 1.

Table 1. Comparison of Analytical and FEA results

<table>
<tr><th rowspan="2">Entity</th><th rowspan="2">Location on the Plate</th><th colspan="2">Analytical</th><th rowspan="2">FEA</th><th colspan="2" rowspan="2">Difference in %</th></tr>
<tr><th>Timoshenko</th><th>Empirical</th></tr>
<tr><td>w_{Max} (mm)</td><td>Centre</td><td>0.1264</td><td>0.1278</td><td>0.1302</td><td>-2.98</td><td>-1.81</td></tr>
<tr><td>σ_{xx} (MPa)</td><td>Centre</td><td>7.3698</td><td>-</td><td>6.5506</td><td colspan="2">11.12</td></tr>
<tr><td>σ_{yy} (MPa)</td><td>Centre</td><td>21.5625</td><td>-</td><td>21.7600</td><td colspan="2">-0.92</td></tr>
<tr><td>σ_{xx} (MPa)</td><td>Edge</td><td>29.7396</td><td>-</td><td>27.9197</td><td colspan="2">6.12</td></tr>
<tr><td>σ_{yy} (MPa)</td><td>Edge</td><td>43.2813</td><td>-</td><td>41.8395</td><td colspan="2">3.33</td></tr>
<tr><td>σ_{Max} (MPa)</td><td>Edge</td><td>-</td><td>43.3657</td><td>41.8395</td><td colspan="2">3.52</td></tr>
</table>

The deflection at the centre of the plate calculated using finite element analysis is almost equal to that obtained based on analytical solutions. The percentage error is less than 3%. The stress intensity also shows similarity like deflection with an error less than 4%. The maximum variation between FEA results and analytical formulations exist for stress in X axis. The values obtained at the centre and edges are showing a difference of 11% and 6 % respectively. The analytical solutions are over estimated the stress at centre and edges of the plate. The thickness of the plate can be reduced as the maximum stress in the structure is smaller than the yield stress of the material which is 350 MPa. In conclusion, the result from finite element analysis shows good agreement with analytical calculations.

3.2.5. Validation of Coupling between ANSYS®- modeFRONTIER®

The previously analysed plate is used to verify the link between ANSYS® and modeFRONTIER®. The results of optimisation process are highly depending on the connection between various software and tools included in the loop. Therefore a validation study should be carried out to make sure that the coupling between optimisation software and finite element analysis tool is perfect to continue with further optimisations.

The optimisation of a plate with an aim to reduce the weight is carried out using two methods. First one uses ANSYS® as a tool for the calculation of stresses, deflection and weight. Second one uses previously explained empirical relations instead of ANSYS®. The constraint imposed for the optimisation loop is the stress intensity, which should be lower than 270 MPa and the objective function is to minimise the weight of the structure. The lower bounds and upper bounds for the plate considered are 2 mm and 7 mm respectively. Non-dominated Sorting Genetic Algorithm (NSGA-II) employed for the optimisation. The batch script to access ANSYS® is written inside DOSBatch node. The analytical calculations are carried out using the Microsoft Excel node provided in the loop. The optimisation loops for the methods are shown in Fig. 9 and Fig. 10.

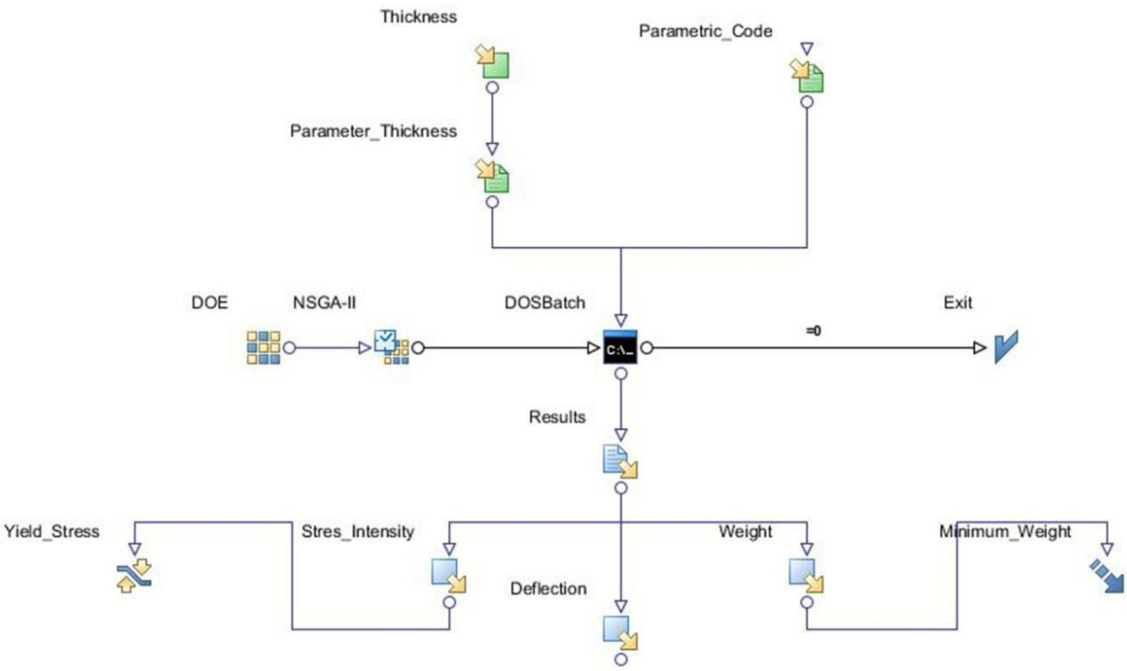


Fig. 9 Optimisation Loop using ANSYS® APDL

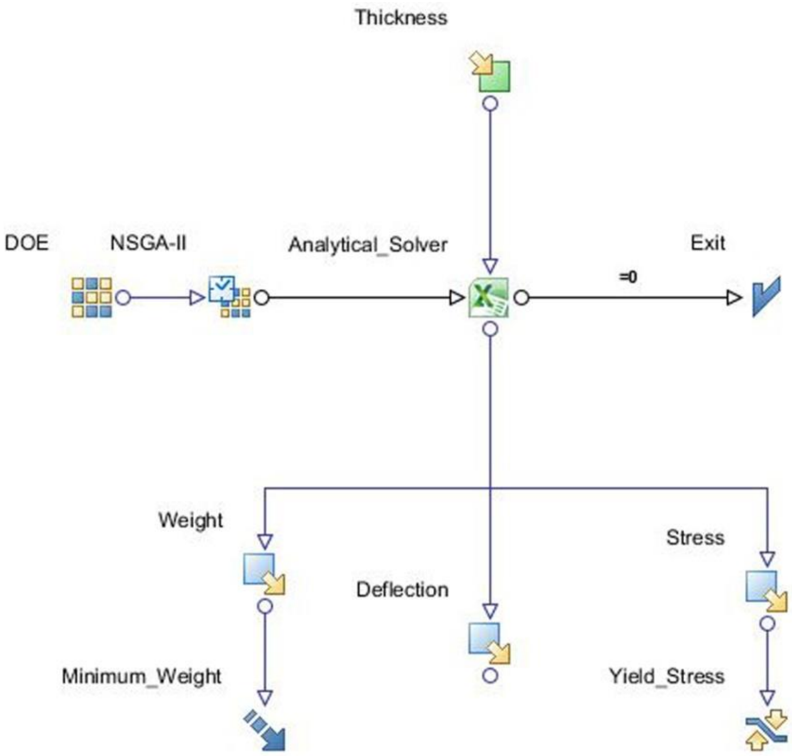


Fig. 10 Optimisation Loop using Analytical Method

The optimisation results from both methods are compared for the stress induced on the plate due to lateral pressure. The converged value of thickness and the corresponding stress value are given in Table 2. The expected thickness is calculated using (11) by assuming maximum stress as -270.00 MPa.

Table 2. Comparison of Optimisation Result

Expected Thickness (mm)	Stress (MPa)	Thickness (mm) Theoretical	Stress (MPa) Theoretical	Thickness (mm) ANSYS	Stress (MPa) ANSYS
2.4056	-270.00	2.4050	-269.91	2.3781	-267.94

The stress against thickness is plotted in Fig. 11. The curves are almost overlapping each other. This indicates that the finite element analysis based optimisation can be carried out by coupling ANSYS® - modeFRONTIER® and the coupled tool provides adequate results.

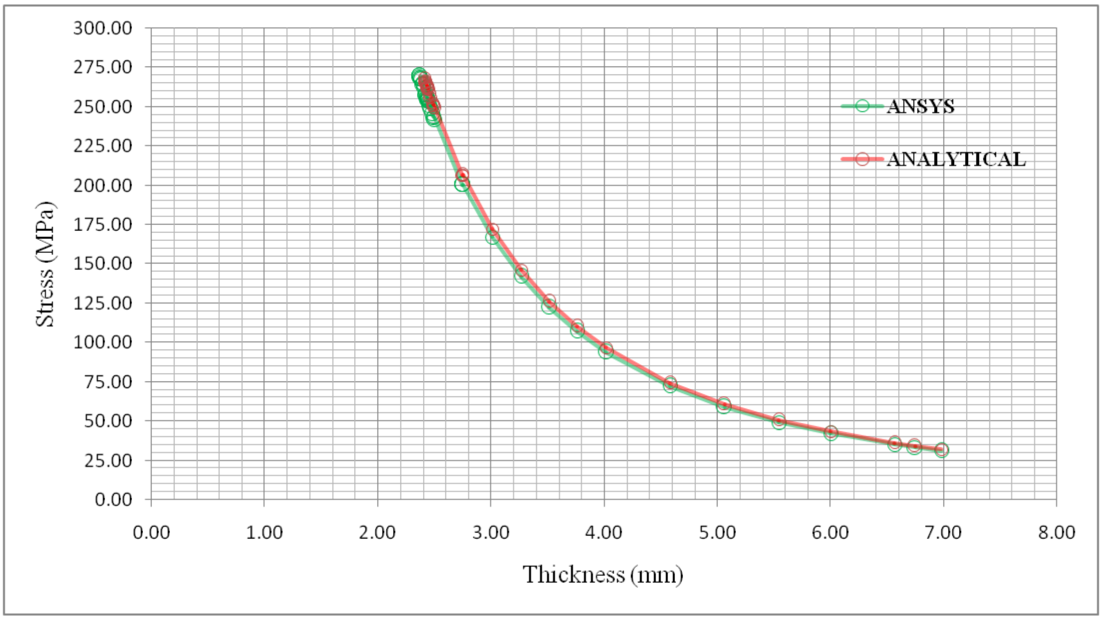


Fig. 11 Comparison between Analytical and FEA results

3.3. Stiffened Panel Optimisation

Structural optimisation of a typical stiffened panel, described in the beginning of this chapter, carried out with an aim to reduce the weight. There are seven design variables considered including plate thickness, longitudinal and transverse stiffener scantlings, and their numbers. The initial design of the stiffened panel has the following scantlings. The plating is 2400 mm long, 800 mm wide and 6 mm thick. The plate is supported by longitudinal stiffeners and transverse frames. Their dimensions are given in Table 3 and the geometry is shown in Fig.7. There are three longitudinal stiffeners and four transverse frames in the initial design.

Table 3. Stiffener Dimensions

h_{wx} (mm)	t_{wx} (mm)	h_{wy} (mm)	t_{wy} (mm)
120	5.5	200	6.5

The yield stress for all materials is 315 MPa. Typical values of upper and lower bound constraints for the plate and stiffener scantlings (Ma et al., 2013) are;

$$\begin{cases} 5\ mm \le t_p \le 40\ mm \\ 100\ mm \le h_{wx} \le 500\ mm \\ 5\ mm \le t_{wx} \le 25\ mm \\ 200\ mm \le h_{wy} \le 800\ mm \\ 5\ mm \le t_{wy} \le 25\ mm \\ 1 \le N_x \le 10 \\ 1 \le N_y \le 10 \end{cases}$$

The objective function and the constraints for the problem can be written as;

$$W = \rho_p\, t_p\, B\, L + \rho_{sx}\, N_x\, L\, h_{wx}\, t_{wx} + \rho_{sy}\, N_y\, B\, h_{wy}\, t_{wy} \tag{12}$$

$$h_{wy} > h_{wx} + 50 \tag{13}$$

A workflow is created in the modeFRONTIER® to carry out the optimisation process. The optimisation loop for the stiffened panel is shown in Fig. 12.

