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Structural Assessment of Small Craft Hulls: Integration of ISO 12215-5 Analytical Methods and Finite Element Analysis

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March 23, 2026

Abstract

Recreational vessels are boats designed for leisure activities, like fishing, competition, or general navigation. Technological advancements yield new and increasingly sophisticated models. Nowadays, new vessels must comply to various standards to be commercialized. In Europe, ISO 12215 is the standard related to the structural validity of vessels up to 24 meters.

This work performs a study and follows the steps towards the development of a new craft by incorporating numerical modeling capabilities to the traditional methodology. Riamar has proposed this project, which main goal is the structural design of the hull of said new recreational vessel, in accordance with the ISO 12215 standard requirements.

A crucial phase of this work was the definition of the mechanical properties of the materials used by Riamar company. This stage included an experimental step in which representative composite specimens of the two possibilities of material for the new vessel were manufactured, measured, and subjected to tensile and three-point bending tests. Furthermore, the mechanical properties for the considered laminates were also calculated analytically with the formulation provided by the ISO 12215-5 standard. The comparison between the results obtained from these two methods lead to the validation of the properties calculated analytically with the results obtained experimentally.

Two approaches to perform the structural design validation were pursued: an analytical one and a numerical one. The first one performs structural calculation, based on the mechanical properties previously defined and analytical equations obtained from ISO 12215-5. The second one bases itself in the use of FEM packages from software, like SolidWorks and Abaqus to analyze the stress in the vessels' panels when certain boundary conditions are applied. In both cases, applied pressures and limit stresses are defined by ISO 12215-5. Furthermore, optimization analyses were conducted to determine the most suitable combination of element type and element size for both software packages, considering a representative geometric model. Additional studies were also performed to evaluate the influence of variations in the elastic modulus and laminate thickness on the numerical results.

With these methodologies, two vessels were tested. The first one, *Pescador 500*, is an already validated model produced by Riamar. The results previously obtained by the certifying entity are compared to the ones obtained by both methods. The second vessel, the *Evolution 5.0*, is currently under development. The materials studied in this work are intended for its manufacture and therefore require verification of compliance with ISO 12215-5.

Results show that the finite element modeling methodologies are an important complement to traditional methods of vessel design, thus permitting a better optimization and more accurate validation of the product and all the available resources.

Keywords: Composite materials; Finite element method; Boat designing; Structural optimization; ISO 12215-5.

Resumo

Embarcações de recreio são barcos concebidos para atividades de lazer, como pesca, competição ou navegação em geral. Os avanços tecnológicos têm permitido o desenvolvimento de modelos novos e cada vez mais sofisticados. Atualmente, as novas embarcações devem cumprir diversas normas para poderem ser comercializadas. Na Europa, a ISO 12215 é a norma relativa à validação estrutural de embarcações até 24 metros de comprimento.

O presente trabalho realiza um estudo e descreve as etapas conducentes ao desenvolvimento de uma nova embarcação, incorporando capacidades de modelação numérica à metodologia tradicional. A empresa Riamar propôs este projeto, cujo principal objetivo é o dimensionamento estrutural do casco da nova embarcação de recreio, em conformidade com a norma ISO 12215-5.

Uma fase crucial deste trabalho consistiu na definição das propriedades mecânicas dos materiais utilizados pela empresa Riamar. Esta etapa incluiu uma componente experimental, na qual foram fabricados provetes compósitos representativos das duas soluções de material consideradas para a nova embarcação, posteriormente medidos e submetidos a ensaios de tração e de flexão em três pontos. Adicionalmente, as propriedades mecânicas dos laminados considerados foram também determinadas analiticamente com base na formulação fornecida pela norma ISO 12215-5. A comparação entre os resultados obtidos por estes dois métodos conduziu à validação das propriedades calculadas analiticamente com base nos resultados experimentais.

Foram seguidas duas abordagens para a validação do dimensionamento estrutural: uma analítica e uma numérica. A abordagem analítica baseia-se em cálculos estruturais efetuados a partir das propriedades mecânicas previamente definidas e das equações analíticas presentes na ISO 12215-5. A abordagem numérica recorre à utilização de programas com capacidades de análise de elementos finitos, como o SolidWorks e o Abaqus, para analisar o estado de tensão nos painéis da embarcação sob determinadas condições de fronteira. Em ambos os casos, as pressões aplicadas e as tensões limite são definidas pela ISO 12215-5. Adicionalmente, foram realizadas análises de otimização para determinar a combinação mais adequada de tipo e dimensão de elemento em ambos os programas, considerando um modelo geométrico simplificado e representativo. Foram ainda efetuados estudos complementares para avaliar a influência de variações no módulo de elasticidade e na espessura do laminado nos resultados numéricos.

Com base nestas metodologias, foram analisadas duas embarcações. A primeira, Pescador 500, é um modelo já validado e produzido pela Riamar. Os resultados previamente obtidos pela entidade certificadora foram comparados com os resultados alcançados através das duas abordagens adotadas neste trabalho. A segunda embarcação, a Evolution 5.0, encontra-se atualmente em desenvolvimento. Os materiais estudados neste trabalho destinam-se ao seu fabrico e, por esse motivo, requerem a verificação da conformidade com a ISO 12215-5.

Os resultados demonstram que as metodologias de modelação por elementos finitos constituem um complemento relevante aos métodos tradicionais de projeto de embarcações, permitindo uma melhor otimização e uma validação mais rigorosa do produto e dos recursos disponíveis.

Palavras-chave: Materiais compósitos; Método dos elementos finitos; Projeto de embarcações; Otimização estrutural; ISO 12215-5.

Acknowledgements

I would like to express my sincere gratitude to Professor Doctor Jorge Lino for accepting to supervise this thesis and for his valuable guidance throughout this work. I am also deeply grateful to Professor Doctor Abel dos Santos and Professor Doctor Augusto Barata da Rocha for their knowledge, patience, and continuous support. Without their guidance, the completion of this work would not have been possible.

I would also like to thank Francisco Cálão, manager and co-founder of Nautav/Riamar, and Engineer Paulo Matos for the opportunity to contribute to the design of the new vessel and for their hospitality during my visits to the company in Aveiro.

My sincere thanks also go to FAST (FEUP Academic Solar Team) for allowing me to be part of their “family”. Being part of this team allowed me to share many memorable moments with remarkable people, learn valuable concepts, and open many doors for my future.

I would also like to express my appreciation to Doctor Rui Amaral, Doctor Daniel Cruz, and Engineer Ricardo Rocha, from INEGI (Institute of Science and Innovation in Mechanical and Industrial Engineering), for their support and availability in assisting with the experimental testing procedures.

A heartfelt thank you to my family for their unwavering support throughout my life. To my parents, for always supporting me and giving me the opportunity to pursue my dreams, and to my brother, who was always there when I needed him the most.

Finally, I would like to thank my friends for their encouragement and for always being there during this journey. Their support, companionship, and the many moments we shared provided the motivation and energy needed to complete this important chapter of my life.

Contents

Abstract	i
Resumo	ii
Acknowledgements	iii
List of Figures	ix
List of Tables	xii
List of Acronyms	xiii
1 Introduction	1
1.1 Background and Context	1
1.1.1 Riamar Company	1
1.1.2 FEUP Academic Solar Team	3
1.2 Motivation	4
1.3 Objectives	4
1.4 Thesis Outline	5
2 Bibliographic Review	7
2.1 Small Craft Industry	7
2.1.1 Economic Overview	7
2.1.2 Historical Overview	9
2.2 Regulatory and Standard Frameworks	10
2.2.1 ISO 12215 Series Overview	11
2.3 Components and Terminology of Small Craft	12
2.3.1 Nautical Terminology	12
2.4 Product Development applied to the Small Craft Industry	15
2.4.1 New Product Development	15
2.4.2 Rule-of-Thumb & Design Errors	16
2.4.3 New Product Development in the Small Craft Industry	17
2.4.4 Simulation-Driven Design	18
2.4.5 V-model Approach	19
2.5 Riamar Vessels Development Progress	21
2.6 Chapter Summary	24

3	Materials, Mechanical Characterization and Methods	25
3.1	Materials and Mechanical Characterization	25
3.1.1	Composite Materials Description	25
3.1.2	Experimental Tests	28
3.1.3	Properties Definition according to ISO 12215-5	46
3.1.4	Comparison between Material Properties	52
3.2	Application Procedure of ISO 12215-5 Standard	54
3.2.1	Geometric and Design Parameters	54
3.2.2	Pressure Adjusting Factors and Design Pressures	57
3.2.3	Design Stresses	59
3.2.4	Methods for Structural Analysis and Scantlings Determination	61
3.3	Numerical Modeling	65
3.3.1	Numerical Model Creation	66
3.3.2	Mesh Parameters Optimization	70
3.3.3	Elastic Modulus and Thickness Sensitivity Analysis	76
3.4	Chapter Summary	81
4	Applications and Results	83
4.1	Case Study 1 - Pescador 500	83
4.1.1	ISO 12215-5 Analytical Scantling	85
4.1.2	Finite Element Analysis	88
4.1.3	Results and Comparison	97
4.2	Case Study 2 - Evolution 5.0	101
4.2.1	ISO 12215-5 Analytical Scantling	104
4.2.2	Finite Element Model Analysis	108
4.2.3	Results and Comparison	116
4.3	Chapter Summary	118
5	Conclusions and Future Work	121
5.1	Conclusions	121
5.2	Future Work	123
	References	124
A	Material Characterization: additional data	129
A.1	Measurements of Specimens	129
A.2	Three-point Bending Test — Numerical Approach	132
B	Pescador 500 Scantling Data	139
C	Evolution 5.0 Scantling Data	144

List of Figures

1.1	Side-by-side of an existing and an in-development boats designed by Riamar company.	2
1.2	'Atlântico' vessel prototype developed by FAST.	3
2.1	European recreational market growth [5].	8
2.2	Boat environment terms and components [24].	13
2.3	Displacement hull, semi-displacement hull, and planning hull representations [26].	14
2.4	Stage-Gate model steps [27].	15
2.5	Hybrid Agile Stage-Gate model steps [28].	16
2.6	Design Spiral description used in ship design [35].	18
2.7	V-model applied to ship designing [35].	20
2.8	Integrated design model obtained with the application of the V-model [35]. . . .	21
2.9	Final CAD model of the Evolution 7.0 vessel [46].	22
2.10	Wake of Riamar Evolution 7.0 vessel [46].	22
2.11	3D printing of the Riamar Evolution 7.0 vessel [46].	23
2.12	3D printing of the molds developed for the the production of the Riamar Evolution 7.0 vessel.	23
2.13	Deck design of Riamar Evolution 7.0 vessel, developed by Joana dos Santos [48].	23
3.1	Dry fiber of chopped strand mat [48].	26
3.2	Dry fiber of woven roving [49].	26
3.3	Dry fiber of rovimat [48].	26
3.4	CAD representation of the mold of the Evolution 5.0 hull; the side walls of the hull, region (A), are laminated with "Mat" composite material, whereas the hull bottom, region (B), is laminated with "Rovimat" material.	27
3.5	Manufactured laminate plates used in experimental tests; the nominal thicknesses measured for each laminate were 3.9 mm for "Mat" material and 5.1 mm for "Rovimat" material.	29
3.6	Samples manufactured for tensile and three-point bending tests; the top ones, of dimensions 250x25 mm were used for the tensile tests and the bottom ones, of dimensions 150x45 mm, for three-point bending tests.	30
3.7	Thickness dispersion graph for the tensile samples.	31
3.8	Hydraulic grips and extensometer used in the tensile tests.	32
3.9	Global view of the setup for tensile tests; to be highlighted the hydraulic grips and their control equipment (on the left of the tensile machine), besides the computer numerical control of the tensile machine (on the right side).	33
3.10	Tested tensile specimens and their failure identification designation.	34
3.11	Stress-strain curves considering the two examined materials and fiber directions. .	35

3.12	Bar graph with the ultimate tensile strength results.	36
3.13	Bar graph with the ultimate tensile strain results.	37
3.14	Bar graph with the Young's modulus results.	37
3.15	Thickness dispersion graph for the three-point bending samples.	39
3.16	Global view of the setup used for the three-point bending tests.	40
3.17	Three-point bending samples after testing.	41
3.18	Composite laminate specimens after three-point bending tests performed until failure.	42
3.19	Stress-strain curves of three-point bending tests until failure.	43
3.20	Stress-strain curves of three-point bending tests until 5 mm nose displacement for "Rovimat" and <i>MB0#1</i> , <i>MB0#7</i> and <i>MB0#8</i> specimens and until 8 mm nose displacement for specimens <i>MB0#5</i> and <i>MB0#6</i>	43
3.21	Bar graph with the ultimate flexural strength results.	44
3.22	Bar graph with the ultimate flexural strain results.	45
3.23	Bar graph with the flexural modulus of elasticity results.	45
3.24	Panel average dimensions l and b ; adapted from ISO 12215-5 [20].	55
3.25	Camber c_1 measurement in panel with a shorter dimension of b_1 ; s_1 , s_2 and s_3 represent natural stiffeners that limit the panels; adapted from ISO 12215-5 [20].	56
3.26	Chine beam B_c and deadrise angle β measurements; adapted from ISO 12215-5 [20].	56
3.27	Schematics of the considered 1 mm wide strip; adapted from ISO 12215-5 [20].	63
3.28	Optimist boat used as reference model for the optimization of numerical variables for the finite element analysis.	67
3.29	Geometry developed based on the Pescador 500 vessel and Optimist boat.	67
3.30	Shell Abaqus geometric model.	68
3.31	Location and identification names of nodes considered for the results comparison.	68
3.32	Location of supports imposed on the reference model.	69
3.33	Mesh discretization of the model, in SolidWorks and Abaqus software, for 25 mm of element size; the SolidWorks-generated mesh contains 129 307 nodes and 64 400 elements, and the Abaqus-generated mesh contains 81 529 nodes and 27 034 elements.	71
3.34	Von Mises distribution results using an average element size of 50 mm.	72
3.35	Displacement results on the Y-direction using an element size of 50 mm.	73
3.36	Von Mises stress results comparison for all models.	74
3.37	Displacement results comparison in the Y-direction for all models.	74
3.38	Computational time cost in logarithmic scale for all simulations performed.	76
3.39	Graphs showing the influence of E on the displacement and its sensitivity to the E variation.	78
3.40	Graphs showing the influence of t on the von Mises stress and its sensitivity to the t variation.	79
3.41	Graphs showing the influence of t on the displacement and its sensitivity to the t variation.	80
4.1	Vessel to be analyzed in this section.	83
4.2	CAD model imported from Rhino to SolidWorks of the Pescador 500 vessel.	84
4.3	CAD model developed of the Pescador 500 vessel.	85
4.4	Side view of Pescador 500 CAD model.	85
4.5	Rear (left) and front (right) views of Pescador 500 model.	86
4.6	Evaluated panels identification and location of the Pescador 500 vessel.	86

4.7	Model considered for the FEA of Pescador 500 vessel.	89
4.8	Top-hats considered for the FEA of Pescador 500 vessel.	89
4.9	Partitioning and panels geometry considered for the FEA of Pescador 500 vessel.	90
4.10	Bonding locations between the hull and the top-hat structure of the model representing Pescador 500 vessel.	91
4.11	Supports location and area where pressure was applied in the model representing Pescador 500 vessel.	91
4.12	Mesh discretization of SolidWorks model of Pescador 500	92
4.13	Von Mises stress and displacement in Z-direction contour plots for Pescador 500 vessel's analysis.	92
4.14	Components of Abaqus model for Pescador 500 vessel.	93
4.15	Tie location in the Abaqus model of Pescador 500 vessel.	94
4.16	Boundary conditions applied to Abaqus model for Pescador 500 vessel.	94
4.17	Mesh discretization for the Abaqus model of Pescador 500 vessel.	95
4.18	Stress distribution in the 1-direction (left) and in the 2-direction (right), obtained from the Abaqus analysis of the Pescador 500 vessel.	95
4.19	Displacement in the normal direction of the hull surface, obtained from the Abaqus analysis of the Pescador 500 vessel.	96
4.20	Stress contour plot, in the panel A1, in the 1-direction (left) and 2-direction (right), from the Pescador 500 vessel.	96
4.21	Displacement in the normal direction of the panel A1, from the Pescador 500 vessel.	96
4.22	Panel D7 of Pescador 500 vessel and its structural support.	99
4.23	Stress contour plot in panel D7 of the Pescador 500 vessel along the 2-direction, calculated with the Abaqus software, with the location of the critical node shown with a red dot.	99
4.24	Panel E9 of Pescador 500 vessel and its structural support.	100
4.25	Stress contour plot in panel E9 of the Pescador 500 vessel along the 2-direction, calculated with the Abaqus software, with the location of the critical node shown with a red dot.	100
4.26	CAD model of the Evolution 5.0 vessel.	101
4.27	Mold of the hull of the Evolution 5.0 vessel.	102
4.28	Mold of the deck of the Evolution 5.0 vessel.	102
4.29	Mold of the internal structure of the Evolution 5.0 vessel.	103
4.30	Internal structure designs of Evolution 5.0 vessel.	103
4.31	Hull designs of the Evolution 5.0 vessel.	104
4.32	Canoe body draft T_C measurement of the Evolution 5.0 vessel.	105
4.33	Length of waterline L_{WL} and beam of waterline B_{WL} of the Evolution 5.0 vessel.	105
4.34	Deadrise angle β and chine beam B_C of the Evolution 5.0 vessel.	106
4.35	Evaluated panels identification and location of the Evolution 5.0 vessel.	107
4.36	Model considered for the FEA of the Evolution 5.0 vessel.	109
4.37	Partitioning and panels geometry considered for the FEA of the Evolution 5.0 vessel.	109
4.38	Bonding locations between the hull and the developed internal structure of the model of the Evolution 5.0 vessel.	110
4.39	Supports location and area where pressure is applied in the model of the Evolution 5.0 vessel.	111
4.40	Mesh discretization of the model Evolution 5.0 vessel, in the SolidWorks software.	111
4.41	Von Mises stress contour plot for Evolution 5.0 vessel, obtained with SolidWorks FEM package.	112

4.42	Components of Abaqus model for Evolution 5.0 vessel.	112
4.43	"Tie" location in the Abaqus model of the model Evolution 5.0 vessel.	113
4.44	Boundary conditions applied to Abaqus model for Evolution 5.0 vessel.	114
4.45	Mesh discretization for the Abaqus model of the Evolution 5.0 vessel.	114
4.46	Stress distribution in the 1-direction (left) and in the 2-direction (right), obtained from the Abaqus analysis of the Evolution 5.0 vessel.	115
4.47	Displacement in the Y-direction, obtained from the Abaqus analysis of the Evolution 5.0 vessel.	115
4.48	Stress contour plot, in the panel A1, in the 1-direction (left) and 2-direction (right), from the Evolution 5.0 vessel.	116
4.49	Displacement in the Y-direction of the panel A1 from the Evolution 5.0 vessel.	116
4.50	Stress contour plot in the 1-direction of panel F8 of Evolution 5.0 vessel, with the location of the critical node shown with a red dot.	118
A.1	Developed model for the three-point bending numerical analysis.	133
A.2	Meshed model for the three-point bending numerical analysis.	134
A.3	Force-displacement curves produced with numerical data from Abaqus simulations for "Mat" material.	135
A.4	Force-displacement curves produced with numerical data from Abaqus simulations for "Rovimat" material.	135
A.5	Overlapped force-displacement curves based on numerical data (dashed lines) and experimental data (solid lines) for "Mat" laminates.	137
A.6	Overlapped force-displacement curves based on numerical data (dashed lines) and experimental data (solid lines) for "Rovimat" laminates.	137
B.1	Analytical tool spreadsheet performing the scantling determination of the panel D7 from the Pescador 500 vessel.	141
C.1	Analytical tool spreadsheet performing the scantling determination of the panel D6 from the Evolution 5.0 vessel.	146

List of Tables

2.1	Terminology used in marine environment [23].	12
2.2	Main structural components of small motorboats and their functions [24].	13
2.3	Terms relevant for the application of ISO 12215 standard.	14
2.4	Typical cost impacts associated with boat design deficiencies [33].	17
3.1	Layers of dry fiber of composite material used in Pescador 500 vessel's hull.	27
3.2	Layers of dry fiber of composite material options to be used in Evolution 5.0 vessel's hull.	28
3.3	Averages of measured width and thickness of tensile test specimens.	31
3.4	Failure mode coding system for tensile testing, which is composed of three characters: the first for the failure type, the second for the failure area, and the third for the failure location.	33
3.5	Results obtained for the ultimate tensile strength σ_{ut}	36
3.6	Results obtained for the ultimate tensile strain ϵ_{ut}	37
3.7	Results obtained for the of Young's modulus E	37
3.8	Average width and thickness of three-point bending specimens.	39
3.9	Failure mode coding system for three-point bending testing, which is composed of three characters: the first for the failure mode, the second for the failure area, and the third for the failure location.	41
3.10	Results summary of ultimate flexural strength σ_{uf}	44
3.11	Results summary of ultimate flexural strain ϵ_{uf}	45
3.12	Results summary of flexural modulus of elasticity E_f	45
3.13	Properties of E-Glass dry fiber and Polyester matrix; adapted from Table C.3. of Annex C (ISO 12215-5).	47
3.14	Equations to calculate the elastic properties of CSM and WR layers.	48
3.15	Tensile (ϵ_{ut}) and flexural (ϵ_{uf}) ultimate strains for CSM and WR layers.	48
3.16	Fiber contents, thickness per dry fiber mass per square meter, and composite density obtained with the laminate plates thicknesses manufactured by Riamar company.	49
3.17	Elastic properties of CSM and WR layers.	50
3.18	Tensile (σ_{ut}) and flexural (σ_{uf}) strength properties of CSM and WR layers.	50
3.19	Relation between fiber weight, thickness per fiber weight, and laminate thickness for CSM 0.45 kg/m^2 , WR 0.5 kg/m^2 and RVM 0.95 kg/m^2	50
3.20	RVM layer elastic properties and fiber content in volume calculated based on the thickness of CSM and WR layers.	51
3.21	Ultimate strain values for RVM layer based on CSM and WR layers.	51
3.22	Calculated strength properties of CSM, WR, and RVM layers.	51
3.23	Individual layers and laminate properties of the "Rovimat" material, calculated with the analytical method described by ISO 12215-5.	52

3.24	Individual layers and laminate properties of the "Mat" material, calculated with the analytical method described by ISO 12215-5.	52
3.25	Comparison of tensile properties obtained from different characterization methods for "Mat" material.	53
3.26	Comparison of flexural properties obtained from different characterization methods for "Mat" material.	53
3.27	Comparison of tensile properties obtained from different characterization methods and fiber orientations for "Rovimat" material.	53
3.28	Comparison of flexural properties obtained from different characterization methods for "Rovimat" material.	53
3.29	Design categories and associated wave conditions [46].	57
3.30	Builder quality classification and corresponding values of the factor k_{BB} ; adapted from Table 15 of ISO 12215-5 [20].	60
3.31	Determination of the assessment method factor k_{AM} , representing Table 16 of ISO 12215-5 [20].	60
3.32	Reference thickness and material properties used for the initial numerical model.	69
3.33	Boundary conditions imposed on the model.	70
3.34	Summary of the analyses conducted to evaluate the sensitivity of the mesh parameters, using the reference model.	71
3.35	Computational cost in seconds for each combination of element type and mesh size.	75
3.36	Elastic modulus and thickness values group to be analyzed.	77
3.37	Sensitivity analysis using SolidWorks with respect to Young's modulus variation.	77
3.38	Sensitivity analysis using Abaqus with respect to Young's modulus variation.	77
3.39	Sensitivity analysis using SolidWorks with respect to the thickness variation.	79
3.40	Sensitivity analysis using Abaqus with respect to the thickness variation.	79
4.1	Main characteristics of the Pescador 500 vessel.	86
4.2	Design pressure and panel-dependent coefficients for the Pescador 500 vessel.	87
4.3	Individual layers and laminate properties for the material considered for the hull of Pescador 500 vessel.	88
4.4	Design stresses calculated for each assessment method of the Pescador 500's hull material.	88
4.5	Comparison of calculated compliance factors for the Pescador 500 vessel.	97
4.6	Main characteristics of the Evolution 5.0 vessel.	106
4.7	Design pressure and panel-dependent coefficients for the Evolution 5.0 vessel.	107
4.8	Design stresses calculated for each assessment method of the Evolution 5.0's hull material, the "Rovimat" material.	108
4.9	Lamina material properties defined in Abaqus for the RVM layer of "Rovimat" in the FEA of the Evolution 5.0 vessel, adapted from Table 3.23.	113
4.10	Comparison of the obtained compliance factors for the Evolution 5.0 vessel.	117
A.1	Width and thickness measurements of "Mat" material specimens used for the tensile tests.	129
A.2	Width and thickness measurements of "Rovimat" material specimens used for the tensile tests.	130
A.3	Measured width and thickness values of "Mat" material specimens for three-point bending tests.	130
A.4	Measured width and thickness values of "Rovimat" material specimens for three-point bending tests.	131

A.5	Comparison of material properties obtained from the analytical and experimental methods for "Mat" material, regarding the tensile properties.	131
A.6	Comparison of material properties obtained from the analytical and experimental methods for "Mat" material, regarding the flexural properties.	131
A.7	Comparison of material properties obtained from the analytical and experimental methods for "Rovimat" material, regarding the tensile properties and fiber orientations.	132
A.8	Comparison of material properties obtained from the analytical and experimental methods for "Rovimat" material, regarding the flexural properties.	132
A.9	Geometrical, and loading parameters used in the three-point bending simulation .	133
A.10	Simulation groups and corresponding material and contact parameters.	134
A.11	Slopes obtained for the force-displacement curves obtained through FEM simulations for the "Mat" material.	136
A.12	Slopes obtained for the force-displacement curves obtained through FEM simulations for the "Rovimat" material.	136
A.13	Force-displacement slopes and corresponding Young's modulus values for MB0 specimens.	138
A.14	Force-displacement slopes and corresponding Young's modulus values for RB0 specimens.	138
B.1	Geometrical properties and curvature parameters of panels from the Pescador 500 vessel.	139
B.2	Design stresses calculated for each layer of the Pescador 500's hull material, regarding the Enhanced method and FEM.	140
B.3	Comparison between the results obtained analytically using the Enhanced and Simplified methods for the Pescador 500 vessel.	142
B.4	Stress results in 1- and 2-directions and compliance factors, obtained with Abaqus software, for Pescador 500.	142
B.5	Von Mises stress results and compliance factors, obtained with SolidWorks software for Pescador 500 vessel.	143
C.1	Geometrical properties and curvature parameters of panels from the Evolution 5.0 vessel.	144
C.2	Design stresses calculated for the application of the "Rovimat" material in the Evolution 5.0 vessel.	145
C.3	Comparison between the results obtained analytically using the Enhanced and Simplified methods for the Evolution 5.0 vessel.	147
C.4	Stress results in 1- and 2-directions and compliance factors, obtained with Abaqus software.	147
C.5	Von Mises stress results and compliance factors, obtained with SolidWorks software for Evolution 5.0 vessel.	147

List of Acronyms

CAD	Computer Aided Design
CAGR	Compound Annual Growth Rate
CFD	Computational Fluid Dynamics
CLT	Classical Laminate Theory
CNC	Computer Numerical Control
CSM	Chopped Strand Mat
CV	Coefficient of Variation
EU	European Union
FAST	FEUP Academic Solar Team
FEA	Finite Element Analysis
FEM	Finite Element Method
ISO	International Standard Organisation
NPD	New Product Development
RVM	Rovimat
SME	Small and Medium Enterprises
WR	Woven Roving

Chapter 1

Introduction

This dissertation is part of a proposal by the company Nautav/Riamar and it was developed with the aim of completing the Master's degree in Mechanical Engineering, in the field of General Mechanical Engineering, at the Faculty of Engineering of the University of Porto.

1.1 Background and Context

In the maritime industry, the small craft sector is characterized by the manufacture of vessels up to 24 meters in length. These boats may be recreational, used for watersports or leisure activities, or intended for commercial purposes, such as small-scale transport or fishing operations.

Nowadays, there is strong competition in the maritime market, especially in the recreational vessel manufacturing industry, where new models must differentiate themselves through their attractive designs, innovative technologies, and efficiency. Therefore, companies need to design new models to stay afloat in order to maintain themselves as competitive and relevant in the industry. In this case, new vessels, in addition to complying with the standards imposed by the International Standard Organisation (**ISO**), in order to be commercialized without legal concerns, must be as efficient as possible in terms of material usage, which ensures that manufacturing costs and time are minimized.

1.1.1 Riamar Company

Riamar is a company that focuses on the design, manufacture, and sale of recreational boats. It is a Portuguese company dedicated to the design and construction of small recreational boats. Founded in the 1960s in Aveiro by Manuel Alves Barbosa, a former motorboat racer and European champion in 1961/1962, it was one of the first shipyards in Portugal to use fiberglass in boatbuilding, a pioneering step that marked the beginning of modern composite construction in the country. Over the decades, the company has earned recognition for producing durable and accessible craft suited to both leisure and small-scale fishing, combining traditional craftsmanship with practical, user-oriented design. Figure 1.1 shows an example of a vessel produced by Riamar (on the left) and a representation of one that is being developed (on the right). Riamar currently

operates from Ílhavo, continuing to supply both domestic and export markets, with a significant share of its production going to international clients.

Currently, Riamar is collaborating with INEGI to enhance the efficiency and sustainability of its future vessels while advancing the company's digitalization processes and adoption of new technologies [1].



(a) Riamar Pescador 500 boat [2]



(b) Riamar Evolution 5.0 Open boat [3]

Figure 1.1: Side-by-side of an existing and an in-development boats designed by Riamar company.

1.1.1.1 Designing Phase

Riamar applies modern digital methods throughout its design workflow. They use Computer Aided Design (**CAD**) tools to represent digitally all physical components of their vessels, enabling designers and engineers to specify dimensions, materials, and geometry in a coherent model. From this 3D model, they also create scale prototypes via 3D printing, which assist in assessing the aesthetics, geometry, and ergonomics of the design before full-scale manufacturing.

Nowadays, Riamar, in collaboration with academic institutions like FEUP and INEGI, is starting to incorporate advanced engineering techniques into the designing phases of their boats. In terms of fluid dynamics, for example, hulls with steps are being researched to determine if they provide a better performance in terms of efficiency.

1.1.1.2 Manufacturing Process

After the design phase of a new vessel is completed, the manufacturing process begins. The hull and the deck are each produced as a single component. The process starts with the fabrication of the mold, which represents the negative geometry of the vessel's hull. This mold may be produced either manually or using a CNC machining process.

Once the mold is finished, an initial layer of gel coat is sprayed onto its surface to provide the hull exterior with surface protection, impermeabilization, UV resistance, and a polished finish. Following this step, the lay-up process begins, during which successive layers of fiberglass fiber and polyester resin are placed inside the mold.

After completion of the hand lay-up process, the mold is left to rest until the resin has fully cured. Demolding can then take place, during which the hull or deck is separated from the mold.

1.1.2 FEUP Academic Solar Team

The motivation for the current work, besides the cooperation with Riamar company and its objectives, has connections with an academic project at FEUP, joining students from different engineering areas, with the objective of designing and manufacturing vessels for student competitions.

The FEUP Academic Solar Team (**FAST**) serves as an instructive project because it demonstrates the experimental and research-oriented aspects of small boat designing. FAST was established in 2023 to develop sustainable maritime solutions through solar-powered boat design and construction. The first prototype from FAST, called 'Atlântico' combines solar-powered racing capabilities with hydrofoil technology and a lightweight hull structure that uses flax fiber reinforced with cork-core sandwich materials, as shown in Figure 1.2.



Figure 1.2: 'Atlântico' vessel prototype developed by FAST.

The use of flax fiber in boat construction requires a detailed and repeated design process since these materials have lower stiffness and ultimate strength than non-natural reinforcement fibers. The team conducted multiple Finite Element Method (**FEM**) analyses to find the optimal laminate thickness, fiber orientation, and internal stiffener arrangement for their design. The simulation process enabled the team to achieve equilibrium between weight, rigidity, and strength performance, which resulted in a vessel that could handle operational forces.

The structural optimization and numerical simulation techniques shown by FAST prove that engineering analysis brings essential value to early design stages which matches the requirements of Riamar and other small- to medium-sized manufacturers.

1.2 Motivation

Small and Medium Enterprises (**SME**) that are dedicated to boat designing and manufacturing often use empirical knowledge as their main foundation for structural design instead of performing systematic analyses to improve the boat's design and performance. The lack of access and knowledge to use advanced engineering tools are some of the reasons that restrict their ability to optimize their designs and meet certification standards.

Riamar company demonstrates the current situation by uniting skilled craftsmanship with production expertise yet facing challenges to validate their structures through accessible methods. The FAST demonstrates how academic teams with limited members and resources can successfully merge simulation and optimization techniques into their design workflow.

Taking over this project was a natural step to further develop and consolidate the knowledge acquired at FAST, being responsible for designing the internal structure of the prototype 'Atlântico'. This study reveals itself as a great opportunity to understand the design methodologies adopted by boatbuilding companies such as Riamar, apply composite manufacturing concepts in the maritime sector acquired during the course and in the manufacturing of 'Atlântico', and enhance numerical modeling techniques regarding FEM software.

1.3 Objectives

The research aims to establish a functional design approach that unites conventional boatbuilding methods with analytical and numerical tools to help small craft manufacturers improve their structural designs, ensure compliance with safety standards, and maintain market competitiveness.

The main objectives of this project are:

- Characterization of materials used by Riamar and comparison of the results with the properties proposed by ISO 12215-5;
- Development of a methodology to be integrated in the design process of a boat that allows the analysis of the vessel's structure;
- Development of an analytical tool based on the ISO 12215-5 methodology to perform structural analysis and scantlings determination of small craft vessels;
- Development of a methodology for implementing FEM, according to ISO 12215-5 guidelines, to validate the results obtained by the computational tool;
- Understanding the viability of FEM software as a structural designing tool, comparing the results obtained by FEM software with ISO 12215-5 scantling results;
- With the developed methodologies, verify if the material that will be applied to the bottom of the hull of the new vessel of the Riamar company is in compliance with the requirements of the ISO 12215-5 standard.

1.4 Thesis Outline

This document is divided into 5 chapters.

Chapter 1 is the current one and presents the context and motivation of the work within the small craft manufacturing industry. Riamar is presented as the industrial case study, and FAST shows the importance of including optimization techniques in the design workflow. Furthermore, the main objectives of the dissertation are defined, establishing the basis for the methodologies and analyses developed in the following chapters.

Chapter 2 presents a review of the small craft industry, addressing its economic and historical context and showcasing difficulties faced by small and medium-sized enterprises. It also covers the regulatory and standard frameworks governing recreational vessels, focusing on the ISO 12215 standard as the reference for structural design and the main components and terminology used in the small boat environment. Additionally, it examines product development methodologies applied to boat design, demonstrating the limitations of traditional rule-of-thumb approaches and the relevance of simulation-driven and integrated design models within the small craft industry.

Chapter 3 addresses the material characterization and the development of the methodologies to be applied. With the necessary background knowledge established, it is essential to define the material basis and methodological framework that will be followed and applied to the case studies under consideration. This includes an initial characterization of the composite materials used by Riamar, both analytically and experimentally, followed by the presentation of the analytical procedure workflow described in the ISO 12215-5 standard and its guidelines regarding FEM implementation. Subsequently, using a representational hull model, the finite element analysis methodology is presented through the application of the FEM packages of two numerical codes, Abaqus and SolidWorks. Particular attention is devoted to identifying the most appropriate mesh discretization parameters, as well as to evaluating the sensitivity of the numerical results to variations in laminate thickness and elastic modulus.

Chapter 4 presents the application of the previously defined methodologies to two different vessel models. The first is an already validated recreational boat manufactured by Riamar, known as Pescador 500, which already has some previously made scantling calculations done by a credited company that will be useful to validate the obtained results. The second is a new model, still being developed, referred to as Evolution 5.0.

Chapter 5 presents the conclusions drawn from the work carried out and outlines important future work that may be developed following this project.

Chapter 2

Bibliographic Review

To achieve the goals that are proposed in this work, some research has to be performed regarding the small craft industry, regulatory and standard frameworks, and general knowledge about vessel-related terminology. In addition to these contents, this chapter presents an investigation of the new product development methodologies applied to the small craft industry, the advancements made by the Riamar company in the development of its new vessel, and the gaps and opportunities identified through this research.

2.1 Small Craft Industry

To develop efficient approaches for the structural design of small boats, it is mandatory to examine the economic and historical framework in which this industry is contained. Understanding the evolution of the sector and its current situation is vital for an adequate framing of structural design methodologies and the application of regulatory and normative requirements.

2.1.1 Economic Overview

The maritime industry is a growing market that already represents a considerable part of the global economy. According to recent market research, the global maritime market reached approximately 2.26 trillion USD in 2024 and is projected to grow significantly, with a Compound Annual Growth Rate (**CAGR**) of around 6.3 % from 2025 to 2034 [4].

More specifically, the recreational boat industry is also a growing market. The Europe Recreational Boating Market, which covers leisure craft including small boats, is estimated at about USD 16.78 billion in 2025 and is projected to grow to around USD 24.67 billion by 2031, as shown in Figure 2.1, with a steady CAGR of roughly 6.7 % during the forecast period. This reflects continued expansion in demand for smaller recreational vessels across EU markets [5].

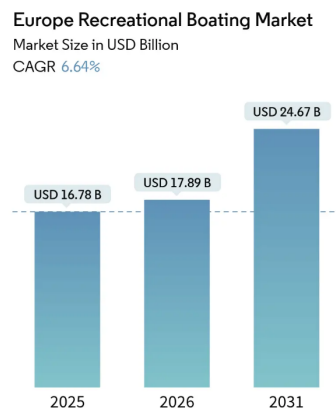


Figure 2.1: European recreational market growth [5].

The European Boating Industry consists almost entirely of small and medium-sized manufacturers (about 97 % SMEs) and includes around 3 600 companies employing over 82 000 people in boatbuilding and related activities, reflecting both the economic size and employment footprint of the small boat sector in Europe [6].

Industry members in the European recreational boating sector, where most manufacturers are SMEs, have specifically identified the need for stronger market surveillance and a stable regulatory environment. Small manufacturers often struggle more than larger competitors with the costs and administrative burden of compliance with, for example, safety and environmental standards under the Recreational Craft Directive, which can be relatively higher for smaller production volumes [7].

Also, European industry associations have warned that trade measures such as tariffs on recreational boats and equipment can jeopardize jobs and harm SMEs more than larger companies. Changes in trade policy or retaliatory duties can have a disproportionate effect on their competitiveness and cost structures [8].

In other recreational boating market reports for Europe, persistent supply-chain bottlenecks and regulatory reasons that increase costs and complexity are emphasized. These factors tend to reduce business profitability and put additional pressure on smaller builders who cannot easily absorb inefficiencies or pass costs to end buyers [5].

Research comparing companies of different sizes consistently finds that SMEs encounter greater financial and capability barriers when adopting advanced technologies like those required for digital transformation, including simulation tools. Larger firms have greater financial resources to invest in innovative methodologies, while SMEs struggle with budget limits and other operational expenses [9].

A study on innovation adoption highlights that larger companies face fewer financial and structural barriers to technology adoption compared with SMEs, meaning they are more likely to absorb the costs and risks of investing in advanced design and simulation technologies [9].

Surveys conducted among SMEs indicate that financial pressures and cost increases are key barriers to digital and technology investment, reinforcing that smaller companies have a harder

time allocating budgets to such tools [10]. Furthermore, a report regarding the digitalization of SMEs in Portugal concluded that "productivity remains a challenge for Portuguese companies' competitiveness," which can be improved by digital adoption. This technological implementation is slightly lower in Portugal than the average in the EU, especially "in traditional sectors where companies' abilities to develop digital assets tend to be more limited". Regarding the reasons defined for this digital adoption delay, beyond the financial costs, the report mentions the lack of awareness, as in the owners and managers often do not know how and where to apply the digital tools, and the lack of capabilities, because the employees would "need technical know-how to integrate such digital tools" [11]. Therefore, the development of methodologies that are simple to apply and cost-effective to implement represents a significant advantage within smaller companies' operating environments.

Furthermore, an investigation conducted by Vítor Carneiro et al. [12] examined the obstacles faced by SMEs in the development of new products. The study highlights previously identified difficulties, such as limited human and financial resources available for investment, and also points out inadequate design practices and the lack of structured methodological frameworks to better organize the product development process [12].

Based on this research, Vítor Carneiro et al. [13] created a model that structures the complex parameters identified into a systematic methodological approach that SMEs can implement with no prior design experience. With this established conceptual framework, a methodology was developed in the form of an integrated design tool with a straightforward user interface, aimed at supporting SMEs "in implementing effective design management practices" within their product development activities. This tool was implemented in a recreational craft company and resulted in "tangible improvements in areas such as design management, strategic integration, and project planning and execution" [14].

2.1.2 Historical Overview

Compared to the small craft origin, with evidence that dates back to the 6th millennium BC, the concept of recreational craft is very recent. Prior to the two waves of development witnessed in the 20th century, the first recreational craft was documented in the 19th century [15].

First, after World War I and the advancements in the automotive industry, the 1930s profited from quick advances in engine power, reduced weight, and more affordable prices. Additionally, the introduction of paid holidays throughout Europe increased the popularity of small crafts, just like it did for all other leisure sectors. In that time, boats are still modest (8/9 m), mostly made of wooden runabouts, and have a top speed of 30 knots [15].

In the 1960s, the second wave of development in small craft was observed. The emerging use of composites, large-scale production, and faster boats with accommodations started to appear. Furthermore, social developments and longer paid holidays helped this progress [15].

During this time, small leisure craft was unregulated. It was until the 1979 Fastnet Race, where the fleet met a violent storm, leading to 15 yachtsmen losing their lives, with 75 capsizes

and 5 sinkings [16]. This tragedy has shown the world the need to impose regulation in small recreational boats, especially in terms of stability and structure of the vessels.

In that regard, the Recreational Craft Directive was approved in 1994, promoting the development of various supporting standards, such as the ISO 12217 for stability and ISO 12215 for structural regulation [17].

2.2 Regulatory and Standard Frameworks

As previously mentioned attention to certification is vital to designing. This section shows some examples of requirements that, according to the Recreational Craft Directive 2013/53/EU [18], small recreational vessels must meet:

1. Stability, buoyancy and structure:

- Remain stable under foreseeable conditions (ISO 12217);
- Have adequate buoyancy (ISO 12217);
- Withstand design loads (ISO 12215).

2. Handling and safety:

- Steering systems (ISO 11592 and ISO 13929);
- Visibility from helm (ISO 11591);
- Protection against falling overboard (ISO 15085);
- Emergency exits (ISO 9094 and ISO 11812);
- Fuel systems and fire protection (ISO 8847, ISO 8848 and ISO 9094).

3. Environmental requirements:

- Exhaust emissions limits (ISO 8178);
- Noise emission limits (ISO 14509).

4. Documentation:

- CE marking - a legal compliance mark, stating that the boat meets safety, environmental and design requirements (ISO 14945);
- Builder's plate - contains key technical limits like the manufacturer name and the CE mark;
- Hull Identification Number - a unique serial number assigned to each boat (ISO 10087);
- Owner's Manual - carries important information for the user to know like the operating speed and safety instructions (ISO 10240).