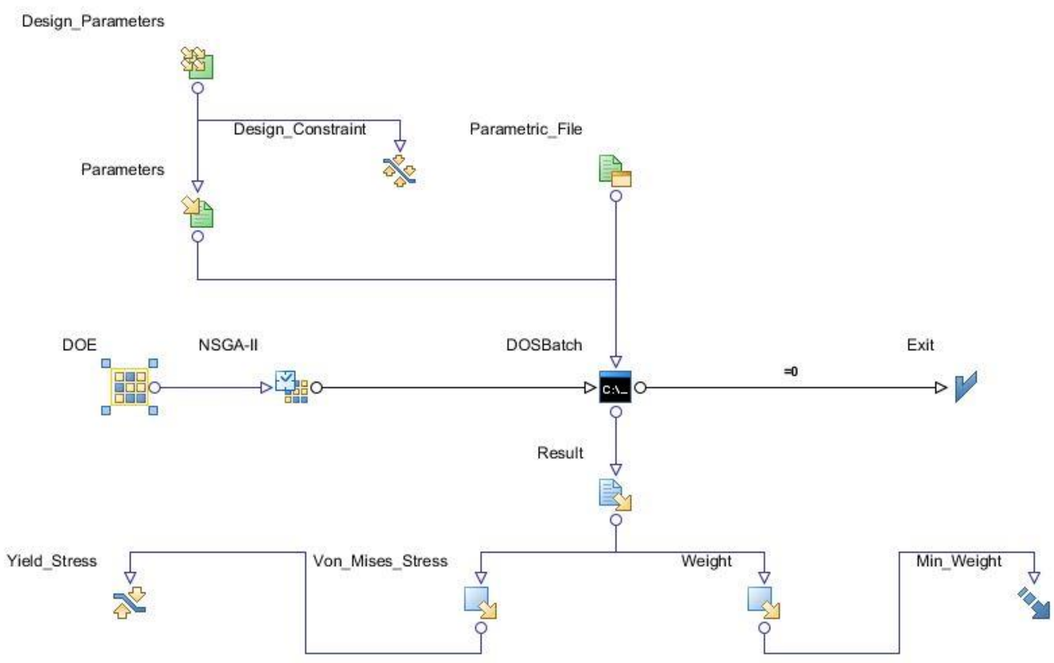


Fig. 12 Stiffened Panel Optimisation Loop using ANSYS® APDL

The DOSBatch node in the loop executes ANSYS in batch mode and analyse the stiffened panel in a similar way as it did in plate optimisation problem. The stiffened panel is created in ANSYS using the macro created in ANSYS Parametric Design Language. The parametric code is capable to modify the design parameters and provide the required results which are von Mises stress and weight of the stiffened panel. SHELL181 element is used to mesh all members of the stiffened panel. A convergence study is carried out to decide on the mesh size and a mesh size of 25x30 mm selected. The convergence study report is given in Appendix A1. NSGA-II algorithm employed to optimise the panel (Braydi, 2017). In addition to the constraints for the stiffener dimensions, two more constraints are employed. First one is that height of the transverse stiffener is always greater than height of the longitudinal stiffeners. This makes sure that the primary supporting member will be frames in the stiffened panel model. The maximum von Mises stress should be less than 274 MPa considering a factor of safety of 1.15 for the material.

The optimisation process started with 11 designs of experiments generated randomly. The first design of experiment is provided according to the initial scantlings of the stiffened panel. The controlling parameters for the algorithm are defined with 11 initial individuals, 55 generations, crossover probability of 0.9 and mutation probability of 1.0. The Optimisation process converged after completing around 305 iterations. The time for one run using a machine with Intel® Core i3, 2.0 GHz CPU and 4GB RAM took approximately 30 seconds. The total process took around 2.5 hours to provide a converged solution. The history of optimisation process is shown in Fig. 13 and the initial and optimised scantlings for the stiffened panel are given in Table 4.

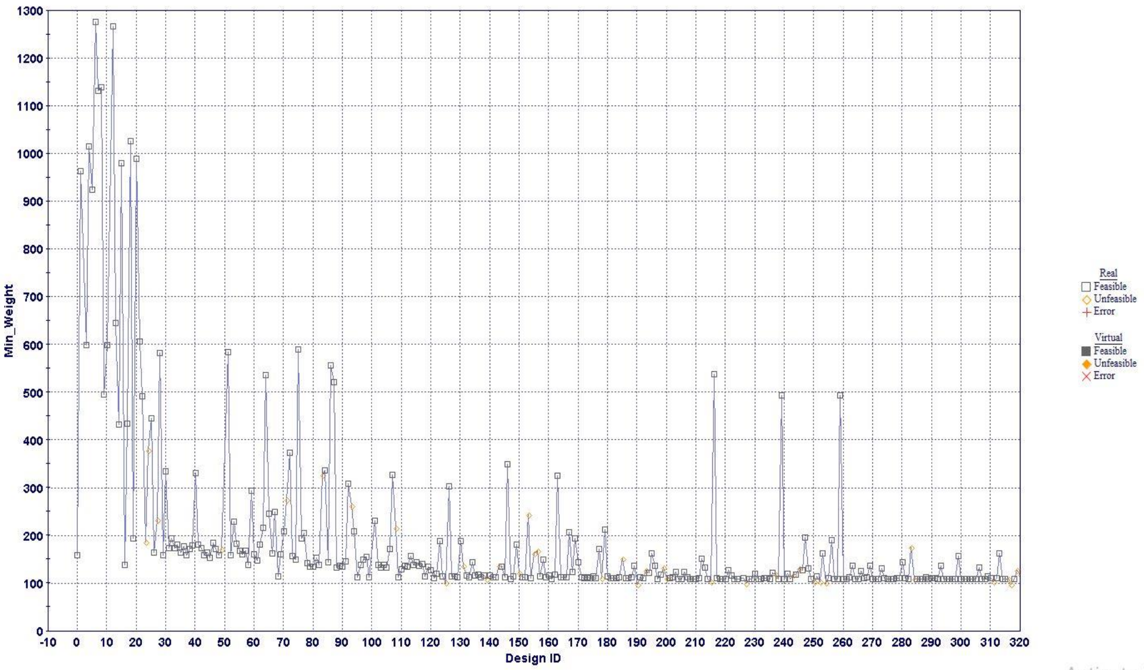


Fig. 13 Convergence History of the Objective Function

Table 4. Comparison of Initial and Optimum Scantling

Variables	Initial Design	Optimum Design
t_p	6.0 mm	5.8 mm
N_x	3	1
h_{wx}	120 mm	100 mm
t_{wx}	5.5 mm	5.0 mm
N_y	4	2
h_{wy}	200 mm	210 mm
t_{wy}	6.5 mm	5.0 mm
Objective: W	159.3696 kg	109.3248 kg

The weight of the stiffened panel is reduced from 159.3696 kg to 109.3248 kg after optimisation. The maximum stress in the optimised panel is 271.6279 MPa. It is evident from the above table that the optimisation provided approximately 31% reduction in the weight of the panel without affecting the structural integrity of the stiffened panel. This will also results in reduction of total cost of the structure. This indicates that the optimisation can play an important role in the reduction of weight in ship and marine structures. Therefore it is advisable to employ structural optimisation process in the early design stage of ship and other marine structures. The optimised stiffened panel is shown in Fig. 14 and the von Mises stress in the stiffened panel before and after optimisation is shown in Fig. 15. The ANSYS® parametric code for the stiffened panel analysis is given in Appendix A2.

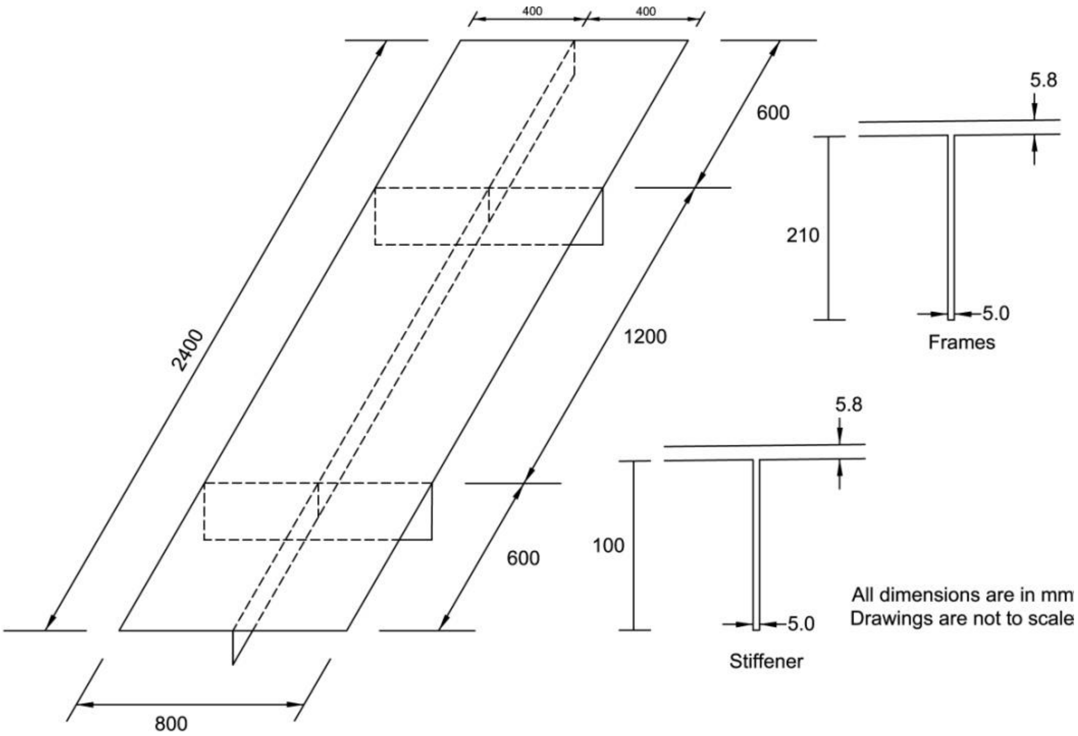


Fig. 14 The optimised stiffened panel

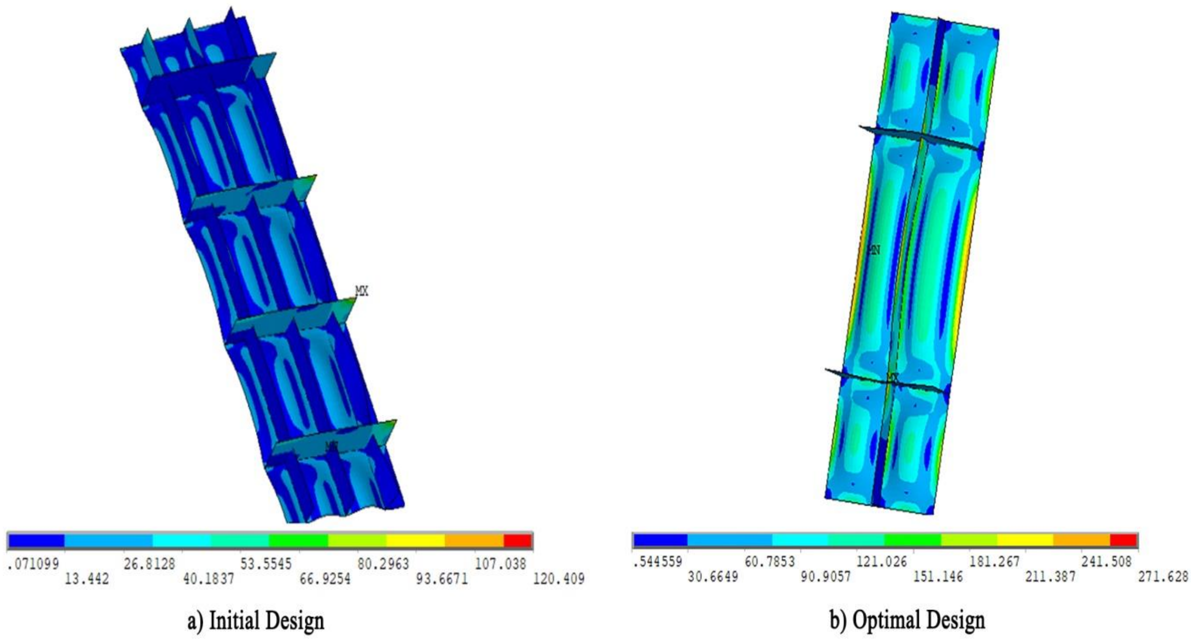


Fig. 15 The von Mises stress contour for the initial and final design

3.4. Response Surface Generation

Surrogate models or meta-models or response surface models can be used as substitutes for complex numerical models which are cheaper to evaluate (Blanning, 1975; Kourakos and Mantoglou, 2013). The response surfaces are capable to represent the relationship between several design variables and one or more response variables. The usage of response surfaces simplifies the complicated simulation optimisations by reducing the computational efforts and time. R is an integrated suite of software facilities for data manipulation, calculation and graphical display (Venables et al., 2018). R has a large collection of packages for data analysis. One of the packages named ‘rsm’, which is used for generating response surfaces in this thesis.

The aim is to create response surfaces representing the minimum weight obtained from the optimisation process done with ANSYS® - modeFRONTIER® loop. The finite element analysis can be replaced with response surface if they can accurately represent the objective of the optimisation process.

3.4.1. Normalisation of Data

One of the preliminary tasks in most of the statistical analysis is data normalisation. The magnitudes of various input variables can be spread over a wide range of values. The final response of the statistical evaluation depends on this spread. So it is necessary to carry out a normalisation procedure to bring all the variables within a specified range of values.

The input data are extracted from the optimisation process carried out earlier. The design variables and the output/response variables are also included in the data extracted. Input data for the R tool from the ANSYS® - modeFRONTIER® loop for the stiffened panel optimisation is shown below. The first few lines are included here.

x1	x2	x3	x4	x5	x6	x7	von_Mises_Stress	Min_Weight
6.00	5.50	6.50	120.00	200.00	3.00	4.00	120.40	159.37
30.60	13.20	9.10	230.00	790.00	1.00	10.00	4.39	963.69
9.00	20.40	18.20	160.00	430.00	2.00	7.00	12.14	598.83
23.90	24.50	9.90	260.00	330.00	5.00	3.00	40.87	1015.32
12.20	12.50	14.30	230.00	470.00	6.00	10.00	6.37	925.02
27.10	23.20	15.20	300.00	460.00	4.00	8.00	6.97	1276.06
38.70	9.20	8.40	320.00	530.00	6.00	8.00	9.83	1132.49

There are seven input variables corresponding to the scantlings of the stiffened panel. They are represented by variables x1 to x7. The response or output variables are named as von_Mises_Stress and Min_Weight corresponds to von Mises stress and minimum weight respectively. The input variables are normalised using the following relations.

Y1 ~ (x1 - 21.85)/16.85; Y2 ~ (x2 - 14.75)/9.75; Y3 ~ (x3 - 12.15)/7.15; Y4 ~ (x4 - 210)/110; Y5 ~ (x5 - 495)/295; Y6 ~ (x6 - 5)/4; Y7 ~ (x7 - 6)/4.

Where Y1, Y2..., Y7 are normalised variable corresponding to input variables x1, x2..., x7 respectively. All the input variable values lie in a range between -1 and 1 once the normalisation procedure is completed. The procedure used for normalisation of variables is explained below.

Step 1: Identify the minimum and maximum values of variables in each column

Step 2: Calculate the average of sum of maximum and minimum value in each column

$$\text{Average Sum} = \frac{\text{Maximum value} + \text{Minimum value}}{2} \tag{14}$$

Step 3: Calculate the average of difference of maximum and minimum value in each column

$$\text{Average Difference} = \frac{\text{Maximum value - Minimum value}}{2} \tag{15}$$

Step 4: Use the following relation for creating normalised variable for each column

$$\text{Normalised Variable} = \frac{(\text{Design Variable - Average Sum})}{\text{Average Difference}} \tag{16}$$

3.4.2. Response Surface using Polynomial Regression

The package called ‘rsm’ can be used for standard response surface methods based on first or second order polynomial regression models. Special functions like FO, TWI, PQ or SO (‘First-Order’, ‘Two-Way Interaction’, ‘Pure Quadratic’ and ‘Second-Order’ respectively) are included in the package and which specifies the response surface portion of the model. Multiple response optimisations are not covered in the package. Response surfaces are established for each variable separately in case of multiple response variable problems based on the same design data.

The ‘rsm’ package is used to establish a response surface for the minimum weight, which is the output/response variable. Also a different response surface generated to accurately represent the von Mises stress in the stiffened panel. The polynomial expressions representing the Min_Weight and von_Mises_Stress are generated as second order to achieve more accuracy than first order polynomial. R tool has no GUI. Appropriate coding is necessary to run the data analysis in R. The codes for the generation of response surfaces are given in Appendix A3 and the execution of code will provide results in the following format.

	Estimate	Std. Error	t value	Pr(>|t|)	
(Intercept)	836.09904	3.71442	225.095	< 2.2e-16	***
Y1	233.42986	5.91932	39.4353	< 2.2e-16	***
Y2	168.72417	3.81434	44.2342	< 2.2e-16	***
Y3	116.81434	5.01505	23.2928	< 2.2e-16	***
Y4	155.13404	5.14019	30.1806	< 2.2e-16	***
Y1:Y4	-32.46640	7.07821	-4.5868	6.98E-06	***
Y1:Y5	16.12237	3.15572	5.1089	6.25E-07	***
Y1:Y6	5.33439	2.4532	2.1745	0.0305652	*
Y1:Y7	2.50389	1.89946	1.3182	0.1885829	
Y2:Y6	90.76182	4.03981	22.4669	< 2.2e-16	***
Y2:Y7	-10.15464	3.08892	-3.2874	0.0011492	**
Y3:Y4	-0.34486	6.79096	-0.0508	0.9595379	
Y3:Y5	76.85179	3.54236	21.6951	< 2.2e-16	***
Y3:Y6	-16.71419	2.09977	-7.96	5.23E-14	***
Y1^2	-3.47749	2.48775	-1.3978	0.1633412	
Y2^2	-11.18839	2.17847	-5.1359	5.49E-07	***

The full result is attached with the codes in Appendix A3. The result provides the correlation between various variables and their significance in the value of output variable. The significant variables and their correlation influencing the output variable are identified by checking the *p*-value, which is *Pr (>|t|)* in the result. The significant terms are the ones with p value less than 5%. Estimate provides the weightage of each variable on the output variable. The p value is the probability of obtaining a result equal to or more extreme than what was actually observed under the null hypothesis. Smaller the p value higher is the significance. The column ‘t value’ shows the *t*-test associated with testing the significance of the parameter listed in the first column and is obtained by dividing the ‘Estimate’ by ‘Std. Error’. The response polynomial generated for estimating the weight of the structure can be written as:

W = 836.09904 + 233.42986*Y1 + 168.72417*Y2 + 116.81434*Y3 + 155.13404*Y4 + 114.26292*Y5 + 190.34547*Y6 + 136.51625*Y7 - 19.93504*Y1*Y2 + 12.16043*Y1*Y3 - 32.46640*Y1*Y4 + 16.12237*Y1*Y5 + 5.33439*Y1*Y6 - 13.15543*Y2*Y3 + 114.30040*Y2*Y4 + 90.76182*Y2*Y6 - 10.15464*Y2*Y7 + 76.85179*Y3*Y5 - 16.71419*Y3*Y6 + 41.21212*Y3*Y7 - 54.10183*Y4*Y5 + 70.96034*Y4*Y6 + 19.71127*Y4*Y7 + 58.45976*Y5*Y7 - 11.18839*Y2^2 + 21.54268*Y4^2 + 3.36939*Y7^2

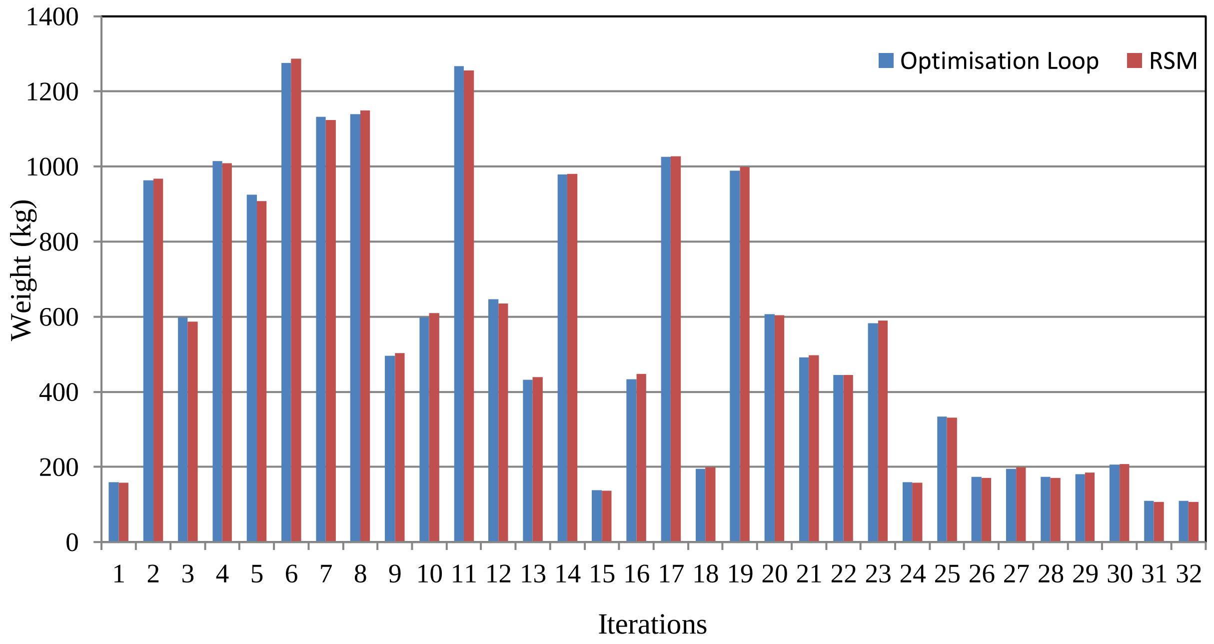
The weight of the stiffened panel is calculated with the above expression and compared with that of the values obtained from ANSYS® - modeFRONTIER® optimisation loop in Table 5. The differences between the values are also mentioned in the Table.

Table 5. Comparison of Weight based on RSM and Optimisation Loop

Weight (kg)		Difference (%)
Optimisation Loop	RSM	
159.37	158.21	0.73
963.69	968.25	-0.47
598.83	587.58	1.88
1015.32	1008.58	0.66
925.02	908.48	1.79
1276.06	1286.87	-0.85
1132.49	1123.49	0.79
1139.52	1150.19	-0.94
496.45	503.14	-1.35
598.74	609.89	-1.86
1267.53	1256.47	0.87
646.03	634.95	1.71
431.83	438.64	-1.58
979.40	980.77	-0.14
138.84	136.55	1.65
434.19	447.75	-3.12
1025.62	1027.43	-0.18
194.79	199.06	-2.19

989.02	998.90	-1.00
606.32	604.46	0.31
492.21	498.05	-1.19
444.94	445.55	-0.14
582.57	590.08	-1.29
159.37	158.21	0.73
334.10	331.49	0.78
174.35	170.28	2.33
194.79	199.06	-2.19
174.35	170.28	2.33
180.94	185.00	-2.25
205.67	207.89	-1.08
109.32	106.52	2.57
109.32	106.52	2.57

It is evident from the table that the percentage error between the weight estimation between FEM and RSM is small. The maximum percentage error in the estimation is less than 4%. The results prove that the RSM is a good alternate for the weight estimation of stiffened panel. The advantage of replacing RSM in the optimisation loop is that the calculation time can be reduced significantly. Response polynomials similar to weight estimation can be generated for the stress calculation also. The results tabulated are plotted below.



4. MIDSHIP REGION OPTIMISATION OF RO-PAX VESSEL

The Ro-Pax vessels are designed to transport vehicles and passengers efficiently. The shipyards and owners of the vessels are always desire for improvements in design, weight reduction, constructional cost, operational cost, etc. These requirements force the engineer to look for an optimal ship structural design. Once the general characteristics such as main dimensions, coefficients of form of a ship design have been finalised, the midship structural design can be worked out as a basic and initial structural problem. The major part of the hull section will follow the pattern of the midship section if the midship structural section is specified in early design stage. The scantlings toward each ends of the ship can be given as modifications to the midship section as the bending moments and shear forces are highest between quarter points of the hull. The midship section design of a Ro-Pax vessel gives an approximate estimation of hull weight. It can also be used for the approximate estimation of cost for the bidding.

4.1. Structure

A typical Ro-Pax vessel is considered and its midship region is optimised. The main particulars (approximate values) of the ship are given in Table 6. Due to the confidentiality of the design and drawings, layout of the midship with approximate dimensions is presented in Fig. 16.

Table 6. Main particulars of the Ro-Pax vessel

Length overall	220.00 m
Length between perpendiculars	210.00 m
Breadth	30.00 m
Depth	9.0 m
Draft	6.50 m
Block coefficient	0.629
Displacement	28000.00 t

The ship has 8 decks and a double bottom. The web frame spacing is 2400 mm. There are three accommodation decks and four cargo decks. The longitudinal bulkhead is situated approximately at 9500 mm from the centre line and extending between the inner bottom and the 2nd deck. HP profiles are used for stiffening decks and longitudinal bulkheads. The materials for the construction are ordinary mild steel and high strength steel. The yield stresses for the mild steel and high strength steel are 235 and 355 MPa respectively.

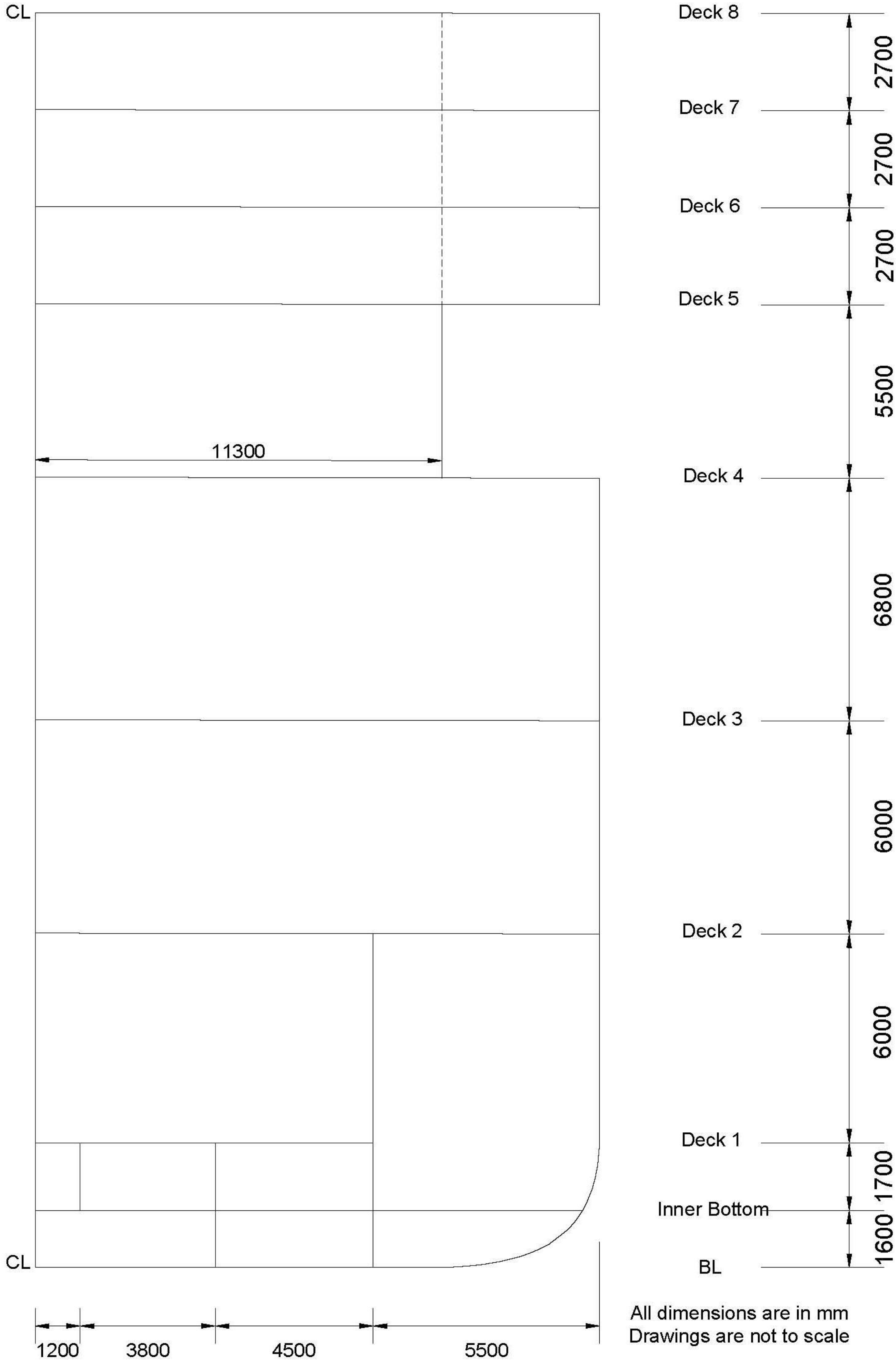


Fig. 16 Layout of the midship for the Ro-Pax vessel

4.2. Design Loads

The structural strength of ships can be realistically determined only if the loads are assessed properly. The loading conditions for the Ro-Pax vessels are identified based on BV rules for steel ships. There are eleven loading conditions are critical for the vessel. Each load case includes three types of loads; structure weight, dead weight items such as trailers, trucks, cars as deck pressure and buoyancy loading and dynamic sea pressure. The eleven load cases are summarised in Table. 7.