Not meeting this requirements means that the vessel cannot be legally sold or put into service in the European Union (EU), as insurance and registration may be invalid. The obligation to comply with these conditions is the boat manufacturer, or the importer if the boat was not manufactured in the EU, that is the manufacturer or the importer are legally liable if something happens with the vessel, as stated in Article 44 of Recreational Craft Directive 2013/53/EU [18].

2.2.1 ISO 12215 Series Overview

The "ISO 12215 - Small craft — Hull construction and scantlings" series are the main standards in regard to structural designing certification of small boats. This series is composed by the following parts [19]:

ISO 12215-1 — Materials: Thermosetting resins, glass reinforcement, reference laminates - defines reference materials and laminate properties used throughout the standard;

ISO 12215-2 — Materials: Core materials for sandwich construction - covers core materials used in sandwich panels;

ISO 12215-3 — Materials: Steel, aluminum alloys, wood, other materials - defines properties for non-composite structural materials;

ISO 12215-4 — Workshop and manufacturing - specifies workshop conditions, material storage and handling, and requirements for the manufacturing of the craft;

ISO 12215-5 — Design pressures for monohulls, design stresses, scantlings determination - defines the dimensions, design local pressures, mechanical properties and design stresses for the scantlings determination of monohull small craft;

ISO 12215-6 — Structural arrangements and details - concerns structural details and structural components not explicitly included in ISO 12215-5, ISO 12215-7, ISO 12215-8 and ISO 12215-9;

ISO 12215-7 — Determination of loads for multihulls and of their local scantlings using ISO 12215-5 - defines the dimensions, local design pressures and global loads acting on multihull craft;

ISO 12215-8 — Rudders - gives requirements on the scantlings of rudders fitted to small craft;

ISO 12215-9 — Sailing craft appendages - defines the loads and specifies the scantlings of sailing craft appendages on monohull sailing craft;

ISO 12215-10 — Rig loads and rig attachment in sailing craft - defines the design loads and stresses on rig elements and loads and scantlings of rig attachments and mast steps/pillars of monohulls and multihulls.

Part 5 of this series is the part, as stated above, responsible for the structural design and scantlings itself. This part of the standard presents the design pressures, allowable stresses, and verification criteria to determine the structural validity of the various components of the monohull vessels, like hull plating, stiffeners, and bulkheads. It applies to a lot of building materials, like metals, wood, and fiber-reinforced plastics [20].

This standard gives more importance to local loads rather than global loads for monohull craft. While global loads refer to the phenomenon that produces overall deformation of the entire vessel, such as, for example, longitudinal bending from hogging and sagging, local loads act on individual structure components, such as hull panels, deck panels, and stiffeners [21]. These loads are often caused by hydrostatic pressure on the hull plating, hydrodynamic pressure due to vessel speed or slamming, and impact loads [22]. In contrast, global loads are of major importance and analyzed for multi-hull vessels.

2.3 Components and Terminology of Small Craft

Knowing the appropriate denomination of the boat's common terminology and components is of extreme relevance to understand the language presented in the standards and also to recognize the role of each part of the vessel.

Therefore, a small research about this type of expressions was made to establish a clear understanding of the terminology used throughout the standards and to ensure correct interpretation of the vessel's components and their functions.

2.3.1 Nautical Terminology

Table 2.1 and Figure 2.2 show and explain the terminology used in the marine environment to refer to directions of the boat.

Table 2.1: Terminology used in marine environment [23].

Expression	Meaning
Stern	The rear or aft-most part of a boat or yacht, located opposite to the bow.
Aft	At or near the stern.
Port	The left hand side of the boat looking forward.
Starboard	Right-hand side of the boat looking forward.
Bow	Front-most part of the boat, opposite to stern.
Fore	At or near the bow.

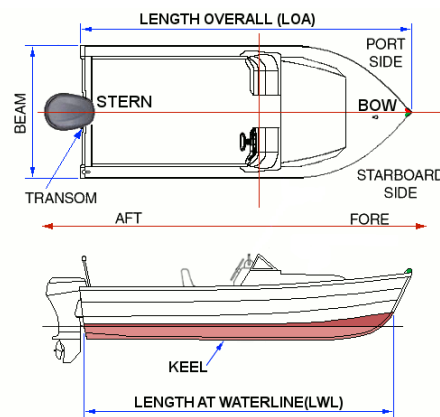


Figure 2.2: Boat environment terms and components [24].

Regarding the structural components, Table 2.2 shows the names of such components and provides a brief explanation of their role in achieving the structural stability of boats.

Table 2.2: Main structural components of small motorboats and their functions [24].

Component	Description and Structural Role
Hull	The main watertight body of the vessel. It forms the outer skin, resisting hydrostatic pressure and supporting all other components. The hull bottom sustains the highest loads from slamming and buoyancy forces.
Deck	The upper closure of the hull that adds torsional stiffness and provides a working platform. Often reinforced with deck beams or sandwich panels.
Transom	The flat or slightly curved surface closing the stern. It supports outboard engines or sterndrives and transfers thrust into the hull structure.
Chine	The distinct line or edge where two surfaces of a boat's hull meet, most commonly where the bottom meets the side shell.
Stringers	Longitudinal stiffeners bonded to the inner hull surface to increase resistance to bending between frames. Often run parallel to the keel.
Bulkheads	Vertical structural walls dividing the interior into compartments. They provide torsional rigidity and improve safety by compartmentalization.
Top-hats	Common type of component of the internal structure of a boat, can be used parallel to the keel or transverse.

Concerning the hull, vessels may present different hull types. A displacement hull moves

through the water by pushing it aside, and almost all of the vessel's weight is supported by buoyancy. A planing hull, on the other hand, is designed to create hydrodynamic lift the faster it goes. When the speed is high enough, the hull rises partially out of the water and is held up by a mix of buoyancy forces and dynamic pressure, entering planing mode [25]. Geometrically, displacement hulls are more rounded, which results in them being more efficient at lower speeds and planing hulls have a more pronounced V-shape with flatter surfaces to increase the hydrodynamic lift, making them more efficient at higher speeds, as shown in Figure 2.3.

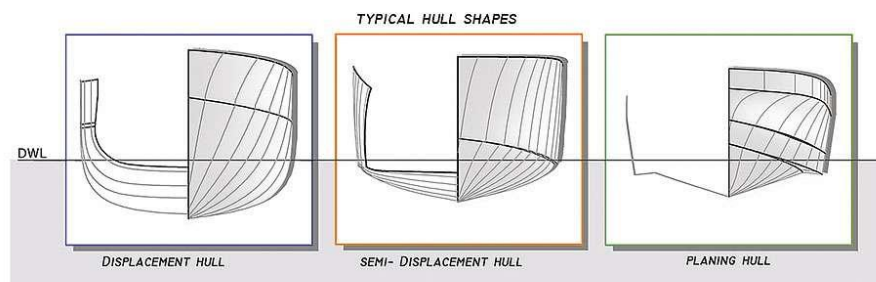


Figure 2.3: Displacement hull, semi-displacement hull, and planing hull representations [26].

Furthermore, there are also other definitions useful specifically for the understanding and application of the contents presented in the standards (Table 2.3).

Table 2.3: Terms relevant for the application of ISO 12215 standard.

Term	Description
Panel	Portions of the hull or deck delimited by supporting members.
Stiffener	Longitudinal or transverse structural member attached to plating to provide additional rigidity, reduce panel spans, and transmit loads to primary supporting structures.
Scantling	Specified dimensions and material properties of structural components, including plate thicknesses, stiffener section properties, and spacing.
Design Loads	Calculated loads applied to the structural elements to be analysed.
Design Stress	Maximum allowable stress used for structural dimensioning, derived from material properties and appropriate safety factors.
Beam	Maximum width of the boat.
Lenght Overall	Lenght from the fore-most point of the boat to the aft-most
Length at Waterline	Horizontal measurement of a vessel's hull from bow to stern at the exact point where it intersects with the water surface.

2.4 Product Development applied to the Small Craft Industry

When designing a new boat, this is similar to designing a new product that must be appealing to the customer in terms of functionality, efficiency, and even aesthetics. Therefore, the development of a boat should follow a well-organized plan in order to maximize its potential.

In this context, a product development study was carried out to understand how new methodologies can improve the design process of small boats.

2.4.1 New Product Development

The New Product Development (NPD) is the process as a sequence of activities extending from the generation of the new idea to the launch of the product to the market place. Robert G. Cooper was one of the most influential people in terms of defining this new concept, being responsible for the development of the "Stage-Gate model" in the 1980s [27].

According to Cooper, new products are prone to fail as only 1 in 10 new product concepts succeed. Furthermore, "44% of new-product projects fail to meet their profit objectives, and 49% are launched late to market" [27]. Stage-Gate objective is to transform this high-risk, often disorganized activity into a disciplined, manageable business process.

The Stage-Gate model is divided in 5 steps:

1. **Scoping** - research to access the technical and market merits of the idea;
2. **Build Business Case** - primary research including market research, concept testing and technical appraisals;
3. **Development** - implementation of the development plan and creation of the product;
4. **Testing & Validating** - validation of the viability of project with in-house tests and trials;
5. **Launch** - full commercialization of the product.

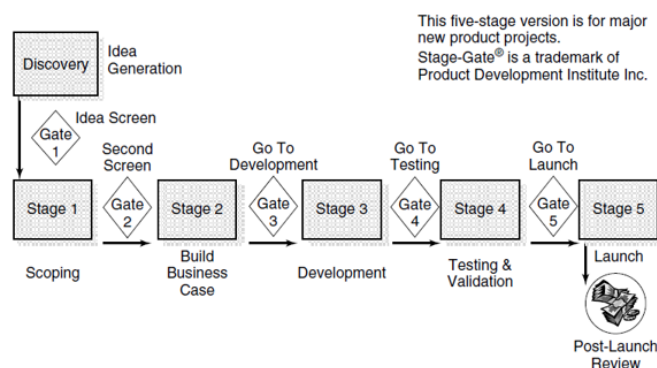


Figure 2.4: Stage-Gate model steps [27].

Due to the linearity of this model, meaning multiple steps could not be taken simultaneously, and since the modern world is characterized by a fast-paced market, new variations of this model have been developed. For example, the Hybrid Agile Stage-Gate model, depicted in Figure 2.5 has a few implementations that deserve to be noted, such as:

- Iterative "spiral" - repeatedly putting something in front of customer to receive feedback;
- Fluid product definitions - product evolves with feedback while developing;
- Tangible deliveries - prototypes, virtual models, 3D prints or CADs.

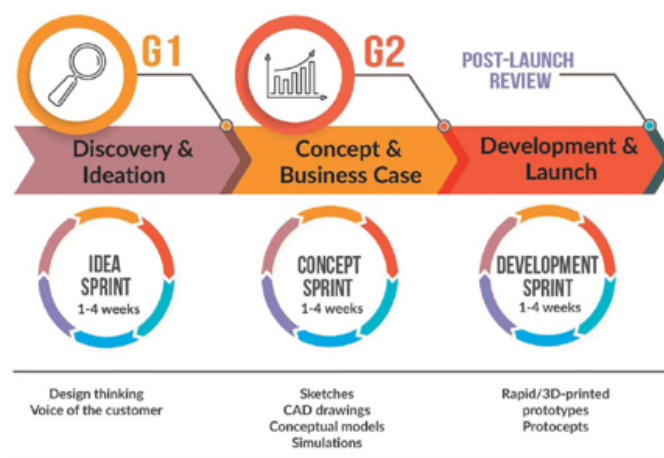


Figure 2.5: Hybrid Agile Stage-Gate model steps [28].

2.4.2 Rule-of-Thumb & Design Errors

SMEs, that have yet to adapt to the above-mentioned ways of designing new products, often design the product and sent it directly to manufacturing due to not having the resources or knowledge to do a proper simulation on how the product would perform functionally, meeting the required criteria [29].

In the boat-building industry, these companies rely on their "rule-of-thumb" methodologies to design their new boats. "Rule-of-thumb" in engineering means the use of a heuristic or experience-based guideline to estimate design parameters quickly, based on past practice, rather than using theoretical analysis [30].

A few curious examples of these rules for fiberglass construction are described by Dave Gerr [31], who states that:

- basic shell thickness is the length of hull divided by 1000;
- bottom is 15% thicker;
- for planning hulls, increase the bottom thickness by another 10%;

- deck is 10% thinner than the basic thickness.

These types of rules are commonly very conservative, thus making them very safe for both the buyer and the manufacturer, since the boat is likely to not have structural problems. However, not performing specific analysis, hydrostatic and structural for example, could lead to non-optimized designs [32].

Furthermore, these methods, "though time-tested, struggle to cope with the increasing complexity of marine engineering challenges, especially in the face of stringent environmental regulations and the demand for higher efficiency," as stated by Pawlusik et al. [32]. Ultimately this methodology is very prone to design errors.

Design errors in the boat industry, as in all other industries, are something to avoid at all costs. According to Balakumar Ramaswamy [33], design errors could lead to significant added costs in terms of working hours, monetary losses and even environmental effects, as can be seen in Table 2.4. Some reasons that commonly lead to design errors are the use of empirical shortcuts. This come into agreement with what was researched about the rule-of-thumb method and insufficient iteration [33]. The domains of the design that are most vulnerable to errors are the hull form design and resistance prediction, the weight and stability engineering and structural scantlings, among others.

Table 2.4: Typical cost impacts associated with boat design deficiencies [33].

Category	Typical Cost Impact
Design rework (engineering)	+10–30% of design hours
Structural retrofit	+5–20% of material budget
Stability correction (ballast, layout change)	+3–10% of overall cost
Operational penalty (fuel, speed)	+5–15% fuel cost per year
Reputational / warranty exposure	Not directly measurable

2.4.3 New Product Development in the Small Craft Industry

There are also companies that have employed NPD knowledge in their boat's designing process. BK Yacht's for instance, which is a boat designing company, define their whole design process in these stages [34]:

1. Conceptual Design - brainstorming, research and sketches;
2. Preliminary Design - initial 3D modelling and renders;
3. Detailed Design & Construction Drawings - hull shape optimization, weight estimate, stability and structural calculations, Computer Numerical Control (CNC) and manufacturing files;
4. Manufacturing;

5. Validation and Sea Trials.

In the "Detailed Design" phase, numerous companies in the boat manufacturing industry employ the "Design Spiral" methodology to implement this phase, shown in Figure 2.6. It is distinguished by the continuous enhancement of all design categories in accordance with the previous stages of its spiral. The goal is to constantly improve the product until it achieves a nearly flawless final outcome.

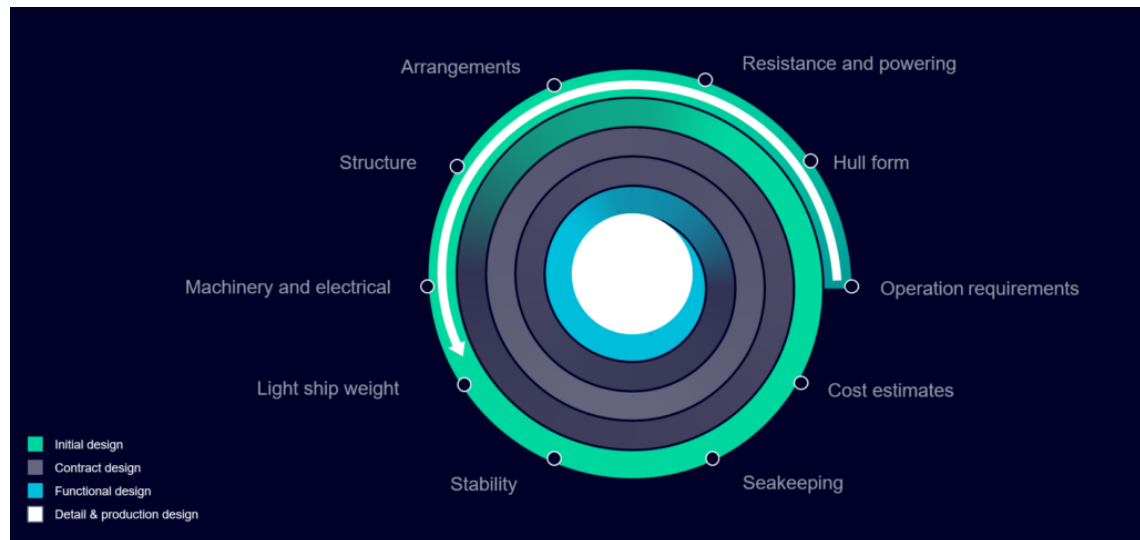


Figure 2.6: Design Spiral description used in ship design [35].

According to Balakumar Ramaswamy, boat designer at BK Yacht, they specifically include structural integrity analysis using ISO 12215 and stability analysis using ISO 12217 in their "Design Spiral" as their elective method to design their yachts [36].

2.4.4 Simulation-Driven Design

The Simulation-Driven methodology is a design and development approach that uses numerical simulation to inform and validate decisions from early concept stages onward. According to Massobrio [37], "is an engineering approach that leverages advanced (3D) virtual technology to drive the design and development process by getting, for a product or assembly, qualitative answers (such as a 3D color map showing pressure distribution) and quantitative details (such as the point-by-point values, extractions of averages or min/max values) before the physical product is built". Additionally, the methodology combines conventional CAE simulation techniques, artificial intelligence projections, system modeling, and process simulation within product development, enabling engineers to put their conceptions into practice and address client needs and limitations.

This type of approach has already seen a lot of benefits in various industries. Simulation tools allow engineers to test many design variations in a virtual environment, accelerating the overall development timeline and lowering time-to-market compared to reliance on physical prototypes

alone. In fact, reports show that simulation use can reduce development time by up to 75% when integrated early in the design process [38].

Additionally, simulation techniques enable optimization of material usage and performance before physical production, reducing waste and saving materials through virtual exploration of design alternatives [39]. Furthermore, Charles McLean and Swee Leong, members of the Manufacturing Simulation and Modeling Group of the National Institute of Standards and Technology of the United States of America, state that simulation technology can reduce overall product development costs, improve quality, and shorten time-to-market, which reinforces that earlier “virtual testing” is less expensive than relying primarily on later physical tests [40].

Obviously, simulation-driven design is already present in the boatbuilding industry. On one hand we have, for example, Computational Fluid Dynamics (CFD) simulations that are used to predict airflow/waterflow interactions, drag and other performance metrics essential for efficient marine vessel design [41]. On the other hand, Finite Element Analysis (FEA) can be used as a validation and optimization method for the structural development of a boat. It can be utilized to compare with results obtained with scantling tools software, which are programs that allow the importation of the CAD model and perform the scantling calculations, based on ISO 12215, for instance, to ensure the compliance with the standards.

To perform FEA, companies value software that integrates CAD with CAE to avoid imports and exports of data. Engineering research indicates that traditional CAD and CAE systems have different geometric representations and data structures, requiring manual intervention when transferring models. This manual work is time-consuming and can introduce errors when preparing geometry for simulation (for example, meshing issues or model refinement) [42].

Moreover, studies on CAD/CAE integration specifically point out that the lack of compatibility between design and analysis systems results in extra manual processing steps, time delays, and risk of geometric inconsistencies. Integrated platforms aim to eliminate these translation steps by maintaining a consistent model from design through simulation [43].

2.4.5 V-model Approach

The V-model approach, recently adapted to ship design, combines the best of both worlds from the design spiral and the simulation-driven design method.

The V-model originated in the software industry during the 1980s, evolving from the conventional waterfall paradigm. Engineers determined that intricate projects necessitated a more systematic methodology for overseeing testing and validation phases [44].

Siemens is currently implementing the V-model into the ship design, shown in Figure 2.7. According to Marie Loison [35], “Shipyards and designers are under increasing pressure to create efficient designs in shorter time frames. Failure to meet the required targets leads to financial penalties”, meaning that the designing phase of a ship must be the minimum time-length possible to get the advantages versus the other competition. One of the downsides of the spiral is its linearity, a problem also present in the original Stage-Gate model, curiously. Once again, the fact

that each step in the designing phase must wait for the end of the prior leads to inefficiency during this stage.

In that regard, the V-model has been developed to eliminate those inefficiencies. This approach is "based on digital data handling and simulation tools available today", according to Marie Loison [35]. It acknowledges that decisions made at each stage affect the options for subsequent selections and that the process is iterative and collaborative. It also seeks to reassess procedures, technologies, and organizational workflows to enhance traceability and synchronization during the design phases.

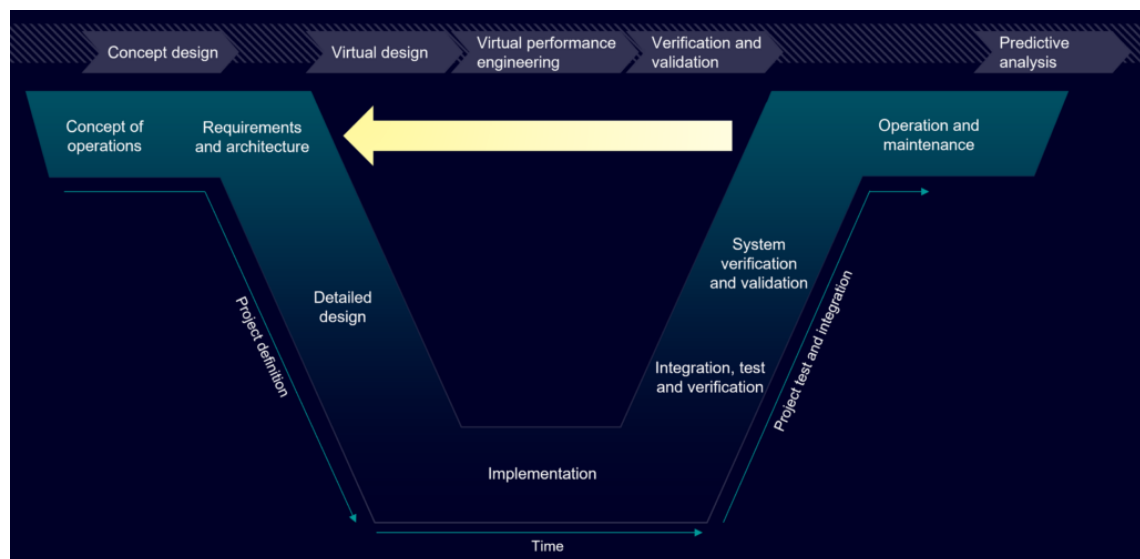


Figure 2.7: V-model applied to ship designing [35].

This model originates a wholly integrated design process, shown in Figure 2.8, where CAE simulation is directly connected to CAD models, performance metrics, and design optimization instruments. This methodology enables naval architects to concurrently assess many operational criteria. Shipyards might explore numerous alternatives through simulation and swiftly concentrate on particular design variants that fulfill the task criteria.

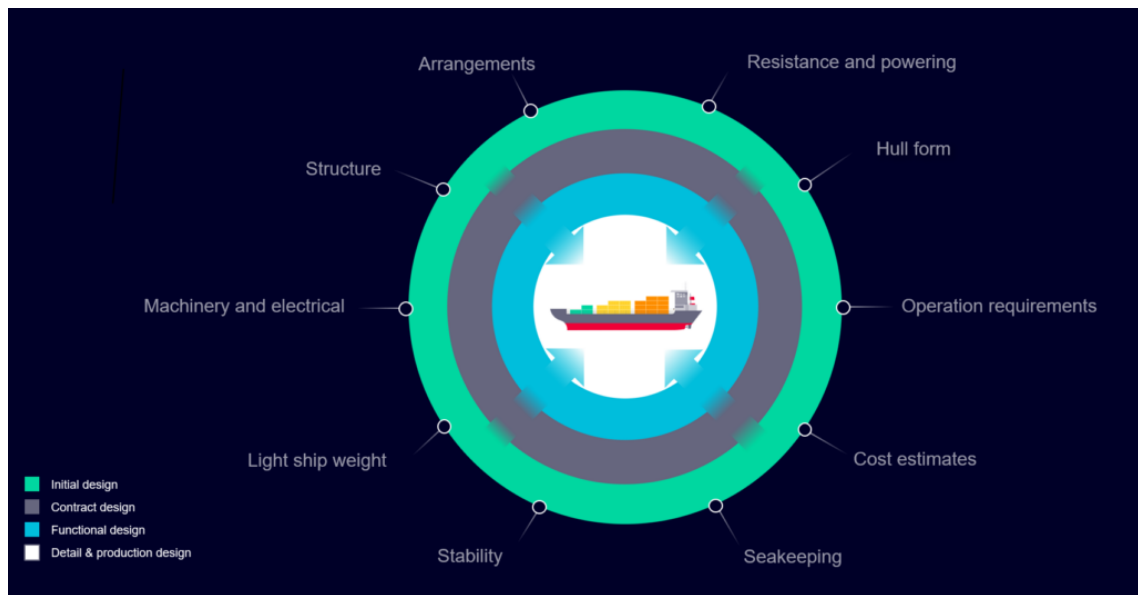


Figure 2.8: Integrated design model obtained with the application of the V-model [35].

Euan Freeman [45], engineer of the company Cox Marine, regarding the implementation of the model developed by Siemens in their projects, claimed that they "significantly reduced cost and time on the front-end as we got the right design answer in the shortest possible time. It is massively important because we can get people to make the right design decision, thereby reducing development time by 50 to 80 percent". Naturally, Siemens projected this approach for ship designing, but it can easily be implemented in smaller boat development.

2.5 Riamar Vessels Development Progress

Riamar has been developing their new models for years. Orlando Silva, in his master thesis [46], has developed a preliminary 3D model of a vessel, using the CAD functionalities of Solid-Works software, which was called Riamar Evolution 7.0, shown in Figure 2.9. Additionally, 3D models of the mold of the hull and deck and their negatives were created to help the planning of the manufacturing phase.

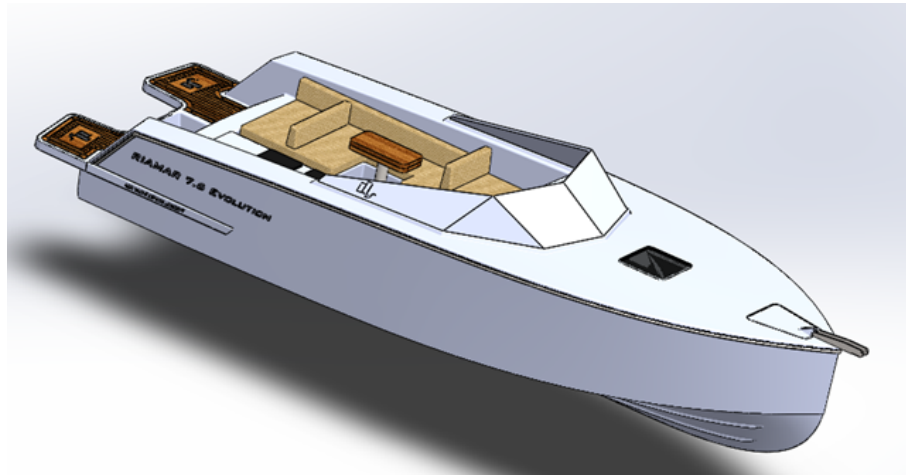


Figure 2.9: Final CAD model of the Evolution 7.0 vessel [46].

Also, stability tests, using an add-in for SolidWorks called Flow Simulation, were performed to ensure the validity of the design in terms of performance and to foresee its behavior in open waters, as demonstrated by Figure 2.10.

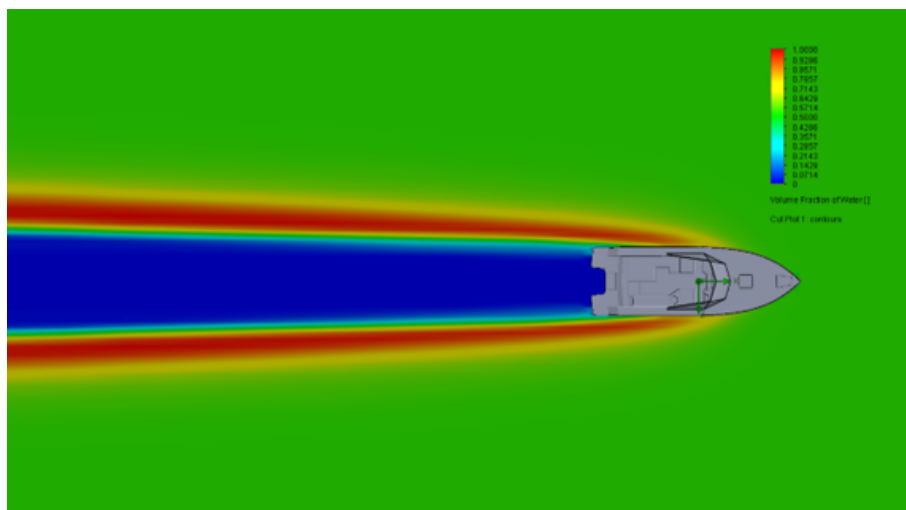


Figure 2.10: Wake of Riamar Evolution 7.0 vessel [46].

Furthermore, prototypes of the boat, shown in Figure 2.11, and the molds, in Figure 2.12, were made using additive manufacturing. This process allows the designers to see errors and potential problems that would be invisible in the software, enabling once again the validation of the vessel's design.

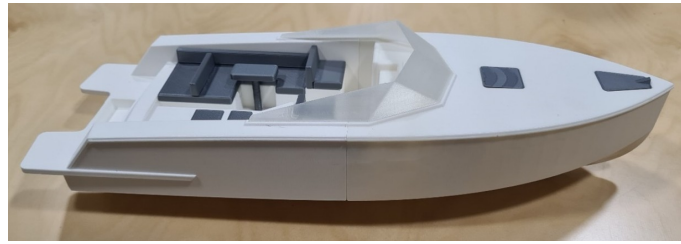
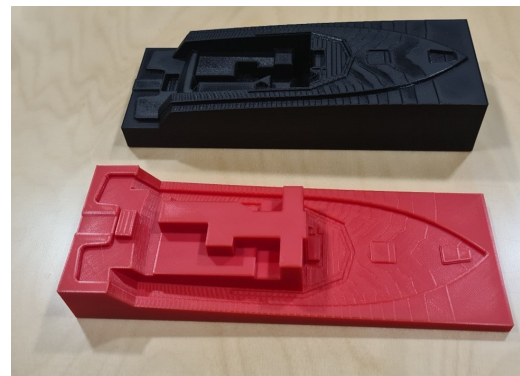


Figure 2.11: 3D printing of the Riamar Evolution 7.0 vessel [46].



(a) 3D printing of the hull molds [47]



(b) 3D printing of the deck mold [47]

Figure 2.12: 3D printing of the molds developed for the the production of the Riamar Evolution 7.0 vessel.

In another thesis work, performed by Joana dos Santos [48], an extensive investment was made into the designing of a new cabin to be used for the vessel, as depicted in Figure 2.13.

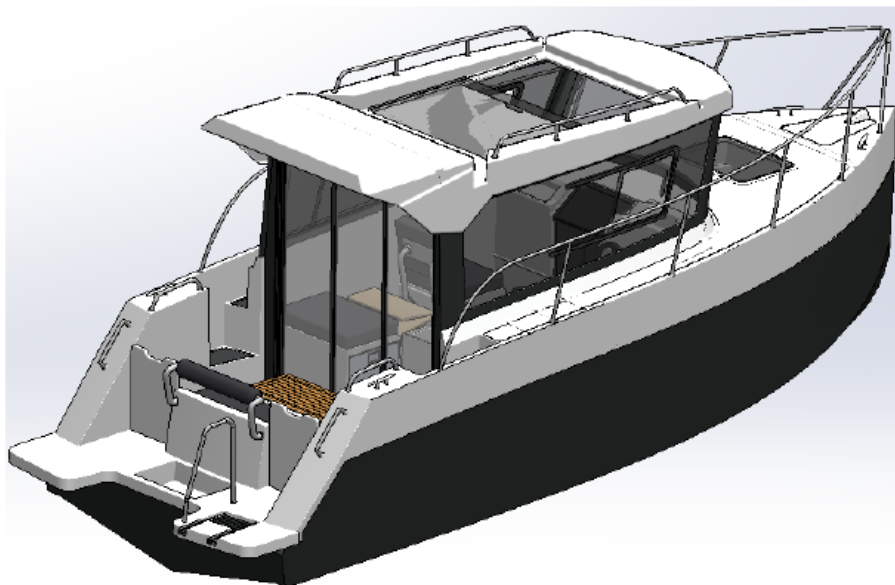


Figure 2.13: Deck design of Riamar Evolution 7.0 vessel, developed by Joana dos Santos [48].

Nowadays, the Riamar company is developing a vessel with its geometry based on the one designed by Orlando Silva, but with only 5 m length called Riamar Evolution 5.0.

2.6 Chapter Summary

This chapter laid the groundwork for the development of methodologies to structural design small recreational boats by looking at the industry context, the rules that govern it, and the different methodologies used to design a boat. The small craft sector was shown to be economically important and mostly made up of small and medium-sized businesses.

The development of recreational boats over time and the use of composite materials made it clear that structured design approaches were needed. This was made even clearer by the introduction of international standards after major maritime accidents. In this context, the ISO 12215 series, especially ISO 12215-5, was chosen as the main reference for structural design.

The study of structural elements and maritime terminology provides the necessary technical groundwork for the precise comprehension and application of these requirements. Also, looking at traditional design methods showed that they were not very efficient and optimized. This led to the establishment of new development frameworks like the Design Spiral, Simulation-Driven Design and the V-model.

Finally, the way processes are designed at Riamar right now demonstrates that they are shifting from using traditional methodologies to using procedures that are backed up by simulations. This demonstrates the need for a methodology that integrates both analytical and numerical approaches. This bibliographic study thus validates the methodological framework established in the next chapters.

Chapter 3

Materials, Mechanical Characterization and Methods

This chapter presents the characterization of the materials used by Riamar, performed by experimental tests and analytical calculation. Additionally, the methodology applied to the scantling determination of the boat components, according to ISO 12215-5, is also explained, as well as some guidelines to perform the **FEA**. Furthermore, a numerical model was created to understand the workflow of the two selected software FEM packages and optimize the choice of numerical variables that will be most appropriate in efficiency and accuracy for the proposed applications.

3.1 Materials and Mechanical Characterization

The Riamar company will employ two glass fiber composite materials in the production of its new vessel. To properly assess their mechanical properties and laminate thickness, both analytical and experimental approaches will be applied. The results obtained from these methods will then be compared to define the set of material properties to be adopted in the structural validity assessment. The laminate used in the Pescador 500 vessel is also presented in this section.

3.1.1 Composite Materials Description

As previously stated, Riamar uses composite materials reinforced with glass fibers in the production of its vessels. Two types of glass fiber reinforcement structures are used by the company:

- Chopped Strand Mat (**CSM**) is a fiberglass reinforcement composed of randomly oriented chopped glass fibers held together by a binder, shown in Figure 3.1; due to the random fiber orientation, composites manufactured with this reinforcement can be classified as quasi-isotropic;
- Rovimat (**RVM**), shown in Figure 3.3 is a fiberglass reinforcement that combines CSM with Woven Roving (**WR**), which fiber structure can be observed from Figure 3.2; owing to the

presence of WR fibers, composites reinforced with this material are expected to exhibit similar mechanical properties in the 0° and 90° directions.



Figure 3.1: Dry fiber of chopped strand mat [48].

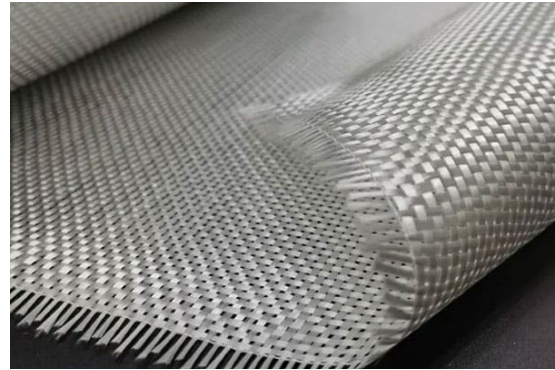


Figure 3.2: Dry fiber of woven roving [49].

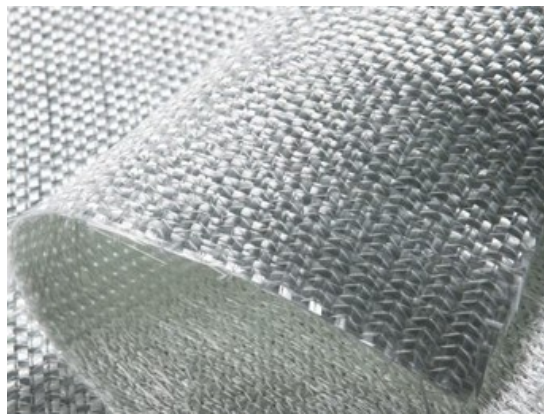


Figure 3.3: Dry fiber of rovimat [48].

Composite materials composed exclusively of CSM layers can be considered quasi-isotropic, meaning that they exhibit nearly identical strength and stiffness in all in-plane directions due to the random orientation of the fibers [50]. In contrast, composites reinforced with WR or RVM present anisotropic behavior, and 0° and 90° being usually defined when establishing mechanical properties. Composites reinforced with RVM layers will be characterized in a single direction, with the understanding that the results are representative of both directions.

The resin used is a general-purpose polyester, commonly employed in the boatbuilding industry due to its adequate mechanical properties and compatibility with the manufacturing process adopted, namely hand lay-up. Although this process is not the most time-efficient, it is simple, cost-effective, and well suited to the geometry of marine vessels.

The composite material used in the hull of the Pescador 500 vessel consists of five CSM layers: three with a dry fiber mass per square meter of 0.6 kg/m^2 , one of 0.45 kg/m^2 , and one of 0.3 kg/m^2 . This material composition is shown in Table 3.1.

Table 3.1: Layers of dry fiber of composite material used in Pescador 500 vessel's hull.

Composite Material	Fiber layer	Dry fiber mass per square meter [kg/m ²]
Pescador 500 Hull	Layer 1 - CSM	0.30
	Layer 2 - CSM	0.45
	Layer 3 - CSM	0.60
	Layer 4 - CSM	0.60
	Layer 5 - CSM	0.60
	Total	2.55

As previously stated, two composite laminates will be used in the manufacture of the hull of the new Evolution 5.0 vessel. The first, referred to as "Mat" for simplicity, is composed of four CSM glass fiber layers: three with a dry fiber mass per square meter of 0.60 kg/m² and one with 0.45 kg/m². This material will be applied to the side walls of the hull and to the internal structure of Riamar's new vessel, Evolution 5.0. The second, designated "Rovimat", consists of the same CSM layers with an additional RVM reinforcement layer composed of 0.45 kg/m² of CSM and 0.50 kg/m² of WR. This material will be used on the bottom of the hull of the Evolution 5.0 vessel. The composition of both laminates is shown in Table 3.2. Figure 3.4 presents a CAD model representing the mold of the Evolution 5.0 hull, highlighting the regions where the two materials are applied.

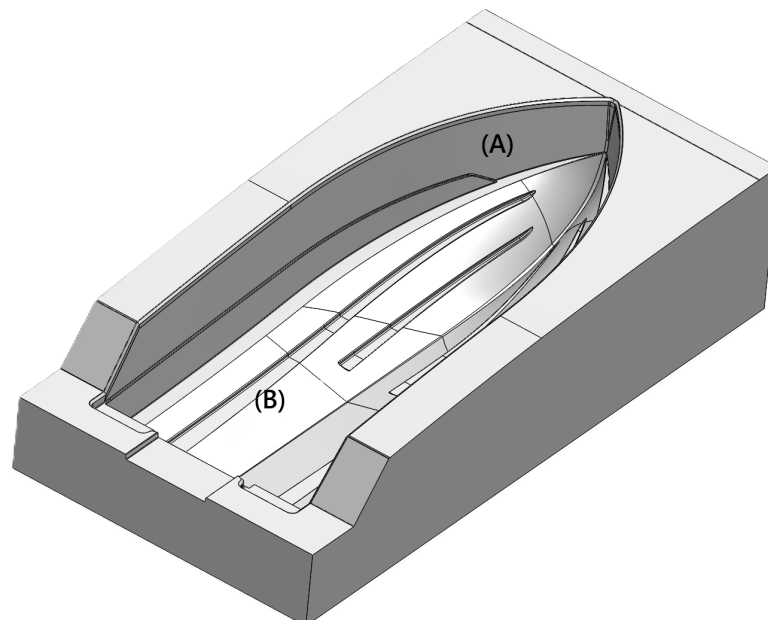


Figure 3.4: CAD representation of the mold of the Evolution 5.0 hull; the side walls of the hull, region (A), are laminated with "Mat" composite material, whereas the hull bottom, region (B), is laminated with "Rovimat" material.

Table 3.2: Layers of dry fiber of composite material options to be used in Evolution 5.0 vessel's hull.

Composite Material	Fiber layer	Dry fiber mass per square meter [kg/m ²]
(A) "Mat"	Layer 1 - CSM	0.45
	Layer 2 - CSM	0.60
	Layer 3 - CSM	0.60
	Layer 4 - CSM	0.60
	Total	2.25
(B) "Rovimat"	Layer 1 - RVM	0.95
	Layer 2 - CSM	0.45
	Layer 3 - CSM	0.60
	Layer 4 - CSM	0.60
	Layer 5 - CSM	0.60
	Total	3.20

The ISO 12215-5 standard allows material characterization either through experimental testing of representative samples or by analytical calculation, as it includes sections describing the process to characterize the properties of metals, wood, and fiber-reinforced plastics, including glass fiber composites.

3.1.2 Experimental Tests

Tensile and 3-point bending tests were performed regarding the experimental part. Since the main objective of this work is the structural design of the new vessel, only "Mat" and "Rovimat" materials were studied.

The performed tensile tests were based on the ASTM-D3039/D3039M [51] standard, and the three-point bending tests were executed in accordance with the ASTM-D7264/D7264M [52] standard.

Two different plates were manufactured containing the composite composition of the two studied materials, presented in Figure 3.5.

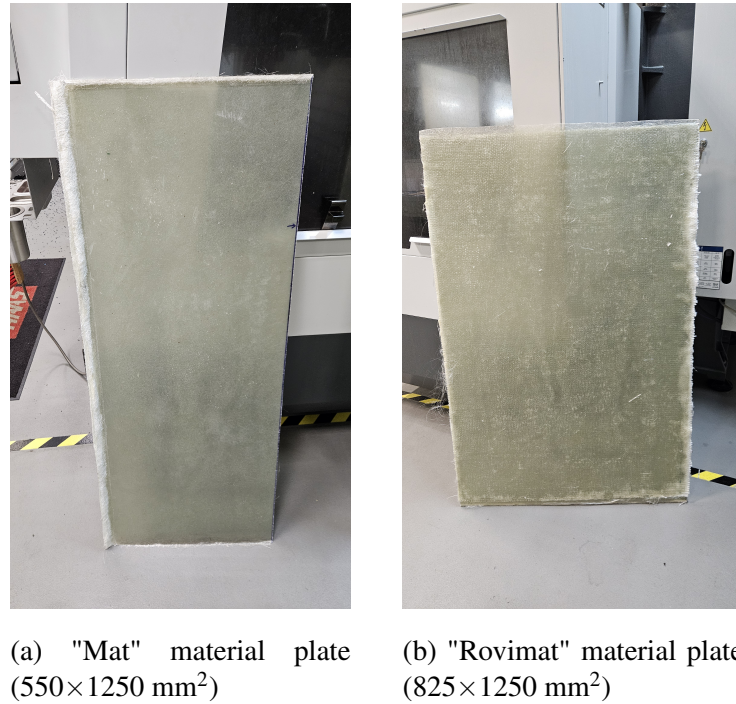


Figure 3.5: Manufactured laminate plates used in experimental tests; the nominal thicknesses measured for each laminate were 3.9 mm for "Mat" material and 5.1 mm for "Rovimat" material.

From these plates, regular rectangular specimens with dimensions of 250×25 mm² and 150×45 mm² were cut for the tensile and three-point bending tests, respectively, as shown in Figure 3.6. These plates have nominal thicknesses of 3.9 mm and 5.1 mm, respectively, for "Mat" and "Rovimat" materials.

The specimens were cut using a diamond circular saw (model DV 25, manufactured by Wendt GmbH, Germany). The machine allows a maximum blade diameter of 335 mm and a maximum blade rotational speed of 2 900 rpm. No delamination was observed in any specimen after the cutting procedure.

Regarding the number of specimens, a total of 14 samples were cut from the "Mat" material plate: 6 for tensile testing and 8 for three-point bending tests. From the "Rovimat" material plate, three specimens were manufactured for each fiber orientation (0°, 45°, and 90°), resulting in a total of 9 specimens for tensile testing. For three-point bending tests, due to time constraints, only the 0° fiber orientation was analyzed, with 8 specimens produced for this direction.

Each specimen was assigned a name or acronym. The first letter denotes the material (M for "Mat" and R for "Rovimat"), and the second letter indicates the test type (T for tensile and B for bending), followed by 0, 45, or 90, corresponding to the fiber orientation. Finally, the last number serves as the specimen identifier. For example, the first tensile specimen of the "Rovimat" material with a 0° fiber orientation is designated as *RT0#1*.

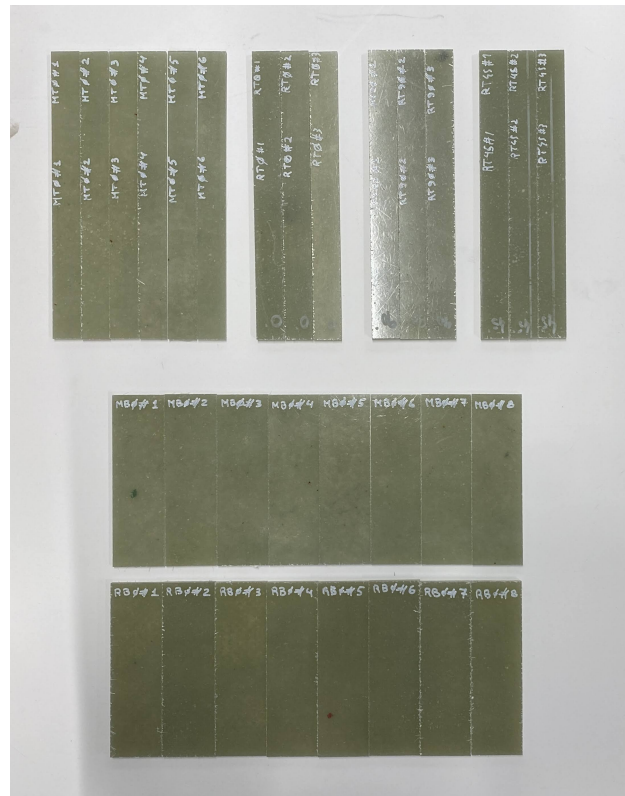


Figure 3.6: Samples manufactured for tensile and three-point bending tests; the top ones, of dimensions 250x25 mm were used for the tensile tests and the bottom ones, of dimensions 150x45 mm, for three-point bending tests.

3.1.2.1 Tensile Tests

The first step to perform the tensile tests was to measure the width and thickness of the specimens. These measurements were carried out using a digital micrometer (Mitutoyo Digimatic) with an accuracy of $2\text{ }\mu\text{m}$ and a resolution of 0.001 mm . The average width and thickness were then calculated from the recorded measurements. The averages are shown in Table 3.3.

Also, a thickness dispersion graph, shown in Figure 3.7, was made using the thickness measurements. Additional details about the measured values are displayed in Appendix A, in Table A.1. Only specimens #1, #2 and #3 of "Mat" material were tested.

Table 3.3: Averages of measured width and thickness of tensile test specimens.

Type	Spec.	Average width	Average thickness
		[mm]	[mm]
MT0	#1	24.4	3.7
	#2	24.4	3.7
	#3	24.4	3.5
	#4	24.1	4.1
	#5	24.3	4.1
	#6	24.4	4.0
RT0	#1	24.2	4.9
	#2	24.3	4.5
	#3	24.3	3.9
RT45	#1	24.3	5.3
	#2	24.4	5.2
	#3	24.4	5.2
RT90	#1	24.4	5.0
	#2	24.2	4.9
	#3	24.3	5.2

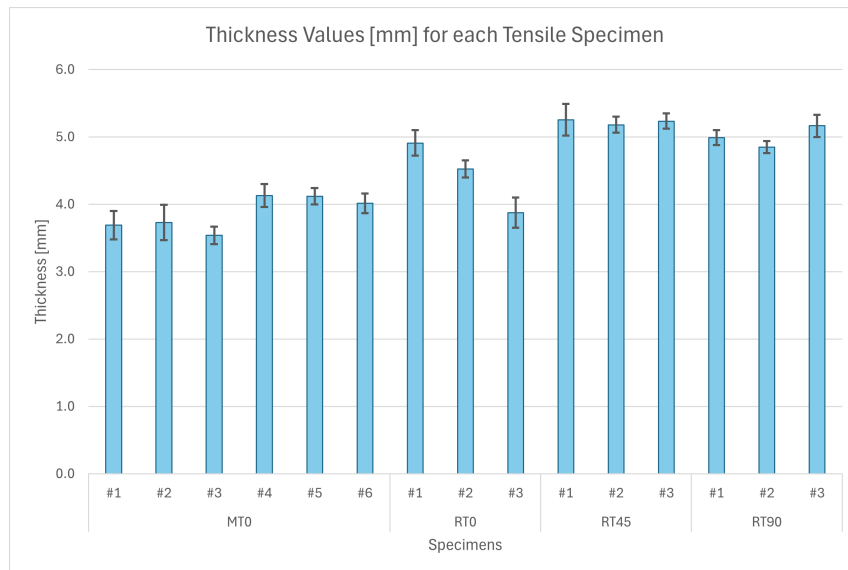


Figure 3.7: Thickness dispersion graph for the tensile samples.

With the required geometric data available, the test setup could be prepared. The testing machine used was an INSTRON 5900R, which has a maximum load capacity of 300 kN.

For the tensile tests, hydraulic grips were used, as shown in Figure 3.8, with a grip pressure of 200 bar, from a 692 bar maximum. This type of grip allows the specimen to be positioned and secured without the application of manual force, as it can be held in the correct alignment while

the grips are hydraulically clamped. The gripping force is controlled and adjusted through the control panel located on the left side of the machine, shown in Figure 3.9. These grips offer several advantages compared to conventional gripping mechanisms. They provide evenly distributed pressure across the grip area, reducing stress concentrations at the grip interface and minimizing specimen slippage during loading due to the high and stable clamping force.

Regarding other important test parameters, the displacement rate used was 5 mm/min, and an extensometer with a gauge length of 50 mm was utilized to measure the specimen strain.

Each sample was put under tensile load until there was obvious failure. Force-displacement data was obtained for each specimen with an acquisition frequency of 20 Hz, or every 0.05 seconds. These curves were used to obtain the stress-strain curves, in accordance with ASTM D3039/D3039M.

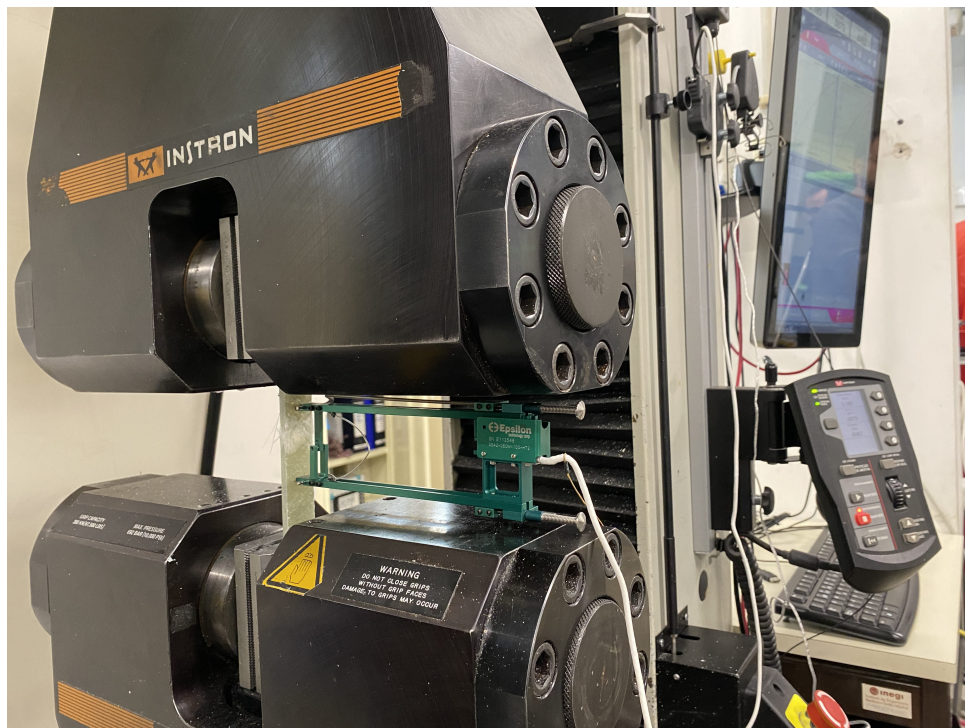


Figure 3.8: Hydraulic grips and extensometer used in the tensile tests.



Figure 3.9: Global view of the setup for tensile tests; to be highlighted the hydraulic grips and their control equipment (on the left of the tensile machine), besides the computer numerical control of the tensile machine (on the right side).

The ASTM tensile standard [51] suggests an identification code regarding the failure type, area and location. For each specimen, three letters are attributed, the first concerning the failure type, the second the failure area, and the third the failure location, as shown in Table 3.4 [51]. Figure 3.10 illustrates the failure locations for each specimen and the failure identification designation for each specimen tested.

Table 3.4: Failure mode coding system for tensile testing, which is composed of three characters: the first for the failure type, the second for the failure area, and the third for the failure location.

First Character		Second Character		Third Character	
Failure Type	Code	Failure Area	Code	Failure Location	Code
Angled	A	Inside grip/tab	I	Bottom	B
Edge delamination	D	At grip/tab	A	Top	T
Grip/tab	G	<1w from grip/tab	W	Left	L
Lateral	L	Gauge	G	Right	R
Multi-mode	M(xyz)	Multiple areas	M	Middle	M
Long. splitting	S	Various	V	Various	V
Explosive	X	Unknown	U	Unknown	U
Other	O				

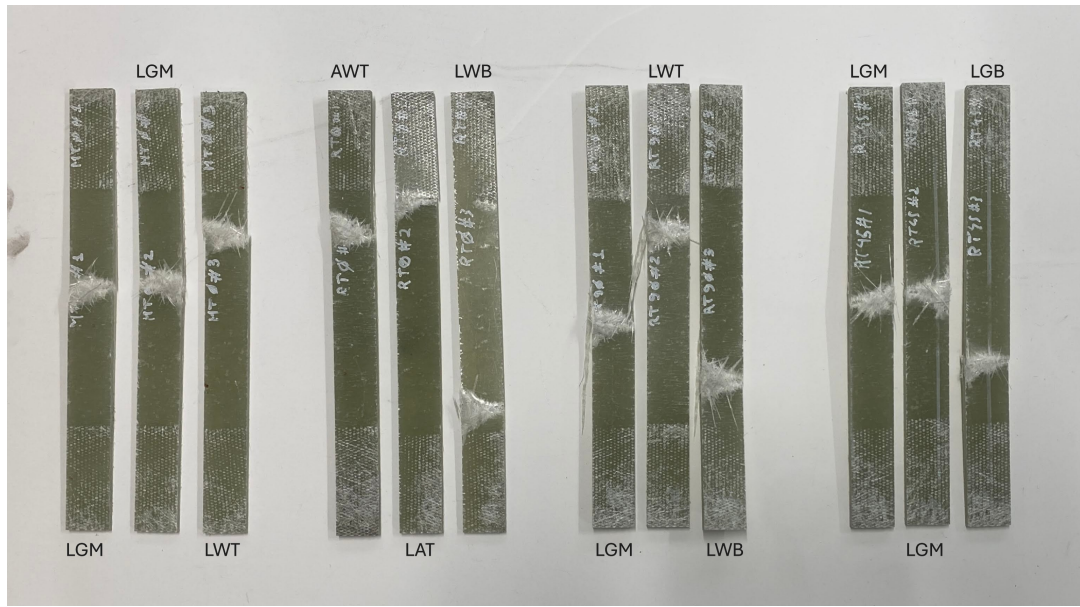


Figure 3.10: Tested tensile specimens and their failure identification designation.

The stress at each data point (σ_{it}) is defined by:

$$\sigma_{it} = \frac{P_i}{A} = \frac{P_i}{w \times t}, \quad (3.1)$$

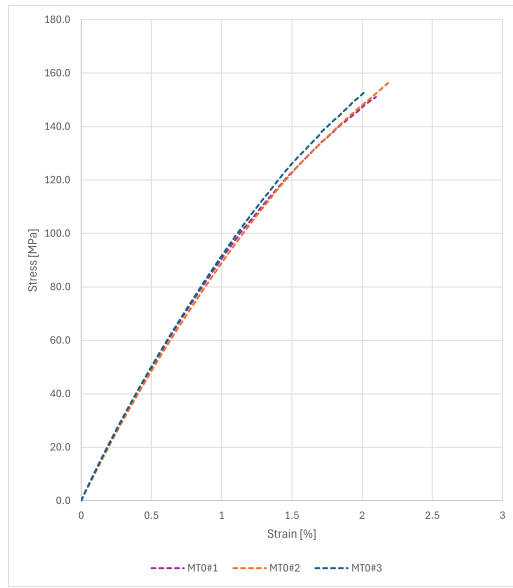
where, P_i is the load at each data point, and A is the section area of the analyzed specimen, which corresponds to the product of the measured width (w) and thickness (t).

The strain at each point (ϵ_{it}) is determined by:

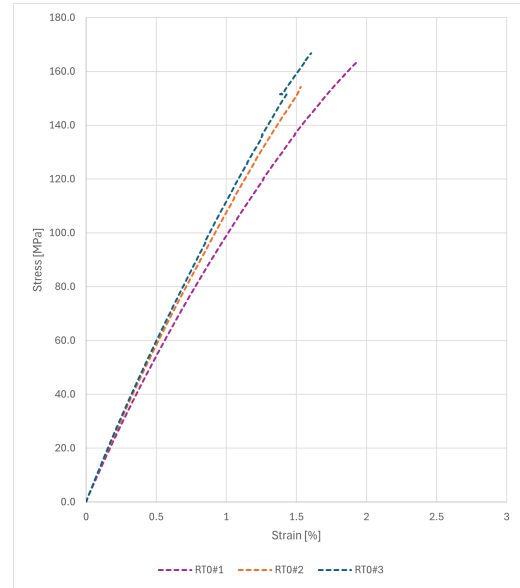
$$\epsilon_{it} = \frac{\delta_{it}}{L_g}, \quad (3.2)$$

where δ_i is the displacement measured with the extensometer and L_g the extensometer gauge length.

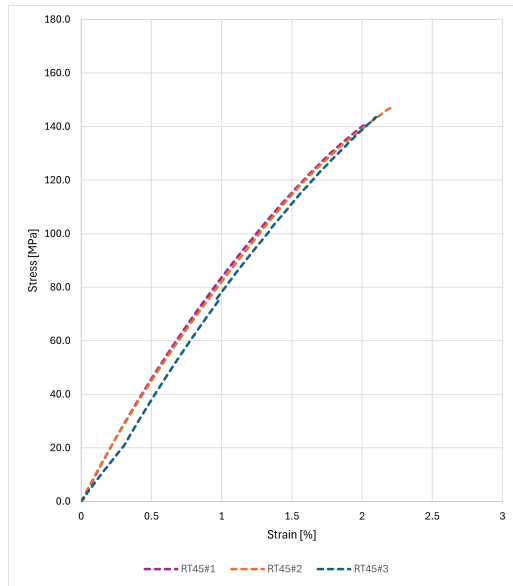
The stress-strain curves can then be obtained, as shown in Figure 3.11. From the observation of these curves, specimens *RT45#3* and *RT90#1* exhibit indications of slippage during the tensile tests. Consequently, the results obtained from these specimens are not considered representative of the material behavior and were therefore excluded from the calculation of the mechanical properties.



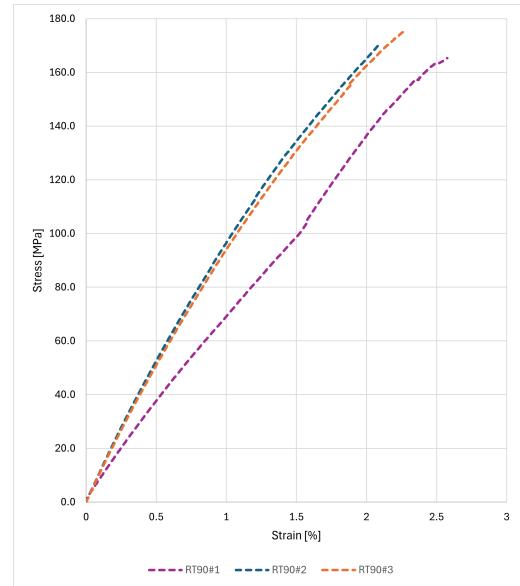
(a) "Mat" material



(b) "Rovimat" material in the 0° direction



(c) "Rovimat" material in the 45° direction



(d) "Rovimat" material in the 90° direction

Figure 3.11: Stress-strain curves considering the two examined materials and fiber directions.

To determine the elastic modulus, the ASTM standard recommends selecting two points on the stress-strain curve (ideally at 0.1% and 0.3%) and extracting the corresponding tensile stresses. Using Hooke's Law, the elastic modulus of the chord between these two points can then be calculated. This value is assumed to represent the elastic modulus of the material.

In addition, the ultimate tensile strength (σ_{ut}) and the ultimate tensile strain (ϵ_{ut}) were determined for each specimen.

To calculate the mechanical properties from the tensile test results, the average value ($\bar{x}$) represented in Equation 3.3, standard deviation (s_{n-1}) in Equation 3.4, and Coefficient of Variation (CV) in Equation 3.5, were calculated in accordance with the ASTM standard [51], as follows:

$$\bar{x} = \frac{1}{n} \sum_{i=1}^n x_i, \quad (3.3)$$

$$s_{n-1} = \sqrt{\frac{\sum_{i=1}^n x_i^2 - n\bar{x}^2}{n-1}}, \quad (3.4)$$

$$CV = 100 \times \frac{s_{n-1}}{\bar{x}}, \quad (3.5)$$

where n is the number of specimens and x_i is the measured or derived property.

Tables 3.5, 3.6 and 3.7 show the results for each property that was examined, and Figures 3.12, 3.13 and 3.14 represent the results in bar graphs.

Table 3.5: Results obtained for the ultimate tensile strength σ_{ut} .

	Ultimate Tensile Strength σ_{ut}		
	Average	Standard deviation	Coefficient of variation
	$\bar{x}$ [MPa]	s_{n-1} [MPa]	CV [%]
MT0	153.7	3.0	1.93
RT0	161.6	6.6	4.09
RT45	143.9	3.6	2.52
RT90	170.8	1.1	0.62

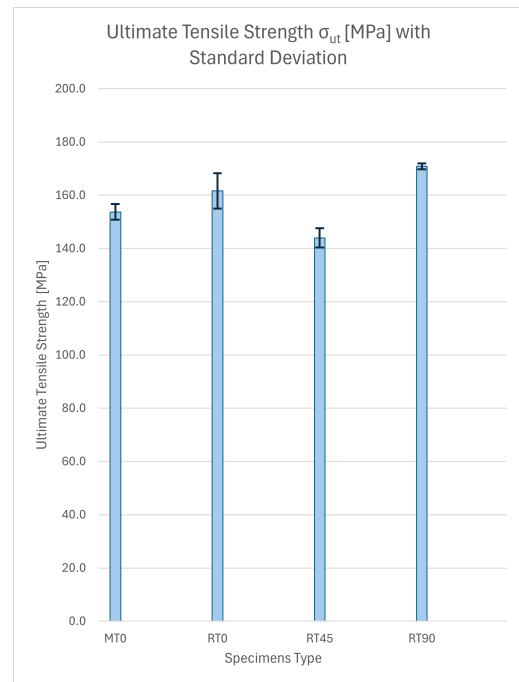


Figure 3.12: Bar graph with the ultimate tensile strength results.

Table 3.6: Results obtained for the ultimate tensile strain ϵ_{ut} .

	Ultimate Tensile Strain ϵ_{ut}		
	Average	Standard deviation	Coefficient of variation
	$\bar{x}$	S_{n-1}	CV
	[%]	[%]	[%]
MT0	2.11	0.093	4.39
RT0	1.69	0.220	12.99
RT45	2.12	0.106	5.01
RT90	2.18	0.128	5.89

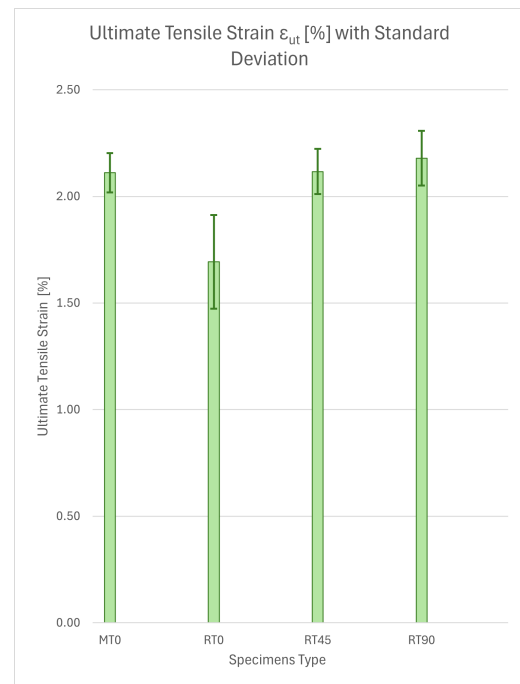


Figure 3.13: Bar graph with the ultimate tensile strain results.

Table 3.7: Results obtained for the of Young's modulus E .

	Young's Modulus E		
	Average	Standard deviation	Coefficient of variation
	$\bar{x}$	S_{n-1}	CV
	[MPa]	[MPa]	[%]
MT0	10082	136	1.35
RT0	11669	590	5.06
RT45	9340	247	2.65
RT90	10534	224	2.12

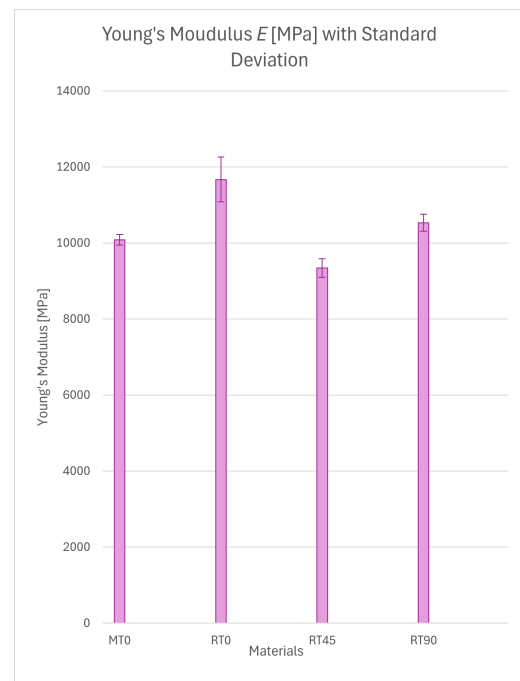


Figure 3.14: Bar graph with the Young's modulus results.

From these results one can understand that the "Mat" material (MT0) exhibits stable and repeatable tensile behavior, with relatively low CV for strength, strain, and Young's modulus.

The introduction of a RVM layer leads to an increase in Young's modulus compared to the "Mat" material laminate. Both principal material directions show higher stiffness values ($RT0$: 11 669 MPa, $RT90$: 10 534 MPa) relative to $MT0$ (10 082 MPa). Although the RVM layer is bidirectional, the difference observed between the 0° and 90° specimens suggests that laminate-level effects, such as crimps or processing-related variability, may have influenced the measured stiffness.

The "Rovimat" material results show a higher ultimate tensile strength compared to the "Mat" material laminate for both 0° and 90° orientations. $RT0$ and $RT90$ specimens show ultimate tensile strengths of 161.6 MPa and 170.8 MPa, respectively, compared to 153.7 MPa for $MT0$. While the measured strength in the 90° direction is higher than in the 0° direction, this difference can be interpreted once again as process-related variability.

Specimens tested at 45° of "Rovimat" material ($RT45$) exhibit the lowest tensile modulus and strength among the "Rovimat" material specimens. This behavior is expected, as the added directional fibers from the RVM layer are being solicited by shear forces.

The "Rovimat" material laminates exhibit higher variability in some measured properties, particularly strain and Young's modulus, compared to the "Mat" material laminate. This increased variability may indicate greater sensitivity to manufacturing parameters such as fiber alignment, impregnation quality, and fabric placement.

3.1.2.2 Three-Point Bending Tests

Regarding the three-point bending tests, apart from tests until failure, additional tests were made to observe the elastic response of the material. Therefore, some tests were performed until 5 mm and 8 mm deflections of the specimen. The results obtained would be useful later to compare to a supplementary analysis made using numerical tools to try to obtain the elastic modulus through inverse analysis. These numerical simulations are explained in Appendix A.

Similarly to the tensile tests, an initial procedure of measuring the thickness and width of the samples was conducted. The obtained averages of said measurements are shown in Table 3.8. More details about the measured values are shown in Appendix A, in Table A.2.

To have a better understanding about the variation of thickness of the three-point bending samples, a thickness dispersion graph was also made, as shown in Figure 3.15.

Table 3.8: Average width and thickness of three-point bending specimens.

Type	Spec.	Average width	Average thickness
		[mm]	[mm]
MB0	#1	44.2	3.9
	#2	44.1	3.8
	#3	44.1	4.0
	#4	44.1	4.0
	#5	44.1	4.0
	#6	44.0	4.0
	#7	44.2	3.8
	#8	44.1	4.0
RB0	#1	44.1	4.8
	#2	45.3	5.3
	#3	44.1	4.6
	#4	44.1	5.2
	#5	44.1	5.4
	#6	44.2	5.5
	#7	44.1	5.7
	#8	44.2	5.2

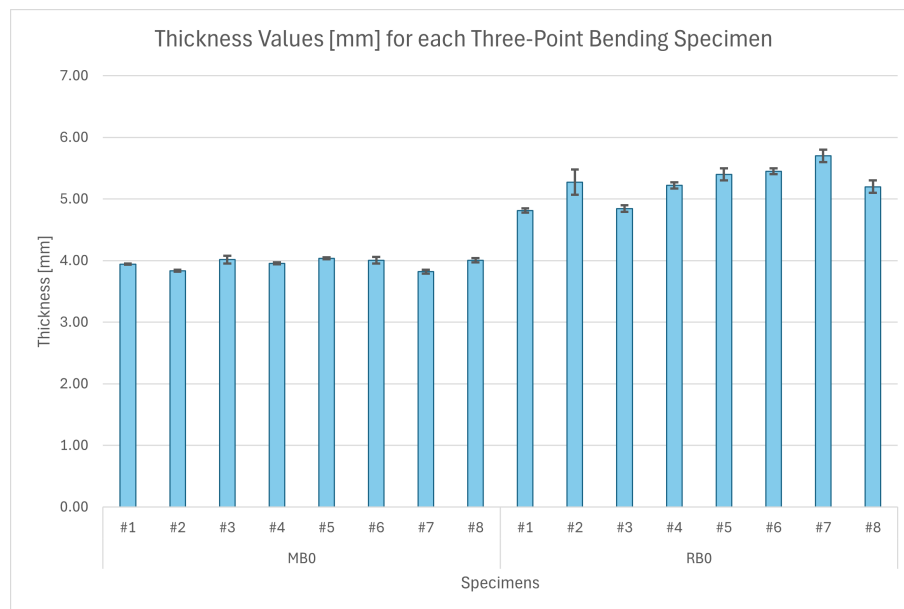


Figure 3.15: Thickness dispersion graph for the three-point bending samples.

To perform the three-point bending tests, the same testing machine was used. However, instead of hydraulic grips, it was equipped with cylindrical contact surface supports, one at the top as the loading nose and two at the bottom as the specimen supports. All the cylindrical surfaces had a

diameter of 15 mm, and the span between the centers of the bottom supports was 100 mm. Each specimen was positioned at mid-span using reference marks previously drawn on the samples.

As in the tensile tests, a displacement rate of 5 mm/min was employed. The data acquisition rate was set to 20 Hz, and the force-displacement data obtained from the tests were exported to Microsoft Excel files for subsequent processing.

Videos of the three-point bending tests were taken with a setup to allow posterior observation of the specimen's mechanical behavior in the three-point bending test. The setup built to perform and record these tests is shown in Figure 3.16.



Figure 3.16: Global view of the setup used for the three-point bending tests.

Only three specimens from each material were tested until failure. For the "Mat" material, these were specimens *MB0#2*, *MB0#3*, and *MB0#4*, while for the "Rovimat" material, specimens *RB0#1*, *RB0#2*, and *RB0#3* were tested until failure. Specimens *MB0#1*, *MB0#7*, and *MB0#8*, as well as *RB0#4*, *RB0#5*, and *RB0#6*, were tested until the loading nose reached a specimen deflection of 5 mm. Additionally, specimens *MB0#5* and *MB0#6* were tested up to a specimen deflection of 8 mm. Specimens *RB0#7* and *RB0#8* were not tested. Figure 3.17 shows the specimens after completion of all tests.

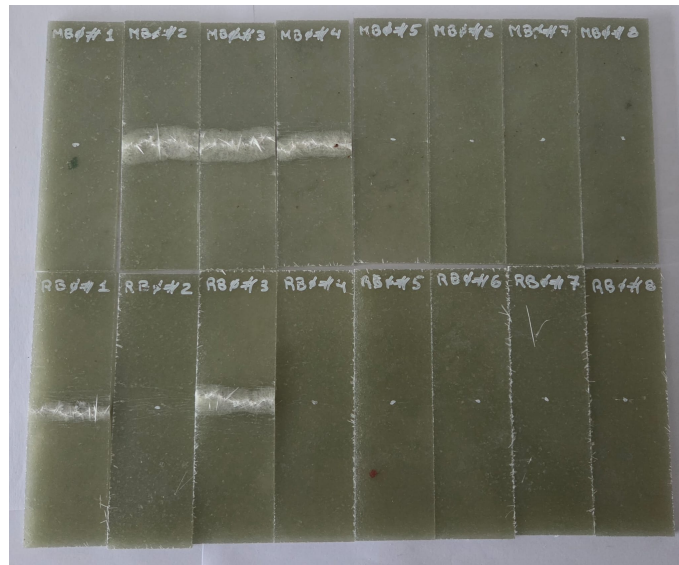


Figure 3.17: Three-point bending samples after testing.

The ASTM D 7264/D 7264M standard [52] also includes a failure categorization method described by Table 3.9.

Table 3.9: Failure mode coding system for three-point bending testing, which is composed of three characters: the first for the failure mode, the second for the failure area, and the third for the failure location.

First Character		Second Character		Third Character	
Failure Mode	Code	Failure Area	Code	Failure Location	Code
Tension	T	At loading nose	A	Top	T
Compression	C	Between loading noses	B	Bottom	B
Buckling	B	At support nose	S	Left	L
Interlaminar shear	S	Between load and support nose	L	Right	R
Multi-mode	M(xyz)	Unknown	U	Middle	M
Other	O			Various	V
				Unknown	U

The failure categorization was performed by observing Figure 3.18. Regarding the failure area and location, it is straightforward to observe that all specimens failed at the loading nose area, with the failure occurring at mid-span. This corresponds to the failure area code "A" and failure location code "M" for all specimens.

Drawing conclusions about the failure mode in this type of test is more challenging. Based on the observations, all specimens, except *RB0#2*, exhibited a similar failure mode. Since the loading nose for all specimens was positioned on the surface shown in Figure 3.18 (a), the failure appears

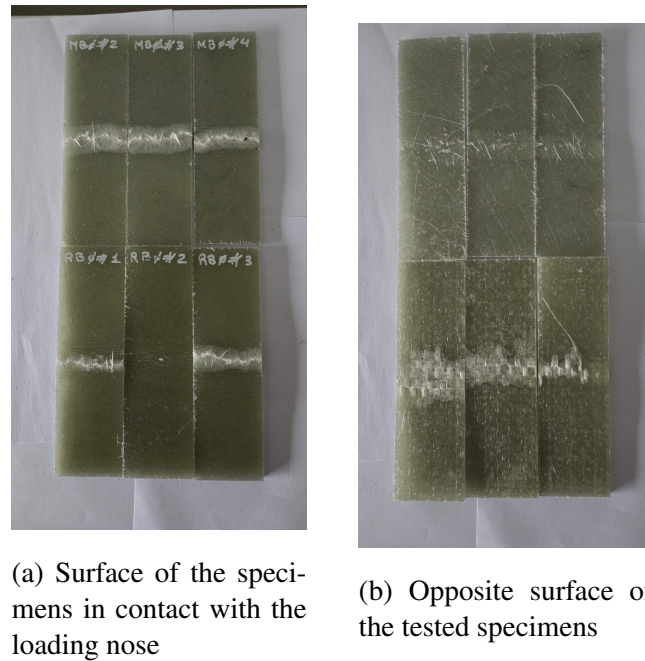


Figure 3.18: Composite laminate specimens after three-point bending tests performed until failure.

to have occurred under compression. Accordingly, the failure identification for these specimens is "CAM".

For the remaining specimen, the failure occurred in the outer layer of the laminate, indicating that this layer was subjected to tensile loading. The corresponding failure identification for this specimen is therefore "TAM".

As previously stated, the direct output of this test is the force-displacement curve. According to Daniel Cruz [53], the stiffness (compliance) of the testing machine, which is influenced by the tested materials and the load cell, must be accounted for. Therefore, the compliance of the testing system, K_s , was determined following the methodology outlined by Mujika [54] and the stress-strain curves were adapted according to it.

With the corrected curves of force-displacement and following the procedures defined in ASTM [52], the relevant mechanical properties can be determined. First, the flexural stress is calculated using the Equation 3.6 :

$$\sigma_{if} = \frac{3P_i L}{2bh^2}, \quad (3.6)$$

where σ_{if} is the flexural stress at a given instant [MPa], P_i is the applied force [N], L is the support span [mm], b is the sample width [mm] and h is the sample thickness [mm].

The flexural strain is obtained using a method different from that employed in tensile testing and is given by:

$$\varepsilon_{if} = \frac{6\delta_i h}{L^2}, \quad (3.7)$$

where ε_{if} is the flexural strain at the outer surface at a given instant [mm/mm], and δ_i is the mid-span deflection [mm].

These equations allow the development of the stress-strain curves for the three-point bending tests. Figure 3.19 presents the stress-strain curves for the three-point bending tests performed until failure and Figure 3.20 presents the tests conducted until the previously identified deflections.

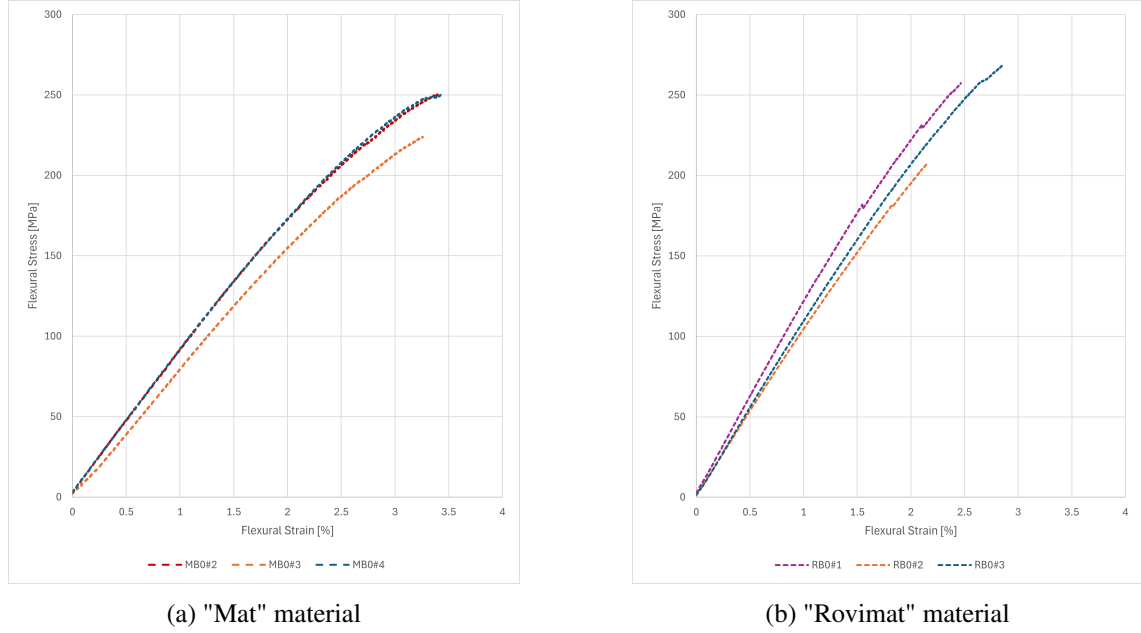


Figure 3.19: Stress-strain curves of three-point bending tests until failure.

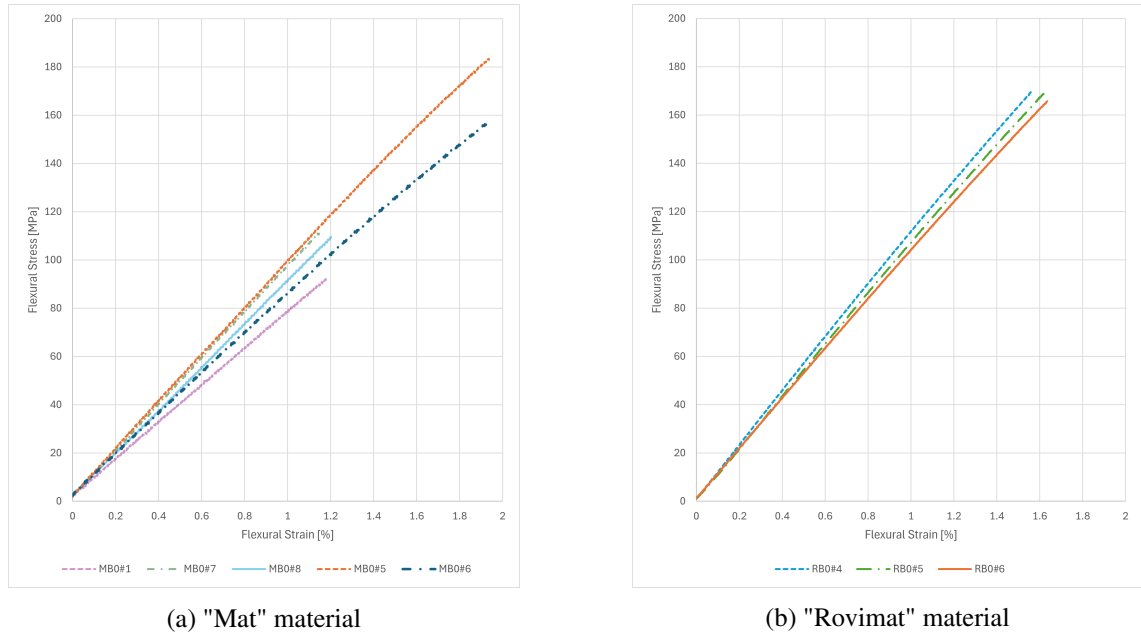


Figure 3.20: Stress-strain curves of three-point bending tests until 5 mm nose displacement for "Rovimat" and MB0#1, MB0#7 and MB0#8 specimens and until 8 mm nose displacement for specimens MB0#5 and MB0#6.

The flexural modulus of elasticity is determined using Hooke's law by selecting two points on the stress-strain curve and calculating the corresponding slope. As recommended by the standard, strain levels of 0.1% and 0.3% were used for this calculation.

However, there are situations in which the use of alternative strain points becomes necessary. In the case of specimen *MB0#3*, an initial toe region can be identified, which does not represent the intrinsic material behavior. According to ASTM [52], this toe region “is an artifact caused by take-up of slack and alignment, or seating of the specimen.” Consequently, this initial slack must be compensated. To prevent the results from being affected by this phenomenon, strain levels higher than those recommended by the standard were selected. In this specific case, the strain points chosen to calculate the flexural modulus of elasticity were 0.5% and 0.8%.

The ultimate flexural strength (σ_{uf}) and ultimate flexural strain (ϵ_{uf}) values of each specimen were also determined.

The averages, standard deviations, and CV of the three-point bending results are shown in Tables 3.10, 3.11 and 3.12, which are accompanied by the respective bar graphs in Figures 3.21, 3.22 and 3.23.

Table 3.10: Results summary of ultimate flexural strength σ_{uf} .

	Ultimate Flexural Strength σ_{uf}		
	Average	Standard deviation	Coefficient of variation
	$\bar{x}$	S_{n-1}	CV
	[MPa]	[MPa]	[%]
Mat	242.3	14.7	6.05
Rovimat	244.7	32.2	13.16

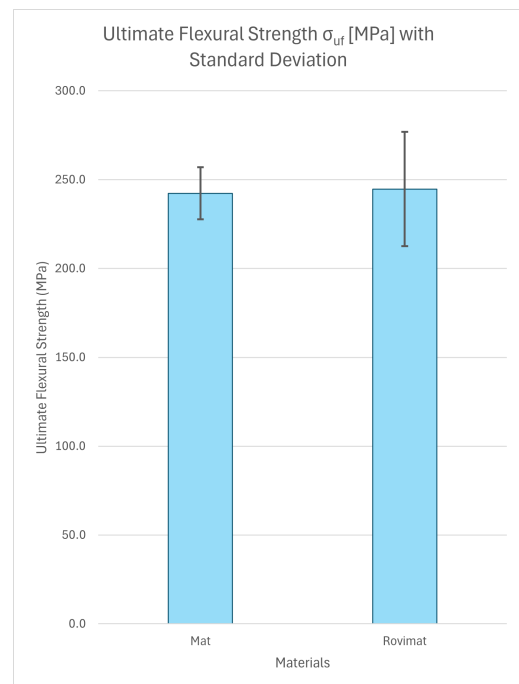


Figure 3.21: Bar graph with the ultimate flexural strength results.

Table 3.11: Results summary of ultimate flexural strain ϵ_{uf} .