Table 7. Loading conditions

Load Conditions	Description
1	Full load on decks + (a+) + Sagging
2	Full load on decks + (a+) + Hogging
3	Full load on decks + (a-) + Sagging
4	Full load on decks + (a-) + Hogging
5	Full load on decks + (b) + Sagging
6	Full load on decks + (b) + Hogging
7	Full load on decks + (c+) + Sagging
8	Full load on decks + (c+) + Hogging
9	Full load on decks + (d+) + Sagging
10	Full load on decks + (d+) + Hogging
11	Ballast Condition + (a+) + Hogging

4.2.1. Hull Girder Loads

Hull girder loads can be divided into static and dynamic components. The static component consists of still water bending moments and shear forces. They are induced due to the difference between the distributions of the light ship weight, cargo, etc. and the opposing buoyancy forces along the ship length. Hydrodynamic loads due to wave, sloshing, slamming, etc. are come under dynamic component. The wave induced dynamic loads include the vertical and horizontal shear forces, bending moments and torsional moments. The classification societies are published various rules to calculate the hull girder bending moments and shear forces. BV Rules NR467 Part B, Chapter 5, Sections 2 provides the equations to calculate the hull girder loads for steel ships.

4.2.2. Local Loads

Local loads are directly applied to individual structural members such as plate panels, ordinary stiffeners and primary supporting members. They include pressure and deck loads.

i. *Sea Pressure Loads*

The pressure experienced on the ship hull is mainly due to sea pressures. This can be divided into two; still water pressure and wave pressure. The calculation of still water and wave pressure are done using BV Rules NR467 Part B, Chapter 5, Section 5. The distribution of still water pressure is shown in Fig. 17.

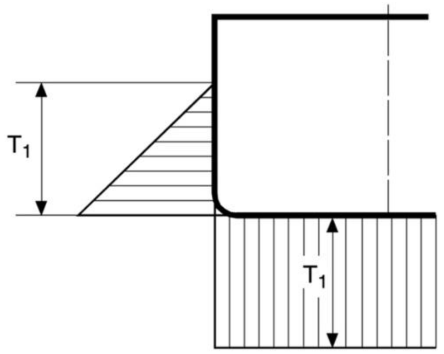


Fig. 17 Still water pressure on the hull [8]

The wave pressure components defined in the Table 7 are explained below.

1. Load case "a"

It is considered when ship is at rest and encountered with a wave which produces a relative motion of the sea waterline, both positive and negative, which is symmetric on both sides. Load case "a" is divided into two; load case "a+" and load case "a-". Load case "a+" is applicable when the ship encounters with a wave crest and results in a positive relative motion of sea waterline with a relative increase in height (h_1) from the still waterline. When the ship encounters with a wave trough and produces a negative relative motion of sea waterline with a relative decrease in height (h_1) from the still waterline.

2. Load case "b"

The wave is considered to induce heave and pitch motions in addition to the motions considered in load case "a". Load case "b" is used when the ship encounters with a wave and produces a positive relative motion of sea waterline with a relative increase in height ($0.5*h_1$) from the still waterline. Still water and wave pressure loads are to be considered for load case "a" whereas inertial loads are also applicable in load case "b" in addition to still water and wave pressure loads. The wave pressure distribution in load cases "a+", "a-" and "b" are shown in Fig. 18.

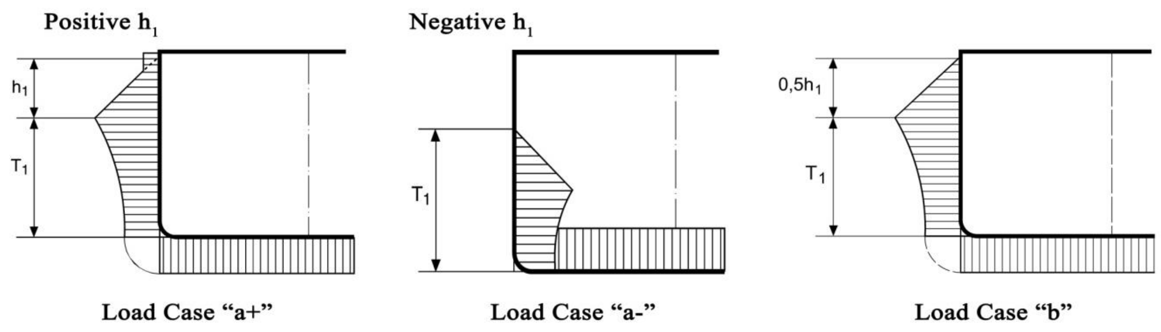


Fig. 18 Wave loads in Load case "a+", "a-" and "b" [8]

3. Load case "c" and "d"

When the ship is encountered with a wave that induces sway, roll and yaw motions and also a relative motion of the sea waterline which is anti-symmetric on the ship sides, then the load cases "c" and "d" are to be considered. These load cases induces vertical and horizontal wave bending moments and vertical shear forces in the hull girder. Load case "c" also induces torque in the hull girder in addition to the moments and shear forces in the hull girder. When a wave crest is considered at one side of the ship, there will be a negative sea waterline on the other side of the ship in these load cases.

Load case "c+" and "c-"are considered when the ship encounters a wave and produces a positive relative motion of sea waterline on the side of ship under consideration with a relative increase in height (h_2) from still water line and a negative relative motion of sea waterline on the side with a relative decrease in height (h_2) from still waterline respectively.

Load case "d+" is applied when the ship encounters a wave crest and results in a positive relative motion of sea waterline on the side of consideration with a relative increase in height ($0.5*h_2$) from still waterline. Whereas load case "d-" is applied when the ship encounters a wave crest and results in a negative relative motion of sea waterline on the side of consideration with a relative decrease in height ($0.5*h_2$) from still waterline. The wave pressures for load cases "c" and "d" are shown in Fig. 19.

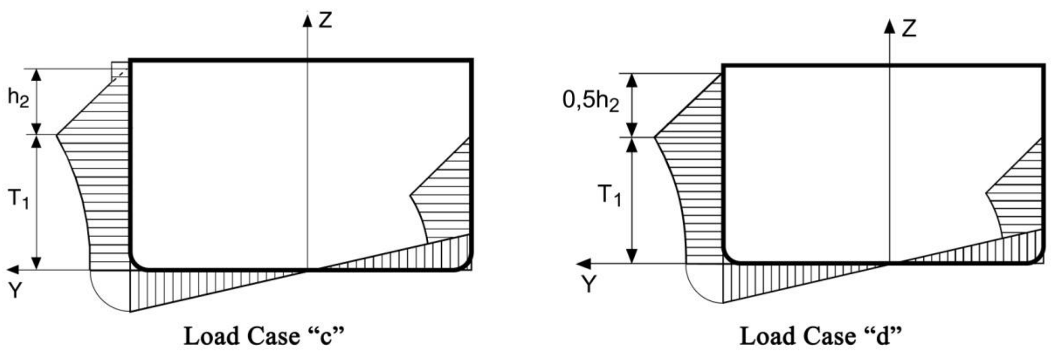


Fig. 19 Wave loads in load case "c" and "d" [8]

ii. Wheeled and Accommodation Loads

The loads on the deck are found out by using the BV rules NR467 Part B, Chapter 7, Sections 3. BV MARS2000 software provided the values based on the requirements. The assumption is that the forces transmitted through the tyres are comparable to the uniformly distributed pressure on the tyre print. These forces are considered to be applied as concentrated load in the tyre print centre.

The requirement for the application of deck loads as concentrated loads will be a fine mesh in finite element analysis. Since the present study focussing on the feasibility of developing an integrated platform for the optimisation using finite element analysis, the deck loads are applied as uniformly distributed loads over the section considered. The accommodation loads are not available at this point of study.

4.3. Finite Element Model

The CAD model of the Ro-Pax vessel is provided by AVEVA Marine. The tool is capable to generate the initial design of the ship only. The code is modified to adapt to the changes which arise during the optimisation process. The CAD model of the ship (between 80m from aft to 147.20m) generated using the ANSYS® APDL compatible code obtained from AVEVA Hull Structural Design software is shown in Fig. 20.

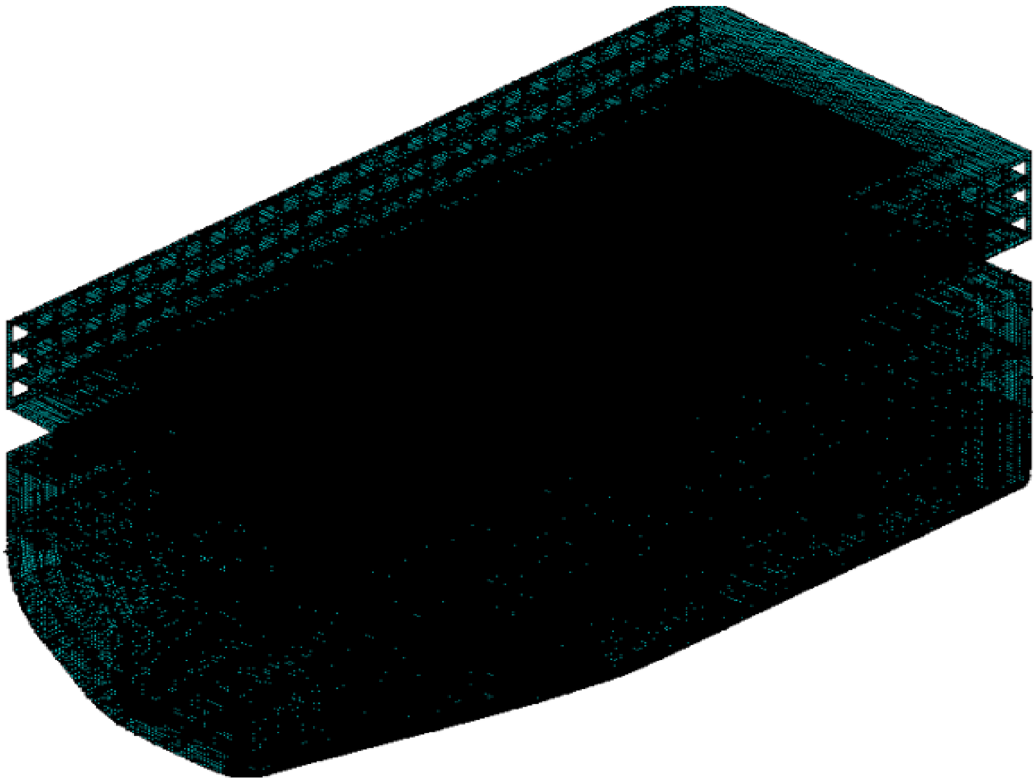


Fig. 20 CAD model of the Ro-Pax vessel

A segment between 2 main frames is extracted from the ship model using custom made codes to optimise the structural scantling of the midship section. The extracted model is 4.8 m in length and is shown in Fig. 21. Ma et al. (2014) used a segment between two frames of a double hull oil tanker to optimise the structural scantling of the midship section. The numerical calculations carried out by Ma et al. on the segment provided good results and the proposed approach is capable to generate better midship section designs with reasonable time. The author's codes are capable to modify the parameters such as plate thickness, stiffener spacing and stiffener profiles. They are considered as the design parameters for the optimisation process. The mesh size considered for the analysis is 400x400 for plates. The mesh size requirement from the classification society is that the mesh size should as close as stiffener spacing [9]. The minimum stiffener spacing considered in the model is 400mm. The plates and stiffeners are modelled using SHELL181 and BEAM188 elements respectively. The girders are also modelled using SHELL181 elements.