	Ultimate Flexural Strain ϵ_{uf}		
	Average	Standard deviation	Coefficient of variation
	$\bar{x}$	S_{n-1}	CV
	[%]	[%]	[%]
Mat	3.43	0.04	1.16
Rovimat	2.51	0.35	13.99

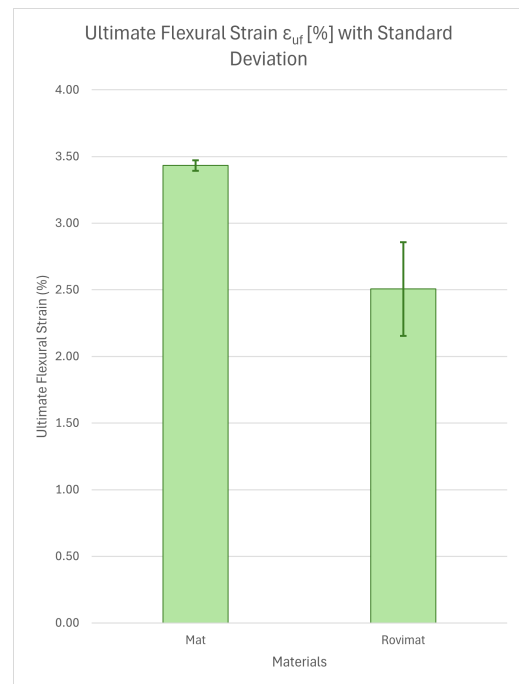


Figure 3.22: Bar graph with the ultimate flexural strain results.

Table 3.12: Results summary of flexural modulus of elasticity E_f .

	Flexural Modulus of Elasticity E_f		
	Average	Standard deviation	Coefficient of variation
	$\bar{x}$	S_{n-1}	CV
	[MPa]	[MPa]	[%]
Mat	8 776	800.0	9.12
Rovimat	11 018	672.7	6.11

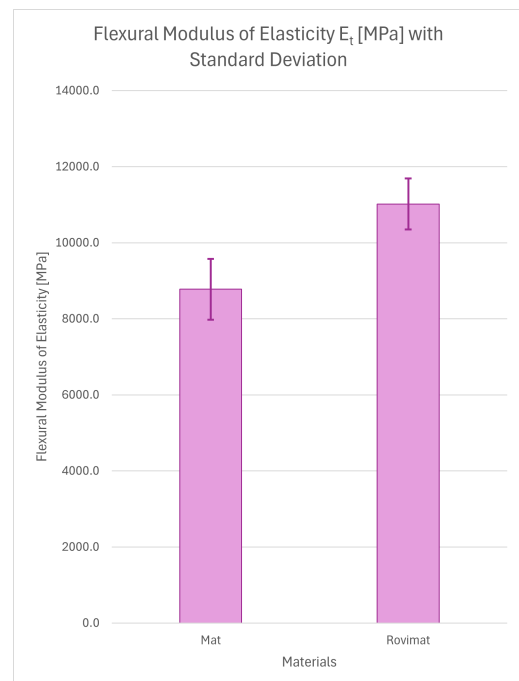


Figure 3.23: Bar graph with the flexural modulus of elasticity results.

The results of the three-point bending tests indicate that the addition of the RVM layer reinforcement did not lead to a significant increase in ultimate flexural strength. The average flexural

strength of the "Mat" laminate was measured at approximately 242 MPa, while the "Rovimat" laminate exhibited a similar average value of approximately 245 MPa. The *RB0#2* test results have greatly influenced the average value of the ultimate flexural strength, as the other two "Rovimat" material specimens present much higher flexural strength values. This variation in experimental strength may be associated with the fact that specimen *RB0#2* exhibited a different failure mode compared to the other specimens.

The ultimate flexural strain between the two laminates shows a bigger difference. The "Mat" material reached a higher average strain at failure, approximately 3.43%, compared to 2.51% for the "Rovimat" material. This reduction in strain capacity indicates that the addition of RVM layer decreases the laminate's ability to undergo deformation before failure.

The inclusion of the RVM layer has a significant effect on the flexural stiffness of the laminate. The average flexural modulus increases from 8 776 MPa for the "Mat" material to about 11 018 MPa for the "Rovimat" material.

The variability of the flexural test results differs notably between the two laminates and across the measured properties, similarly to the results seen in tensile testing. For the ultimate flexural strength, the "Rovimat" laminate exhibits a substantially higher CV than the "Mat" laminate. A similar trend is observed for the ultimate flexural strain, where the "Rovimat" laminate again shows significantly higher variability. In contrast, the "Mat" material demonstrates more consistent behavior in both strength and strain.

For the flexural modulus of elasticity, however, the variability is lower for the "Rovimat" laminate. This could mean that the stiffness is less affected by local defects and manufacturing inconsistencies. The higher number of specimens taken under consideration could also be a factor in this occurrence.

3.1.3 Properties Definition according to ISO 12215-5

ISO 12215-5 provides tables and equations that allow the definition of the mechanical properties required to perform the structural calculations. Part of this data is supplied directly by industry sources, while the remaining values are calculated using the Halpin-Tsai formulations [20].

Before describing the procedure used to determine the material properties in accordance with ISO 12215-5, it is important to note that the properties of RVM layers cannot be obtained directly. Instead, they must be derived from thickness-weighted averages of the properties of the corresponding CSM and WR layers. The global mechanical properties of the composite laminate are also determined using thickness-weighted averages of its constituent layers. This requirement is therefore taken into account in the methodology here demonstrated. It begins by introducing several key definitions. The standard presents equations based on the law of mixtures for composite materials, as shown by:

$$\psi = \frac{\phi \rho_{df}}{\phi \rho_{df} + (1 - \phi) \rho_m}, \quad (3.8)$$

$$\phi = \frac{\psi}{\psi + (1 - \psi) \frac{\rho_{df}}{\rho_m}}, \quad (3.9)$$

$$\frac{t}{w} = \frac{1}{\phi \rho_{df}}, \quad (3.10)$$

$$\rho_c = \rho_{df} \phi + \rho_m (1 - \phi), \quad (3.11)$$

$$\rho_c = \frac{\rho_{df} \rho_m}{\rho_{df} + \psi (\rho_m - \rho_{df})}, \quad (3.12)$$

$$t = \frac{w}{\rho_{df} \phi} \left(\frac{\rho_{df}}{\psi} + \rho_m - \rho_{df} \right), \quad (3.13)$$

where, t is the thickness of the laminate [mm], w is the dry fiber reinforcement mass per square meter [kg/m²], ψ is the fiber content by mass in the laminate [1], ϕ is the fiber content by volume in the laminate [1] and ρ_{df} , ρ_m , ρ_c are the density of the dry fiber, matrix and composite, respectively [kg/m³].

Furthermore, properties of the reinforcement fibers and matrices are also provided, as seen in Table [3.13].

Table 3.13: Properties of E-Glass dry fiber and Polyester matrix; adapted from Table C.3. of Annex C (ISO 12215-5).

Properties	Unit	E-Glass fiber	Polyester
Density (ρ_{df} or ρ_m)	[kg/m ³]	2560	1200
Elastic modulus (E_{df} or E_m)	[MPa]	73 000	3 300
Shear modulus (G_{df} or G_m)	[MPa]	30 000	1 222
Poisson's ratio (ν_{df} or ν_m)	[1]	0.22	0.32

Moreover, equations to calculate the elastic modulus, the shear modulus and the Poisson's ratio of composite layers are defined, derived from Halpin-Tsai formulas [20], and taken from Table C.5 of Annex C of ISO 12215:5. It is important to note that, even though composite layers with unidirectional fiber arrangements are not being considered for this study, it is necessary to calculate their properties to retrieve the mechanical properties of CSM and WR. The ISO 12215-5 defines that:

$$E_{UD1} = 0,975 \times [E_{f1} \times \phi + E_m \times (1 - \phi)], \quad (3.14)$$

$$E_{UD2} = E_m \times \frac{1 + \zeta \eta_E \phi}{1 - \eta_E \phi}, \quad \text{with } \zeta = 1 \text{ and } \eta_E = \frac{\frac{E_{f2}}{E_m} - 1}{\frac{E_{f2}}{E_m} + \zeta}, \quad (3.15)$$

$$G_{UD12} = G_m \times \frac{1 + \zeta \eta_G \phi}{1 - \eta_G \phi}, \quad \text{with } \zeta = 1 \text{ and } \eta_G = \frac{\frac{G_{f2}}{G_m} - 1}{\frac{G_{f2}}{G_m} + \zeta}, \quad (3.16)$$

$$\nu_{UD12} = \nu_f \times \phi + \nu_m \times (1 - \phi), \quad (3.17)$$

$$\nu_{UD21} = \nu_{UD12} \times \frac{E_{UD2}}{E_{UD1}}, \quad (3.18)$$

where E_{UD1} and E_{UD2} are the elastic moduli for unidirectional fiber composites for directions 1 (fiber direction) and 2 (transverse to the fiber), respectively, and G_{UD12} is the shear modulus for unidirectional fiber composites in the 1-2 plane (corresponding to shear at 45° to directions 1 and 2). The terms ν_{UD12} and ν_{UD21} correspond to the Poisson's ratios for unidirectional fiber composites associated with the 1-2 and 2-1 directions, respectively.

Using the mechanical properties of the unidirectional fiber composite, it is possible to determine the corresponding properties of CSM and WR laminates, as previously described and as presented in Table 3.14.

Table 3.14: Equations to calculate the elastic properties of CSM and WR layers.

Young's modulus	Shear modulus	Poisson's ratio
$E_{\text{CSM}} = \frac{3}{8}E_{\text{UD1}} + \frac{5}{8}E_{\text{UD2}}$	$G_{\text{CSM}} = \frac{1}{8}E_{\text{UD1}} + \frac{1}{4}E_{\text{UD2}}$	$\nu_{\text{CSM}} = \frac{E_{\text{CSM}}}{G_{\text{CSM}}} - 1$
$E_{\text{WR}} = 0.5(E_{\text{UD1}} + E_{\text{UD2}})$	$G_{\text{WR}} = G_{\text{UD12}}$	$\nu_{\text{WR}} = \nu_{\text{UD12}} \frac{E_{\text{UD2}}}{E_{\text{WR}}}$

Regarding the ultimate strain values, they do not require calculation, as the standard provide their values. The retrieved data that is required for this work is depicted in Table 3.15.

Table 3.15: Tensile (ϵ_{ut}) and flexural (ϵ_{uf}) ultimate strains for CSM and WR layers.

Fiber Layer	Strain	Value [%]
CSM	ϵ_{ut}	1.35
	ϵ_{uf}	1.88
WR	ϵ_{ut}	1.55
	ϵ_{uf}	1.84

The ultimate strains allow the calculation of ultimate stress, following the known Hooke's Law:

$$\sigma_u = E \times \epsilon_u, \quad (3.19)$$

where E and ϵ_u are associated properties. For instance, to calculate the ultimate tensile strength of a CSM layer, the values to multiply are E_{CSM} and ϵ_{ut} . Note that for the shear strength (τ_{UST}), the shear modulus (G) is used.

Determining the fiber content values, either in terms of mass fraction or volume fraction, is the critical step to compute the mechanical properties needed for the scantling calculations. From the equations shown in this section, it can be understood that these ratios govern the mechanical properties of the composite layers and, consequently, the laminates. For an accurate representation of the properties of the materials manufactured by the Riamar company, the thicknesses of representative laminate plates will be used to calculate these values.

The global laminate thicknesses were obtained from the average of the measured specimen thicknesses for each material. The resulting values were defined as $t_{Mat} = 3.9$ mm for the "Mat" material and $t_{Rovimat} = 5.1$ mm for the "Rovimat" material.

Using Equations 3.9, 3.10, and 3.11, the updated fiber content values were calculated and are presented in Table 3.16.

Table 3.16: Fiber contents, thickness per dry fiber mass per square meter, and composite density obtained with the laminate plates thicknesses manufactured by Riamar company.

Composite layer	Fiber content in volume	Fiber content in mass	Thickness per dry fiber mass per square meter ratio	Composite density
	ϕ [%]	ψ [%]	t/w [mm/(kg/m ²)]	ρ_c [kg/m ³]
CSM	0.225	0.382	1.740	1506
WR	0.466	0.651	0.838	1834

Following the described methodology in this subsection, the elastic- and strength-related properties were determined for the CSM and WR layers, shown respectively in Tables 3.17 and 3.18, respectively.

Table 3.17: Elastic properties of CSM and WR layers.

Composite layer	E [MPa]	G [MPa]	ν [1]
CSM	10 071	3 566	0.41
WR	21 756	3 056	0.10

Table 3.18: Tensile (σ_{ut}) and flexural (σ_{uf}) strength properties of CSM and WR layers.

Composite layer	σ_{ut} [MPa]	σ_{uf} [MPa]
CSM	136	164
WR	305	400

The mechanical properties of the RVM layer will be calculated with weighted averages from the properties of CSM and WR, as previously indicated, having the thickness as reference for each property weight. In that regard, it is necessary to define the thickness for the three types of composite layers, according to the fiber contents values obtained with the thicknesses of the manufactured laminates (Table 3.19).

Table 3.19: Relation between fiber weight, thickness per fiber weight, and laminate thickness for CSM 0.45 kg/m², WR 0.5 kg/m² and RVM 0.95 kg/m².

Composite layer	w [kg/m ²]	t/w [mm/(kg/m ²)]	t [mm]
CSM	0.45	1.74	0.8
WR	0.50	0.84	0.4
RVM	0.95	1.26	1.2

Having the values of thickness for each composite layer, it is possible to calculate the elastic properties and the fiber content in volume (ϕ) of the RVM layer. Equation 3.20 shows the process to determine RVM's elastic modulus (E_{RVM}), as an example:

$$E_{RVM} = \frac{(t_{CSM} \times E_{CSM} + t_{WR} \times E_{WR})}{t_{RVM}} = \frac{0.8 \times 10071 + 0.4 \times 21756}{1.2} = 14149 \text{ MPa}. \quad (3.20)$$

Regarding the strain-related properties, they are defined by choosing the minimum strain for each type of behavior, as shown in Table 3.21.

Table 3.20: RVM layer elastic properties and fiber content in volume calculated based on the thickness of CSM and WR layers.

Composite layer	w [kg/m ²]	t [mm]	E [MPa]	G [MPa]	ν [1]	ϕ [%]
CSM	0.45	0.8	10 071	3 566	0.40	0.225
WR	0.50	0.4	21 756	3 056	0.11	0.466
RVM	0.95	1.2	14 149	3 388	0.30	0.309

Table 3.21: Ultimate strain values for RVM layer based on CSM and WR layers.

Composite layer	ϵ_{ut} [%]	ϵ_{uf} [%]
CSM	1.35	1.88
WR	1.55	1.84
RVM	1.35	1.84

The strength properties are obtained following Equation 3.19, and they are depicted in Table 3.22.

Table 3.22: Calculated strength properties of CSM, WR, and RVM layers.

Composite layer	σ_{ut} [MPa]	σ_{uf} [MPa]
CSM	136	164
WR	337	400
RVM	191	260

After calculating the properties for each individual layer, it is possible to obtain the necessary mechanical properties of the laminate. The elastic and strength-related properties of the laminates are determined through thickness-weighted averages of the individual laminate layers, while the strain-related properties are defined as the minimum strain value among the layers, similarly to the way of defining the RVM layer. Tables 3.23 and 3.24 present the properties of the individual layers and global laminates of the "Rovimat" material and "Mat" material, respectively, calculated with the described analytical method.

Table 3.23: Individual layers and laminate properties of the "Rovimat" material, calculated with the analytical method described by ISO 12215-5.

Composite layers	w [kg/m ²]	t [mm]	E [MPa]	G [MPa]	ν [1]	σ_{ut} [MPa]	σ_{uf} [MPa]
Layer 1 - RVM	0.95	1.2	14 149	3 380	0.30	191	260
Layer 2 - CSM	0.45	0.8	10 071	3 566	0.41	136	189
Layer 3 - CSM	0.60	1.0	10 071	3 566	0.41	136	189
Layer 4 - CSM	0.60	1.0	10 071	3 566	0.41	136	189
Layer 5 - CSM	0.60	1.0	10 071	3 566	0.41	136	189
"Rovimat" material	3.20	5.1	11 030	3 522	0.38	149	206

Table 3.24: Individual layers and laminate properties of the "Mat" material, calculated with the analytical method described by ISO 12215-5.

Composite layers	w [kg/m ²]	t [mm]	E [MPa]	G [MPa]	ν [1]	σ_{ut} [MPa]	σ_{uf} [MPa]
Layer 1 - CSM	0.45	0.8	10 071	3 566	0.41	136	189
Layer 2 - CSM	0.60	1.0	10 071	3 566	0.41	136	189
Layer 3 - CSM	0.60	1.0	10 071	3 566	0.41	136	189
Layer 4 - CSM	0.60	1.0	10 071	3 566	0.41	136	189
"Mat" material	2.25	3.9	10 071	3 566	0.41	136	189

3.1.4 Comparison between Material Properties

Tables 3.25 to 3.28 show the comparison between the tensile and flexural mechanical properties obtained from the utilized methods. The variations presented for the experimental results correspond to the standard deviation values, as they show good agreement with the maximum and minimum values obtained for the respective properties.

Table 3.25: Comparison of tensile properties obtained from different characterization methods for "Mat" material.

Characterization method for "Mat" material tensile properties	t [mm]	E [MPa]	ϵ_{ut} [%]	σ_{ut} [MPa]
ISO 12215-5	3.9	10 071	1.35	136.0
Tensile tests	3.7	10 081 $\pm$ 136	2.11 $\pm$ 0.09	153.7 $\pm$ 3.0

Table 3.26: Comparison of flexural properties obtained from different characterization methods for "Mat" material.

Characterization method for "Mat" material flexural properties	t [mm]	E_f [MPa]	ϵ_{uf} [%]	σ_{uf} [MPa]
ISO 12215-5	3.9	–	1.88	189.0
Three-point bending tests	4.0	8 776 $\pm$ 800	3.43 $\pm$ 0.04	242.3 $\pm$ 16.9

Table 3.27: Comparison of tensile properties obtained from different characterization methods and fiber orientations for "Rovimat" material.

Characterization method for "Rovimat" material tensile properties	t [mm]	E [MPa]	ϵ_{ut} [%]	σ_{ut} [MPa]
ISO 12215-5	5.1	11 030	1.35	148.9
Tensile tests – 0° fiber angle	4.4	11 669 $\pm$ 590	1.69 $\pm$ 0.22	161.6 $\pm$ 6.6
Tensile tests – 45° fiber angle	5.2	9 340 $\pm$ 247	2.12 $\pm$ 0.11	143.9 $\pm$ 3.6
Tensile tests – 90° fiber angle	5.0	10 534 $\pm$ 224	2.18 $\pm$ 0.13	170.8 $\pm$ 1.1

Table 3.28: Comparison of flexural properties obtained from different characterization methods for "Rovimat" material.

Characterization method for "Rovimat" material flexural properties	t [mm]	E_f [MPa]	ϵ_{uf} [%]	σ_{uf} [MPa]
ISO 12215-5	5.1	–	1.84	205.7
Three-point bending tests	5.2	11 018 $\pm$ 673	2.51 $\pm$ 0.35	244.7 $\pm$ 32.2

Comparing the analytical and experimental results, the predicted elastic modulus calculated by the method described in ISO 12215-5 shows very good agreement with the values obtained from

tensile testing. This indicates that the analytical framework of the standard is capable of accurately predicting stiffness when realistic fiber contents are used.

The analytically calculated strength-related properties are close to the experimental values, although they are slightly lower. A larger discrepancy can be observed for the ultimate strain values. This difference reflects the conservative nature of the values prescribed by the standard, which are intentionally defined to ensure safety and robustness across a wide range of manufacturing conditions.

The thickness values presented for the analytical calculations correspond to the average thickness considering all manufactured specimens of each material, as previously discussed, and they should be seen as the nominal thickness of the materials. Concerning the thickness values reported for the experimental tests, they represent the average thickness of only the specimens used in the respective test.

Variations in thickness within specimens of the same laminate should not be disregarded, as they may significantly influence the calculated mechanical properties. These variations can be attributed to the manufacturing process, namely manual hand lay-up. Since this process is performed manually, it may introduce irregularities that affect laminate thickness. For instance, uneven resin distribution can result in resin-rich areas, locally increasing the laminate thickness. Such effects could be mitigated through the use of vacuum-assisted processes to remove excess resin. However, this technique was not employed in the manufacture of the laminates considered in this work.

For the application of these materials in the Evolution 5.0 vessel, the properties presented in Tables 3.23 and 3.24 will be adopted. The experimental results were particularly valuable for validating the analytically obtained values; however, certain aspects limit their direct use in this case.

As discussed in the following section, the ISO 12215-5 standard significantly penalizes material properties associated with high standard deviation values, such as the σ_{uf} obtained for the "Rovimat" material. Moreover, the experimental procedure conducted does not allow for the individual characterization of each composite layer, which is required for the application of the analytical Enhanced method described subsequently.

3.2 Application Procedure of ISO 12215-5 Standard

ISO 12215-5 describes different procedures to perform the structural calculations necessary to ensure the boat is safe and can be fully certified. This standard describes a series of steps that allows the scantling determination.

3.2.1 Geometric and Design Parameters

To perform the scantling calculations, the first step is to determine the geometric and design parameters of the vessel. All dimensions shall be measured in accordance with ISO 8666, with the craft in the fully loaded condition, being m_{LDC} the mass at the fully loaded condition.

The vessel is divided into several areas: bottom, sides, decks, and superstructures. All regions located below the waterline are classified as bottom areas. The portions of the hull above the waterline correspond to the side area. Deck areas include the cockpit sole, bench tops, seating surfaces, and any other open deck regions exposed to environmental loads and foot traffic. Finally, all parts located above the deck level are classified as superstructure areas. In this work, only the bottom area is considered, as the object of analysis is the vessels' bottom of the hull.

These areas are composed of panels and stiffeners, which must be characterized in terms of geometry, material, and location within the vessel. Focusing on panels, they are geometrically defined by their shorter dimension b , and their longer dimension l , shown in Figure 3.24. In most cases, panels are rectangular, making this determination straightforward. However, ISO 12215-5 also accounts for non-rectangular geometries, transforming them into equivalent rectangular surfaces.

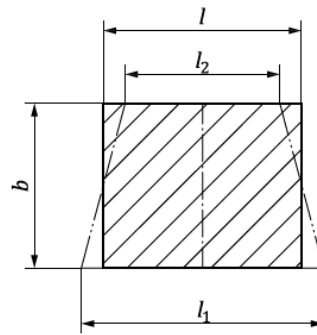


Figure 3.24: Panel average dimensions l and b ; adapted from ISO 12215-5 [20].

Furthermore, the longitudinal position x of each panel must be defined, as it directly influences the corresponding design pressure. This position is measured from the aft end of flotation, defined as the point where the waterline intersects the stern, to the center of the panel. Curvatures, in both l and b directions, also influence the scantling calculations as they increase the panel's capability of sustaining loads. They are characterized by the camber of the panel c , which is measured as shown in Figure 3.25, and are utilized to calculate the curvature correction factor k_c , using Table A.3 of ISO 12215-5 [20].

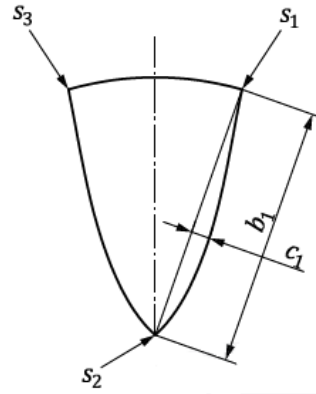


Figure 3.25: Camber c_1 measurement in panel with a shorter dimension of b_1 ; s_1 , s_2 and s_3 represent natural stiffeners that limit the panels; adapted from ISO 12215-5 [20].

There are other important values that must be defined before performing the scantling calculations:

- Waterline length (L_{WL});
- Chine beam (B_c) is the width of the vessel, from chine to chine, measured at $0.4L_{WL}$ from the aft end of the vessel;
- Deadrise angle (β) is the angle between the bottom of the boat's hull, the chine used as reference as depicted in Figure 3.26, and a horizontal plane, also measured at $0.4L_{WL}$;
- Maximum expected speed at maximum load condition (V);
- Maximum depth of the canoe body (T_c) also known as the boat draft, which is the vertical distance between the lowest point of the boat and the waterline.

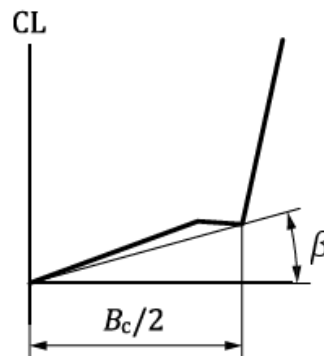


Figure 3.26: Chine beam B_c and deadrise angle β measurements; adapted from ISO 12215-5 [20].

Finally, the aspect ratio factors for bending moment k_2 and for shear force k_{SH} must also be calculated. These factors adjust the required rigidity and strength of the panels, depending on the l and b values of the panel. For instance, two panels with the same area, thickness, and material but

different lengths and widths, do not experience the same maximum stress and deflection. These factors are determined using Table A.2 of ISO 12215-5 [20].

3.2.2 Pressure Adjusting Factors and Design Pressures

Chapters 8 and 9 of ISO 12215-5 are responsible for the support regarding the calculation of the design pressures. There are a few factors that influence the design pressures that are considered by the ISO standard.

Design category factor k_{DC}

The design category factor is associated with the environmental conditions for which a vessel is designed to operate. It accounts for pressure variations resulting from wave slamming expected in the corresponding sea states for each category [20]. Vessels intended for calmer conditions are assigned a lower design category factor, as their structural components do not need to be as heavily reinforced as those designed for more severe conditions. Table 3.29 presents the design category factors and the corresponding average wave heights for each operating environment.

Table 3.29: Design categories and associated wave conditions [46].

Design category	Definition	k_{DC} value	Significant wave height
A	Ocean	1.0	Above 4 meters
B	Offshore	0.8	Up to 4 meters
C	Inshore	0.6	Up to 2 meters
D	Sheltered waters	0.4	Up to 0.3 or 0.5 meters

Dynamic load factor k_{DYN}

The vertical acceleration that the boat sustains when crashing into a wave at high speed or when descending from a wave's crest into its trough is named the "dynamic load factor" in the ISO 12215-5 standard. This component, which is expressed in g 's, is an important variable when determining the hull bottom design pressures of planing hull motor vessels. The k_{DYN} value to be taken is the lesser value of k_{DYN1} ,

$$k_{DYN1} = 0.32 \left(\frac{L_{WL}}{10B_C} + 0.084 \right) (50 - \beta_{0.4}) \frac{V^2 B_C^2}{m_{LDC}}, \quad (3.21)$$

and k_{DYN2} ,

$$k_{\text{DYN2}} = \frac{0.5 V}{m_{\text{LDC}}^{0.17}}, \quad (3.22)$$

not to be taken > 6 nor < 3 , where L_{WL} is the waterline length [m], B_C is the chine beam [m], $\beta_{0.4}$ is the deadrise angle measured at $0.4 L_{\text{WL}}$ from the aft end of the vessel [°], V is the maximum expected speed of the vessel [knots] and m_{LDC} is the mass in maximum load condition [kg].

Longitudinal pressure distribution factor k_L

The longitudinal pressure distribution factor considers the variation of pressure loads according to the lengthwise position on the craft. This factor is determined by:

$$k_L = (1.667 - 0.222 k_{\text{DYN}}) \frac{x}{L_{\text{WL}}} + 0.133 k_{\text{DYN}}, \quad (3.23)$$

not to be taken > 1 .

Area pressure reduction factor k_{AR}

This coefficient is used to adjust the design pressures based on the size of the structural component. According to ISO 12215-5, a panel's local pressure decreases with increasing area because it experiences an average of high slamming loads on small sections and reduced sea loads on greater areas. This means that small panels are more likely to experience "peak" slamming pressure uniformly across the surface, resulting in a k_{AR} closer to 1, and large panels have the ability to distribute the "peak" loads, leading to a smaller k_{AR} . The equation to calculate this factor is:

$$k_{AR} = \frac{k_R \times m_{\text{DC}}^{0.15}}{A_D^{0.3}}, \quad (3.24)$$

where, A_D is the design area (m^2) of the component, and k_R is 1 for the bottom, side and deck areas of planing motorcraft in planing mode, or,

$$k_R = 1.5 - 3 \times 10^{-4} b, \quad (3.25)$$

for bottom, side and deck plating of sailing craft, displacement motor craft and planing motor craft in displacement mode.

ISO 12215-5 contains other pressure adjusting factors like superstructures and deckhouse pressure reduction factor k_{sup} and pressure correcting factor for slamming of light and stable sailing craft k_{SLs} , but will not be explained here, as the studied vessels are not sailing boats.

Regarding the design pressures, which are described in kN/m^2 , there are two different calculation methods for the design pressures applied to the bottom of the hull for motor craft: the design pressure for vessels in displacement mode P_{BMD} and for vessels in planning mode P_{BMP} .

For the plating of motorcraft bottom in displacement mode, ISO 12215-5 imposes that:

$$P_{BMD} = \max(P_{BMD, \text{BASE}} \times k_{AR} \times k_{DC} \times k_L; P_{BMD, \text{MIN}}), \quad (3.26)$$

where $P_{BMD, \text{BASE}}$ is:

$$P_{BMD, \text{BASE}} = 2.4 \times m_{LDC}^{0.33} + 20, \quad (3.27)$$

and $P_{BMD, \text{MIN}}$, which is the minimum value to be taken for this design pressure is:

$$P_{BMD, \text{MIN}} = \max[(0.45 \times m_{LDC}^{0.33} + 0.9 \times L_{WL}) \times k_{DC} \times k_L; 10T_c; 7], \quad (3.28)$$

which implies that the minimum design pressure for this area in this condition is 7 kN/m^2 .

For the plating of motorcraft bottom in planning mode, ISO 12215-5 states that:

$$P_{BMP} = \max(P_{BMP, \text{BASE}} \times k_{AR} \times k_L; P_{BMD, \text{MIN}}), \quad (3.29)$$

where $P_{BMP, \text{BASE}}$ is:

$$P_{BMP, \text{BASE}} = \frac{0.1 \times m_{LDC}}{L_{WL} \times B_C} (1 + k_{DC}^{0.5} \times k_{DYN}), \quad (3.30)$$

and $P_{BMD, \text{MIN}}$ is the same as the one used for the calculation of P_{BMD} .

3.2.3 Design Stresses

As explained in Section 3.1, the material properties used for the scantling calculations can be obtained by monitoring fiber content from manufactured samples and calculating the properties with the analytical methods, or by just assuming the fiber contents from Table C.7 of ISO 12215-5, and applying the analytical methods, or by applying the material properties obtained in experimental tests. For the latter, the ISO 12215-5 standard advises that the strength-related properties shall be corrected either to 90% of the mean value or the mean value minus two standard deviations, whichever is less. Regarding the elastic modulus, no correction is needed, and the mean value can be used.

For the design stresses definition, there is also factors that aim to regulate the conservativeness of the scantling calculations, considering the method of acquiring the mechanical properties and the method of scantling assessment.

The boatbuilding quality factor k_{BB} is defined by the choice of approach taken in the acquirement of the mechanical properties. This factor is only for fiber-reinforced plastic materials and reflects the quality of the "as-built" material obtained by the boatbuilder. Table 3.30 shows the values of k_{BB} for each described method of mechanical property retrieval.

Table 3.30: Builder quality classification and corresponding values of the factor k_{BB} ; adapted from Table 15 of ISO 12215-5 [20].

Build quality	Builder characteristics	Value of k_{BB}	
		Hand laid	Infused, prepreg or similar
TESTED	Mechanical properties determined on finished laminates by testing.	1.0 (tested value)	1.0 (tested value)
HIGH	Fiber mass content monitored using representative layups (for example, thickness control).	0.95 of calculated / table value	1.0 of calculated / table value
LOW	No direct measurement of fiber mass content. Material properties taken from reference tables according to the manufacturing process.	0.75 of calculated / table value	0.80 of calculated / table value

The goal of the assessment method factor k_{AM} is to balance the outcomes of the different assessment techniques so that the more straightforward techniques yield more conservative conclusions than the more scientifically advanced ones. Table 3.31 shows the value of k_{AM} depending on the scantling method utilized.

Table 3.31: Determination of the assessment method factor k_{AM} , representing Table 16 of ISO 12215-5 [20].

Assessment method	Value of k_{AM}	
	FRP and wood	Metal
Method 1: Simplified method	0.9	1.0
Method 2: Enhanced (ply-by-ply) method	0.95	1.0
Method 3: Developed and Method 5: Finite element method	1.0	1.0
Method 4: Direct test	1.0	1.0

For the calculation of the design stress itself for single-skin plating of fiber reinforced plastics material, ISO 12215-5 proposes Equation 3.31 for the direct stress σ_d :

$$\sigma_d = 0.5 \times \sigma_{uf} \times k_{BB} \times k_{AM}, \quad (3.31)$$

and Equation 3.32 for the shear stress τ_d :

$$\tau_d = 0.5 \times \tau_{il} \times k_{BB} \times k_{AM}. \quad (3.32)$$

3.2.4 Methods for Structural Analysis and Scantlings Determination

To perform the structural analysis and scantlings determination, ISO 12215-5 provides 6 methods:

1. The **Simplified method** is intended for single-skin or sandwich glass-reinforced plastics with same properties in b and l directions, as well as to metals and wood; it uses basic algebraic formulas to determine the required thickness for plating;
2. The **Enhanced method** is intended for the same materials as the Simplified method, but extends to orthotropic materials, like carbon and aramid fiber reinforcements; it performs a ply-by-ply analysis, calculating the stress and strain in each individual layer of the material;
3. The **Developed method** is intended for any type of laminate, including non-balanced ones; it uses Classical Laminate Theory (CLT) to analyze stress and strain in both principal directions, typically applying failure criteria such as Tsai-Hill or Tsai-Wu [50];
4. The **Direct test method** relies on physical mechanical testing of the built components, which are deemed compliant if its tested strength is at least equal to the required design value;
5. The **Finite Element Method (FEM)** allows the use of 3D numerical procedures to assess structural integrity; it allows more realistic analysis of actual geometries than relying on rectangular equivalents;
6. The **Drop test method** is an alternative test specifically for craft with a hull length less than 6 m; the fully loaded craft is lifted to a specific height and released into water; the boat passes if an inspection after the drop shows no cracks.

In this work, three of these methods will be put into practice to perform the scantling calculations of the bottom hull panels: the Simplified method, the Enhanced method and FEM.

3.2.4.1 Simplified Method

As previously stated, through the Simplified method, it is possible to calculate directly the required thickness for each panel t_d using Equation 3.33,

$$t_d = b \times k_c \times \sqrt{\frac{P \times k_{2b}}{1000 \times \sigma_{d,Simp}}}, \quad (3.33)$$

where b is the shorter dimension [mm], k_c is the curvature factor [1], P is the design pressure calculated for that panel [kN/mm²], k_{2b} is the aspect ratio for the bending moment [1] and $\sigma_{d,Simp}$ is the design stress calculated for the Simplified method, for material of the plate being analyzed [MPa]. According to this method, the laminate behaves as only one layer, so the flexural strength used to calculate the design stress $\sigma_{d,Simp}$ is the equivalent of the whole laminate.

Furthermore, it is also possible to define the required value of dry fiber reinforcement per squared meter w_f , as defined in Equation 3.34:

$$\frac{t_d}{w_f} = \frac{t}{w} \equiv w_f = \frac{t_d}{t} \times w, \quad (3.34)$$

where w is the existing dry fiber reinforcement per square meter [kg/m^2], t is the existing thickness in the panel [mm]. According to this method, the panel is compliant with the ISO 12215-5 standard if the fiber reinforcement per square meter ratio ($w_{f, ratio}$), also designated in this work as compliance factor calculated with the Simplified method (CF_{Simp}), is bigger then 1, which is defined by Equation 3.35:

$$CF_{Simp} = w_{f, ratio} = \frac{w}{w_f}. \quad (3.35)$$

3.2.4.2 Enhanced Method

This method procedure is described on Annex H of ISO 12215-5 [20]. After concluding the determination of the geometry of the panels, the design loads and the material properties of each individual layer, the Enhanced method leads to a ply-by-ply analysis of the laminate.

The first step is the calculation of the design loads acting on a 1 mm wide strip of the panel. They divide into two loads: a shear force F_d and bending moment M_d imposed on the panel, originated by the design pressure, for directions b and l . Considering these values for the b direction, Equation 3.36 shows the method to calculate the shear force:

$$F_{db} = k_c \times k_{SHb} \times P \times b \times 10^{-3}, \quad (3.36)$$

where F_{db} is the shear force in b direction [N/mm] and k_{SHb} is the aspect ratio for shear force in the b direction [1]. Equation 3.37 demonstrates the method to calculate the bending moment:

$$M_{db} = -1/6 \times k_{2b} \times b^2 \times P \times 10^{-3}, \quad (3.37)$$

where M_{db} is the bending moment in the b direction [N.mm/mm], and k_{2b} is the aspect ratio factor for bending moment in the b direction [1]. The M_{db} , must be corrected by multiplying it with the curvature factor k_c to account the curvature of the panel, as seen in Equation 3.38:

$$M_{dbcorr} = M_{db} \times k_c, \quad (3.38)$$

where M_{dbcorr} is the corrected bending moment for the analyzed panel [N.mm/mm].

The next step is the calculation of the neutral axis of the laminate (z_{NA}) and the bending stiffness EI of the laminate strip. Figure 3.27 shows a schematic of the 1 mm wide laminate strip where 1 is the baseline/reference location, which in this case was defined in the inner surface of the laminate, t_i and z_i is a ply's thickness and centroid distance from the baseline, respectively, z_t is the distance from the inner layer to the outer layer, not considering the gel coat, z_{NA} is the

distance from the baseline to the neutral axis location, and z_{crit} is the distance of the neutral axis to the outer surface of the considered ply.

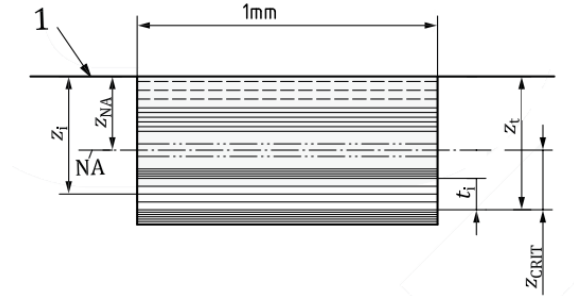


Figure 3.27: Schematics of the considered 1 mm wide strip; adapted from ISO 12215-5 [20].

The neutral axis location z_{NA} is given by Equation 3.39:

$$z_{NA} = \frac{\sum E_i \times t_i \times z_{gi}}{\sum E_i \times t_i}, \quad (3.39)$$

and the bending stiffness for each ply $E_i I_i$ [N.mm] is given by Equation 3.40:

$$E_i I_i = E_i \times \frac{t_i^3}{12} + E_i \times t_i \times z_{gi}^2, \quad (3.40)$$

which is the sum of the second moments of area (inertia) of each ply around its centroid plus, according to the parallel axis theorem [20], the area multiplied by the square of the distance to the base. The sum of $E_i \times I_i$ from all layers gives the value of EI_{BASE} .

The EI value around the neutral axis EI_{NA} is obtained with Equation 3.41:

$$EI_{NA} = EI_{BASE} - z_{NA}^2 \times \sum (E_i \times t_i), \quad (3.41)$$

still according to the parallel axis theorem. The value EI_{NA} is usually also called "D", the rigidity, in laminate theory [20].

Then, with Equation 3.42, the section moduli SM_i is then calculated for each layer,

$$SM_i = \frac{EI_{NA}}{z_{crit} E_i}, \quad (3.42)$$

and with these values, it is possible to predict the expected maximum strains ϵ_i and stresses σ_i for each layer represented in Equations 3.43 and 3.44, respectively:

$$\epsilon_i = \frac{M_{db}}{E_i SM_i}, \quad (3.43)$$

$$\sigma_i = \frac{M_{db}}{SM_i}. \quad (3.44)$$

To analyze the compliance between the calculated values and the mechanical properties of the material, the compliance factor determined with the Enhanced method (CF_{Enh}) is given by:

$$CF_{Enh} = \frac{\sigma_{di,Enh}}{\sigma_{i,Enh}}, \quad (3.45)$$

where $\sigma_{d,Enh}$ is the design stress, or allowed stress, calculated for the Enhanced method application regarding the material of the layer of the panel being analyzed, and $\sigma_{i,Enh}$ is the predicted maximum stress for the layer. Therefore, if the compliance factor is greater than 1, the layer can sustain the predicted bending moments. Regarding the failure logic of the laminate, the ISO 12215-5 standard follows a "first ply to fail" logic, meaning the weakest layer governs the entire laminate's strength [20], which is usually one of the outer layers.

The Enhanced method also allows for an interlaminar shear stress analysis using the previously defined shear force F_d . This analysis is particularly important for assessing the bond integrity of sandwich panels and the adhesion between stiffeners and plating, ensuring that delamination does not occur. Although this analysis may also be applied to single-skin panels, it is generally not considered critical according to ISO 12215-5 [20] and, therefore, will not be performed in this work. Furthermore, the standard ISO 12215-5 focuses on the calculation of stress in the b -direction as it expects it to be higher than the one calculated for the l -direction.

To conduct the analysis using this method, only the ultimate flexural strength and elastic modulus of each layer are necessary as understood from the procedures described in this section.

The Enhanced method is considered a more reliable approach to perform scantling determinations than the Simplified method, as it consists of a much more detailed process with less conservative definitions.

3.2.4.3 Finite Element Method

This method was added to the ISO 12215-5 standard as a way of performing structural calculations in the last standard revision. It recognizes that structural components are best analyzed using 3-D numerical procedures. The recommended methodology, supported by ISO 12215-5, is:

1. apply a unit pressure to the analyzed elements;
2. calculate the stresses σ_i in each analyzed element due to the unit pressure;
3. check on the analyzed elements that $\sigma_i \times P_d \leq \sigma_d$, where P_d is the design pressure for each panel considering the relevant factors.

Regarding the boundary conditions, no explicit rule is specified in the document, with the exception of the indication that the critical area to be analyzed should be situated far from the model boundaries and that loads should be applied as distributed loads.

More research was done regarding the application of boundary conditions for this method. Inertia relief was used by several researchers, such as Grasso et al. [22], Alvarado et al. [43], Mancuso et al. [55], and Kang et al. [56]. This condition allows the balancing of the force differences by creating a state of static equilibrium [43], equivalent to that of a vessel in a marine

environment. The maximum number of simple supports applied with the inertia relief condition is 6, according to Kang [56].

As the objective is for the material not to fail, for the material description in the FEM environment, since the simulation is conducted in the elastic regime, only elastic parameters are necessary to input.

Even though the ISO 12215-5 standard advises the scantlings obtained with the Enhanced method and with the finite element method to be compared, the observation done to numerical and analytical results simultaneously must be done prudently, as these methods consider different types of situations. The analytical methods analyze the panels individually, assuming they are fully fixed at the edges with average dimensions and curvatures. Analyses performed with FEM consider the exact geometry of each panel and allow the load to distribute more realistically since it considers the global model. Naturally, it is expected that the compliance factors calculated with the FEM approach to be higher than the ones calculated with the analytical tool, due to the imposed conservative nature of the analytical methods and the possibility of load distribution inside the model. However, if the reason for the occurrence is adequately explained, the standard permits the opposite to occur.

Regarding the output taken from the FEM software, ISO 12215-5 evaluates laminate compliance based on lamina-level normal stresses derived from classical laminate theory. Since these stresses are defined in the principal material coordinate system of each ply, comparison must be performed using stresses expressed in the same coordinate system.

The compliance factor values calculated from the FEA results (either being CF_{Abaq} for the Abaqus software results or CF_{SW} for the SolidWorks FEM package results) are obtained similarly to the Enhanced method, as it is defined by the following:

$$CF_{FEM} = \frac{\sigma_{di,FEM}}{\sigma_{i,FEM}} \quad (3.46)$$

where $\sigma_{di,FEM}$ is the design stress calculated for the application of FEM regarding the material of the panel being analyzed, and $\sigma_{i,FEM}$ is the considered maximum stress for the panel.

3.3 Numerical Modeling

The methodologies developed in this work are intended to be integrated into the design workflow of boat designers in order to enable more efficient structural calculations and material allocation. In this context, the finite element analyses performed must be expeditious while maintaining adequate accuracy. Consequently, the selected simulation parameters must support this balance between efficiency and reliability.

In that regard, some parameters must be analyzed to optimize the numerical analysis process regarding time consumption and results accuracy and to understand their influence on the final results.

Regarding the software used to perform the FEM simulations, two numerical codes are considered:

- **SolidWorks**, which is primarily a CAD software, but also includes FEM simulation capabilities;
- **Abaqus**, which is a more advanced tool specialized to perform FEA.

The use of SolidWorks as a FEM tool is motivated by the advantage of a rapid preprocessing stage, without the need to import geometric data. Since SolidWorks is the CAD software used by Riamar company, designers would not need to switch software platforms to perform finite element analyses.

Moreover, the transfer of geometric data from CAD to CAE software, particularly for complex geometries, which is often the case for marine structures, can lead to geometric inconsistencies, such as imprecise or even invalid entities [57]. These issues frequently require additional corrective work to eliminate geometric errors and ensure that the generated mesh is free of deficiencies.

Naturally, the SolidWorks software presents certain limitations, particularly with respect to material definition. Nevertheless, SolidWorks should not be disregarded as a useful tool for preliminary finite element analyses.

Regarding the specific parameters that require optimization, they are mesh-related, with the element type, which can be solid or shell, and element size. Due to the variations of material properties exposed on Section 3.1, the sensitivity analysis to the elastic modulus and thickness changes will also be examined.

3.3.1 Numerical Model Creation

To understand and optimize the numerical modeling workflow for both software, an additional geometric model was developed and analyzed in both FEM environments before proceeding to the vessel analyses described in Chapter 4.

The CAD model of the vessel that was developed was inspired by the design of Riamar's Pescador 500, shown in Figure 1.1 (a), as well as by the design of an Optimist boat (Figure 3.28), chosen for its geometric simplicity.



Figure 3.28: Optimist boat used as reference model for the optimization of numerical variables for the finite element analysis.

The developed CAD model is shown in Figure 3.29. It is 5 m long, 2 m wide, and consists of a solid model with a uniform thickness of 5.27 mm.

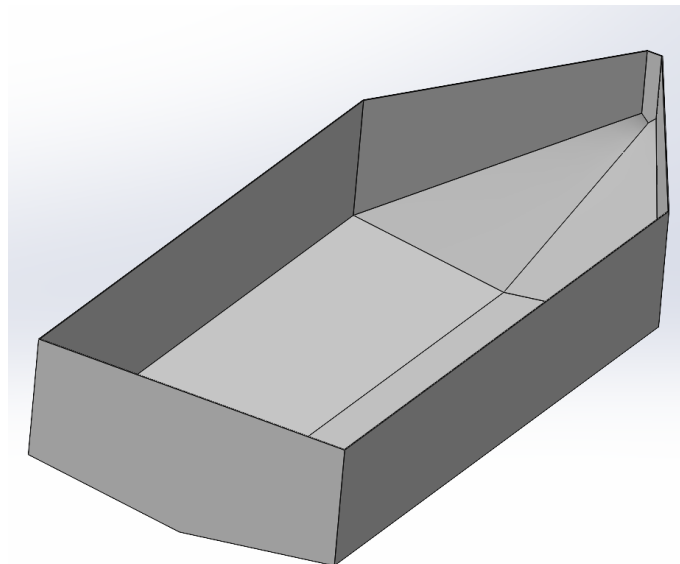


Figure 3.29: Geometry developed based on the Pescador 500 vessel and Optimist boat.

Four different finite element models were created: one solid model and one shell model for each software. To transform the solid model to shell in SolidWorks, the inside surfaces of the model were deleted. Then both solid and shell files were imported to Abaqus with, probably due to its simple geometry, no imprecise or invalid geometry. Figure 3.30 shows the shell geometry of the reference model imported from SolidWorks to Abaqus.

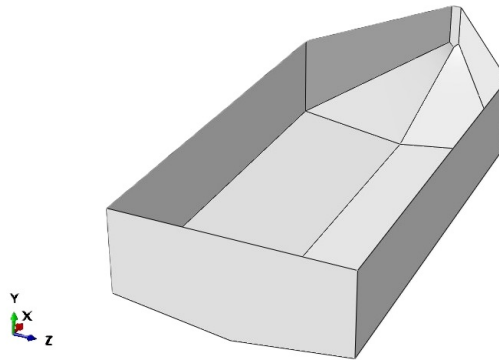


Figure 3.30: Shell Abaqus geometric model.

To ensure a consistent comparison between the results of the different simulations, all results must be extracted from identical locations in each model. For this purpose, geometric partitions were created in all models, enabling the definition of "Sets" in Abaqus and "Selection Groups" in SolidWorks corresponding to the nodes of interest. These sets and selection groups facilitate systematic and consistent data extraction from the global model results. Figure 3.31 shows the location of the considered nodes, as well as the identification name given to each one. The reason for selecting these two nodes is that they correspond to the geometric center of the hull panel in which they are located, being also away from the supports described in Figure 3.32.

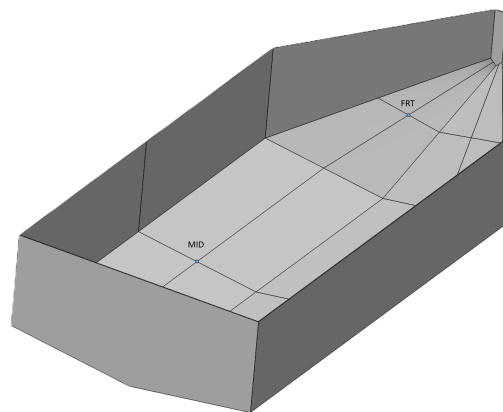


Figure 3.31: Location and identification names of nodes considered for the results comparison.

The reference material chosen for the considered models is defined in Table 3.32. The material properties chosen are those defined for the Pescador 500 vessel, assuming an isotropic material. Since the solid models have a surface thickness of 5.27 mm, the shell thickness was defined as 5.27 mm as well. An important gap noticed in the material definition of the SolidWorks software is the inability to define different ply material properties. This feature is especially helpful when complex composite materials are being modeled, with, for example, different directional elastic properties or different layer thicknesses.

Table 3.32: Reference thickness and material properties used for the initial numerical model.

t	E	ν	ρ
[mm]	[MPa]	[-]	[kg/m ³]
5.27	8 276	0.4	1.41

The boundary conditions defined are the same for all models and are based on the ones advised by the ISO 12215-5 standard. In order to minimize reaction forces at the boundaries [43], the inertia relief option was activated. Furthermore, two supports were defined at the top of the transom, and one fully fixed point was defined at the bow. A pressure (P) of 1 kN/m^2 was applied to the bottom of the hull, also according to the ISO 12215-5 directions. Figure 3.32 shows the location of the supports, and Table 3.33 shows the displacement restrictions imposed by each support, as well as the other boundary conditions imposed in the model.

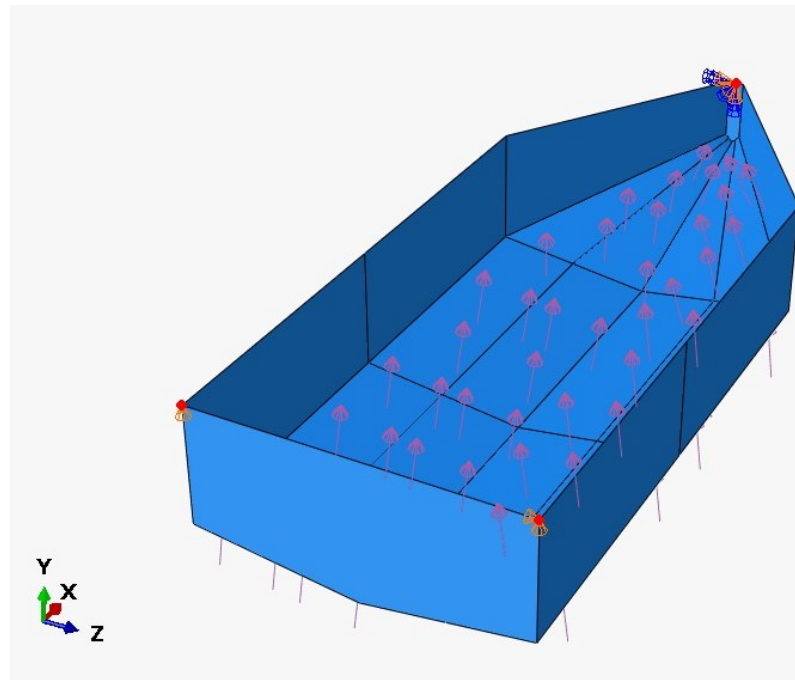


Figure 3.32: Location of supports imposed on the reference model.

Table 3.33: Boundary conditions imposed on the model.

Location of application	Boundary condition
Left side of the transom	$y = 0$
Right side of the transom	$y = z = 0$
Bow	$x = y = z = 0$
Whole model	Inertia relief
Bottom of the hull	$P = 1 \text{ kN/m}^2$

The last parameters to be defined in the finite element models are the mesh-related ones, which will be addressed in subsection 3.3.2.

3.3.2 Mesh Parameters Optimization

To determine the most appropriate combination of element type and element size, several configurations were applied to the developed models. By evaluating the accuracy and computational efficiency of each configuration, the most suitable combination was selected for each software package.

The adoption of a reference representation of the vessels described in Chapter 4 is justified for this procedure, since when applying the same boundary conditions to a similar geometric model, the behavior of the structure under similar loading scenarios could be evaluated, allowing the influence of mesh refinement on the numerical response to be systematically assessed.

The element types can be divided into two groups: solid and shell elements. Models using solid elements discretize the full three-dimensional volume of the structure, allowing the stress and strain fields to be evaluated throughout the material. Shell element models, in contrast, represent the structure by its midsurface with an assigned thickness, reducing the computational cost while accurately capturing membrane and bending behavior in thin-walled components.

For SolidWorks, only one element type was analyzed for each modeling approach. The shell models employ triangular elements with three vertex nodes and three mid-edge nodes, while the solid models use tetrahedral elements with four vertex nodes and six mid-edge nodes [58]. These correspond to quadratic elements, selected through the “High Quality” option in the advanced mesh parameters.

Linear elements, which have equivalent geometries but do not include mid-edge nodes, are selected using the “Draft Quality” option. However, these elements were not considered in this work.

For Abaqus, several element types were analyzed for both shell and solid models:

- S4 is a linear 4-node quadrilateral shell element with full integration;
- S4R is a linear 4-node quadrilateral shell element with reduced integration;

- S8R is a quadratic 8-node quadrilateral shell element with reduced integration;
- C3D4 is a linear 4-node tetrahedral solid element with full integration;
- C3D10 is a quadratic 10-node tetrahedral solid element with full integration.

Furthermore, regarding element sizes, simulations were performed with five different sizes for the Abaqus shell model: 150, 100, 50, 25, and 10 mm. For the other models, the 10 mm element size was not included due to the time necessary to conclude the simulations. Figure 3.33 shows the mesh discretization of the model in SolidWorks and Abaqus software for an average of 25 mm of element size.

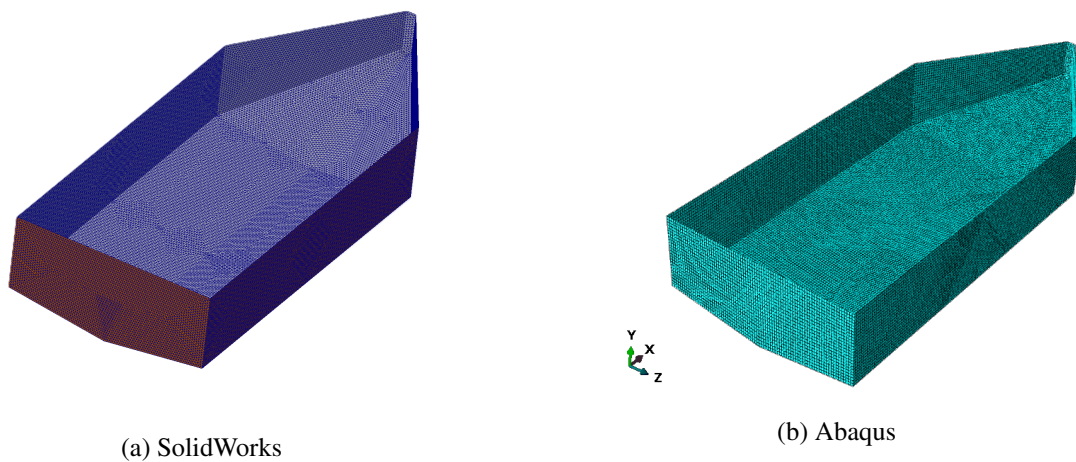


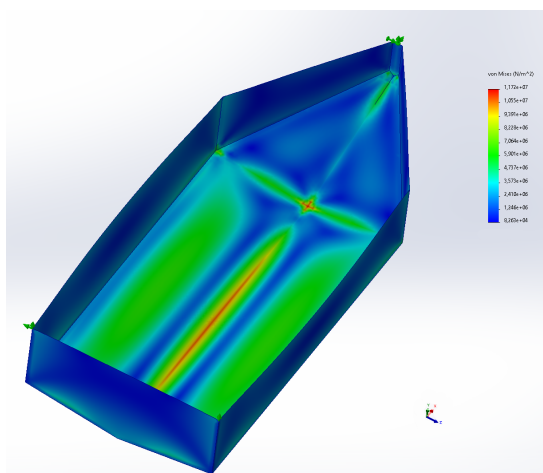
Figure 3.33: Mesh discretization of the model, in SolidWorks and Abaqus software, for 25 mm of element size; the SolidWorks-generated mesh contains 129 307 nodes and 64 400 elements, and the Abaqus-generated mesh contains 81 529 nodes and 27 034 elements.

A series of simulations were performed using the mesh configurations previously described. Table 3.34 summarizes the analyses with the different mesh parameters.

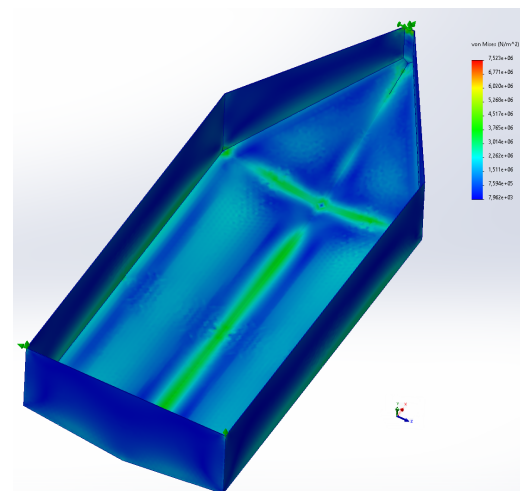
Table 3.34: Summary of the analyses conducted to evaluate the sensitivity of the mesh parameters, using the reference model.

FEM model		Element sizes in mm	Element types	No. of simulations
SolidWorks	Shell	150, 100, 50, 25	Shell “High Quality”	4
	Solid	150, 100, 50, 25	Solid “High Quality”	4
Abaqus	Shell	150, 100, 50, 25, 10	S4, S4R, S8R	15
	Solid	150, 100, 50, 25	C3D4, C3D10	8

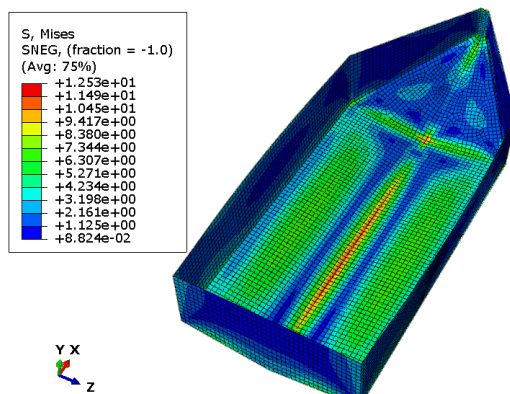
Figures 3.34 and 3.35 present the corresponding von Mises stress and Y-direction displacement results, respectively, for the four FEM models analyzed.



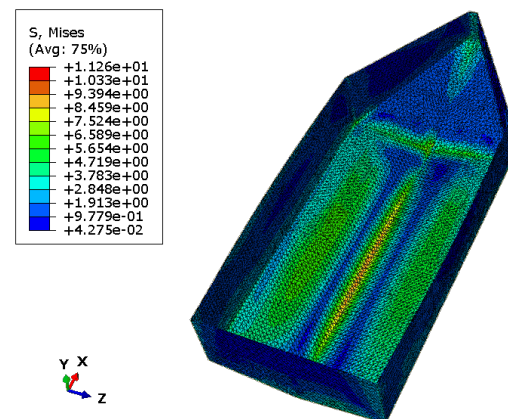
(a) SolidWorks shell model



(b) SolidWorks solid model



(c) Abaqus shell model using S8R element type



(d) Abaqus solid model using C3D10 element type

Figure 3.34: Von Mises distribution results using an average element size of 50 mm.

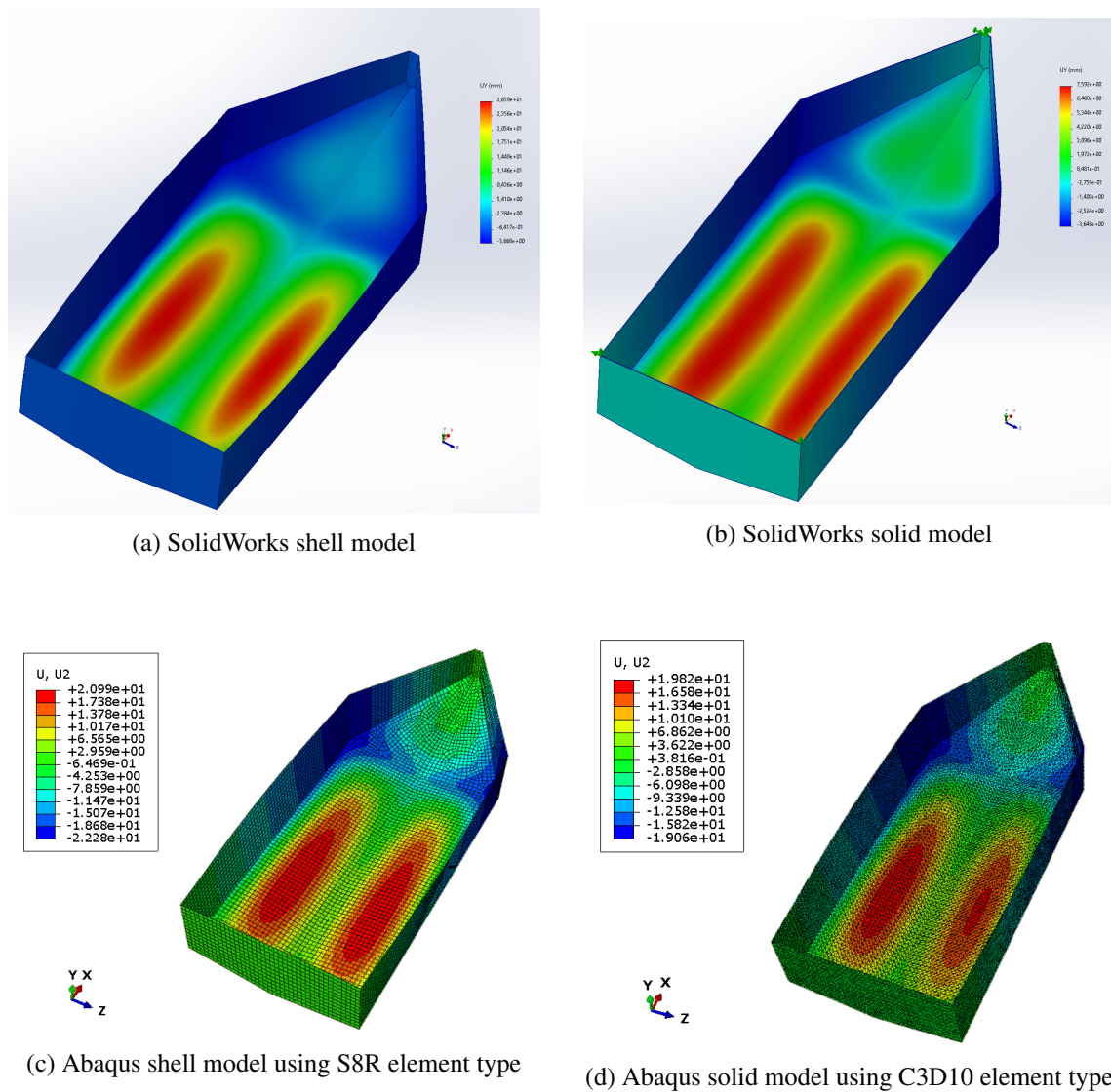


Figure 3.35: Displacement results on the Y-direction using an element size of 50 mm.

To understand which is the most efficient combination, the von Mises stress and the displacement in the Y -direction were extracted at the selected nodes for all mesh parameter combinations.

In SolidWorks, the nodal results were obtained using the “Probe Value” feature, each simulation at a time. In Abaqus, a Python script was developed to automatically extract the corresponding data from the selected nodes across all simulation output files.

Using the collected results, convergence graphs were generated to illustrate the variation of von Mises stress and displacement with mesh refinement. Figures 3.36 and 3.37 present a direct comparison of the results obtained with different element types as a function of element size.

It should be noted that the data identified in the legend as “SWshell” and “SWsolid” correspond to the results obtained from the SolidWorks shell and solid models, respectively.

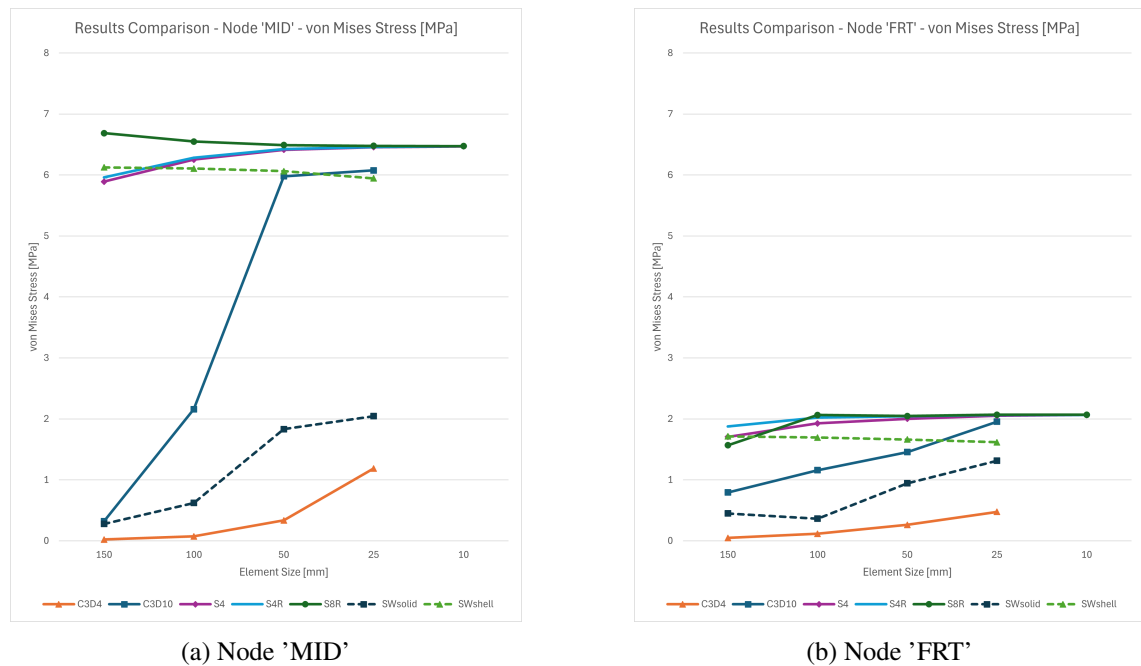


Figure 3.36: Von Mises stress results comparison for all models.

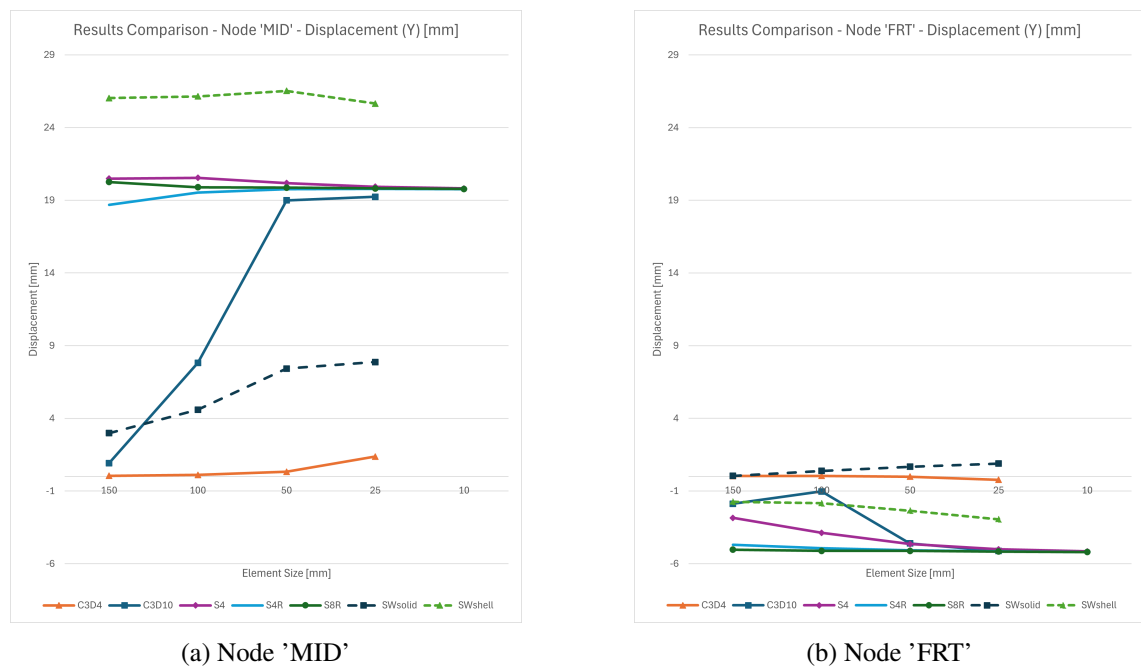


Figure 3.37: Displacement results comparison in the Y-direction for all models.

The Abaqus shell elements (S4, S4R, and S8R) show consistent and stable convergence behavior. As the mesh is refined from 150 mm to 25 mm, both displacement and stress values, for the two nodes, progressively stabilize, with very small differences observed between 50 mm and 25 mm element sizes. This indicates that the shell modeling approach in Abaqus provides mesh-independent results.

In contrast, the solid element formulations show a significantly different response. The linear tetrahedral element C3D4 consistently underestimates both displacement and stress when compared to the shell solutions. A similar tendency is observed for the SolidWorks solid model, whose results remain noticeably lower than those obtained with shell elements. Even with mesh refinement, these solid linear formulations do not approach the converged shell values.

The quadratic tetrahedral element C3D10 presents a more favorable behavior. For coarse meshes (150 mm and 100 mm), the results deviate considerably from the shell solutions; however, as the mesh is refined to 50 mm and particularly to 25 mm, both displacement and stress values begin to converge toward those obtained with Abaqus shell elements.

Regarding the comparison between shell models in Abaqus and SolidWorks, the results show partial agreement. In terms of von Mises stress, the SolidWorks shell model produces values that are relatively close to those obtained with Abaqus shell elements. Although exact convergence is not observed, the discrepancies remain moderate, and the overall trend with mesh refinement is consistent.

To determine the most suitable modeling approach for Abaqus and SolidWorks, the computational time required to complete each analysis was also taken into account. Table 3.35 summarizes the execution time for each simulation, and Figure 3.38 presents the corresponding graphical representation.

Overall, shell elements are presented as the most accurate modeling strategy for thin composite hull structures subjected primarily to bending. Abaqus shell elements, in particular, show clear convergence and numerical stability with moderate mesh refinement. Linear solid elements (C3D4 and SWsolid) are not suitable for this type of structural behavior, while quadratic solid elements (C3D10) can achieve comparable accuracy but at the expense of increased computational effort.

Table 3.35: Computational cost in seconds for each combination of element type and mesh size.

		Element sizes in mm				
		150	100	50	25	10
Element types	Swshell	2	2	4	14	–
	Swsolid	3	3	10	55	–
	S4	3.6	7.3	25	97	449
	S4R	2.9	5.2	16	63	313
	S8R	7.4	16	64	274	3350
	C3D4	4.2	8.2	31	111	–
	C3D10	9.6	22	100	496	–

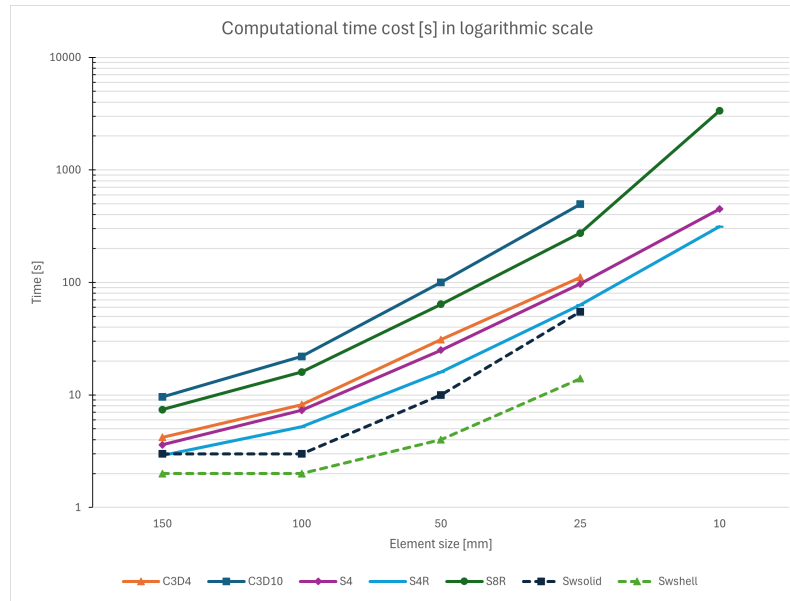


Figure 3.38: Computational time cost in logarithmic scale for all simulations performed.

In SolidWorks, an element size of 100 mm was selected. The convergence results imply that further mesh refinement does not result in considerable converging variations in stress or displacement values, while the 100 mm mesh provides the shortest execution time among the assessed configurations.

For the Abaqus analyses, greater emphasis was placed on numerical accuracy, as this software is intended to provide the reference results. Among the evaluated element formulations, the S8R element was considered the most suitable, as it exhibited the lowest sensitivity to mesh refinement and demonstrated stable convergence behavior. With respect to mesh density, an element size of 25 mm was selected. This size corresponds to the finest mesh that offers improved precision while maintaining a reasonable computational cost. Consequently, the S8R element with a 25 mm characteristic length was adopted for the subsequent analyses.

3.3.3 Elastic Modulus and Thickness Sensitivity Analysis

The sensitivity of the simulation to the variation of the elastic modulus and thickness was also investigated. From the experimental tests performed previously in this work, it is perceived that these material- and manufacturing-related properties can fluctuate, even from samples from the same laminate. In that regard, a combination of 5 simulations were performed for each parameter and for each software.

These analyses were performed using a finite difference approach to estimate local sensitivities, and these were converted to normalized sensitivity coefficients to enable comparison of the relative impact of different parameters on the model output [59]. The equation to calculate the sensitivity value is:

$$S_X^Y = \frac{\Delta Y}{\Delta X} \times \frac{X_{ref}}{Y_{ref}} = \frac{Y_n - Y_{ref}}{X_n - X_{ref}} \times \frac{X_{ref}}{Y_{ref}}, \quad (3.47)$$

where S_X^Y is the sensitivity of the output Y to the input X , X_{ref} and X_n are the reference and new inputs, and Y_{ref} and Y_n are the reference and new outputs, respectively.

These analyses employed the same model characteristics as the one for the mesh parameter optimization procedure, only changing the elastic modulus and thickness, one parameter at a time. Table 3.36 shows the group of values used of the elastic modulus and thickness.

Table 3.36: Elastic modulus and thickness values group to be analyzed.

E [MPa]	t [mm]
7 500, 8 000, 8 276, 8 500, 9 000	4.5, 5, 5.27, 5.5, 6

Both software were used to perform these analyses, with the mesh discretization combinations defined in subsection 3.3.2. The node 'MID', with location shown in Figure 3.31, was used to compare the simulation results data.

First, with respect to the elastic modulus sensitivity analysis, Tables 3.37 and 3.38 present the calculated sensitivity values together with the corresponding terms used in their evaluation. Figure 3.39 illustrates the influence of the elastic modulus (E) on the displacement results in the Y-direction of the 'MID' node, as well as the sensitivity of the displacement response to variations in E .

Table 3.37: Sensitivity analysis using SolidWorks with respect to Young's modulus variation.

E [MPa]	$\Delta E/E_{ref}$	U_2	$\Delta U_2/U_{2,ref}$	S_{U_2}
7500	0.093	28.8	-0.103	-1.11
8000	0.032	27.0	-0.034	-1.07
8267	0.000	26.1	0.000	–
8500	-0.028	25.5	0.026	-0.94
9000	-0.089	24.0	0.081	-0.91

Table 3.38: Sensitivity analysis using Abaqus with respect to Young's modulus variation.

E [MPa]	$\Delta E/E_{ref}$	U_2	$\Delta U_2/U_{2,ref}$	S_{U_2}
7500	0.093	21.8	-0.102	-1.10
8000	0.032	20.5	-0.033	-1.03
8267	0.000	19.8	0.000	–
8500	-0.028	19.3	0.027	-0.97
9000	-0.089	18.2	0.081	-0.92

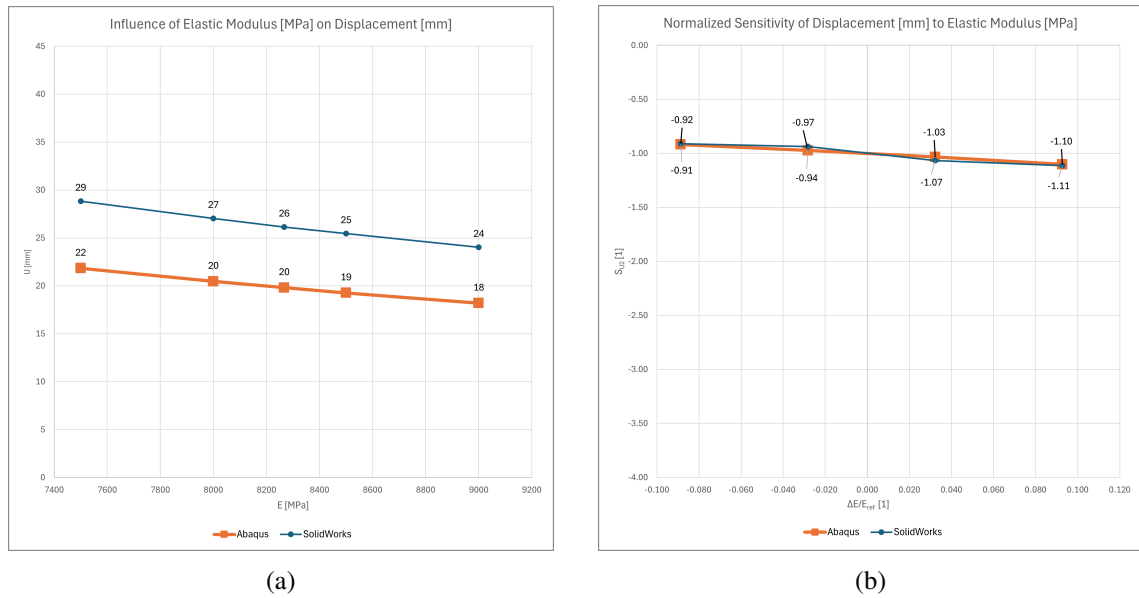


Figure 3.39: Graphs showing the influence of E on the displacement and its sensitivity to the E variation.

The sensitivity analysis with respect to elastic modulus shows that the displacement in the Y-direction (U_2) is inversely proportional to E in both SolidWorks and Abaqus software results. The normalized sensitivity values (S_{U2}) are consistently close to -1, between approximately -0.9 and -1.1. This indicates a nearly linear inverse relationship between stiffness and displacement. A percentage increase in elastic modulus leads to an almost equivalent percentage decrease in displacement.

The sensitivity of stress results to variations in the elastic modulus is not presented, as the stress results remain unchanged with variations in E , meaning that E does not influence stress results. This occurs because the model is analyzed in the linear elastic regime under load-controlled conditions; stresses depend primarily on geometry and applied load, not on stiffness [60].

In addition, the sensitivity to laminate thickness fluctuations was evaluated. Tables 3.39 and 3.40 present the parameter variations together with the corresponding sensitivity values.

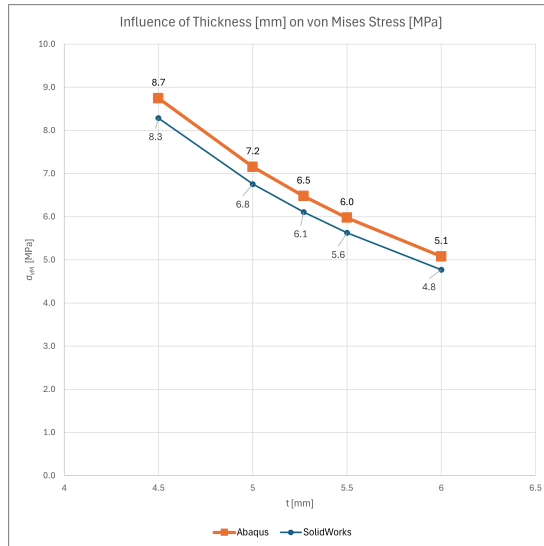
Figures 3.40 and 3.41 illustrate the influence of thickness on both von Mises stress and displacement, as well as the sensitivity of these response quantities to variations in t .

Table 3.39: Sensitivity analysis using SolidWorks with respect to the thickness variation.

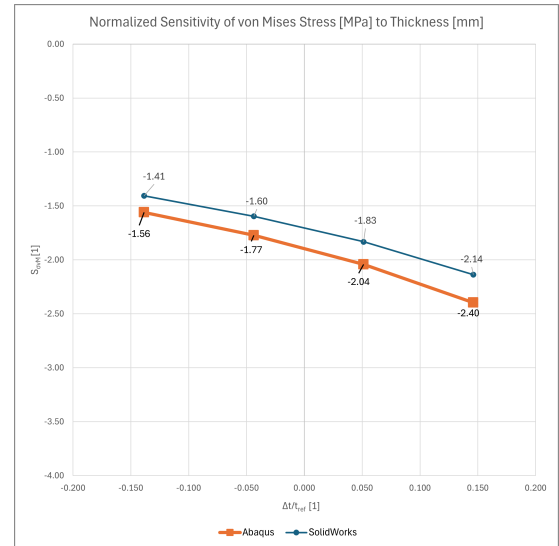
t [mm]	$\Delta t/t_{\text{ref}}$	σ_{vM}	$\Delta\sigma_{vM}/\sigma_{vM,\text{ref}}$	$S_{\sigma_{vM}}$	U_2	$\Delta U_2/U_{2,\text{ref}}$	S_{U_2}
4.50	0.146	8.3	-0.357	-2.44	40.1	-0.533	-3.65
5.00	0.051	6.8	-0.106	-2.08	30.1	-0.152	-2.96
5.27	0.000	6.1	0.000	—	26.1	0.000	—
5.50	-0.044	5.6	0.079	-1.80	23.4	0.106	-2.43
6.00	-0.139	4.8	0.219	-1.58	18.6	0.288	-2.08

Table 3.40: Sensitivity analysis using Abaqus with respect to the thickness variation.

t [mm]	$\Delta t/t_{\text{ref}}$	σ_{vM}	$\Delta\sigma_{vM}/\sigma_{vM,\text{ref}}$	$S_{\sigma_{vM}}$	U_2	$\Delta U_2/U_{2,\text{ref}}$	S_{U_2}
4.50	0.146	8.7	-0.350	-2.40	30.4	-0.492	-3.37
5.00	0.051	7.2	-0.105	-2.04	22.8	-0.142	-2.77
5.27	0.000	6.5	0.000	—	19.8	0.000	—
5.50	-0.044	6.0	0.077	-1.77	17.7	0.102	-2.33
6.00	-0.139	5.1	0.216	-1.56	14.1	0.277	-2.00



(a)



(b)

Figure 3.40: Graphs showing the influence of t on the von Mises stress and its sensitivity to the t variation.

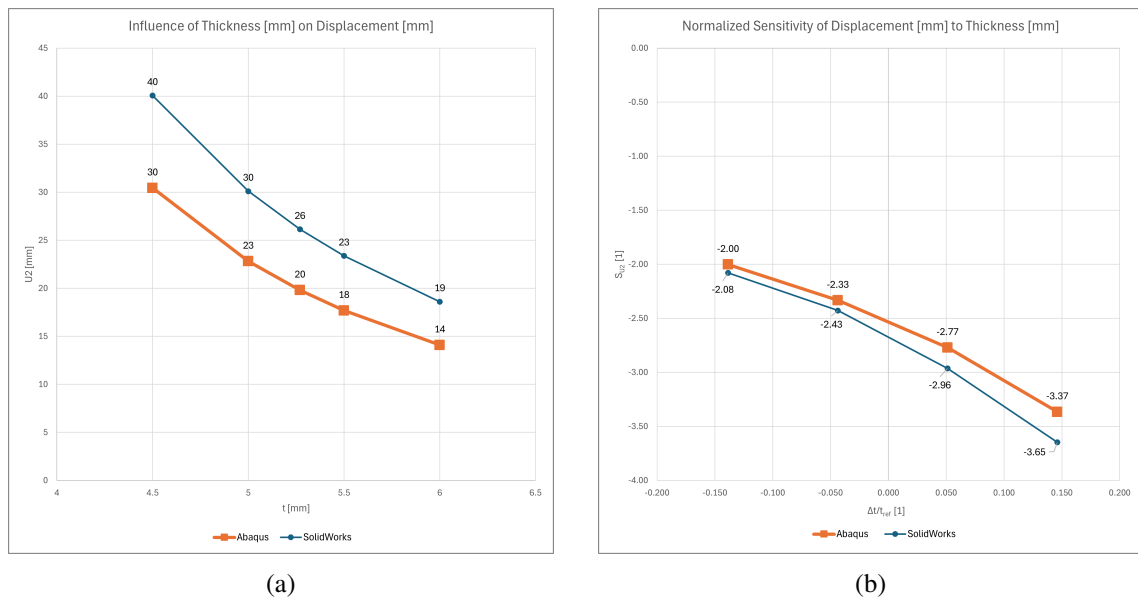


Figure 3.41: Graphs showing the influence of t on the displacement and its sensitivity to the t variation.