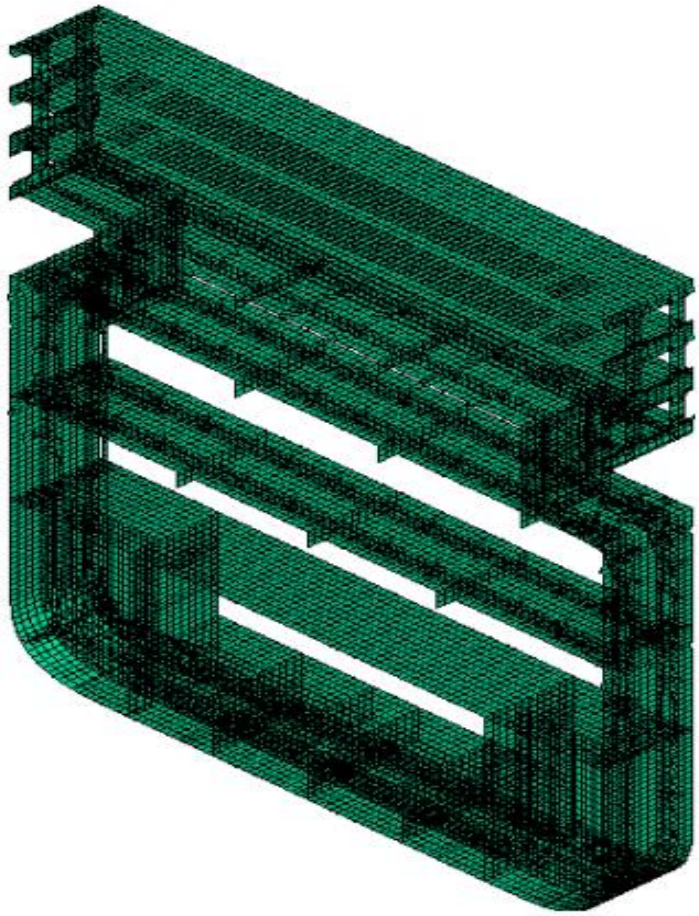


Fig. 21 Midship section model

Finite element analysis carried out on the midship section model considering the load condition 2, involving deck loads, sea pressures and hogging bending moment. This is because the hogging bending moment is higher than the sagging moment for the vessel considered. Other load conditions have to be considered in the future works. Hull girder bending moments and deck loads calculated are given in Appendix A4. Loads obtained from CFD analyses will be used to replace the loads calculated based on rules in future developments. Two rigid regions are created using CERIG command to apply the vertical bending moment to the model. The super structure is not attached with the rigid elements. The super structure consists of only accommodation decks and they are assumed to be not contributing to the hull girder bending moment. The model is shown in Fig. 22.

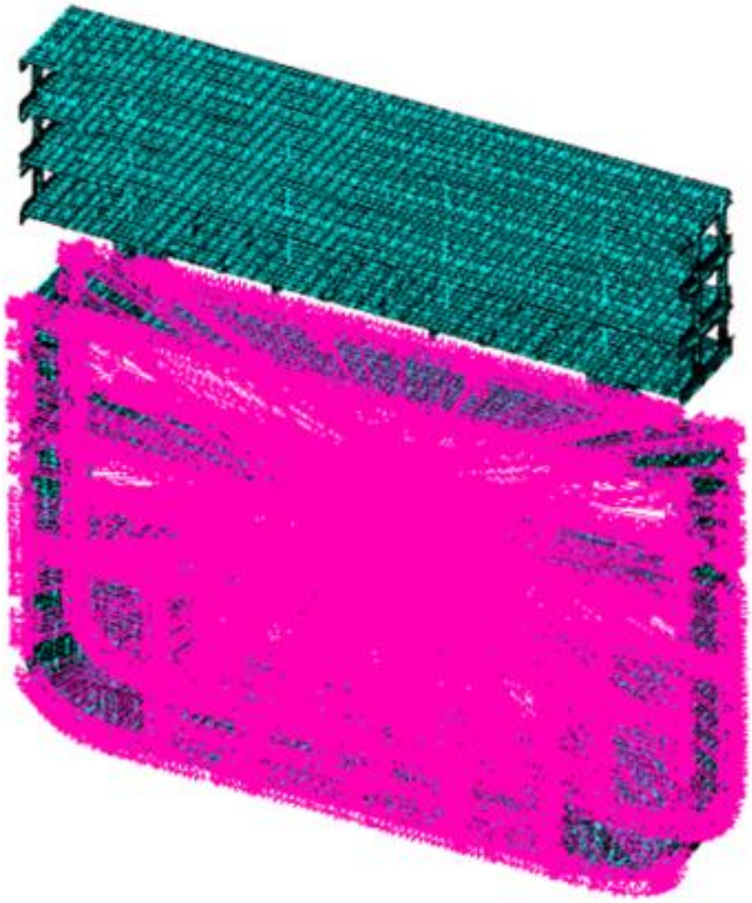


Fig. 22 Midship section model with two end rigid regions

The material properties used in the finite element model are given in Table 8. There are two different materials used in the model.

Table 8. Material properties

Material	Young’s modulus (MPa)	Poisson’s ratio	Yield Strength (MPa)
Mild Steel	206000	0.3	235
High Strength Steel	206000	0.3	355

4.4. Midship Region Scantling Optimisation

The optimisation of the midship region of Ro-Pax vessel carried out using the above segment selected from the CAD model. The midship structural optimisation involves large number of variables such as plate thickness, scantlings of stiffener and stiffener spacing. The schematic of optimisation flow is shown in Fig. 23.

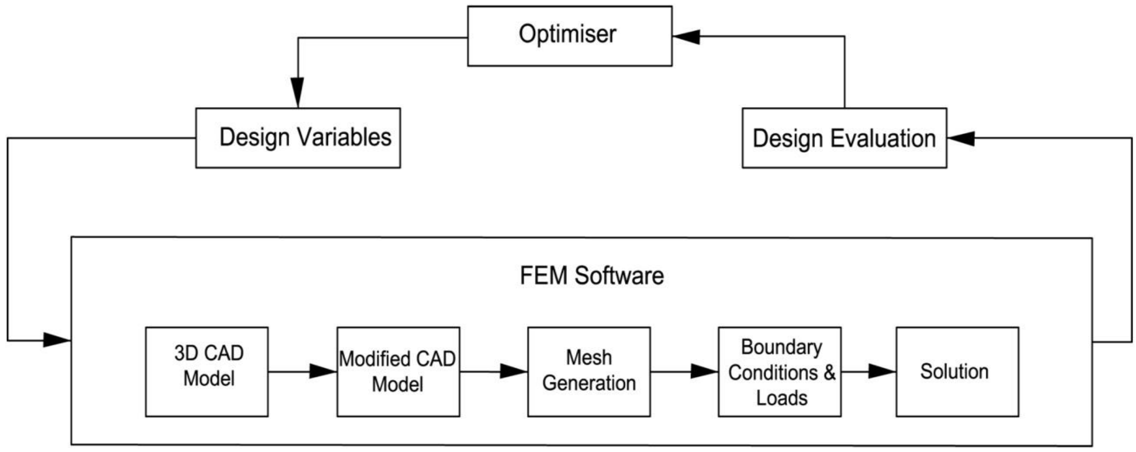


Fig. 23 Schematic of optimisation flow

The initial CAD model is transferred to the FEM software and modifications are done with the design variables values using author's codes. The structural model is meshed and loaded and analysed in the FEM software. The structural integrity of the model is always investigated during the optimisation process. The finite element software provides the results for the objective function (minimum weight) and the constraints (stresses) specified. These values are transferred to the optimiser tool and evaluated in order to modify the design variables (plate thicknesses, stiffener spacing and stiffener profiles) and to create a new structural model. The optimisation process continues until the converged solution is obtained.

Ship design is a typical mathematical optimisation problem involving many design variables objectives and constraints. It makes the ship design optimisation a complex process. Further complications arise when the plates and stiffeners are constrained by yielding under various load conditions and subject to practical design rules (Ma et al. 2014). Typical objectives in ship design optimisation process are to minimise weight, ship building, and operational cost and take care of environmental criteria like accidental oil outflow, EEDI (Energy Efficiency Design Index), etc. Ship design optimisation is usually carried out with multiple objectives (Papanikolaou, 2015).

The current study focused only on one objective, which is to reduce the weight of the model selected. The optimisation results give an idea about the total weight reduction possible for the ship, which in turn helps the designer to calculate the savings in terms of quantity of steel and production cost. Fig. 24 shows the loop set up in modeFRONTIER® for optimising the midship region. Different nodes included in the loop are explained below.

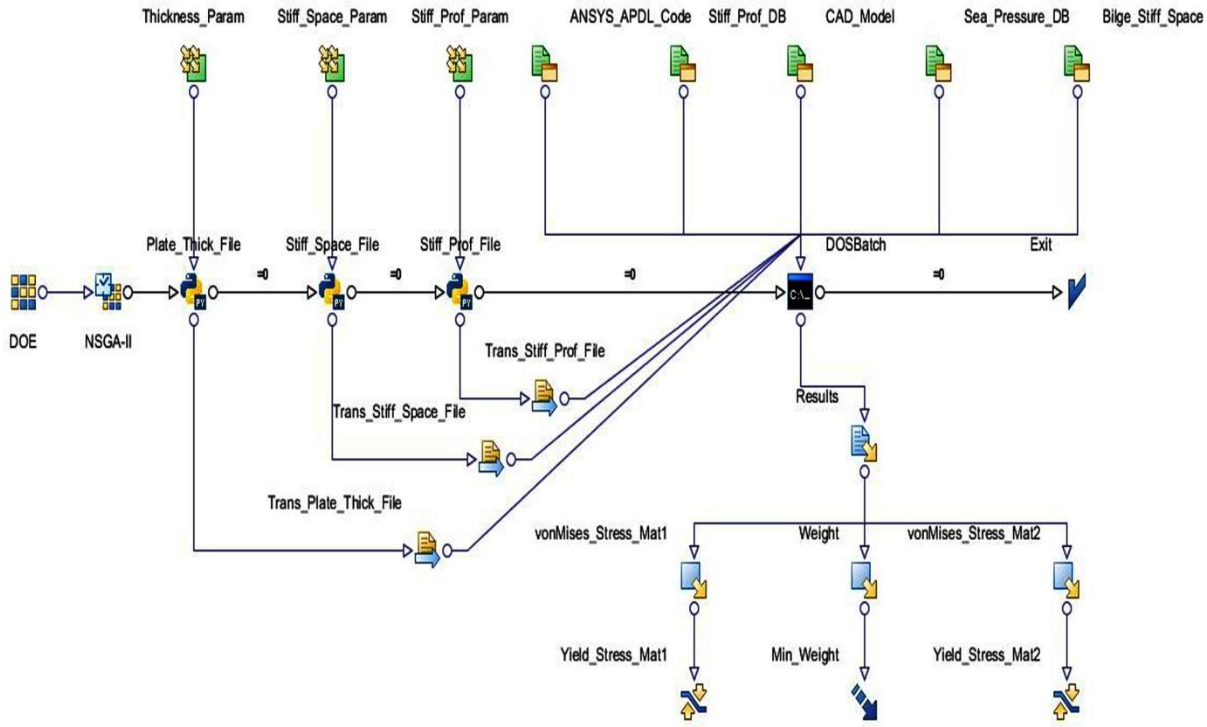


Fig. 24 Midship structural optimisation loop

There are three input vectors, Thickness_Param, Stiff_Space_Param and Stiff_Prof_Param, defined in the loop to include the design variables namely plate thicknesses, stiffener spacing and stiffener profiles respectively. The upper and lower limit of the design variables are given by equation (17). There are 58 design variables considered in the optimisation; 30 design variables to represent plate thicknesses, 10 variables for stiffener geometry and 18 variables for stiffener spacing.

$$\begin{cases} 6\,mm \leq t_p \leq 17\,mm \\ 400\,mm \leq S_s \leq 700\,mm \end{cases} \tag{17}$$

The prefabricated stiffener profiles data is given by the shipyard and they are used in the loop. They are defined in the support file node Stiff_Prof_DB. The node named ANSYS_APDL_Code contains the codes to access the CAD file, modify the parameters, mesh the model, apply the boundary conditions, loads and carry out the finite element analysis. The FEA results are written into an output file by the code defined inside the node. Sea_Pressure_DB node provides the sea pressures on the hull surface. The sea pressure includes the static and wave pressure components. Load case “a” is only considered in the wave pressure calculation. CAD_Model support file node is defined to supplies the initial CAD model for the optimisation.

There are three python script nodes and transfer nodes are implemented to transfer the input files into the working folder. In-house tools are developed, such as for creating bilge stiffener locations, identifying the sea pressure corresponds to nodes, etc. and used in the optimisation loop. The DOSBatch node executes the ANSYS® in batch mode and carries out the finite element analysis. The in-house tools are also called through the DOSBatch node.

The Results node contains the output data received from ANSYS after FEA. They are the vonMises stress values and weight of the structure. The objective node is used to minimise the weight of the structure. The constraints nodes named Yield_Stress_Mat1 and Yield_Stress_Mat2 restrict the vonMises stress in the structural components made of Material 1 and Material 2 to go beyond the maximum allowable stress values for the corresponding materials. The maximum allowable stress value for the materials is calculated using equation (18). The equation is defined in BV Rules NR467, Part B, Chapter 7, Section 3.

$$\sigma_{VM} = \frac{R_y}{\gamma_R \gamma_M} \tag{18}$$

The resistance partial safety factor, γ_R and material partial safety factor, γ_M take the values 1.2 and 1.02 respectively [8]. Only structural constraints are included in the present study. More constraints are added in further developments.

NSGA-II algorithm employed for optimisation even though it is a multi-objective evolutionary optimisation algorithm. There are 50 initial design of experiments considered and the first one was the initial scantlings provided by the shipyard. Fig. 25 shows the variation of weight over iterations. It clearly shows that the algorithm is searching for an optimum solution.