The sensitivity analysis performed with respect to laminate thickness demonstrates a stronger influence on the structural response when compared to variations in elastic modulus. The normalized sensitivity of the Y-direction displacement varies approximately between -2.0 and -3.7, while the normalized sensitivity of the von Mises stress ranges between -1.5 and -2.4. Both sensitivities increase with the decrease of thickness. These values are significantly higher in magnitude than those obtained for the elastic modulus, indicating that the model response is considerably more sensitive to thickness variations. Both software show similar behavior regarding this analysis.

The displacement results, in particular, exhibit high sensitivity to thickness. This behavior is consistent with classical plate bending theory [50], in which the bending stiffness of a plate is proportional to the cubic power of thickness ($D \propto Et^3$). Consequently, even modest changes in thickness produce substantial changes in flexural rigidity, directly reducing deflections under the same applied load.

In addition to affecting displacement considerably, thickness variation also has a big effect on stress levels. As thickness increases, bending stresses decrease due to the higher second moment of area and enhanced resistance to deformation [61]. So, unlike the elastic modulus, which mostly impacts displacement in this load-controlled linear elastic analysis, thickness has a direct effect on both global stiffness and stress magnitudes.

Overall, the results clearly indicate that the laminate thickness has the bigger influence governing the structural behavior of this FEM model. From a structural design perspective, the performed finite element analyses predict that controlling the thickness is therefore more critical than small fluctuations in elastic modulus for ensuring adequate stiffness and strength performance.

3.4 Chapter Summary

In this chapter, the mechanical properties of the composite laminates used by the Riamar company were defined and consolidated for structural assessment through experimental and analytical approaches.

The methodologies for applying the analytical procedures described in ISO 12215-5 were detailed, including the definition of panel geometry, loading conditions, laminate properties, and compliance factor evaluation. With the equations and details explained in the standard regarding the Simplified and Enhanced methods, an analytical tool was created in the form of a Microsoft Excel file that allowed a more efficient assessment of the vessel components.

Furthermore, guidelines for the implementation of FEM application were established. This ensured consistency between the analytical and numerical approaches, allowing meaningful comparison of results.

To validate and refine the numerical approach, a reference hull representation was analyzed under conditions equivalent to those later applied to the studied vessels. This preliminary investigation enabled a clear understanding of the modeling procedures in both Abaqus and SolidWorks, the optimization of mesh-related parameters, and the sensitivity of structural response to variations in elastic modulus and laminate thickness. This study demonstrated that shell elements provide superior performance compared to solid elements. Based on these results, the S8R element with an average size of 25 mm was selected for the Abaqus finite element analyses, while a shell element with an average size of 100 mm was adopted for the SolidWorks FEM package application.

Overall, this chapter provided the necessary foundation for the structural assessment and analyses presented in Chapter 4.

Chapter 4

Applications and Results

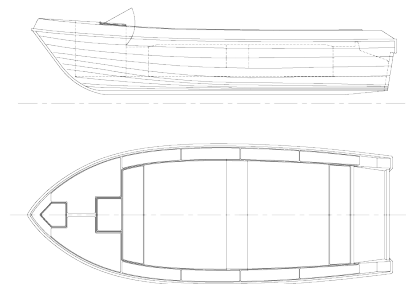
In this chapter, the Simplified method, the Enhanced method, and FEM will be applied as described in Chapter 3. Two case studies will be analyzed: the Pescador 500 vessel, which is a vessel that has already been in production by the Riamar company, with a scantling report already performed for this boat; and the Evolution 5.0, a new vessel project, which is being developed and is the main object of study in this work.

4.1 Case Study 1 - Pescador 500

The first boat to be analyzed will be the Riamar Pescador 500, shown previously in Figure 1.1 and in Figure 4.1 (a) and (b). This boat has been manufactured by Riamar for a long time and has already been subjected to an evaluation of whether it is in compliance with the ISO 12215-5 guidelines, according to the Simplified method. Therefore, it reveals itself as a great opportunity to validate the results obtained with the analytical tool and compare them with the ones collected with the application of FEM.



(a) Riamar Pescador 500 boat



(b) Riamar Pescador 500 schematics

Figure 4.1: Vessel to be analyzed in this section.

Since this model was originally developed several years ago, without the assistance of computer-aided design software, the geometry of this vessel had to be hand-drawn into sketches by a professional to then be passed to CAD software Rhino by a design engineer. Therefore, in order to better understand the geometry of the vessel, the CAD file was imported to SolidWorks before performing any scantling calculation. The original CAD model is shown in Figure 4.2.

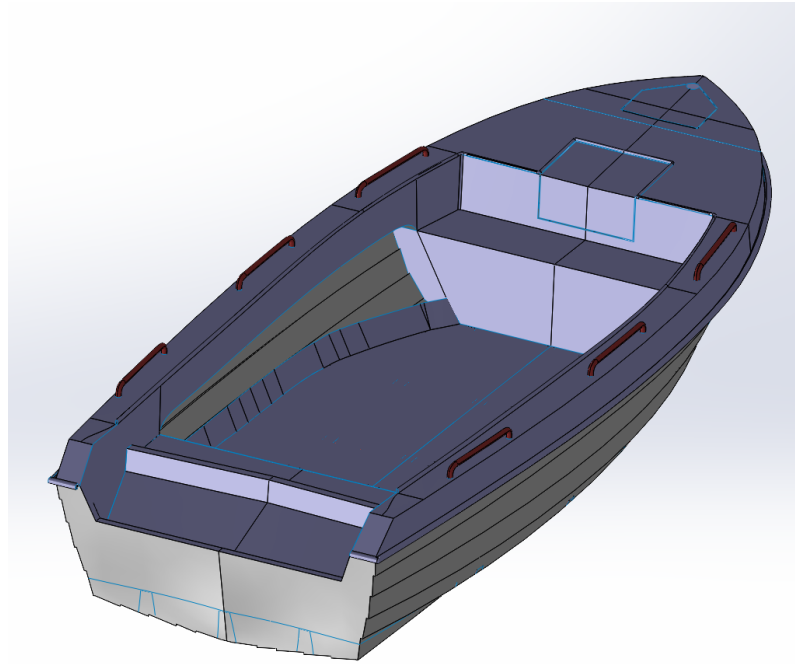


Figure 4.2: CAD model imported from Rhino to SolidWorks of the Pescador 500 vessel.

As observed, this shell model has a considerable number of flaws, including unfinished deck geometry, surface overlapping, and imprecise curvatures, especially in the hull, which will be the object to analyze. An adaptation of this model was then developed, presented in Figure 4.3, with better geometry definition, which will be particularly useful to apply the FEM later in this work.

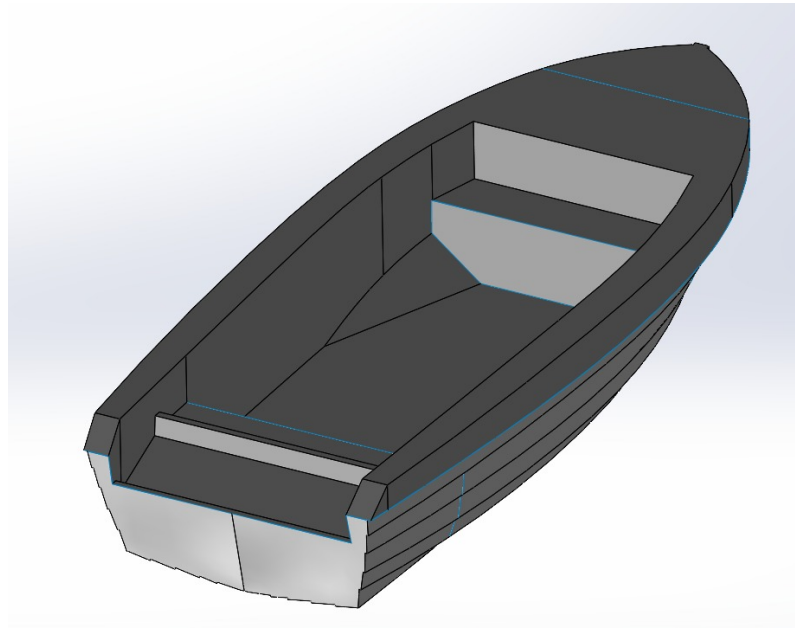


Figure 4.3: CAD model developed of the Pescador 500 vessel.

With the geometry of the vessel established, it is possible to proceed with the structural calculations, using both the analytical and numerical tools.

4.1.1 ISO 12215-5 Analytical Scantling

The application of the analytical tool requires the definition of several input parameters. These variables are determined as described in Section 3.2. In this case, certain data related to the Pescador 500 vessel were extracted from the scantling report to ensure that the validation of the tool was performed using the same input values adopted in the original scantling calculations.

4.1.1.1 Geometric and Design Parameters

To perform the scantling calculations based on ISO standard, the first step, as described in Chapter 3, is to acquire the main data from the vessel. This data was obtained through inspection of the CAD model and is in accordance with the scantling report. The Pescador 500 is a design category C recreational motorcraft made of glass fiber. Additional representations of the vessel are shown in Figures 4.4 and 4.5.

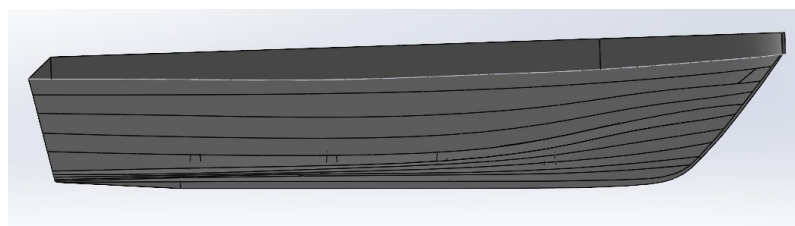


Figure 4.4: Side view of Pescador 500 CAD model.

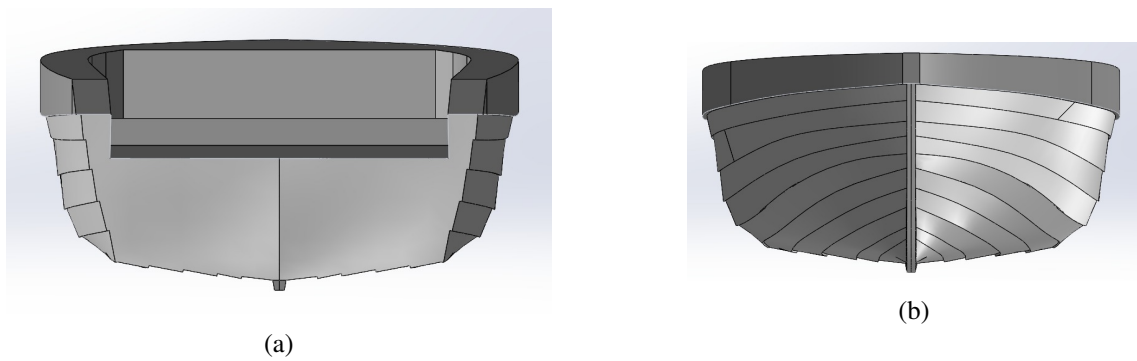


Figure 4.5: Rear (left) and front (right) views of Pescador 500 model.

Additional relevant information regarding this vessel is shown at Table 4.1.

Table 4.1: Main characteristics of the Pescador 500 vessel.

Mass at loaded condition, m_{LDC} [kg]	1162.0
Length of the hull, L_H [m]	4.960
Waterline length, L_{WL} [m]	4.419
Waterline beam, B_{WL} [m]	1.724
Chine beam, B_C [m]	1.708
Canoe body draft, T_C [m]	0.329
Deadrise angle, β [°]	12.7
Maximum speed, V [knots]	29.0

Regarding the panels' identification and dimensions, the same information that is on the scantling report was used. Figure 4.6 shows the panels that were analyzed in this work and their identification letter and number. The information regarding the panels' dimensions is available in Appendix B, in Table B.1. Note that, due to the symmetry of the boat, only one half of the vessel had to be examined.



Figure 4.6: Evaluated panels identification and location of the Pescador 500 vessel.

4.1.1.2 Pressure Adjusting Factors and Design Pressures

The design pressures and pressure adjusting factors were determined using the methodology described in Chapter 3, and the results are displayed in Table 4.2.

Table 4.2: Design pressure and panel-dependent coefficients for the Pescador 500 vessel.

Panel	k_L	k_{AR}	Design Pressure [kN/m^2]
A1	0.665	0.412	18.5
A2	0.661	0.426	19.0
B3	0.809	0.402	22.0
B4	0.809	0.438	23.9
C5	0.936	0.442	27.9
C6	0.937	0.464	29.3
D7	1.000	0.537	36.2
D8	1.000	0.458	30.9
E9	1.000	0.512	34.5

The design category factor $k_{DC} = 0.6$ value was taken from Table 3.29, as this vessel belongs to design category C. The dynamic load factor $k_{DYN} = 4.368$ was calculated using Equations 3.21 and 3.22.

4.1.1.3 Mechanical Properties and Design Stresses

The composite material used in the hull of the Pescador 500 vessel consists of five CSM layers: three with a dry fiber mass per square meter of 0.60 kg/m^2 , one of 0.45 kg/m^2 , and one of 0.30 kg/m^2 .

The composite's properties adopted for the hull of the Pescador 500 vessel were the same as those defined in the scantling report, ensuring that the calculated results are directly comparable with those presented in the report. The properties and the thickness used are shown in Table 4.3.

The composite's properties adopted for the hull of the Pescador 500 vessel were calculated using the fiber content in volume value defined in the scantling report for the CSM layers ($\phi = 0.167$). The properties and the thickness obtained are the same as the ones shown in the scantling report and are shown in Table 4.3.

Table 4.3: Individual layers and laminate properties for the material considered for the hull of Pescador 500 vessel.

Composite Layers	w [kg/m ²]	t [mm]	ρ [kg/m ³]	E [MPa]	ν [1]	σ_{uf} [MPa]
Layer 1 - CSM	0.30	0.7	1430	8267	0.40	156
Layer 2 - CSM	0.45	1.1	1430	8267	0.40	156
Layer 3 - CSM	0.60	1.4	1430	8267	0.40	156
Layer 4 - CSM	0.60	1.4	1430	8267	0.40	156
Layer 5 - CSM	0.60	1.4	1430	8267	0.40	156
Pescador 500 Hull	2.55	6.0	1430	8 267	0.40	156

Regarding the design stress calculations, the values are obtained through Equation 3.31. The boatbuilding factor is $k_{BB} = 0.75$ as it is the same value utilized in the scantling report. The assessment method factor is dependent on each assessment method, as can be seen in Table 3.31. For the application of the Simplified method the assessment method factor is $k_{AM} = 0.9$, for the Enhanced method $k_{AM} = 0.95$ and for FEM is $k_{AM} = 1$.

Table 4.4 presents the design stresses considered for the Pescador 500 hull material for each assessment method. In this table, $\sigma_{d,Simp}$ refers to the design stress used in the Simplified method, $\sigma_{d,Enh}$ represents the weighted average design stress applied in the Enhanced method considering all laminate layers, and $\sigma_{d,FEM}$ corresponds to the weighted average design stress used in the finite element assessment, also accounting for all layers.

In this case, the mechanical properties of all layers are identical. Nevertheless, Table B.2 presents the design stresses calculated individually for each laminate layer for both the Enhanced method and the finite element method.

Table 4.4: Design stresses calculated for each assessment method of the Pescador 500's hull material.

Composite Material	$\sigma_{d,Simp}$ [MPa]	$\sigma_{d,Enh}$ [MPa]	$\sigma_{d,FEM}$ [MPa]
Pescador 500 Hull	52.7	55.6	58.5

4.1.2 Finite Element Analysis

To perform the analysis of the boat's hull, the deck and other unnecessary components were not incorporated in the model and therefore deleted from the CAD model. Figure 4.7 shows the two components that were left, the hull and the top-hats that support the hull. To better understand the geometry of the top-hats, their representation is shown in Figure 4.8.

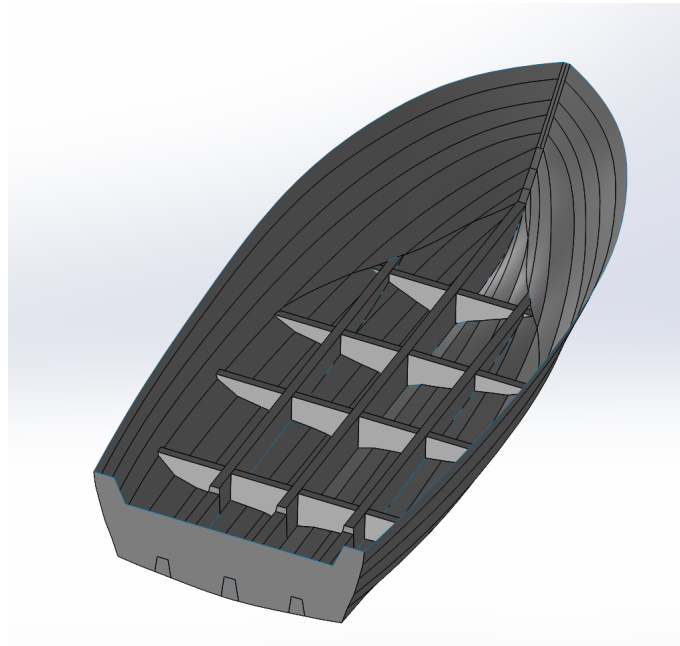


Figure 4.7: Model considered for the FEA of Pescador 500 vessel.

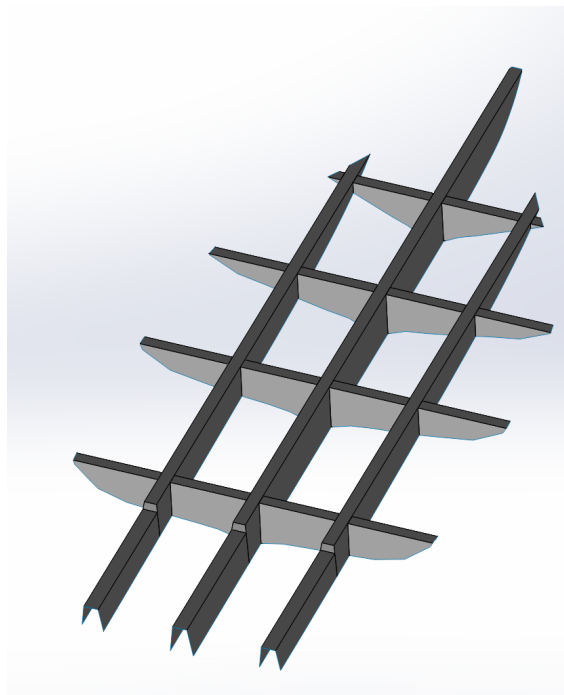


Figure 4.8: Top-hats considered for the FEA of Pescador 500 vessel.

Before creating the FEM model, partitions were set in the hull that correspond to the geometry of the panels considered for the ISO analytical scantling method, shown in Figure 4.9. This would ensure that the geometry of the panels would be the same for both software.

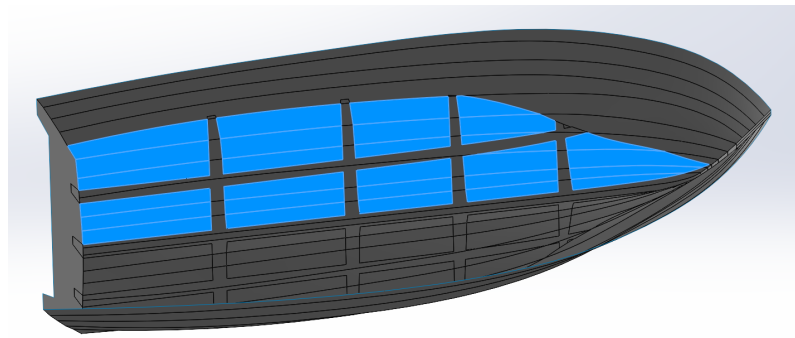


Figure 4.9: Partitioning and panels geometry considered for the FEA of Pescador 500 vessel.

4.1.2.1 SolidWorks Finite Element Analysis

To initiate the SolidWorks FEA procedure, a static study was created using the components shown in Figure 4.7. Since shell models were adopted, the first step consisted of defining the material properties and thickness of the shell surfaces.

The elastic material properties assigned to the hull are in accordance with Table 4.3. Regarding the top-hat stiffeners, their material properties are also shown in the scantling report and present the same elastic properties as the hull. However, they have only three layers of CSM with a total dry fiber reinforcement of $w = 1.5 \text{ kg/m}^2$. This means that the top-hats have a thickness of 3.5 mm. Due to the SolidWorks material definition limitations, the materials were modeled as a uniform isotropic material.

Regarding the interactions, it is necessary to bond the hull to the top-hats structure. SolidWorks software has a useful feature in the "Component interaction" branch that recognizes surfaces that are close to each other and then bonds them automatically. Afterwards, it is possible to inspect the interactions with the "Interaction viewer" option, to be sure that the bonding was properly created in the model. Figure 4.10 shows the interaction viewer screen of the model, where the red line shows the bonding location between the hull and the top-hat structure.

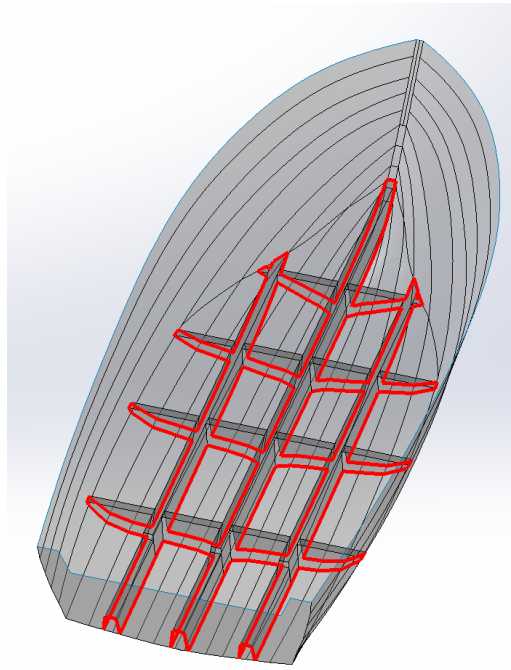


Figure 4.10: Bonding locations between the hull and the top-hat structure of the model representing Pescador 500 vessel.

Similar loads and boundary conditions were applied in this model as the ones shown for the sensitivity analysis in Table 3.33, hull supported at three points with inertia relief. Figure 4.11 shows the location of the supports implemented for this model as well as the surface area where the pressure was applied.

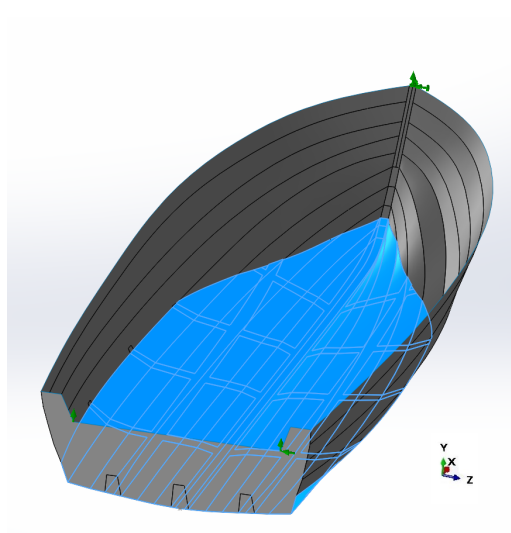


Figure 4.11: Supports location and area where pressure was applied in the model representing Pescador 500 vessel.

Finally, regarding the mesh definition, an element size of 100 mm was adopted based on the results of the study presented in Section 3.3. Figure 4.12 illustrates the discretized model, which

consists of 11 070 nodes and 5 302 elements.

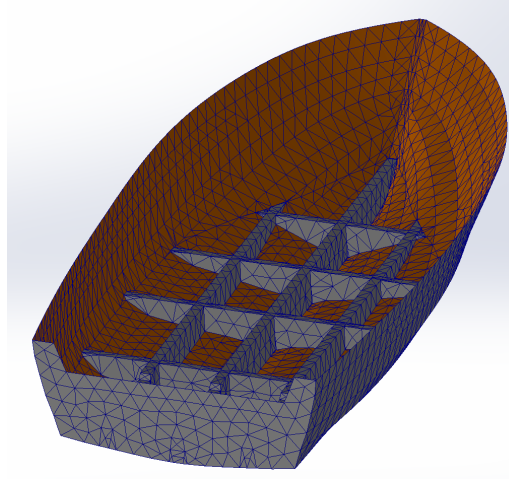


Figure 4.12: Mesh discretization of SolidWorks model of Pescador 500

After finalizing the mesh discretization, the analysis was executed, and the corresponding result files were evaluated. Figure 4.13 presents the contour plot of the von Mises stress distribution, on the left, and the displacement in the Z-direction, on the right. Note that the von Mises stress was used for this comparison because it provides a single scalar measure of the stress state and typically yields conservative values compared to individual stress components. Moreover, the SolidWorks software does not offer a stress output in the coordinate system of the composite material, like the Abaqus FEM package. To provide a clearer visual interpretation, the stress contour plot was capped at 1.5 MPa, with values above this limit represented in grey.

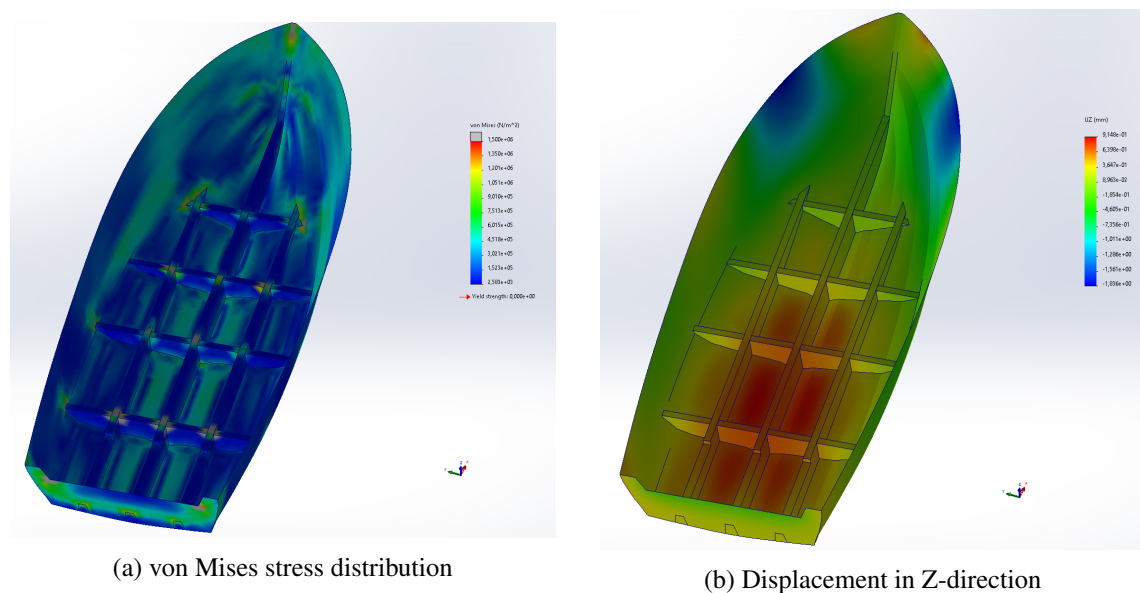


Figure 4.13: Von Mises stress and displacement in Z-direction contour plots for Pescador 500 vessel's analysis.

Using the "Probe value" feature and having the selection sets corresponding to the panels previously created and identified, it was possible to identify the nodes with higher values of stress. To avoid stress concentration nodes, the 5th highest stress values were selected from each panel.

Parti

4.1.2.2 Abaqus Finite Element Analysis

To perform the analysis in Abaqus, the CAD model was imported as a ".step" file. From this file, both the hull and the top-hat stiffeners were generated, as illustrated in Figure 4.14.

Since the hull geometry had already been partitioned, the definition of the panels as sets was carried out immediately. The creation of these sets facilitates subsequent data extraction, particularly for evaluating the maximum stress values in each panel.

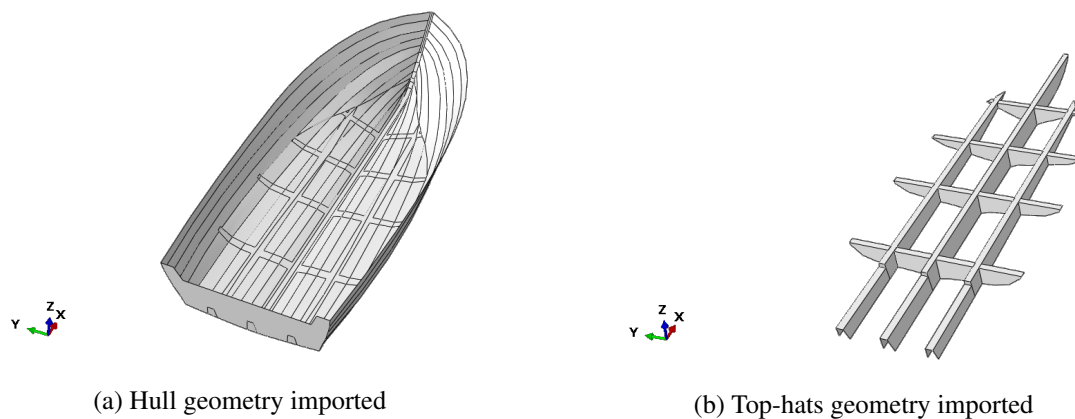


Figure 4.14: Components of Abaqus model for Pescador 500 vessel.

The material properties and total laminate thicknesses are identical to those defined for the SolidWorks models, which can be retrieved from Table 4.3. The only difference in this case is that the material was modeled using the "Composite Layup" feature, which allows the definition of the individual ply thicknesses and fiber orientations. The principal fiber direction was defined along the longitudinal axis of the vessel, lying in the plane of the hull shell, also defined as the 1-direction. The perpendicular to the 1-direction, also lying in the hull shell, is designated by the 2-direction. In this case, as this material is composed of CSM layers only and as they were considered isotropic, the use of this feature is equivalent to assuming a uniform shell material with the same elastic properties and equivalent thickness, as is the case for the SolidWorks FEA.

The definition of interactions does not rely on an automatic detection feature to identify the components to be bonded. Instead, the corresponding surfaces or edges of the hull and top-hat stiffeners must be manually selected and connected using the "Tie" constraint. Figure 4.15 illustrates the edges selected for the application of the "Tie" feature.

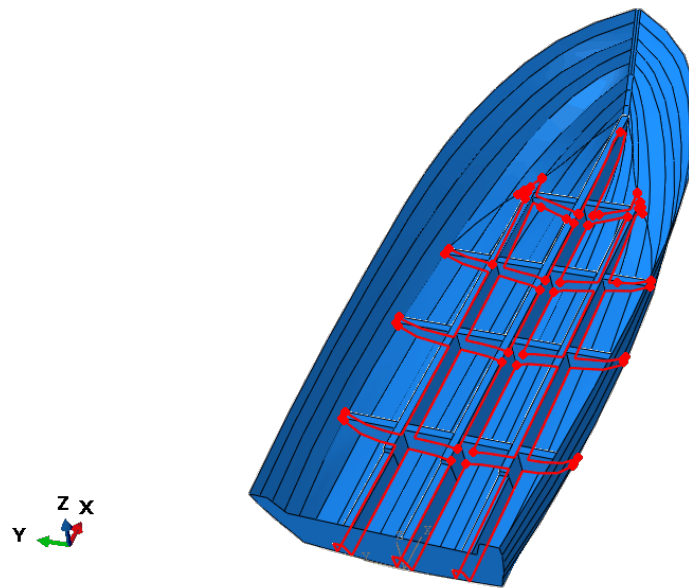


Figure 4.15: Tie location in the Abaqus model of Pescador 500 vessel.

The boundary conditions are identical to those defined for the SolidWorks models. Figure 4.16 illustrates the support locations in accordance with Table 3.33, as well as the applied pressure area.

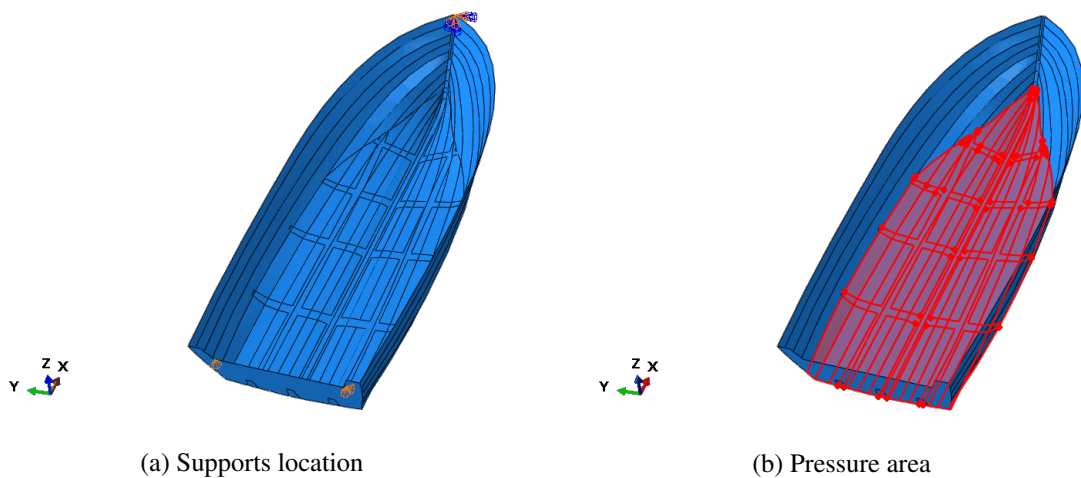


Figure 4.16: Boundary conditions applied to Abaqus model for Pescador 500 vessel.

The finite element discretization was generated using S8R elements with a reference element length of 25 mm. The resulting mesh, shown in Figure 4.17, comprises 67 351 nodes and 22 464 elements.

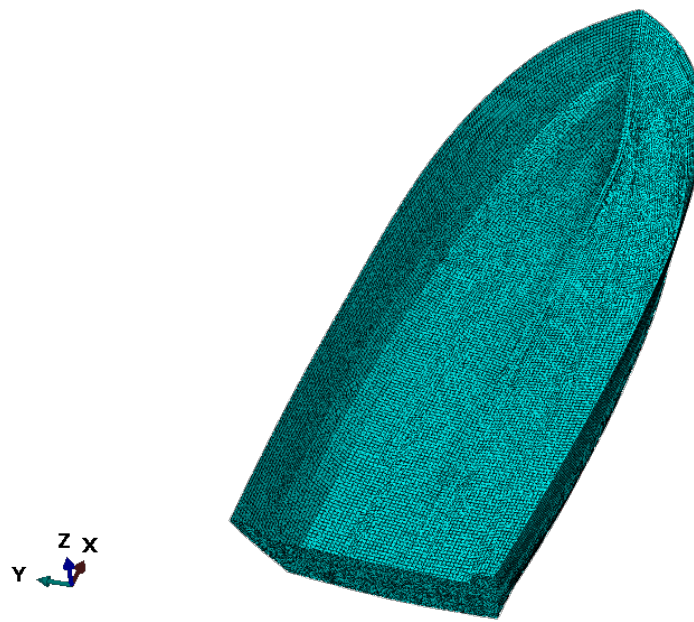


Figure 4.17: Mesh discretization for the Abaqus model of Pescador 500 vessel.

Finally, after meshing was concluded, the analysis could be submitted. For the stress comparison, S11 and S22 were analyzed. These values correspond to the normal stress in the previously attributed fiber direction and the direction perpendicular to the fiber, respectively, which also relate directly to the longitudinal and transverse lamina stresses defined by the ISO methodology.

The stress contour plot comparison is shown in Figure 4.18. The displacement in the Z-direction is also shown in Figure 4.19.

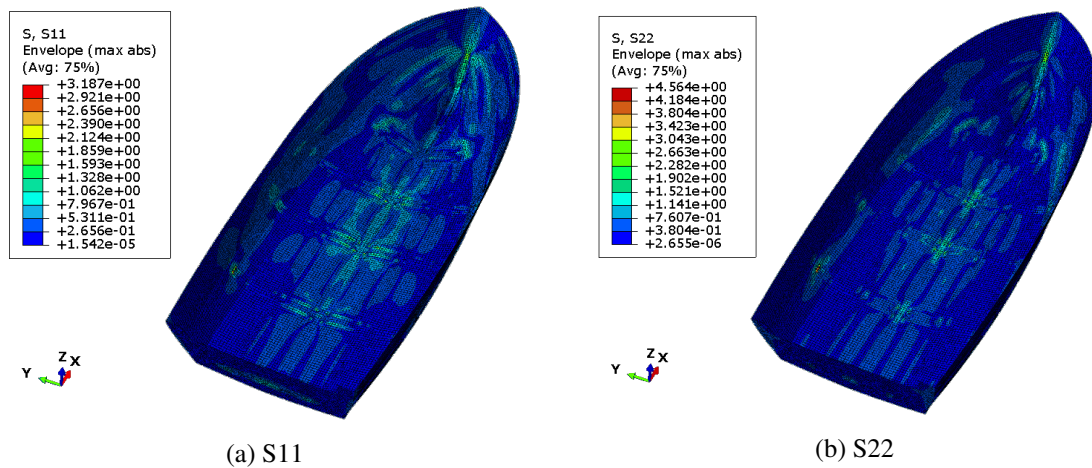


Figure 4.18: Stress distribution in the 1-direction (left) and in the 2-direction (right), obtained from the Abaqus analysis of the Pescador 500 vessel.

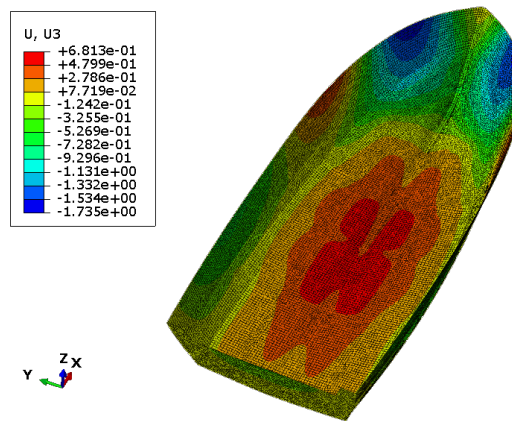


Figure 4.19: Displacement in the normal direction of the hull surface, obtained from the Abaqus analysis of the Pescador 500 vessel.

With the set definition of the panels, made in the beginning of this process, it is possible to visualize the stress distribution and displacement in each panel. Figures 4.20 and 4.21 show the behavior of panel A1 regarding stress and displacement.

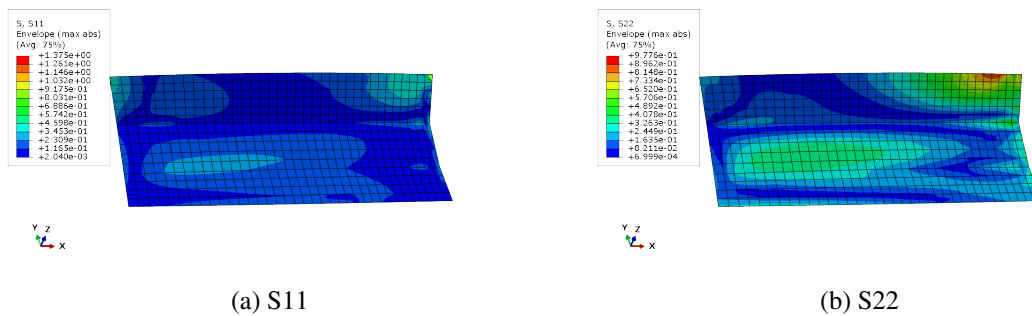


Figure 4.20: Stress contour plot, in the panel A1, in the 1-direction (left) and 2-direction (right), from the Pescador 500 vessel.

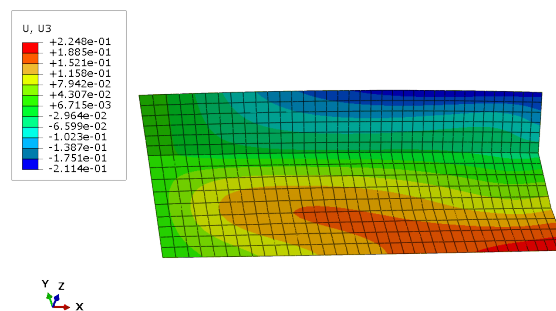


Figure 4.21: Displacement in the normal direction of the panel A1, from the Pescador 500 vessel.

As shown in Figure 4.20, the maximum stress values in both the 1- and 2-directions occur along the edges of the panel. Furthermore, especially in the stress analysis in the 1-direction, additional analyses using progressively refined meshes showed that stress peaks at these locations

increased significantly with mesh refinement. This behavior is characteristic of stress singularities, which occur in linear elastic finite element analyses at sharp geometric discontinuities or at regions where boundary conditions impose abrupt constraints [57].

In the present model, these stress concentrations occur at shell-supported intersections, where idealized constraints and “Tie” interactions enforce displacement compatibility between components. Such modeling assumptions may restrict load transfer in a manner that does not fully represent the physical behavior of the real structure. In practice, connections between internal structural members and the hull include geometric radii and load redistribution mechanisms that mitigate sharp stress gradients. The absence of these features in the numerical model, combined with the presence of sharp corners, contributes to artificially elevated and mesh-dependent stress values that are not representative of the real behavior of the structure.

Therefore, in order to obtain meaningful stress results, a Python script was developed to extract the maximum stresses in the 1- and 2-directions of all panels while excluding the outer ring of elements. This procedure ensures that the reported stress values correspond to the interior region of the panel, where the stress field should be smooth, mesh-convergent, and representative of the actual bending and membrane behavior of the structure.

4.1.3 Results and Comparison

To present the results, Table 4.5 summarizes the results obtained using the different methodologies for each considered panel. In this table, CF_{Simp} represents the compliance factor determined using the Simplified method. The term CF_{Enh} represents the compliance factor calculated using the Enhanced method. The parameters $CF_{Abaq,S11}$ and $CF_{Abaq,S22}$ correspond to the compliance factors obtained from the Abaqus analysis for stresses in the 1- and 2-directions, respectively. Finally, $CF_{SW,vM}$ represents the compliance factor calculated using SolidWorks, based on the von Mises stress criterion. Appendix C presents detailed tables (Tables B.3, B.4 and B.5) containing the stress values determined analytically or extracted from the software, together with the calculated terms used in the determination of the compliance factors.

Table 4.5: Comparison of calculated compliance factors for the Pescador 500 vessel.