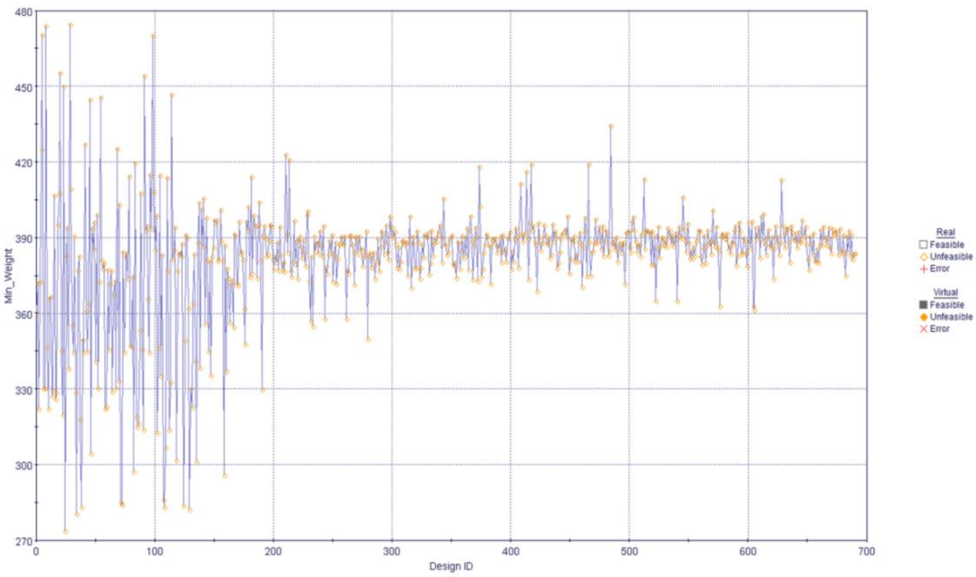


Fig. 25 Convergence history of the objective function: Structural weight

The optimisation converged after finishing around 300 iterations. But the optimisation process failed to give a feasible solution. The von Mises stress in the material 2 resulted in FEA was higher than the yield stress. The stiffeners specified using beam elements are failed to participate in force transfer. This is due to the problems involved in the initial CAD model. The von Mises stress in the material 1 always satisfied the constraint implemented. There is no meaning in comparing the initial and optimised design as the optimisation process failed to provide a feasible solution.

The convergence of optimisation shows the ability to couple modeFRONTIER®-ANSYS® software for structural optimisation of mid-region of Ro-Pax vessel, which was one of the primary objectives.

4.5. Response Surface Generation

Response surface is created for the objective function using the data obtained from optimisation process carried out using modeFRONTIER®-ANSYS® loop even though the optimisation didn't provide feasible solutions. The response surface is created in order to show the efficiency of response surface polynomials to represent the processes in the loop. The same procedure used to create weight function for stiffened panel is employed for the creation of response surface for midship region also. The response surface for the weight is created using polynomial regression. Around 300 designs are considered for the generation of response surface. The polynomial relation for the structural weight and the input variables developed using R is added below. The full code and results are given in Appendix A5.

	Estimate	Std.Error	t value	Pr(>|t|)	
(Intercept`	365.924	0.4764	768.166	< 2e-16	***
Y1	4.3065	0.9366	4.598	6.46E-06	***
Y2	1.7331	1.0476	1.654	0.099191	.
Y3	8.4978	1.2098	7.024	1.62E-11	***
Y4	1.6305	0.9874	1.651	0.099793	.
Y5	18.1669	1.2548	14.477	< 2e-16	***
Y6	2.3236	1.0006	2.322	0.020937	*
Y7	-0.2772	0.9696	-0.286	0.775161	
Y8	21.1239	1.0018	21.086	< 2e-16	***
Y9	0.622	0.8806	0.706	0.480535	
Y10	3.3606	1.0449	3.216	0.00145	**
Y11	7.6034	0.9676	7.858	8.35E-14	***
Y12	-2.3262	1.0371	-2.243	0.025682	*
Y13	9.8896	0.9863	10.026	< 2e-16	***
Y14	3.5447	0.9011	3.934	0.000106	***
Y15	-2.2277	1.0706	-2.081	0.038361	*
Y16	1.0269	0.8972	1.145	0.253385	

The structural weight obtained from optimisation loop and the same calculated using response polynomial is compared in Table 9.

Table 9. Comparison of Weight based on RSM and Optimisation Loop

Weight (tonnes)		Difference (%)
Optimisation Loop	RSM	
340.97	339.75	-0.36
399.19	399.76	0.14
330.54	332.43	0.57
372.77	374.20	0.38
445.89	441.34	-1.02
381.11	390.56	2.48
377.65	382.03	1.16
379.94	379.67	-0.07
322.17	319.17	-0.93
323.13	324.41	0.40
377.49	369.49	-2.12
345.68	348.32	0.77
371.68	374.62	0.79
377.21	373.42	-1.00
329.09	332.53	1.05
367.63	371.68	1.10
372.99	376.31	0.89
330.53	330.79	0.08
425.48	432.12	1.56
333.56	334.39	0.25
403.27	402.43	-0.21
284.96	285.88	0.32
284.23	279.41	-1.70
384.37	388.28	1.02
344.55	352.93	2.43
382.56	384.52	0.51
384.24	383.19	-0.27
373.82	376.20	0.64
340.97	339.75	-0.36
399.19	399.76	0.14

The first order response surface is generated to represent the weight of the midship section selected earlier. The second order polynomial generation takes more time than first order, since the calculation involves large number of design variables. The percentage error values in the table show that there is a good agreement between the weights obtained from the optimisation loop and that calculated based on RSM. This proves that the RSM can be used to obtain an initial weight estimate of the vessel considered. The usage of RSM saves the calculation time compared to an optimisation loop with FEA. The time taken to complete one iteration in modeFRONTIER®-ANSYS® automatic loop is approximately 3 minutes in a machine with Intel® Core i7, 2.80 GHz CPU and 12GB RAM whereas the RSM provided the results in 30 seconds once the response surface is generated. The time required to generate a response surface for 58 design variables included problem is approximately 10 minutes. The total time required to run modeFRONTIER®-ANSYS® automatic loop is around 20 hrs to complete approximately 400 iterations where as RSM can provide results in 3.5 hrs for the same number of iterations. This is quite a big saving of time in the initial stage. The time required to create response surfaces can be further reduced using automatically generating response surfaces.

5. CONCLUSIONS AND RECOMMENDATIONS

5.1. Conclusions

The present thesis developed on the work carried out in University of Liege as part of European Union project HOLISHIP.

The preliminary objective was to study the feasibility of an automated structural optimisation loop for Ro-Pax midship region by coupling ANSYS® and modeFRONTIER® software. The automated loop avoids the manual intervention on graphical user interface during optimisation iterations. The optimisation processes explained in the thesis involve only one objective, minimising the weight of the structure, which is achieved through the work. The results from the optimisation processes show that integration of software is possible and can be extended to multi objective optimisation problems too. The optimisation loop can be applied to the optimisation of midship section of Ro-Pax vessel in the early stages of design. The same loop can be implemented for any other kind of ships by modifying only the author's parametric code created using ANSYS parametric design language.

The results in the thesis show that significant reduction of weight is possible through the structural optimisation. There will be a large reduction in ship weight if the entire ship is able to build with optimum scantlings. Based on BV rules for steel ships design constraints and loads are applied and the critical constraint was von Mises Stress. The von Mises stresses obtained from the analysis are limited by yield stress of the materials. Local loads are also considered in addition to the hull girder loads and the structural behaviour of the model and stresses developed in the structure are studied.

The coupling between ANSYS® and modeFRONTIER® software are verified by optimising a simple plate. The results obtained from the coupled loop are in good agreement with the optimisation carried out using analytical solutions for plates. The study also extended to the optimisation of a stiffened panel.

A large number of studies are available in the area of optimisation of ships and especially Ro-Pax vessels. A detailed literature study has been carried out mainly focussing on the structural optimisation of Ro-Pax vessels. The previous studies are mainly focussed on multi objective optimisations and implemented tools that are created specifically for ship structure analyses. Literatures indicate that only few attempts are done to implement optimisation of ship structures using ANSYS® and modeFRONTIER® coupled loop.

A feasibility study to establish reliable surrogate models using response surface methodology are also carried out. Response surfaces are generated for calculating the weight of a stiffened panel and midship region of Ro-Pax vessel. The results from response surface methodology shows that RSM can be a reliable tool to replace the existing optimisation loops, thereby saving calculation time. Response surface methodology, based on polynomial regressions with second order accuracy, give reliable approximate empirical relations between input variables and the output variable.

The research and development on the structural optimisation of midship region of Ro-Pax vessel is still going on under the framework of HOLISHIP and many more targets are to be achieved like the ones explained in Chapter 5.2.

5.2. Recommendations for Future Works

1. The present study considered only one load condition involving load case “a”. Future studies can be performed by considering all load conditions defined in 4.2. Also the accommodation loads are not included here. They can also be part of future studies.
2. The loads considered in the present thesis are based on the rules. The loads calculated from CFD analyses can be implemented in future.
3. Frame spacing can also be considered as a design variable in future studies.
4. The coupling between FEA and optimisation software is achieved. It is interesting to develop a coupled tool including CAD, FEA and optimisation software.
5. Feasibility study to generate surrogate models such as Kriging response instead of response surfaces

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APPENDIX A1 - Convergence Study for Stiffened Panel

A convergence study on the stiffened panel is carried out to decide appropriate mesh size for analysis. The study is done similar to that of plate convergence study explained in 3.2.1. The element selected for the analysis is SHELL181. It is a four node element with six degrees of freedom per node. The element geometry and node locations are shown in Fig. 26.

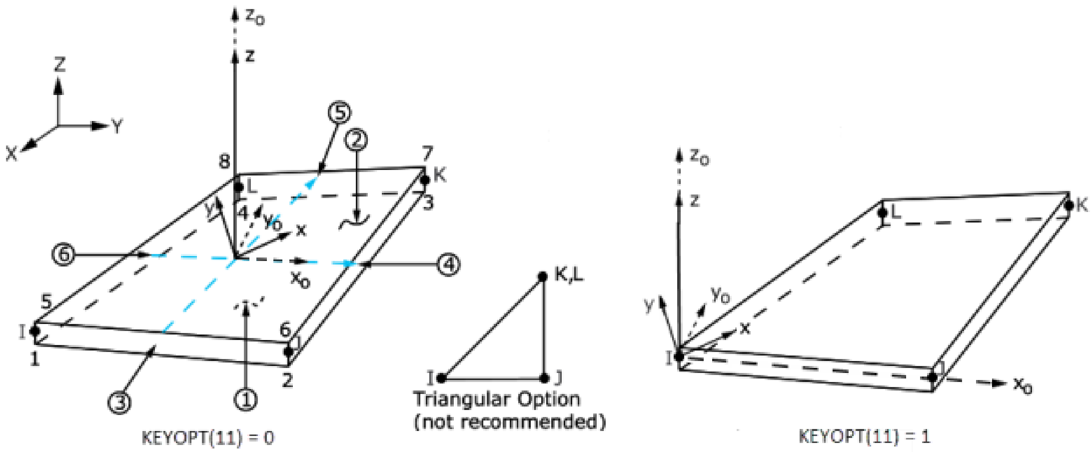

Fig. 26 SHELL181 element geometry (Source: ANSYS® Help)

The convergence study carried out by considering 10 different mesh sizes. The model is assumed to be clamped at all ends and loaded by a uniform pressure of 50 kPa. The deflection against the number of elements plotted in Fig. 27.

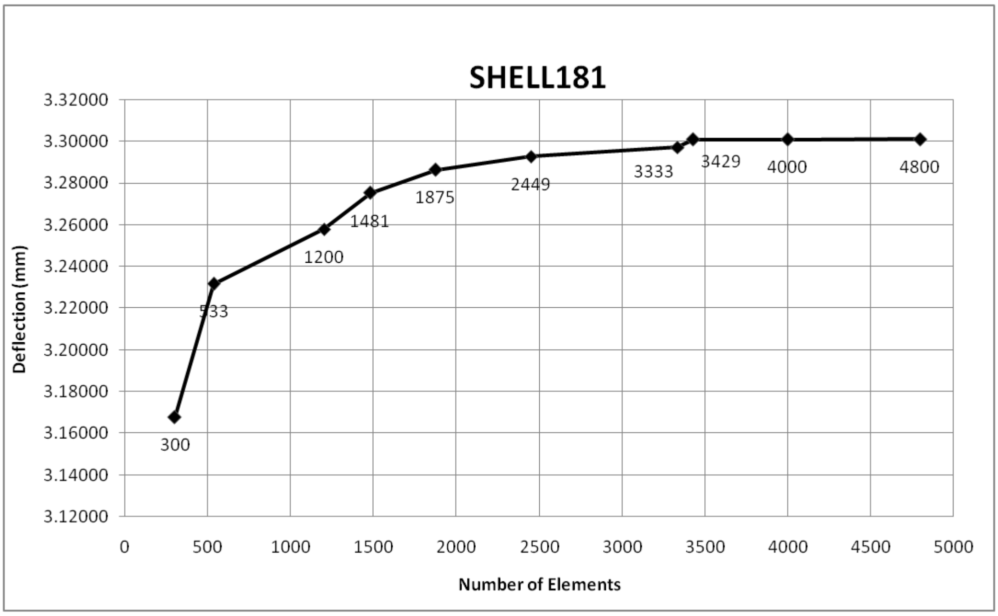

Fig. 27 Convergence representation

It is evident from the figure that the solution is not affecting once the number of elements exceeds 4000. Therefore a mesh size 25x30 mm corresponds to 4000 elements selected for the further studies.