Panel ID	CF_{Simp}	CF_{Enh}	$CF_{Abaq,S11}$	$CF_{Abaq,S22}$	$CF_{SW,vM}$
A1	2.26	3.16	6.72	4.12	9.59
A2	1.48	2.25	5.85	4.42	4.07
B3	1.59	1.75	6.01	3.58	7.64
B4	1.34	1.82	3.32	2.28	1.53
C5	1.38	1.48	4.87	4.02	5.94
C6	1.23	1.44	2.39	1.49	1.28
D7	1.88	2.55	2.95	1.52	2.53
D8	1.12	1.24	2.16	1.36	1.82
E9	1.35	1.75	2.08	1.42	2.40

Regarding the results obtained analytically, the ones calculated using the Simplified method are identical to those presented in the original scantling report. This confirms that the analytical procedure implemented in this work is fully consistent with the methodology previously applied and validates the correct interpretation of the ISO 12215-5 equations, panel geometry, pressure coefficients, and material inputs.

The comparison between the Simplified and Enhanced methods shows that the compliance factors are higher for the Enhanced method. Since the compliance factor is defined as the ratio between allowable stress and calculated stress, higher compliance values indicate that the Enhanced method predicts lower panel stresses than the Simplified method. Therefore, the Enhanced formulation presents itself as a less demanding structural assessment, which is expected due to being a more detailed analysis of the panel behavior in comparison with a more conservative scantling determination method like the Simplified one.

From the Abaqus results, also shown in Table B.4, it can be observed that the stress values in the 2-direction are consistently higher than those in the 1-direction for all analyzed panels. In the case of this vessel, the 2-direction corresponds to the b -direction (the shortest panel dimension). This behavior is in full compliance with the ISO 12215-5 premise, which assumes that bending maximum stresses are governed by the shortest span of the panel. Therefore, the numerical results confirm that the structural response is dominated by bending in the b -direction, as predicted analytically.

The compliance factors obtained with Abaqus are generally higher than those calculated analytically, with the exception of panel D7 and panel E9 in the 2-direction. This indicates that Abaqus predicts slightly lower stresses than the analytical method for most panels, which confirms that, in general, the analytical methods calculate more conservative results than the ones obtained with the a FEM package.

Analyzing the exceptions, the reason behind the behavior of the panel D7 might be due to the location of the panel in the hull, which can be seen better in Figure 4.6, the fact that the panel is not supported with the top-hat structure on the lengthiest of its edges, and the actual geometry of the panel. Figure 4.22 demonstrates D7 panel's stiffener support, and Figure 4.23 shows the stress contour plot in panel D7 along the 2-direction, obtained with Abaqus software, with the location of the critical node shown with a red dot.

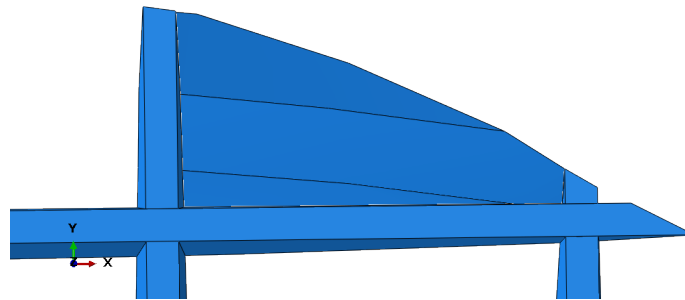


Figure 4.22: Panel D7 of Pescador 500 vessel and its structural support.

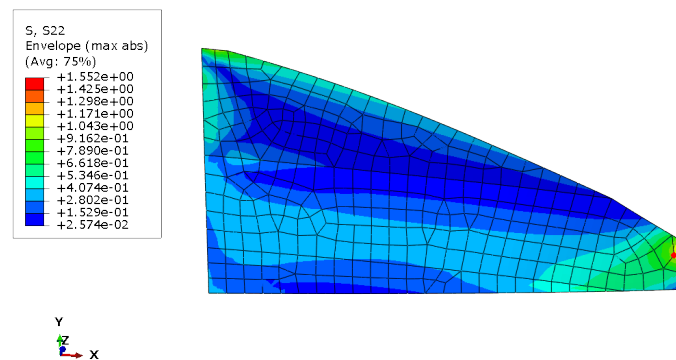


Figure 4.23: Stress contour plot in panel D7 of the Pescador 500 vessel along the 2-direction, calculated with the Abaqus software, with the location of the critical node shown with a red dot.

Upon closer inspection, the geometry of the D7 panel is more complex than the idealized rectangular plate assumed in the ISO 12215-5 analytical formulation. The ISO method simplifies the panel as a uniformly supported rectangular element with well-defined fixed edges. However, in the finite element model, panel D7 is not completely supported.

Furthermore, the extracted S22 normal stress values are taken from a node near the edges of the panel, close to where the top-hat structure ends. This region appears to be a stress concentration zone, where the loads are not being properly distributed from the hull surface, where the pressure is being applied, to the internal structure.

Regarding panel E9 of the Pescador 500 vessel, its geometry and structural configuration are defined in Figure 4.24, while its stress contour plot in the 2-direction is shown in Figure 4.25. This panel location in the vessel is better observed in Figure 4.6.

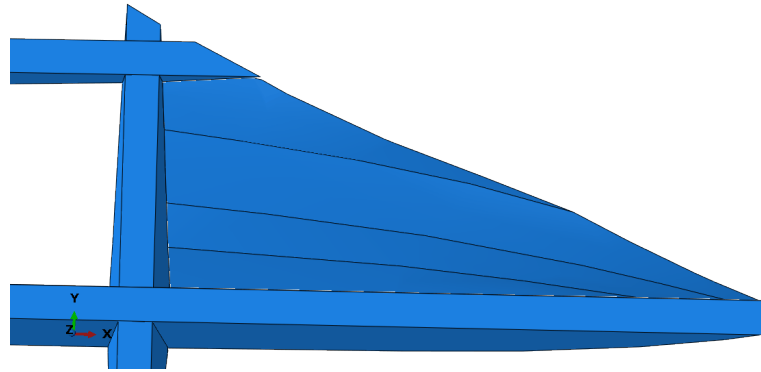


Figure 4.24: Panel E9 of Pescador 500 vessel and its structural support.

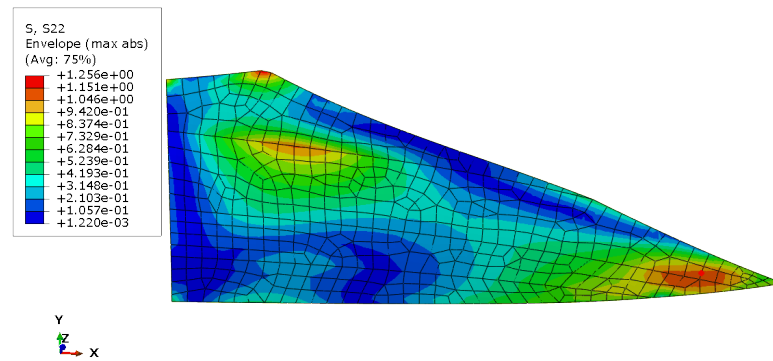


Figure 4.25: Stress contour plot in panel E9 of the Pescador 500 vessel along the 2-direction, calculated with the Abaqus software, with the location of the critical node shown with a red dot.

Panel E9, similarly to panel D7, exhibits a complex geometry, being close to a triangular shape. Furthermore, due to its position in the hull, it has very complicated curvature at the bow end of the panel, which is coincident with the highest stress location and the node from which the S22 values were extracted. Contrary to panel D7, this stress concentration should not be disregarded, as it is not situated at the edge, appearing to be a bending-related phenomenon.

Therefore, the lower compliance factor obtained numerically for panels D7 and E9 does not indicate inconsistency in the ISO method but rather highlights the influence of real geometric complexity. This case demonstrates the ability of FEM to identify localized stress amplification effects that are not fully represented in the simplified analytical model.

The analysis here presented also shows why panels must be analyzed thoroughly when exhibiting lower compliance factors for numerical methods than analytical methods. Even though both panels display complex geometry, panel's D7 numerical results could be attributed to unreliable stress concentration.

Regarding the results obtained with the SolidWorks FEM package, a first general conclusion is that all panels are evaluated as compliant. The compliance values calculated with the stress results from this software are, in general, not very far from the ones obtained with the Abaqus software.

The discrepancies found between both software results could be attributed to solver formulation, stress recovery procedures, and the use of von Mises stress in SolidWorks instead of directional stress components.

Nevertheless, despite these numerical differences, SolidWorks proves capable of correctly capturing the relative panel criticality, confirming its suitability for preliminary structural assessment.

4.2 Case Study 2 - Evolution 5.0

The second vessel, and the primary object of study of this work, is the Evolution 5.0, whose CAD model is shown in Figure 4.26. This vessel is currently under development, and by applying the methodology previously validated through the results obtained for the Pescador 500, it will be possible to provide a recommendation regarding the material to be used in the hull bottom.

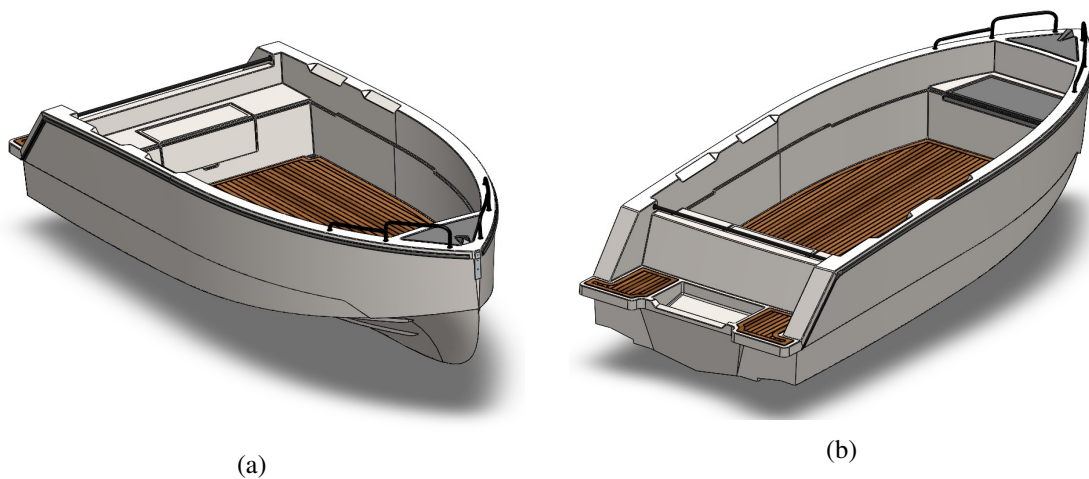


Figure 4.26: CAD model of the Evolution 5.0 vessel.

Although the material allocation is still under investigation, the final geometry has already been defined, as the molds for the vessel have been fabricated (Figures 4.27, 4.28, and 4.29). Three separate molds were produced, since the Evolution 5.0 will be manufactured through the assembly of three primary components: the hull, the internal structure, and the deck.

These components will be joined using structural adhesive, as their design incorporates multiple interfacing surfaces that provide mutual support. The vessel will initially be manufactured using the hand lay-up process; however, a transition to resin infusion is planned for future production.



Figure 4.27: Mold of the hull of the Evolution 5.0 vessel.

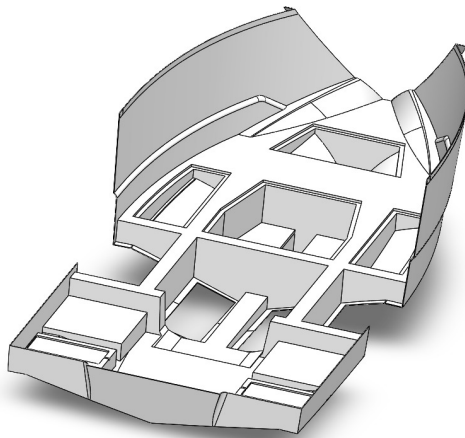


Figure 4.28: Mold of the deck of the Evolution 5.0 vessel.

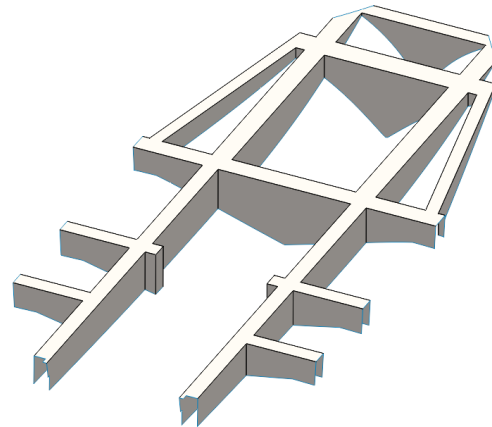


Figure 4.29: Mold of the internal structure of the Evolution 5.0 vessel.

To facilitate the assessment of the most appropriate hull material configuration, a simplified CAD model of the internal structure was developed by the Riamar company design team, shown in Figure 4.30. This simplified configuration was intentionally defined to be structurally less stiff than the actual internal structure.



(a) Representation of internal structure



(b) Simplified internal structure

Figure 4.30: Internal structure designs of Evolution 5.0 vessel.

Additional geometric simplifications were introduced to the hull design in order to facilitate the FEA, as illustrated in the comparison presented in Figure 4.31. This simplified geometry also served as the reference model for the panel geometry assessment. The simplification process involved the removal of natural stiffening features that contribute to the overall rigidity of the hull structure.

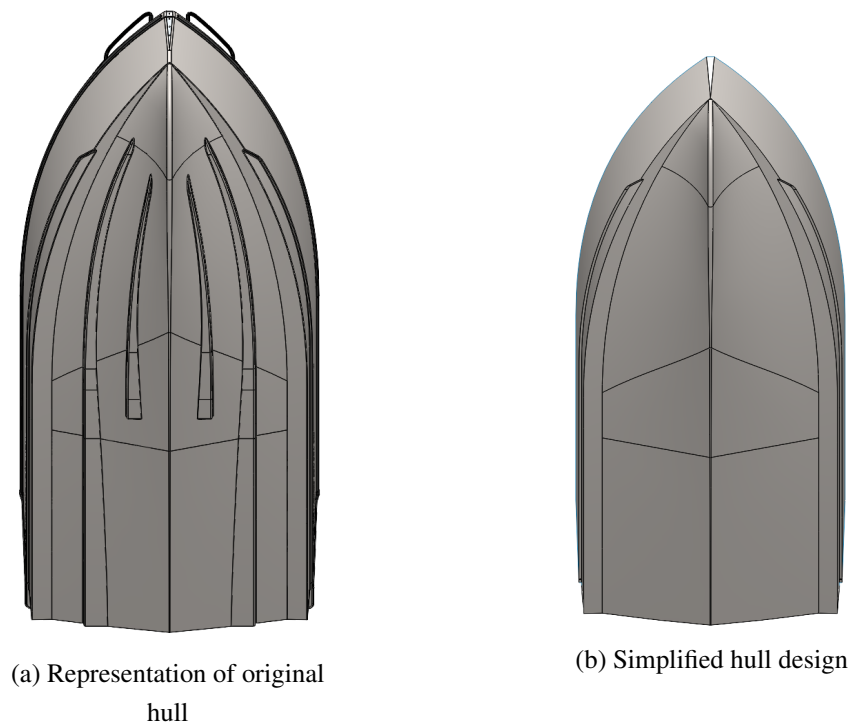


Figure 4.31: Hull designs of the Evolution 5.0 vessel.

By adopting this conservative approach, if the results demonstrate satisfactory compliance for the simplified (weaker) configuration, it can be concluded with confidence that the final, more robust combination of the internal structure and hull will also meet the structural requirements.

4.2.1 ISO 12215-5 Analytical Scantling

Similarly to the assessment of the Pescador 500 vessel, to perform the calculations that validate the use of the "Rovimat" material in the bottom of the hull of Evolution 5.0 analytically, several variables must be defined to use them as inputs in the developed tool. In this case all variables were established by analyzing the CAD of the vessel and applying the procedure described in Section 3.2 or by indications of the Riamar company engineers.

4.2.1.1 Geometric and Design Parameters

Most of the key parameters required for the ISO 12215 assessment depend on the vessel's waterline position. Therefore, an initial estimation of the loaded displacement mass, m_{LDC} , was required. Using the Mass Properties feature in SolidWorks and assigning an average material density of 1600 kg/m^3 to the CAD model shown in Figure 4.26, the structural mass of the vessel was estimated at 638 kg. By adding the mass of the engine and six occupants, assuming an average mass of 75 kg per person, the loaded displacement mass was estimated as $m_{LDC} = 1,183 \text{ kg}$.

With this value defined, the waterline height could be determined by applying Archimedes' principle, which states that a body wholly or partially immersed in a fluid is subjected to an upward

buoyant force equal to the weight of the displaced fluid. To implement this approach, the CAD model of the Evolution 5.0 was modified into a fully solid (non-hollow) representation, and its material density was reassigned to that of seawater ($\rho = 1050 \text{ kg/m}^3$).

Using the "Section View" feature, the model was progressively sectioned until the mass of the submerged volume matched the previously calculated loaded displacement mass. The resulting waterline height, also called canoe body draft T_C , was determined to be 0.295 m, as illustrated in Figure 4.32. Furthermore, the length and beam at the waterline could also be measured from the model, shown in Figure 4.33, with $L_{WL} = 4.854 \text{ m}$ and $B_{WL} = 1.944 \text{ m}$.

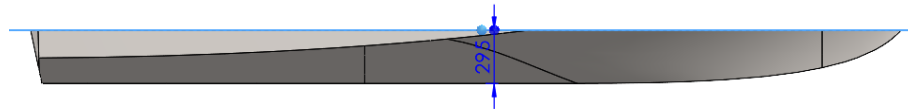


Figure 4.32: Canoe body draft T_C measurement of the Evolution 5.0 vessel.

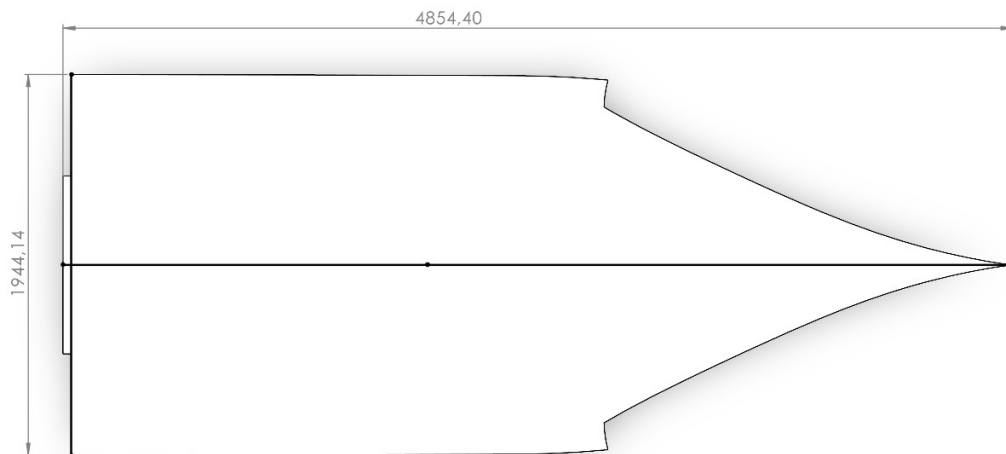


Figure 4.33: Length of waterline L_{WL} and beam of waterline B_{WL} of the Evolution 5.0 vessel.

Additionally, by creating a sectional view of the model at a longitudinal position of $0.4, L_{WL}$ measured from the aft end, the deadrise angle and chine beam were determined. The deadrise angle was measured as $\beta = 12.5^\circ$, and the chine beam was calculated as $B_C = 2 \times 0.9638 = 1.928 \text{ m}$, shown in Figure 4.34.

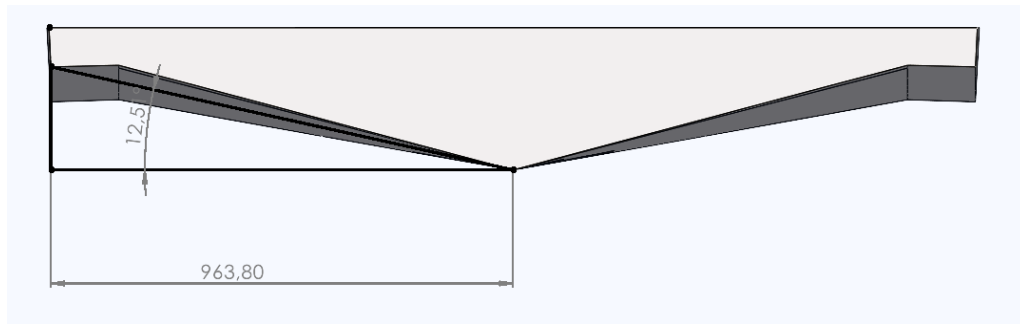


Figure 4.34: Deadrise angle β and chine beam B_C of the Evolution 5.0 vessel.

Table 4.6 summarizes all the relevant information about this vessel. A maximum speed of 29 knots was assumed for this work.

Table 4.6: Main characteristics of the Evolution 5.0 vessel.

Mass at loaded condition, m_{LDC} [kg]	1183.0
Length of the hull, L_H [m]	5.167
Waterline length, L_{WL} [m]	4.854
Waterline beam, B_{WL} [m]	1.944
Chine beam, B_C [m]	1.928
Canoe body draft, T_C [m]	0.295
Deadrise angle, β [°]	12.5
Maximum speed, V [knots]	29.0

The geometric characteristics of the panels were determined individually and are detailed in Appendix C, in Table C.1. Figure 4.35 shows the panel identification and location of the panels analyzed.

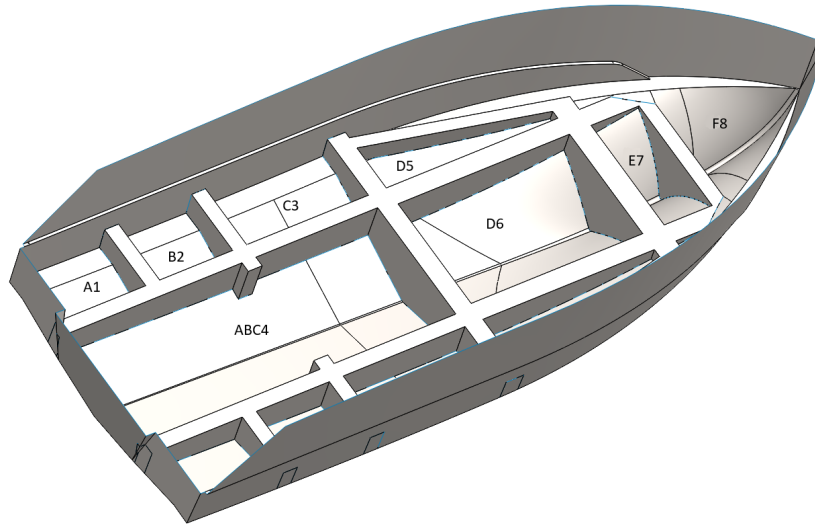


Figure 4.35: Evaluated panels identification and location of the Evolution 5.0 vessel.

4.2.1.2 Pressure Adjusting Factors and Design Pressures

Regarding the pressure adjusting factors and design pressure for each panel, they are shown in Table 4.7.

Table 4.7: Design pressure and panel-dependent coefficients for the Evolution 5.0 vessel.

Panel	k_L	k_{AR}	Design Pressure [kN/m^2]
A1	0.633	0.419	14.7
B2	0.768	0.448	19.0
C3	0.870	0.383	18.4
ABC4	0.750	0.290	12.0
D5	1.000	0.353	24.4
D6	1.000	0.442	19.5
E7	1.000	0.481	26.6
F8	1.000	0.422	23.3

This vessel belongs to design category C, similarly to Pescador 500, being then $k_{DC} = 0.6$. The calculated dynamic load factor is $k_{DYN} = 4.355$.

4.2.1.3 Mechanical Properties and Design Stresses

The material defined for the bottom of the hull of the Evolution 5.0 vessel is the "Rovimat" material. Therefore, this will be the material utilized in the analytical scantling calculations for this vessel. In that order, the necessary data to apply the ISO 12215-5 analytical methods was retrieved from Table 3.23.

Regarding the design stress, Table 4.8 shows the calculated values for the "Rovimat" material, determined similarly to the ones utilized for the Pescador 500 vessel, depending on the assessment method. Table C.2, in Appendix C, presents the design stresses defined for each laminate layer for the Enhanced method and for the finite element method. The boatbuilding factors k_{BB} and the assessment method factors k_{AM} are the same as the ones considered for the Pescador 500 vessel, as these factors are not dependent on the vessel.

Table 4.8: Design stresses calculated for each assessment method of the Evolution 5.0's hull material, the "Rovimat" material.

Composite material	$\sigma_{d_{SIMP}}$ [MPa]	$\sigma_{d_{ENH}}$ [MPa]	$\sigma_{d_{FEM}}$ [MPa]
"Rovimat" Material	87.9	92.8	97.7

4.2.2 Finite Element Model Analysis

To perform the FEA of the Evolution 5.0 vessel model, a very similar procedure was taken to the analysis performed to the Pescador 500 boat. The initial and complex CAD design of the vessel was simplified, and unnecessary boat components were removed from the model, leaving the hull of the vessel and the simplified representation of its internal structure, which are shown in Figure 4.36.

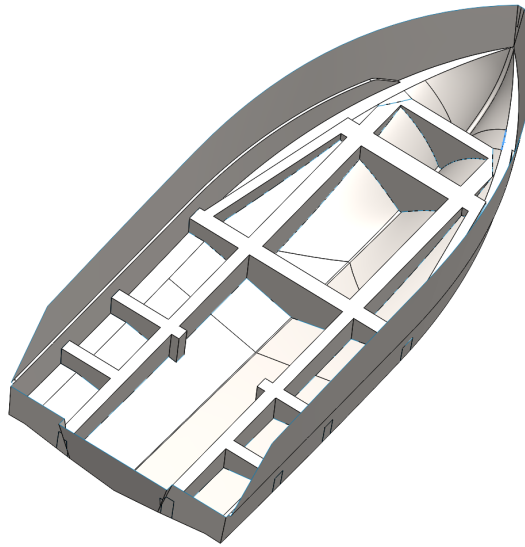


Figure 4.36: Model considered for the FEA of the Evolution 5.0 vessel.

Furthermore, the partitioning of the model to create the panels design in the hull was also carried out, which is shown in Figure 4.37, as it was done for the Pescador 500 vessel study.

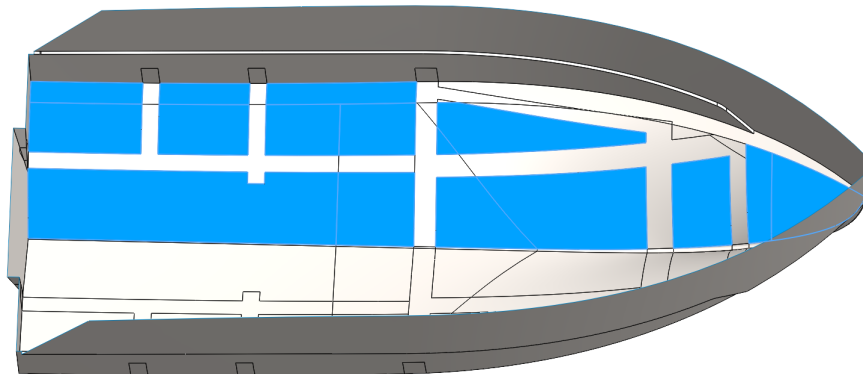


Figure 4.37: Partitioning and panels geometry considered for the FEA of the Evolution 5.0 vessel.

4.2.2.1 SolidWorks Finite Element Analysis

For the SolidWorks FEM package implementation, the first step was once again defining the material properties and thicknesses of the components of the models. For this vessel study, the laminate properties considered for the bottom of the hull are in accordance with Table 3.23. Regarding the internal structure and the side walls of the hull and transom, as it was mentioned in chapter 3, the "Mat" material is the one expected to be used. Therefore, this location's material was modeled based on the properties shown in Table 3.24. Note that due to the software's incapability

of modeling composite materials accurately, both materials were defined as uniform isotropic with only E , ν and ρ defined.

The interactions were set to bond the internal structure to the hull surface. This time additional local interactions add to be defined as the software did not recognize all the bonding edges. Figure 4.38 show the "Interaction viewer" screen that shows the edges that are bonded between the internal structure and the hull.

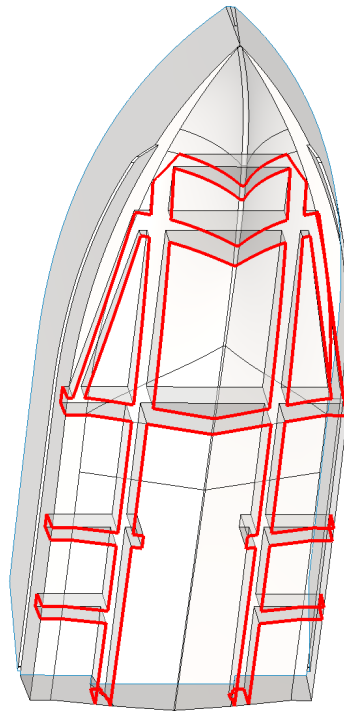


Figure 4.38: Bonding locations between the hull and the developed internal structure of the model of the Evolution 5.0 vessel.

The loads and boundary conditions are like the ones used for Pescador 500 vessel analysis, described in Table 3.33, with three nodes supporting the hull, inertia relief and pressure applied to the bottom of the hull. Figure 4.39 shows the supports location, as well as the area where pressure was applied in the model.

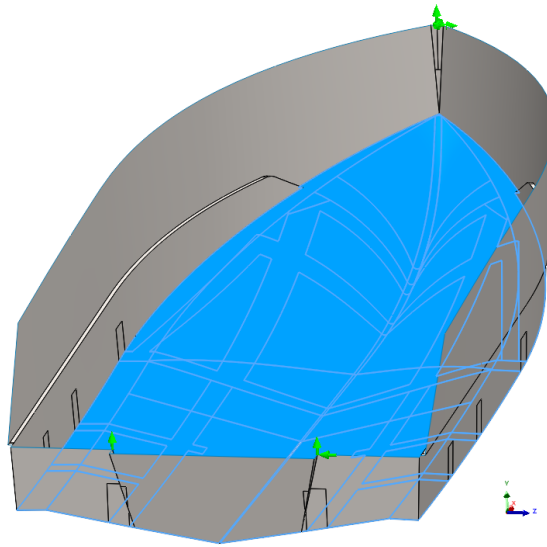


Figure 4.39: Supports location and area where pressure is applied in the model of the Evolution 5.0 vessel.

An average element size of 100 mm was adopted for the mesh discretization, resulting in a total of 30 965 nodes and 15 199 elements, as illustrated in Figure 4.40.

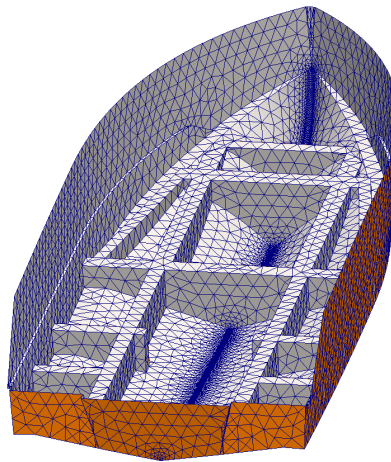


Figure 4.40: Mesh discretization of the model Evolution 5.0 vessel, in the SolidWorks software.

After finalizing the mesh discretization, the analysis could be submitted. Figure 4.41 present the von Mises stress contour plot obtained with the SolidWorks FEM package. Once again, the von Mises stress was used to visualize the stress field, as it provides a single scalar measure of the stress state, facilitating a more straightforward comparison of results. Again, the stress contour plot was capped at 1.5 MPa, with values above this limit represented in grey for clearer visual interpretation.

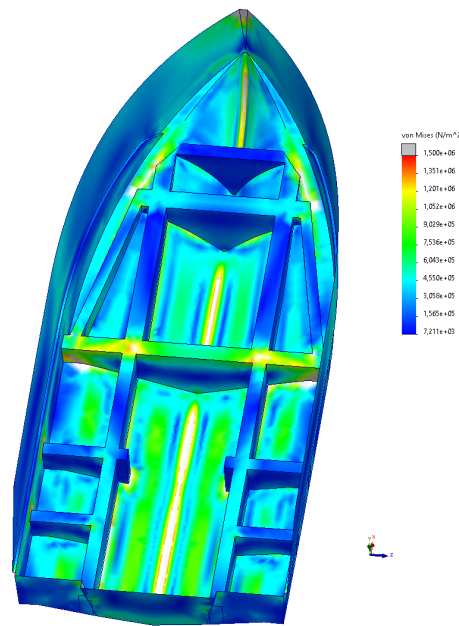


Figure 4.41: Von Mises stress contour plot for Evolution 5.0 vessel, obtained with SolidWorks FEM package.

4.2.2.2 Abaqus Finite Element Analysis

For the application of the Abaqus software the procedure is identical to the one completed for the Pescador 500 vessel. The model was imported to Abaqus, resulting in two parts, shown in Figure 4.42. Also, sets of the panels were created to be easier to retrieve the data later on.

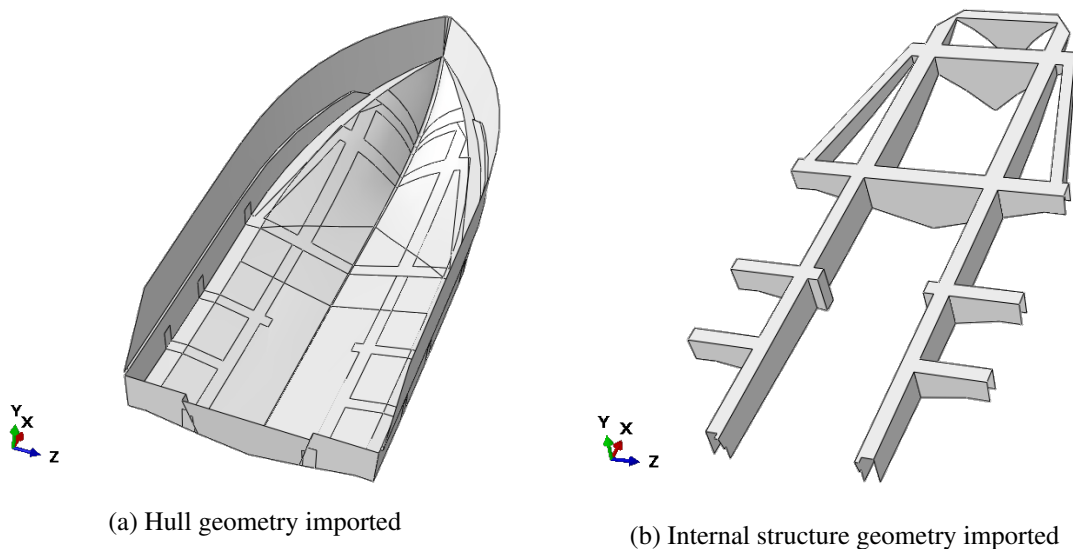


Figure 4.42: Components of Abaqus model for Evolution 5.0 vessel.

As for the SolidWorks FEA of the Evolution 5.0 vessel, the "Mat" material was considered for the side wall of the hull, transom, and internal structure, while "Rovimat" material was applied to

the bottom of the hull region, with both materials' properties retrieved from Tables 3.24 and 3.23, respectively.

Similarly to the other case study, the composite materials were modeled with the "Composite Layup" feature in the Abaqus software. This feature was particularly important to define the RVM layer of the "Rovimat" material, which was defined as an orthotropic lamina material, due to the anisotropy of the layer, with the input properties shown in Table 4.9. The values of $E_1 = E_2$, ν_{12} and G_{12} are equivalent to E , ν and G values shown in Table 3.23 for the RVM layer. As the values of G_{13} and G_{23} were not calculated analytically, the same value as G_{12} will be assumed, which should not affect the results greatly.

Table 4.9: Lamina material properties defined in Abaqus for the RVM layer of "Rovimat" in the FEA of the Evolution 5.0 vessel, adapted from Table 3.23.

Lamina definition properties	E_1 [MPa]	E_2 [MPa]	ν_{12} [1]	G_{12} [MPa]	G_{13} [MPa]	G_{23} [MPa]
RVM layer	14 149	14 149	0.30	3 380	3 380	3 380

The bonding of the hull to the internal structure was performed by selecting the common edges of both parts and using the "Tie" feature. The bonding edges are shown in Figure 4.43.

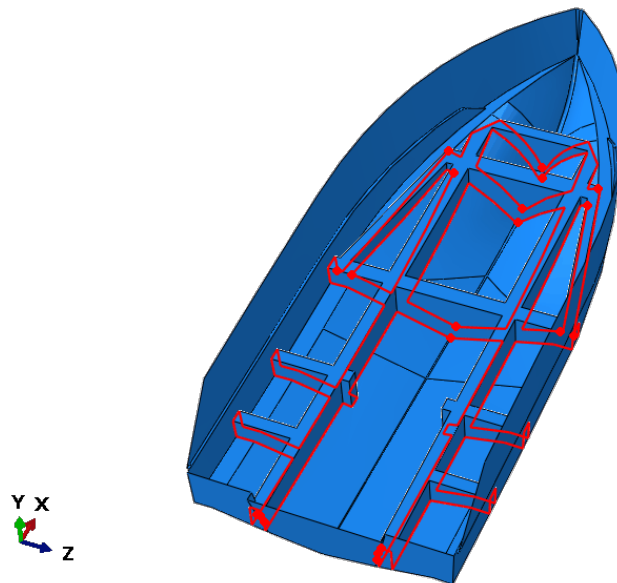


Figure 4.43: "Tie" location in the Abaqus model of the model Evolution 5.0 vessel.

Regarding the boundary conditions, they are depicted in Figure 4.44, consisting of the same 3-node support, two at the transom and one at the bow, the pressure applied at the bottom of the hull, and inertia relief.

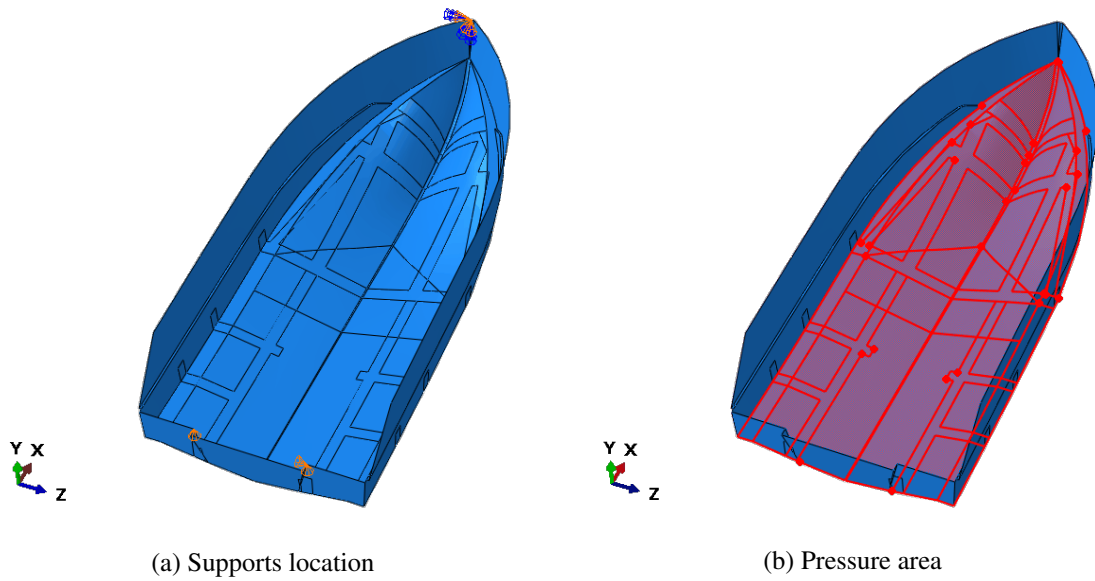


Figure 4.44: Boundary conditions applied to Abaqus model for Evolution 5.0 vessel.

Finally, the mesh discretization, illustrated in Figure 4.45, followed the previously defined combination of element type and size. The model consists of 99 220 nodes and 33 183 S8R elements, with an average element size of 25 mm.

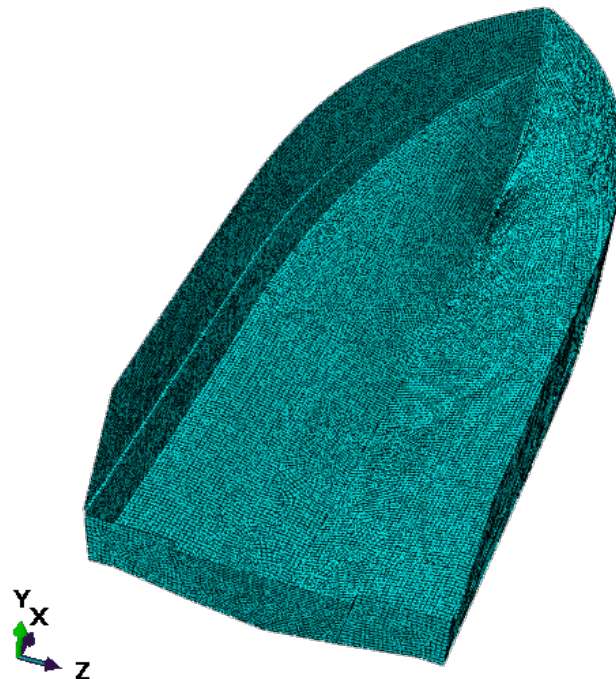


Figure 4.45: Mesh discretization for the Abaqus model of the Evolution 5.0 vessel.

After meshing, the analysis could be submitted. Figure 4.46 shows the stress contour plot of

the hull of the Evolution 5.0 vessel in the 1-direction, on the left, and in the 2-direction, on the right, while Figure 4.47 presents the displacement in the Y-direction, obtained from the Abaqus analysis of the Evolution 5.0 vessel.

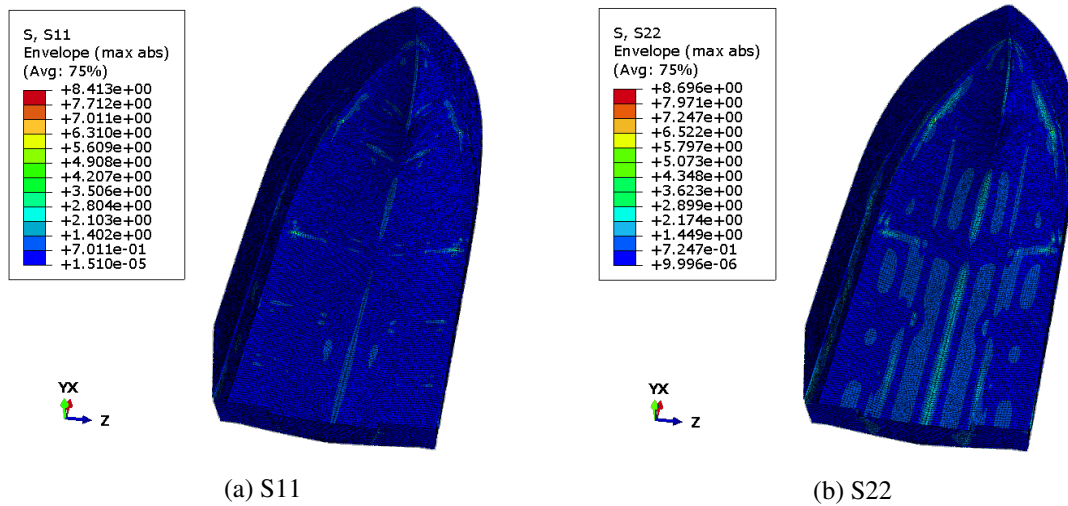


Figure 4.46: Stress distribution in the 1-direction (left) and in the 2-direction (right), obtained from the Abaqus analysis of the Evolution 5.0 vessel.