APPENDIX A2 – ANSYS® Parametric Code for Stiffened Panel

The following code is used to create and modify the design parameters of stiffened panel during optimisation process.

```
! Enter the model creation pre-processor
/PREP7
! Reading Parameters from external file. 7 Parameters are defined in file
*DIM, PARA_OPTI, ARRAY, 7
*VREAD, PARA_OPTI(1), PARAMETERS, TXT,
(F8.4)
KEYW, PR_SET,1
KEYW, PR_STRUC, 1
*SET, L1, 2400
*SET, W1, 800
! Thickness of Plate
*SET, T1, PARA_OPTI(1)
! Thickness of Longitudinal Stiffener
*SET, T2, PARA_OPTI(2)
! Thickness of Transverse Stiffener
*SET, T3, PARA_OPTI(3)
! Web height of Longitudinal Stiffener
*SET, WH2, PARA_OPTI(4)
! Web height of Transverse Stiffener
*SET, WH3, PARA_OPTI(5)
!Number of Longitudinal Stiffener
*SET, NSX, PARA_OPTI(6)
! Number of Transverse Stiffener
*SET, NSY, PARA_OPTI(7)
*SET, SIZE_X, 25
*SET, SIZE_Y, 30
*SET, SIZE_Z, 5
! Element Selection
ET, 1, SHELL181
```

```
! Section Definition for thickness of Plate, Stiffeners and Frames
SECTYPE, 1, SHELL, , S-1
SECDATA, T1, 1, 0, 3
SECTYPE, 2, SHELL, , S-2
SECDATA, T2, 1, 0, 3
SECTYPE, 3, SHELL, , S-3
SECDATA, T3, 1, 0, 3
! Defining Material Properties ABS Grade A Steel
MPTEMP, , , , , , , ,
MPTEMP, 1, 0
MPDATA, EX, 1, , 2.0e5
MPDATA, PRXY, 1, , 0.30
MPDATA, DENS, 1, , 7.8e-6
! Stiffened Plate Generation
! Plate Area
K, 1, 0, 0, 0
K, 2, W1/(2*NSX), 0, 0
K, 3, W1/(2*NSX), L1/(2*NSY), 0
K, 4, 0, L1/(2*NSY), 0
A, 1, 2, 3, 4
! Stiffener Area
K, 5, W1/(2*NSX), L1/(2*NSY), WH2
K, 6, W1/(2*NSX), 0, WH2
A, 2, 3, 5, 6
! Frame Area
K, 7, 0, L1/(2*NSY), WH2
K, 8, 0, L1/(2*NSY), WH3
K, 9, W1/(2*NSX), L1/(2*NSY), WH3
A, 3, 4, 7, 5
A, 5, 7, 8, 9
! Defining mesh attributes
LSEL, S, LENGTH, , W1/(2*NSX)
LESIZE, ALL, SIZE_X, , , , , , , 1
```

```
LSEL, S, LENGTH, , L1/(2*NSY)
LESIZE, ALL, SIZE_Y, , , , , , , 1
LSEL, ALL
LSEL, U, LENGTH, , W1/(2*NSX)
LSEL, U, LENGTH, , L1/(2*NSY)
LESIZE, ALL, SIZE_Z, , , , , , , 1
ALLSEL
! Mesh for plate
ASEL, S, LOC, Z, 0, 0
AATT, 1, , 1, , 1
MSHAPE, 0, 2D
MSHKEY, 1
AMESH, ALL
ALLSEL
! Mesh for Stiffener
ASEL, S, LOC, X, W1/(2*NSX)
AATT, 1, , 1, , 2
MSHAPE, 0, 2D
MSHKEY, 1
AMESH, ALL
ALLSEL
! Mesh for Frame
ASEL, S, LOC, Y, L1/(2*NSY)
AATT, 1, , 1, , 3
MSHAPE, 0, 2D
MSHKEY, 1
AMESH, ALL
ALLSEL
! Replicating the areas and mesh in X direction
ASEL, S, LOC, Z, 0, 0
ASEL, A, LOC, Y, L1/(2*NSY)
AGEN, 2, ALL, , , W1/(2*NSX), , , , 0
ALLSEL
! Replicating the areas and mesh in Y direction
```

```
ASEL, S, LOC, Z, 0, 0
ASEL, A, LOC, X, W1/(2*NSX)
AGEN, 2, ALL, , , , L1/(2*NSY), , , 0
ALLSEL
 ! Replicating the areas into required number in X direction
AGEN, NSX, ALL, , , W1/NSX, , , , 0
ALLSEL
! Replicating the areas into required number in X direction
AGEN, NSY, ALL, , , , L1/NSY, , , 0
! Merging the nodes, keypoints, lines, etc. which are overlapping or very close to each other
NUMMRG, ALL, , , , LOW
! Compressing the numbers of nodes, keypoints, lines, etc.
NUMCMP, ALL, , , , LOW
! Application of BCs and Loads. All edges are clamped
! All DOFs restricted on lines at Y=0
LSEL, S, LOC, Y, 0
DL, ALL, , ALL
LSEL, ALL
! All DOFs restricted on lines at X=0
LSEL, S, LOC, X, 0
DL, ALL, , ALL
LSEL, ALL
! All DOFs restricted on all lines at X=W1
LSEL, S, LOC, X, W1
DL, ALL, , ALL,
LSEL, ALL
! All DOFs restricted on all lines at Y=L1
LSEL, S, LOC, Y, L1
DL, ALL, , ALL,
LSEL, ALL
! Pressure applied on all areas which belong to plate
ASEL, S, LOC, Z, 0, 0
SFA, ALL, 1, PRES, 0.05
ALLSEL
```

```
/SOL
SOLVE
! Writing Weight and Von-Mises Stress in Result.txt file
*CFOPEN, Result, txt,
IRLF,-1
*GET, MASS, ELEM, , MTOT, X
*VWRITE, MASS
%G
/POST1
NSORT, S, EQV, 0, 0,
*GET, VONMISES, SORT, , MAX
*VWRITE, VONMISES
%G
*CFCLOS
FINISH
/EOF
```

APPENDIX A3 – Response Surface Generation Codes and Results

The following code is used in R to generate response surface for the stiffened panel optimisation. The results obtained using R is also given.

```
> library(rsm)
> library(readxl)
> DesignVariables <- read_excel("D:/Internship and Thesis/001_Stiffened Plate
Example_Using Shell/DesignVariables.xlsx", +     sheet = "RSM_Data")
> View(DesignVariables)
> Coded_Designs <- coded.data(DesignVariables, Y1 ~ (x1 - 21.85)/16.85, Y2 ~ (x2 -
14.75)/9.75, Y3 ~ (x3 - 12.15)/7.15, Y4 ~ (x4 - 210)/110, Y5 ~ (x5 - 495)/295, Y6 ~ (x6 -
5)/4, Y7 ~ (x7 - 6)/4)
> Designs_RSM <- rsm(Min_Weight ~ SO(Y1,Y2,Y3,Y4,Y5,Y6,Y7), data =
Coded_Designs)
> summary(Designs_RSM)
Call:
rsm(formula = Min_Weight ~ SO(Y1, Y2, Y3, Y4, Y5, Y6, Y7), data = Coded_Designs)
            Estimate Std. Error  t value  Pr(>|t|)
(Intercept) 836.09904    3.71442 225.0954 < 2.2e-16 ***
Y1          233.42986    5.91932  39.4353 < 2.2e-16 ***
Y2          168.72417    3.81434  44.2342 < 2.2e-16 ***
Y3          116.81434    5.01505  23.2928 < 2.2e-16 ***
Y4          155.13404    5.14019  30.1806 < 2.2e-16 ***
Y5          114.26292    4.83035  23.6552 < 2.2e-16 ***
Y6          190.34547    4.20869  45.2268 < 2.2e-16 ***
Y7          136.51625    2.65169  51.4828 < 2.2e-16 ***
Y1:Y2       -19.93504    7.69170  -2.5918 0.0100844 *
Y1:Y3        12.16043    3.44861   3.5262 0.0004975 ***
Y1:Y4       -32.46640    7.07821  -4.5868 6.981e-06 ***
Y1:Y5        16.12237    3.15572   5.1089 6.248e-07 ***
Y1:Y6         5.33439    2.45320   2.1745 0.0305652 *
Y1:Y7         2.50389    1.89946   1.3182 0.1885829
Y2:Y3       -13.15543    2.52229  -5.2157 3.723e-07 ***
Y2:Y4       114.30040    7.60529  15.0291 < 2.2e-16 ***
```

Y2:Y5	-0.04432	4.16620	-0.0106	0.9915203	
Y2:Y6	90.76182	4.03981	22.4669	< 2.2e-16	***
Y2:Y7	-10.15464	3.08892	-3.2874	0.0011492	**
Y3:Y4	-0.34486	6.79096	-0.0508	0.9595379	
Y3:Y5	76.85179	3.54236	21.6951	< 2.2e-16	***
Y3:Y6	-16.71419	2.09977	-7.9600	5.230e-14	***
Y3:Y7	41.21212	1.25347	32.8784	< 2.2e-16	***
Y4:Y5	-54.10183	8.48186	-6.3785	8.048e-10	***
Y4:Y6	70.96034	4.70664	15.0766	< 2.2e-16	***
Y4:Y7	19.71127	4.31185	4.5714	7.473e-06	***
Y5:Y6	-0.15009	2.47535	-0.0606	0.9516957	
Y5:Y7	58.45976	2.46897	23.6778	< 2.2e-16	***
Y6:Y7	3.04397	2.52036	1.2078	0.2282311	
Y1^2	-3.47749	2.48775	-1.3978	0.1633412	
Y2^2	-11.18839	2.17847	-5.1359	5.486e-07	***
Y3^2	-0.67404	1.49591	-0.4506	0.6526589	
Y4^2	21.54268	7.79669	2.7631	0.0061322	**
Y5^2	1.53047	4.21889	0.3628	0.7170719	
Y6^2	0.44069	1.31678	0.3347	0.7381384	
Y7^2	3.36939	1.09266	3.0837	0.0022633	**

Signif. codes: 0 ‘***’ 0.001 ‘**’ 0.01 ‘*’ 0.05 ‘.’ 0.1 ‘ ’ 1

Multiple R-squared: 0.9997, Adjusted R-squared: 0.9996

F-statistic: 2.152e+04 on 35 and 262 DF, p-value: < 2.2e-16

Analysis of Variance Table

Response: Min_Weight

	Df	Sum Sq	Mean Sq	F value	Pr(>F)
FO(Y1, Y2, Y3, Y4, Y5, Y6, Y7)	7	11029333	1575619	1.0243e+05	< 2.2e-16
TWI(Y1, Y2, Y3, Y4, Y5, Y6, Y7)	21	558106	26576	1.7277e+03	< 2.2e-16
PQ(Y1, Y2, Y3, Y4, Y5, Y6, Y7)	7	731	104	6.7862e+00	2.076e-07
Residuals	262	4030	15		
Lack of fit	162	4030	25	1.0926e+25	< 2.2e-16
Pure error	100	0	0		

Stationary point of response surface:

Y1	Y2	Y3	Y4	Y5	Y6	Y7
-58.790919	-12.194646	7.585606	18.609172	-14.150053	-37.643662	22.143968

Stationary point in original units:

x1	x2	x3	x4	x5	x6	x7
-968.77699	-104.14780	66.38708	2257.00888	-3679.26553	-145.57465	94.57587

Eigenanalysis: eigen() decomposition $`values`

[1] 106.116564 56.982123 1.406415 -15.903218 -20.705535 -47.627508 -68.725532

$vectors

	[,1]	[,2]	[,3]	[,4]	[,5]	[,6]	[,7]
Y1	0.16086218	-0.06947262	0.7770305	0.4833072	-0.3139468	0.18233892	0.01116289
Y2	-0.50957692	-0.18533182	0.1368048	-0.2369491	0.0154681	0.40895896	0.68090850
Y3	0.18614609	-0.52663743	-0.1126967	-0.2260154	-0.6454771	-0.40917926	0.20037671
Y4	-0.63821526	-0.18143264	-0.2350603	0.2553066	-0.3732119	0.20735277	-0.50699745
Y5	0.26655851	-0.56845045	0.1444112	-0.4023453	0.2472558	0.44950480	-0.39985544
Y6	-0.44291342	-0.20169747	0.4567293	-0.1521534	0.3494041	-0.61724181	-0.16829401
Y7	0.07252523	-0.53548399	-0.2830158	0.6395096	0.4025566	-0.07771888	0.22546513

APPENDIX A4 – Design Loads Calculations

The hull girder loads are calculated based on BV Rules for Steel ships NR467 Part B, Chapter 5, Section 2. Still water bending moment in kN-m is given by;

$$M_{SWM,H} = 175n_1CL^2B(C_B + 0.7)10^{-3} - M_{WV,H} \quad (19)$$

$$M_{SWM,S} = 175n_1CL^2B(C_B + 0.7)10^{-3} + M_{WV,S} \quad (20)$$

Where,

$M_{SWM,H}$	-	Still water bending moment in hogging conditions, kN-m
$M_{SWM,S}$	-	Still water bending moment in sagging conditions, kN-m
n1	-	Navigation coefficient, equal to 1
C	-	Wave parameter defined by equation (21)

$$C = 10.75 - \left(\frac{300 - L}{100}\right)^{1.5} \quad for\ 90m \leq L < 300m \quad (21)$$

L	-	Scantling length, m
B	-	Moulded breadth, m
CB	-	Block coefficient
$M_{WV,H}$	-	Vertical wave bending moment in hogging conditions, kN-m
$M_{WV,S}$	-	Vertical wave bending moment in sagging conditions, kN-m

Similarly wave bending moment in kN-m is given by;

$$M_{WV,H} = 190F_MnCL^2BC_B10^{-3} \quad (22)$$

$$M_{WV,S} = -110F_MnCL^2B(C_B + 0.7)10^{-3} \quad (23)$$

Where,

F_M	-	Distribution factor, equal to 1

The vertical wave shear force at a hull transverse section between 0.4L ≤ x ≤0.6L in kN is given by equation (24);

$$Q_{WV} = 30F_QnCLB(C_B + 0.7)10^{-2} \quad (24)$$

Where,

Q_{WV}	-	Vertical wave shear force
F_Q	-	Distribution factor, equal to 0.7 for positive shear force and -0.7 for negative shear force for 0.4L ≤ x ≤0.6L
n	-	Navigation coefficient, equal to 1

Calculated values for the hull girder loads for the Ro-Pax vessel is given in the following table. Horizontal wave bending moment and wave torque are not considered since only load case “a” doesn’t include those two loads.

Table 10. Hull girder loads

Loads	Magnitude
$M_{SWM,H}$	1.430×10^6 kN-m
$M_{SWM,S}$	1.093×10^6 kN-m
$M_{WV,H}$	1.512×10^6 kN-m
$M_{WV,S}$	-1.849×10^6 kN-m
Q_{WV}	1.767×10^4 kN

The sea pressure on the hull is calculated using BV Rules for Steel ships NR467 Part B, Chapter 5, Section 5. The still water pressure at any point on the hull below and at waterline (Refer Fig. 17) in kN/m^2 is given by;

$$p_s = \begin{cases} \rho g(T_1 - z) & \forall\, z \le T_1 \\ 0 & \forall\, z > T_1 \end{cases} \tag{25}$$

Where,

ρ - Sea water density, t/m^3

g - Acceleration due to gravity, m/s^2

T_1 - Ship Draught, m

z - Point at which pressure is being calculated, m

The wave pressure in load case “a” (Refer Fig. 18) at any point on the sides and bottom of the ship below waterline in kN/m^2 is given by;

$$p_W = \rho g h e^{\frac{-2\pi(T_1 - z)}{L}} \quad \forall\, z \le T_1 \qquad \text{for Crest} \tag{26}$$

$$p_W = -\rho g h e^{\frac{-2\pi(T_1 - z)}{L}} \quad \forall\, z \le T_1 \qquad \text{for Trough} \tag{27}$$

without being taken less than $\rho g(z - T_1)$

Where,

h - C_{F1}*h_1, C_{F1} is combination factor to be equal to 1 for load case “a”

h1 - Reference values of the ship relative motions in the upright ship condition

Similarly the wave pressure on the sides above waterline is given by;

$$p_W = \rho g(T_1 + h - z) \ \forall\, z > T_1 \qquad \text{for Crest} \tag{28}$$

without being taken less than $0.15\varphi_1\varphi_2 L$

Where,

φ_1 - Coefficient for pressure on exposed decks

φ_2 - Coefficient equal to 1 if L ≥ 120 m

The sea pressure calculated for load case “a” is given in Table 10. The pressure is calculated for all the node locations on the hull surface. Only few values are given in Table. The calculated distribution of sea pressure is plotted in Fig. 28.

Table 11. Sea Pressure for load case “a”

z	a+	a-
m	Below waterline	Below waterline
0.000	133.601	-133.601
0.001	133.595	-133.595
0.006	133.557	-133.557
0.015	133.483	-133.483
0.030	133.360	-133.360
0.052	133.179	-133.179
0.083	132.935	-132.935
0.100	132.797	-132.797
0.174	132.202	-132.202
0.237	131.689	-131.689
0.315	131.066	-131.066
0.412	130.282	-130.282
0.529	129.343	-129.343
0.682	128.112	-128.112
0.683	128.110	-128.110
0.867	126.634	-126.634
1.097	124.787	-124.787
1.098	124.785	-124.785
1.360	122.687	-122.687
1.650	120.377	-120.377
1.900	118.389	-118.389
2.300	115.216	-115.216
2.622	112.669	-112.669
2.950	110.083	-110.083
3.274	107.533	-107.533
3.600	104.980	-104.980
3.910	102.556	-102.556
4.220	100.139	-100.139
4.530	97.729	-97.729

4.840	95.326	-95.326
5.150	92.930	-92.930
5.460	90.541	-90.541
5.770	88.159	-88.159
6.080	85.784	-85.784
6.390	83.417	-83.417

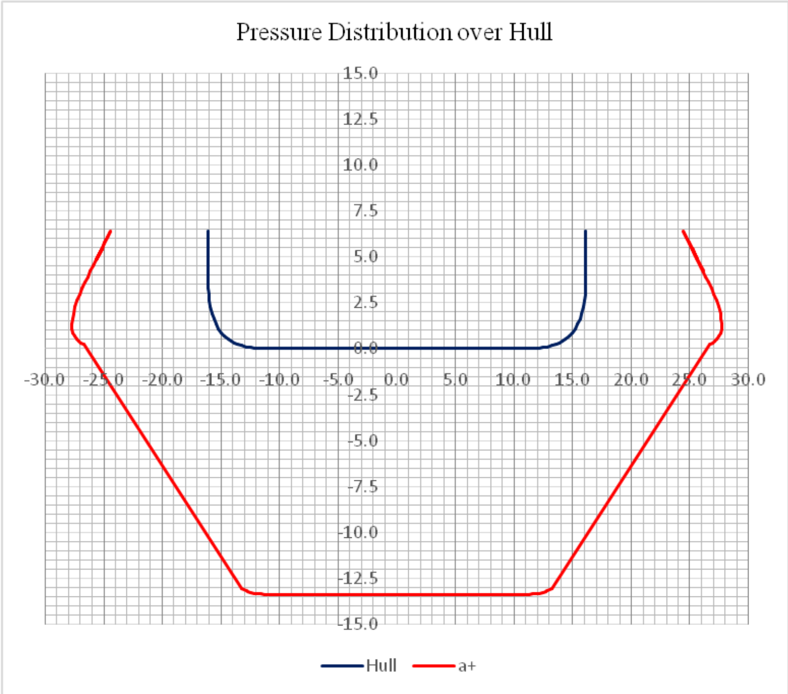

Fig. 28 Pressure distribution over hull for load case “a”

The wheel load data obtained from BV Mars2000 is given in Table 11.

Table 12. Wheel loads on decks

Location	Load (kN/m2)	Number of wheels on the axle
Deck 1	14.00	2
	18.00	4
Deck 3	55.00	2
	25.80	4
Deck 5	14.00	2
	18.00	4
Deck 7	14.00	2
	18.00	4

APPENDIX A5 – Response Surface Generation for Midship Section

The following code is used in R to generate response surface for the midship section of the vessel. The results from the optimisation loop is tabulated and used to obtain response surface polynomial.

```
> library(rsm)
> library(readxl)
>Data_58DV_RSM<-read_excel("D:/Internship and
Thesis/003_Structural_Optimisation_Results/Data_58DV_RSM.xlsx")
> View(Data_58DV_RSM)
> coded_designs <- coded.data(Data_58DV_RSM, Y1 ~ (x1 - 11.5)/5.5, Y2 ~ (x2 - 11.5)/5.5,
Y3 ~ (x3 - 11.5)/5.5, Y4 ~ (x4 - 11.5)/5.5, Y5 ~ (x5 - 11.5)/5.5, Y6 ~ (x6 - 11.5)/5.5, Y7 ~ (x7
- 11.5)/5.5, Y8 ~ (x8 - 11.5)/5.5, Y9 ~ (x9 - 11.5)/5.5, Y10 ~ (x10 - 11.5)/5.5, Y11 ~ (x11 -
11.5)/5.5, Y12 ~ (x12 - 11.5)/5.5, Y13 ~ (x13 - 11.5)/5.5, Y14 ~ (x14 - 11.5)/5.5, Y15 ~ (x15
- 11.5)/5.5, Y16 ~ (x16 - 11.5)/5.5, Y17 ~ (x17 - 11.5)/5.5, Y18 ~ (x18 - 11.5)/5.5, Y19 ~
(x19 - 11.5)/5.5, Y20 ~ (x20 - 13)/7, Y21 ~ (x21 - 11.5)/5.5, Y22 ~ (x22 - 11.5)/5.5, Y23 ~
(x23 - 19)/13, Y24 ~ (x24 - 11.5)/5.5, Y25 ~ (x25 - 11)/6, Y26 ~ (x26 - 11)/6, Y27 ~ (x27 -
12.25)/6.25, Y28 ~ (x28 - 11.5)/5.5, Y29 ~ (x29 - 11.5)/5.5, Y30 ~ (x30 - 11.5)/5.5, Y31 ~
(x31 - 30)/29, Y32 ~ (x32 - 30)/29, Y33 ~ (x33 - 29.5)/28.5, Y34 ~ (x34 - 30)/29, Y35 ~ (x35
- 30)/29, Y36 ~ (x36 - 30)/29, Y37 ~ (x37 - 30)/29, Y38 ~ (x38 - 30)/29, Y39 ~ (x39 - 30)/28,
Y40 ~ (x40 - 30)/29, Y41 ~ (x41 - 550)/150, Y42 ~ (x42 - 550)/150, Y43 ~ (x43 - 550)/150,
Y44 ~ (x44 - 550)/150, Y45 ~ (x45 - 550)/150, Y46 ~ (x46 - 550)/150, Y47 ~ (x47 -
550)/150, Y48 ~ (x48 - 550)/150, Y49 ~ (x49 - 550)/150, Y50 ~ (x50 - 550)/150, Y51 ~ (x51
- 550)/150, Y52 ~ (x52 - 550)/150, Y53 ~ (x53 - 550)/150, Y54 ~ (x54 - 550)/150, Y55 ~
(x55 - 550)/150, Y56 ~ (x56 - 550)/150, Y57 ~ (x57 - 550)/150, Y58 ~ (x58 - 550)/150)
> design_rsm<- rsm(Weight~FO
(Y1,Y2,Y3,Y4,Y5,Y6,Y7,Y8,Y9,Y10,Y11,Y12,Y13,Y14,Y15,Y16,Y17,Y18,Y19,Y20,Y21,
Y22,Y23,Y24,Y25,Y26,Y27,Y28,Y29,Y30,Y31,Y32,Y33,Y34,Y35,Y36,Y37,Y38,Y39,Y40,
Y41,Y42,Y43,Y44,Y45,Y46,Y47,Y48,Y49,Y50,Y51,Y52,Y53,Y54,Y55,Y56,Y57,Y58), data
= coded_designs)
>summary(design_rsm)
Call:
rsm(formula = Weight ~ FO(Y1, Y2, Y3, Y4, Y5, Y6, Y7, Y8, Y9, Y10, Y11, Y12, Y13,
Y14, Y15, Y16, Y17, Y18, Y19, Y20, Y21, Y22, Y23, Y24, Y25, Y26, Y27, Y28, Y29, Y30,
```

Y31, Y32, Y33,Y34, Y35, Y36, Y37, Y38, Y39, Y40, Y41, Y42, Y43, Y44, Y45, Y46, Y47, Y48, Y49, Y50, Y51, Y52, Y53, Y54, Y55, Y56, Y57, Y58), data = Normalised_Variables)

Residuals:

Min	1Q	Median	3Q	Max
-14.6784	-1.7608	-0.1324	1.5864	21.6973

Coefficients: (17 not defined because of singularities)

	Estimate	Std.Error	t value	Pr(>|t|)	
(Intercept)	365.924	0.4764	768.166	< 2e-16	***
Y1	4.3065	0.9366	4.598	6.46E-06	***
Y2	1.7331	1.0476	1.654	0.099191	.
Y3	8.4978	1.2098	7.024	1.62E-11	***
Y4	1.6305	0.9874	1.651	0.099793	.
Y5	18.1669	1.2548	14.477	< 2e-16	***
Y6	2.3236	1.0006	2.322	0.020937	*
Y7	-0.2772	0.9696	-0.286	0.775161	
Y8	21.1239	1.0018	21.086	< 2e-16	***
Y9	0.622	0.8806	0.706	0.480535	
Y10	3.3606	1.0449	3.216	0.00145	**
Y11	7.6034	0.9676	7.858	8.35E-14	***
Y12	-2.3262	1.0371	-2.243	0.025682	*
Y13	9.8896	0.9863	10.026	< 2e-16	***
Y14	3.5447	0.9011	3.934	0.000106	***
Y15	-2.2277	1.0706	-2.081	0.038361	*
Y16	1.0269	0.8972	1.145	0.253385	
Y17	-0.5374	0.7504	-0.716	0.474481	
Y18	3.8099	0.8114	4.696	4.16E-06	***
Y19	8.3646	0.9797	8.538	8.74E-16	***
Y20	4.6476	1.0281	4.521	9.09E-06	***
Y21	-0.105	0.9376	-0.112	0.910954	
Y22	-0.8345	0.9903	-0.843	0.400105	
Y23	4.1471	1.121	3.7	0.00026	***
Y24	-4.007	0.933	-4.295	2.41E-05	***
Y25	24.7047	1.0434	23.677	< 2e-16	***
Y26	8.0861	1.0798	7.488	9.06E-13	***
Y27	-1.8332	0.9375	-1.955	0.051521	.
Y28	-2.3731	0.8595	-2.761	0.006142	**
Y29	-1.6871	0.947	-1.781	0.075923	.
Y30	0.1489	1.0683	0.139	0.889212	
Y31	9.2455	0.8631	10.711	< 2e-16	***
Y32	1.9007	0.9188	2.069	0.039497	*
Y33	2.4002	0.8924	2.69	0.007583	**
Y34	16.3632	1.0005	16.356	< 2e-16	***

Y35	5.0619	1.078	4.696	4.16E-06	***
Y36	0.3546	1.2292	0.289	0.773168	
Y37	7.6309	1.1175	6.828	5.30E-11	***
Y38	17.3196	1.2029	14.398	< 2e-16	***
Y39	10.2807	1.0976	9.367	< 2e-16	***
Y40	31.5779	1.2337	25.596	< 2e-16	***
Y41	-29.7439	0.9346	-31.826	< 2e-16	***
Y42	NA	NA	NA	NA	
Y43	NA	NA	NA	NA	
Y44	NA	NA	NA	NA	
Y45	NA	NA	NA	NA	
Y46	NA	NA	NA	NA	
Y47	NA	NA	NA	NA	
Y48	NA	NA	NA	NA	
Y49	NA	NA	NA	NA	
Y50	NA	NA	NA	NA	
Y51	NA	NA	NA	NA	
Y52	NA	NA	NA	NA	
Y53	NA	NA	NA	NA	
Y54	NA	NA	NA	NA	
Y55	NA	NA	NA	NA	
Y56	NA	NA	NA	NA	
Y57	NA	NA	NA	NA	
Y58	NA	NA	NA	NA	

The design variables from 42 to 58 representing the stiffener spacing design parameters are not influenced the response surface polynomial.