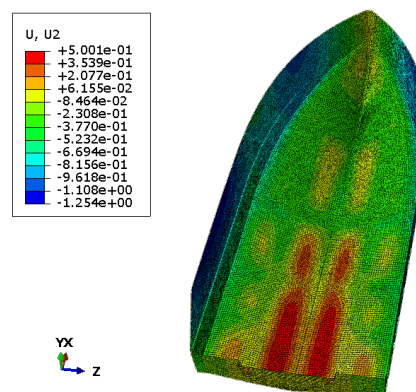


Figure 4.47: Displacement in the Y-direction, obtained from the Abaqus analysis of the Evolution 5.0 vessel.

The individual panel analysis was also enabled by the previously defined sets for data extraction. Figure 4.48 shows the stress contour plot of panel A1 of the Evolution 5.0 vessel in the 1-direction, on the left, and in the 2-direction, on the right, while Figure 4.49 presents the displacement in the Y-direction, obtained from the Abaqus analysis of the Evolution 5.0 vessel.

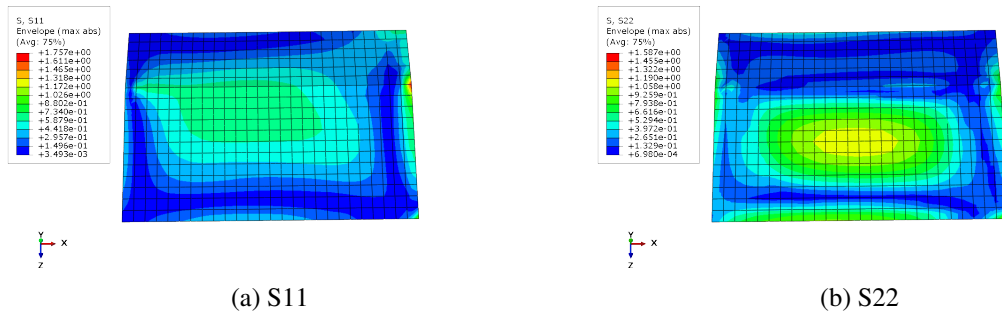


Figure 4.48: Stress contour plot, in the panel A1, in the 1-direction (left) and 2-direction (right), from the Evolution 5.0 vessel.

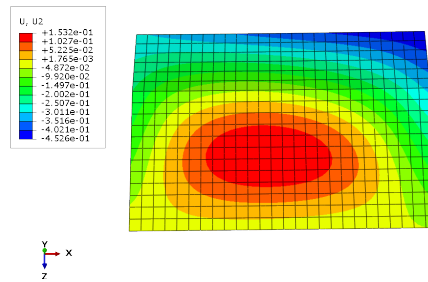


Figure 4.49: Displacement in the Y-direction of the panel A1 from the Evolution 5.0 vessel.

From the panel figures, a bending-dominated behavior can be clearly identified. An outer band of moderate stress is visible along the constrained edges, followed by a gradual transition towards the interior region, where it increases again. The stress field in the central area appears relatively uniform, which is consistent with the location of maximum displacement.

Localized stress peaks are also observed at the boundary nodes. Therefore, the Python script previously developed to exclude the outer ring of elements from the analysis will be reused for the evaluation of the Evolution 5.0 model results, ensuring that boundary-induced numerical artifacts do not influence the interpretation of the panel response.

4.2.3 Results and Comparison

Table 4.10 presents the compliance results obtained with the different methodologies. In Appendix C, Tables C.3, C.4 and C.5 show the calculated terms and stress components used in the determination of the compliance factors.

Table 4.10: Comparison of the obtained compliance factors for the Evolution 5.0 vessel.

Panel ID	CF_{Simp}	CF_{Enh}	$CF_{Abaq,S11}$	$CF_{Abaq,S22}$	$CF_{SW,vM}$
A1	1.11	1.56	9.69	5.86	6.98
B2	1.06	1.43	10.16	5.69	5.63
C3	0.94	1.12	4.67	3.20	2.51
ABC4	1.16	1.73	3.36	2.63	3.65
D5	2.07	4.85	7.58	5.16	6.96
D6	1.22	1.51	4.20	2.68	2.88
E7	1.25	1.65	7.72	9.69	6.60
F8	2.58	3.50	2.81	3.47	3.07

Concerning the analytical results, once again, the compliance factors obtained with the Enhanced method are consistently higher than those calculated with the Simplified method. This confirms that the Simplified procedure behaves more conservatively, systematically predicting higher stress demand relative to the allowable values, as occurred with the Pescador 500 analysis.

According to the Enhanced method, the "Rovimat" laminate satisfies the ISO 12215-5 requirements, as all panels present compliance factors above 1. In contrast, the Simplified method identifies panel C3 as non-compliant. This demonstrates that, for panels operating near the limit state, the choice of analytical method can influence the compliance assessment.

Analyzing the Abaqus results, the use of the "Rovimat" material also satisfies the ISO 12215-5 requirements, as the compliance factors are greater than 1 in both the 1- and 2-directions. Similarly to the Pescador 500 case study, the stress response is generally governed by the 2-direction, resulting in lower compliance factors in this direction. The only exceptions are panels E7 and F8. In the case of panel E7, the shortest span is aligned with the 1-direction, which shifts the governing bending response accordingly. Panel F8, due to its triangular geometry, presents a different load distribution pattern, leading to a deviation from the typical directional behavior observed in the remaining panels.

In general, the compliance factors obtained from Abaqus are higher than those calculated using the analytical methods, with the exception of panel F8, reflecting, once again, the conservative nature of the ISO panel-based formulations when compared to the global numerical model.

Regarding panel F8 of the Evolution 5.0 vessel, its stress contour plot in the 1-direction is shown in Figure 4.50, with the location of the critical node shown with a red dot. This panel location in the vessel, geometry and supports are better observed in Figure 4.35.

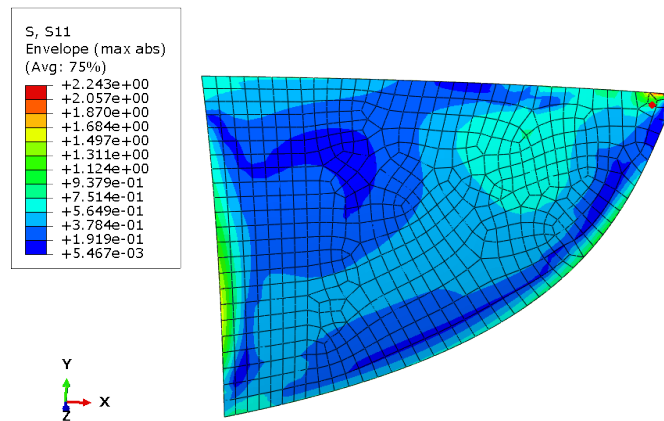


Figure 4.50: Stress contour plot in the 1-direction of panel F8 of Evolution 5.0 vessel, with the location of the critical node shown with a red dot.

This panel is similar to panel F9 of Pescador 500 vessel. It has a complex geometry, a triangular shape with curvatures that are impossible to properly characterize in the analytical tool, explaining the higher compliance factor obtained for the Enhanced method. Furthermore, the node from which the S11 normal stress value is extracted in this panel is located near a stress concentration zone at the edge of the panel, which can influence this stress value.

The SolidWorks software results also show full compliance of the application of the "Rovimat" material with the requirements of the ISO 12215-5 standard, as the $CF_{SW,vM}$ values are all above 1. In fact, these values are relatively similar to the ones calculated using the Abaqus software, which confirms that this software could be an important tool for structural design. The small differences observed between the two numerical code results could be attributed to the different modeling systems of the materials, especially regarding the RVM layer, and due to the use of the von Mises stress values.

4.3 Chapter Summary

This chapter presented the application of the developed methodology for two case studies: the Pescador 500 and the Evolution 5.0 vessels. The objective was to evaluate hull bottom laminate compliance using the ISO 12215-5 analytical methods and to compare these results with numerical simulations performed in Abaqus and SolidWorks. For the Pescador 500 vessel, only one material configuration was analyzed, as the laminate is already defined for this proven design. For the Evolution 5.0 vessel, the "Rovimat" material was evaluated for compliance with the ISO 12215-5 standard.

For each vessel, three methodologies were applied: two analytical approaches and a FEM approach. The results of the analytical Simplified and Enhanced methods were obtained using the developed analytical tool, while the results from FEM approach were computed from Abaqus and SolidWorks software.

For the Pescador 500 case study, the first step was improving the CAD model, as it was imported from Rhino to SolidWorks with some missing and poorly defined surfaces. Then, the analytical procedure described in Section 3.2 was followed to obtain the analytical results for the Simplified and Enhanced methods.

With the CAD model of Pescador 500's hull and top-hats, the FEA was performed using the FEM packages of SolidWorks and Abaqus software, as the CAD had to be imported to Abaqus as well. For the Evolution 5.0 vessel a similar global procedure was adopted, with the additional steps of defining the geometric and design properties of the vessel, like the waterline height and its dependent values.

For the Pescador 500, the analytical implementation was validated by reproducing the results from the original scantling report. The comparison between analytical and numerical approaches demonstrated that the ISO methods generally provide conservative results relative to the finite element results, as the analytically obtained compliance factor values are systematically lower than the numerical ones. Additionally, specific panels such as D7 demonstrated the limitations of the analytical scantlings, as the numerical model captured geometric irregularities and asymmetric boundary conditions that increased local stress demand.

In the case of the Evolution 5.0 analysis, assuming the "Rovimat" material application to the bottom of the hull, the Simplified method continued to show higher conservatism, as this approach consistently produced the lowest compliance factor values, being lower than 1 for panel C3. However, the Enhanced method, with a more detailed assessment, established that all panels, including panel C3, are compliant with the ISO requirements.

Regarding the numerical approach, the FEM assessment performed with the Abaqus software also shown that the laminate is compliant with ISO 12215-5 specifications. Stress response was predominantly governed by the 2-direction, consistent with bending behavior in the shortest span, with minor deviations explained by panel geometry. In general, Abaqus predicted higher compliance factors than the analytical methods, reinforcing the conservative nature of the ISO formulations.

The FEM package from SolidWorks software provided additional numerical validation. The results obtained with this software also confirms that the use of "Rovimat" material is compliant with the ISO structural requirements for small craft vessels. Although based on a scalar von Mises criterion, the compliance factor results are close to the ones obtained with the Abaqus software, with some variation.

Overall, the results demonstrate that ISO 12215-5 provides a conservative and reliable framework for hull panel dimensioning, while finite element analysis offers complementary insight into global load redistribution and localized geometric effects.

Chapter 5

Conclusions and Future Work

5.1 Conclusions

This thesis's main objective was to define the structural design of Riamar company's current new vessel project, the Evolution 5.0. The analysis performed was focused on the assessment of the vessel's hull bottom. To achieve this goal, different methodologies were considered to be integrated in the design process.

A bibliographic research was conducted. Among the topics, there was an economic and historical overview of the small craft industry, useful standards with a focus on comprehending the ISO 12215-5 structural standard, and understanding common terminology and components of small vessels. Also, an investigation was conducted on product development and its correlation with boat designing and an overview of previous work regarding the designing of the new vessel by Riamar company.

Material characterization was performed for selected materials used in the defined vessel. This process involved an experimental part, by performing tensile tests and three-point bending tests. The results were compared with the data defined by ISO 12215-5 standards to achieve the final mechanical properties of the materials used by Riamar company. The results from the experimental tests have shown good alignment with the analytically obtained properties of the materials. Strength- and strain-related properties calculated analytically were slightly lower than the experimental ones, which shows the conservative nature of the standard. The manufactured laminates of the analyzed materials showed high variance in terms of thickness. This phenomenon could be related to the manufacturing process, hand layup, where thickness control is difficult to maintain without the use of vacuum, for example, to extract the extra resin.

Then, methodologies for structural analysis were drawn. Two analytical methods, based on the equations offered by ISO 12215-5 and a numerical one that is established with the ISO standard guidelines and investigation in 3.3 regarding FEM procedure efficiency and parameter influence. The FEM methodology was applied using two numerical codes: Abaqus and SolidWorks. For the analytical methods, a tool in the form of a Microsoft Excel file was developed that performs the

scantling calculation with the inputs of panels dimensions, main data of the vessel, and considered material properties.

The considered methods were applied for two different cases. The first one, the Pescador 500 vessel, is an already proven design that was already evaluated analytically. This vessel analysis validated the developed analytical tool, as the results obtained with it are similar to the ones shown in the scantling report made for the verification of this vessel. FEM results have shown higher compliance factors for the majority of the panels, unveiling the conservative nature of the analytic methods. The exceptions were analyzed in more detail, and complex geometry dimensions can be seen as the major influence in these occasions.

The other case and the main object of study, the Evolution 5.0 vessel, required additional work to define the data regarding the vessel as a whole and individual panel dimensions to apply the analytical tool. Both analytical and numerical results indicate that the application of the "Rovimat" material to the bottom of the Evolution 5.0 hull is compliant with the ISO 12215-5 standard. However, the numerical results showed that the application of the material can be performed with greater safety margins than the analytical results. This shows the importance of incorporating FEA in boat designing, as only applying analytical tools could lead to over-designed vessels, which means a higher cost in unnecessary material quantity in the laminate and higher manufacturing time for the lamination of the extra fiber layers. In cases where the compliance factor values obtained with numerical methods are lower, further research should be performed to understand the reasons behind those results.

The use of CAD software with integrated FEM packages reveals itself as an opportunity to perform FEM analysis in an expedited manner, as it is not necessary to import the model to other software. The SolidWorks software, in specific, lacks the detail that, for example, Abaqus software allows in terms of material definition and results analysis. However, the compliance factors calculated with this software are not immensely far from the ones obtained with Abaqus.

Regarding the Abaqus software, it is seen as a more accurate tool and could be especially useful for a final and more detailed analysis. Therefore, the implementation of all of the discussed methods into the design process of vessels is not to be disregarded, as they complement each other. Firstly, the use of the analytic methods for an initial material definition and allocation. Then, the implementation of a FEM package that allows an expedited analysis to obtain fast results and iterate the simulation with small material changes. Finally, in the last steps of the boat design, a more accurate and detailed FEM tool could be employed to ensure compliance with the ISO 12215-5 standard and refine the last parameters.

It is safe to assume that all objectives that were initially stipulated were completed, since the characterization of the composite material configurations for application in the hull bottom of the Evolution 5.0 vessel has been completed, both analytically and experimentally. A methodological workflow ready to be integrated in the design process of new vessels was developed, which included an analytical tool based on the equations described in ISO 12215-5 and the definition of a FEM application procedure that performs the structural analysis of the hull, based on the ISO guidelines. These methodologies were validated by their application to two vessels, the Pescador

500 and the Evolution 5.0. Finally, conclusions regarding the material selection could be drawn: the "Rovimat" laminate complies with both the numerical and analytical methodologies and can be applied to the bottom of the hull of Evolution 5.0 without concerns.

5.2 Future Work

As future work, the next step is to apply the methodology developed in this study to the scantling calculations of the remaining components of the Evolution 5.0, particularly the hull sides and the deck. The analytical tool developed in this work is capable of performing structural assessments for sandwich panels. Therefore, if such materials are to be adopted in the design, their structural performance can also be evaluated using the same procedure. Furthermore, iterative analysis of the hull's bottom panels may also be performed, as there are certain parameters that are in continuous fluctuation with the development of the vessel, such as the mass of the vessel.

Also, additional investigation should be put into the understanding of the vessel's compliance with other mandatory rules, such as the stability and buoyancy capabilities, according to the ISO 12217. This type of validation must also be performed iteratively as multiple parameters are constantly changing.

Concerning material characterization, additional experimental testing should be conducted on the "Mat" and "Rovimat" composite laminates to better assess the variability in specimen thickness and mechanical properties.

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Appendix A

Material Characterization: additional data

This section aims to provide additional information regarding the material characterization of the materials considered for this work.

A.1 Measurements of Specimens

Tables A.1 and A.2 shows in more detail the data collected regarding the geometrical features of the specimens for the tensile tests.

Table A.1: Width and thickness measurements of "Mat" material specimens used for the tensile tests.

Group	Number	Width [mm]			Average [mm]	Thickness [mm]		Average [mm]
		1	2	3		1	2	
MT0	#1	24.37	24.35	24.42	24.4	3.48	3.90	3.7
	#2	24.36	24.39	24.34	24.4	3.47	3.99	3.7
	#3	24.35	24.32	24.41	24.4	3.41	3.67	3.5
	#4	23.95	24.11	24.22	24.1	3.96	4.30	4.1
	#5	24.27	24.29	24.29	24.3	4.00	4.24	4.1
	#6	24.41	24.39	24.39	24.4	3.87	3.67	4.2

Table A.2: Width and thickness measurements of "Rovimat" material specimens used for the tensile tests.

Group	Number	Width [mm]			Average [mm]	Thickness [mm]		Average [mm]
		1	2	3		1	2	
RT0	#1	24.21	24.18	24.24	24.2	4.72	5.10	4.9
	#2	24.30	24.31	24.31	24.3	4.40	4.65	4.5
	#3	24.32	24.32	24.33	24.3	3.65	4.10	3.9
RT45	#1	24.33	24.34	24.31	24.3	5.02	5.49	5.3
	#2	24.41	24.41	24.41	24.4	5.06	5.30	5.2
	#3	24.38	24.38	24.36	24.4	5.12	5.35	5.2
RT90	#1	24.39	24.41	24.37	24.4	4.88	5.10	5.0
	#2	24.24	24.31	24.12	24.2	4.76	4.94	4.9
	#3	24.34	24.38	24.24	24.3	5.00	5.33	5.2

Tables A.3 and A.4 shows in more detail the data collected regarding the geometrical features of the specimens for the three-point bending tests.

Table A.3: Measured width and thickness values of "Mat" material specimens for three-point bending tests.

Group	Number	Width [mm]			Average [mm]	Thickness [mm]		Average [mm]
		1	2	3		1	2	
MB0	#1	44.20	44.21	44.21	44.21	3.93	3.95	3.94
	#2	44.12	44.13	44.12	44.12	3.82	3.85	3.84
	#3	44.09	44.11	44.17	44.12	3.95	4.08	4.02
	#4	44.08	44.12	44.15	44.12	3.94	3.97	3.96
	#5	44.10	44.09	44.10	44.10	4.02	4.05	4.04
	#6	44.01	44.01	44.02	44.01	3.95	4.06	4.01
	#7	44.18	44.18	44.19	44.18	3.79	3.85	3.82
	#8	44.02	44.06	44.09	44.06	3.97	4.04	4.01

Table A.4: Measured width and thickness values of "Rovimat" material specimens for three-point bending tests.

Group	Number	Width [mm]			Average [mm]	Thickness [mm]		Average [mm]
		1	2	3		1	2	
RB0	#1	44.12	44.15	44.15	44.14	4.78	4.85	4.82
	#2	45.24	45.28	45.30	45.27	5.07	5.48	5.28
	#3	44.10	44.09	44.09	44.09	4.79	4.90	4.85
	#4	44.12	44.13	44.13	44.13	5.17	5.27	5.22
	#5	44.15	44.14	44.13	44.14	5.30	5.50	5.40
	#6	44.25	44.17	44.27	44.23	5.40	5.50	5.45
	#7	44.10	44.06	44.04	44.07	5.60	5.80	5.70
	#8	44.22	44.20	44.17	44.20	5.10	5.30	5.20

Tables A.5 to A.8 present the comparison between the analytically obtained properties with the experimental test results, all calculated considering nominal thickness values, which are 3.9 mm for the "Mat" material and 5.1 mm for the "Rovimat" material.

Table A.5: Comparison of material properties obtained from the analytical and experimental methods for "Mat" material, regarding the tensile properties.

Characterization method for "Mat" material tensile properties	E [MPa]	ε_{ut} [%]	σ_{ut} [MPa]
ISO 12215-5	10 071	1.35	136.0
Tensile tests	$9\,592 \pm 485$	2.11 ± 0.09	144.4 ± 7.4

Table A.6: Comparison of material properties obtained from the analytical and experimental methods for "Mat" material, regarding the flexural properties.

Characterization method for "Mat" material flexural properties	E_f [MPa]	ε_{uf} [%]	σ_{uf} [MPa]
ISO 12215-5	–	1.88	189.0
Three-point bending tests	$9\,007 \pm 860$	3.43 ± 0.04	246.5 ± 10.4

Table A.7: Comparison of material properties obtained from the analytical and experimental methods for "Rovimat" material, regarding the tensile properties and fiber orientations.

Characterization method for "Rovimat" material tensile properties	E [MPa]	ϵ_{ut} [%]	σ_{ut} [MPa]
ISO 12215-5	11 030	1.35	148.9
Tensile tests – 0° fiber angle	$10\,095 \pm 765$	1.69 ± 0.22	140.5 ± 16.3
Tensile tests – 45° fiber angle	$9\,538 \pm 349$	2.12 ± 0.11	147.1 ± 3.7
Tensile tests – 90° fiber angle	$10\,318 \pm 240$	2.18 ± 0.13	155.8 ± 1.5

Table A.8: Comparison of material properties obtained from the analytical and experimental methods for "Rovimat" material, regarding the flexural properties.

Characterization method for "Rovimat" material flexural properties	E_f [MPa]	ϵ_{uf} [%]	σ_{uf} [MPa]
ISO 12215-5	–	1.84	205.7
Three-point bending tests	$11\,291 \pm 792$	2.58 ± 0.46	231.5 ± 10.0

A.2 Three-point Bending Test — Numerical Approach

As previously stated, numerical analyses of the three-point bending tests were also performed using finite element method (FEM) software. The objective of these analyses was to estimate the elastic modulus (E), which differs from the flexural modulus of elasticity (E_f), based on this type of test. By performing several analyses using different values of the elastic modulus, it becomes possible to compare the force-displacement curves obtained experimentally with those from the numerical simulations. This approach has previously been applied to steel specimens, achieving good results in the prediction of both elastic and plastic behavior [53].

Since multiple values of the elastic modulus needed to be evaluated, three Python scripts were developed to improve the efficiency of the analysis process. Compatibility with Python scripting was one of the main reasons for selecting Abaqus as the FEM software. The purpose of the first script was to generate the numerical models with different values of E , while reproducing the experimental test conditions as accurately as possible. The second script sequentially ran all the models generated by the first script. The third script opened the resulting `.odb` files and extracted the force and displacement data of the loading nose for each simulation.

From a geometrical point of view, each model consisted of two rigid two-dimensional planar components, the loading nose and one support, using symmetry boundary conditions, and one deformable two-dimensional planar component representing the specimen. Both "Mat" and "Rovimat" materials were considered, resulting in different specimen thicknesses for each case.

The analyses were carried out using two-dimensional models primarily to reduce computational cost. As a consequence, the resulting load output represents the load per unit width [N/mm]. Figure A.1 illustrates an example of the numerical model employed.

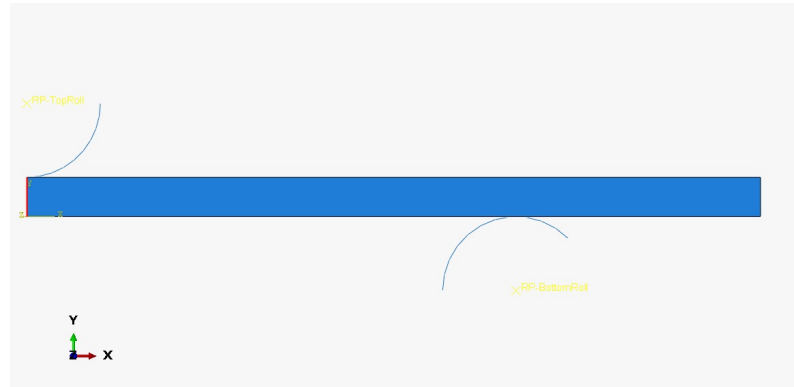


Figure A.1: Developed model for the three-point bending numerical analysis.

This first script allowed several inputs by the user. Table A.9 shows the values that were kept unchanged throughout the different analyses. The models have two steps (static general): the first one to make the nose be in contact with the upper face of the specimen and the second to apply the displacement of 5 mm during 60 s.

Regarding boundary conditions, as previously stated, the left side of the specimen (red line in Figure A.1) is constrained with symmetry across the YZ plane, the support is fully fixed, and the punch is fixed in the X direction, beyond the displacement in the Y direction.

Table A.9: Geometrical, and loading parameters used in the three-point bending simulation

Length	Roller radius	Support span	Punch speed	Punch displacement	Step 2 time	Data points
[mm]	[mm]	[mm]	[mm/min]	[mm]	[s]	[1]
150	7.5	100	5	5	60	500

Regarding the mesh, only the specimen needed meshing, since the other parts are analytic rigid (Figure A.2). The element size used was 0.4 mm and a number of 12 elements through thickness. The element type was CPE4R, which is a 4-node bilinear quadrilateral, plane-strain element with reduced integration. The use of this element is justified by the negligible displacement on the out-of-plane direction when compared with the other two [53, 62].

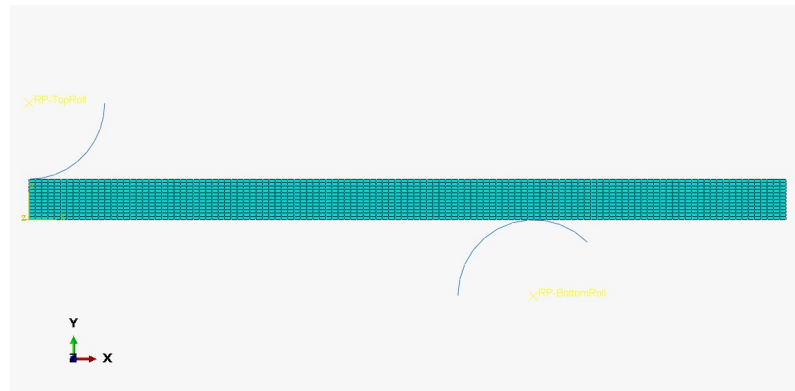


Figure A.2: Meshed model for the three-point bending numerical analysis.

Furthermore, due to the usage of cylindrical polished supports, it is safe to assume frictionless behavior between the specimen and the other parts as a good procedure, as done in other works [63]. For research purposes, additional analyses using a friction coefficient of 0.15 were performed to compare the results.

For the material, the two considered laminates were modeled as regular shell homogeneous elastic materials, each one with its density and Poisson ratio (taken from the ISO 12215 values).

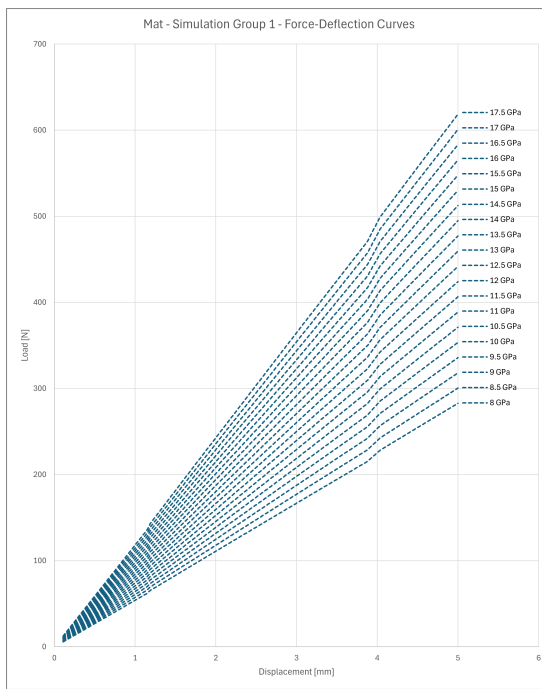
There were 4 groups of simulations made for this analysis (Table A.10). The first two for the "Mat" laminate with the different friction coefficients and the other two for the "Rovimat" laminate, also with different friction coefficients.

Table A.10: Simulation groups and corresponding material and contact parameters.

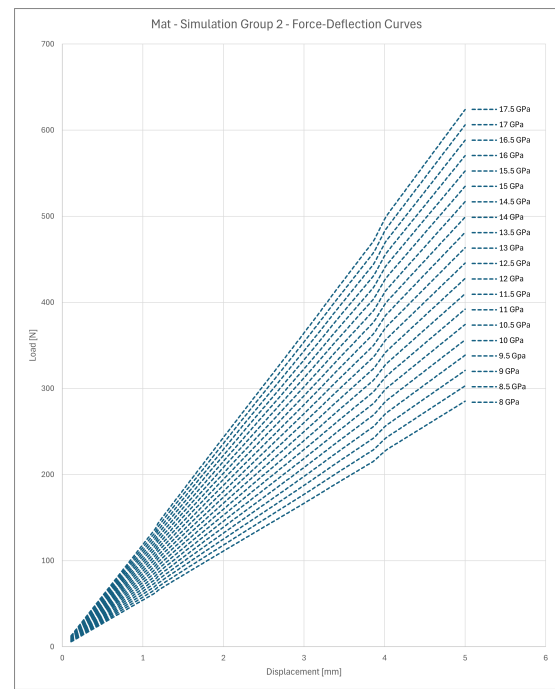
Simulation group	Thickness	Composite density	Poisson ratio	Friction coefficient
	[mm]	[tonne/mm ³]	[1]	[1]
Mat-1	3.9	1.43×10^{-9}	0.4	0
Mat-2	3.9	1.43×10^{-9}	0.4	0.15
Rovimat-1	5.1	1.53×10^{-9}	0.29	0
Rovimat-2	5.1	1.53×10^{-9}	0.29	0.15

Finally, the elastic modulus values were not the same for the two materials. For the "Mat" material, the elastic modulus values considered range from 8 000 to 17 500 MPa, with a constant increment of 500 MPa. For the "Rovimat" material, the range goes from 10 000 to 19 000 MPa, again with a constant increase of 500 MPa.

Figures A.3 to A.4 show the force-displacement curves for the simulations ran for each material and friction coefficient.

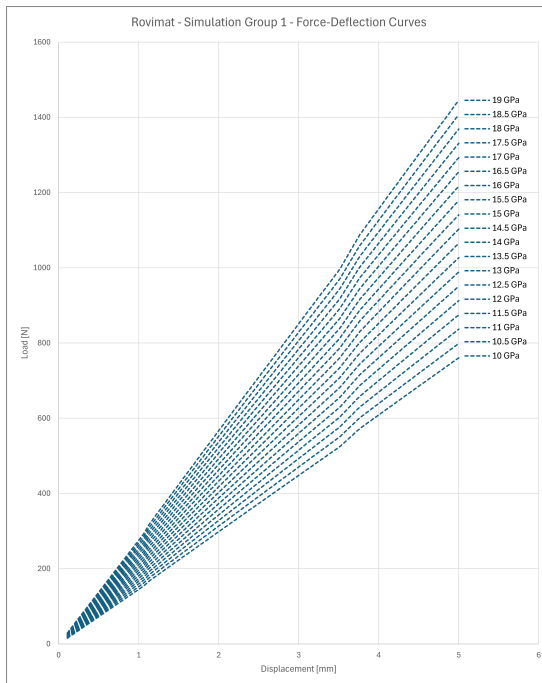


(a)

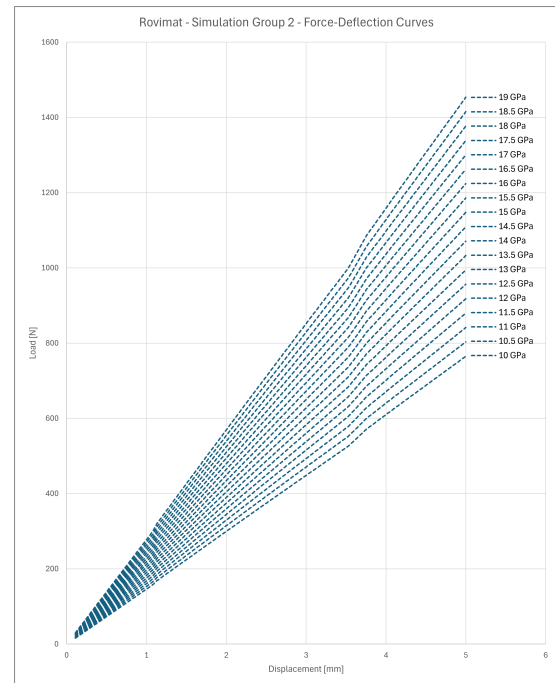


(b)

Figure A.3: Force-displacement curves produced with numerical data from Abaqus simulations for "Mat" material.



(a)



(b)

Figure A.4: Force-displacement curves produced with numerical data from Abaqus simulations for "Rovimat" material.

For an easier analysis, using the "trendline" feature of Excel graphs, the average slope of each force-displacement curve was calculated and is displayed in Tables A.11 and A.12.

Table A.11: Slopes obtained for the force-displacement curves obtained through FEM simulations for the "Mat" material.

<i>E</i>	Slopes – Mat 1	Slopes – Mat 2
[GPa]	[N/mm]	[N/mm]
17.5	124.1	124.9
17.0	120.6	121.4
16.5	117.0	117.8
16.0	113.5	114.2
15.5	109.9	110.7
15.0	106.4	107.1
14.5	102.8	103.5
14.0	99.3	99.9
13.5	95.7	96.4
13.0	92.2	92.8
12.5	88.6	89.2
12.0	85.1	85.7
11.5	81.6	82.1
11.0	78.0	78.5
10.5	74.5	75.0
10.0	70.9	71.4
9.5	67.4	67.8
9.0	63.8	64.2
8.5	60.3	60.7
8.0	56.7	57.1

Table A.12: Slopes obtained for the force-displacement curves obtained through FEM simulations for the "Rovimat" material.

<i>E</i>	Slopes – Rovimat 1	Slopes – Rovimat 2
[GPa]	[N/mm]	[N/mm]
19.0	290.8	291.4
18.5	283.1	283.7
18.0	275.5	276.1
17.5	267.8	268.4
17.0	260.2	260.7
16.5	252.5	253.1
16.0	244.9	245.4
15.5	237.2	237.7
15.0	229.6	230.1
14.5	221.9	222.4
14.0	214.3	214.7
13.5	206.6	207.1
13.0	199.0	199.4
12.5	191.3	191.7
12.0	183.6	184.1
11.5	176.0	176.4
11.0	168.3	168.7
10.5	160.7	161.0
10.0	153.0	153.4

Comparing Tables A.11 and A.12, it is possible to conclude that the change between friction coefficients did not affect the slopes of the force-displacement curves significantly. Therefore, the numerical curves with a friction coefficient of 0 will be used for the comparison between the numerical and experimental data. To perform that comparison, the numerical and experimental curves are overlapped in Figure A.5 for the "Mat" laminate and in Figure A.6 for the "Rovimat" laminate.

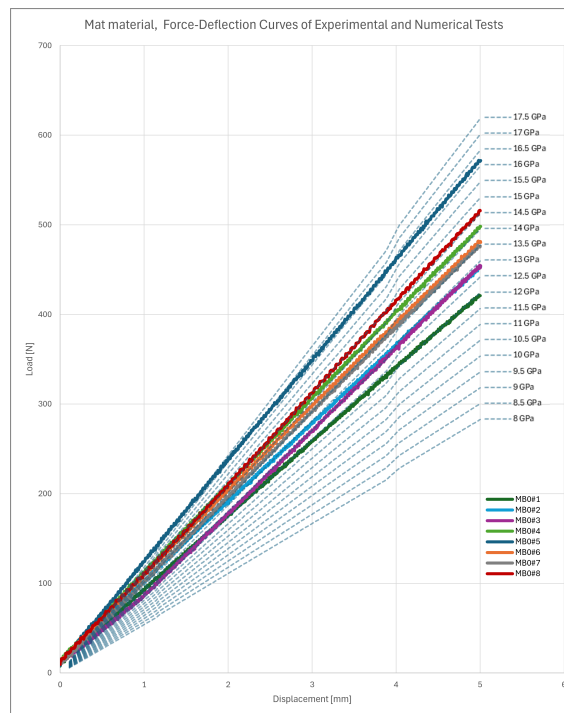


Figure A.5: Overlapped force-displacement curves based on numerical data (dashed lines) and experimental data (solid lines) for "Mat" laminates.

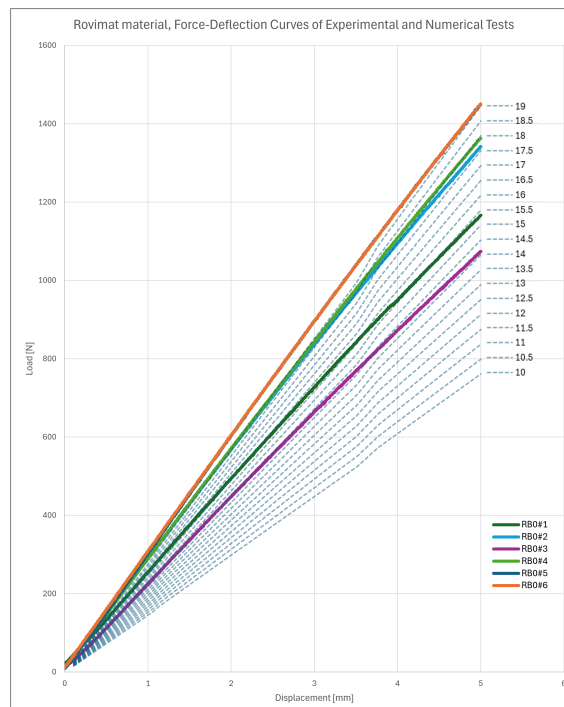


Figure A.6: Overlapped force-displacement curves based on numerical data (dashed lines) and experimental data (solid lines) for "Rovimat" laminates.

With the slope values of the experimental curves, the predicted elastic modulus of each specimen can be defined using linear regression, as shown in Tables A.13 and A.14.

Table A.13: Force–displacement slopes and corresponding Young’s modulus values for MB0 specimens.

Samples	Slope MB0	E
	[N/mm]	[GPa]
MB1	82.7	11.66
MB2	88.0	12.41
MB3	90.5	12.76
MB4	96.7	13.64
MB5	112.3	15.84
MB6	94.2	13.28
MB7	94.0	13.26
MB8	101.2	14.27
Average		13.39

Table A.14: Force–displacement slopes and corresponding Young’s modulus values for RB0 specimens.

Samples	Slope RB0	E
	[N/mm]	[GPa]
RB1	231.2	15.11
RB2	267.6	17.49
RB3	214.9	14.04
RB4	272.3	17.79
RB5	291.2	19.03
RB6	289.4	18.91
Average		17.06

The elastic modulus values obtained through this method are higher than both the experimental tensile results and the analytically calculated values. While this approach demonstrates potential as a complementary stiffness estimation tool, the observed discrepancies indicate that further investigation is required to fully understand the relationship between numerical stiffness predictions and material elastic properties of composite materials.

Appendix B

Pescador 500 Scantling Data

Table B.1 shows the geometrical properties of the panels considered for the Pescador 500 vessel. This data was retrieved from the scantling report done on this vessel.

Table B.1: Geometrical properties and curvature parameters of panels from the Pescador 500 vessel.

ID	Length (mm)	Width (mm)	Long. Pos. (m)	Long. Curv. (mm)	Trans. Curv. (mm)	k_C [1]	k_{2b} [1]
A1	896	340	0.530	0	54	0.587	0.500
A2	879	310	0.506	0	11	0.967	0.500
B3	831	397	1.448	6	48	0.654	0.500
B4	801	307	1.447	0	12	0.957	0.500
C5	655	368	2.248	7	33	0.733	0.500
C6	629	326	2.256	1	16	0.902	0.500
D7	500	252	2.898	33	20	0.685	0.500
D8	635	335	2.977	8	14	0.941	0.500
E9	545	271	3.640	30	9	0.911	0.500

Table B.2 shows the design stress values for each individual layer of the Pescador 500's hull.

Table B.2: Design stresses calculated for each layer of the Pescador 500's hull material, regarding the Enhanced method and FEM.

Composite layers	$\sigma_{d_{ENH}}$ [MPa]	$\sigma_{d_{FEM}}$ [MPa]
Layer 1 - CSM	55.6	58.5
Layer 2 - CSM	55.6	58.5
Layer 3 - CSM	55.6	58.5
Layer 4 - CSM	55.6	58.5
Layer 5 - CSM	55.6	58.5
P500 Hull Composite	55.6	58.5

Figure B.1 shows the calculating table from the developed analytical tool that performs the scantling calculations, applied to a panel from the Pescador 500 vessel.

PRELIMINARY CALCULATIONS											
1	2	3	4	5	6	7	8	9	10	11	12
Design Pressure	Panel short dim	Panel long dim	Curvature Factor	Geometric aspect ratio	EI/EIb From Laminate	Effective aspect ratio	Bending Moment and shear force factors (Table A.2)				
P	b	l	k_c	$A_r=U/b$			along b	along l	along b	along l	
kN/m^2	mm	mm	1	1	1	1	k_{2b}	k_{2l}	k_{SHb}	k_{SHl}	
36.21	252	500	0.685	1.98	1	1.98	0.492	0.338	0.516	0.460	
13	14	15	16	17	18	19	20	21	22	23	24
Shear force (corrected)		Bending mt		Bending mt							
along b	along l	along b	(ini.-correct)	along b	(ini.-correct)						
		Table A.4									
$F_{2b}=k_c \cdot k_{SHb} \cdot P \cdot b/1000$		$M_{2b}=1/6 \cdot k_{2b} \cdot P \cdot b^2 \cdot 10^{-3}$		$M_{2b}=1/6 \cdot k_{2b} \cdot P \cdot (E/E_0)^{0.5} \cdot b^2 \cdot 10^{-3}$							
F_{2b}	F_{2l}	M_{2b}	$M_{2b,cor}$	M_{2b}	$M_{2b,cor}$	k_{SH} (table 15)	k_{SH} (table 16)		k_{SH} and k_{SH} used for mech properties of laminate		
N/mm	N/mm	Nmm/mm	Nmm/mm	Nmm/mm	Nmm/mm	1	1		k_{SH} used for mech properties of core		
3.22	2.88	-188.44	-129.14	-129.72	-88.89	0.75	0.95		tables 15 and 16		
PANEL LAMINATE ANALYSIS											
25	26	27	28	29	30	31	32	33	34	35	36
Ply	Definition			Fibre				Interlam τ_u	σ_{2b}/σ_{2l}		
Nº		dry mass	type* G,C,A	Vol frac ϕ Table C,7	Mass frac ψ Table C,2	EI N/mm ²	σ of, u flexural c > NA, t < N/mm ²	interlam N/mm ²	τ_u/τ_u interlam Table 17	of d N/mm ² (32)/(34)	Interlam td N/mm ² (33)/(34)
		kg/m ² input					σ or $\tau \cdot k_{SH} \cdot k_{SH}$				
5 inner	Mat 600	0.3	G	0.167	0.3	8267	111.2	12.3	0.5	55.6	6.2
4	Mat 600	0.45	G	0.167	0.3	8267	111.2	12.3	0.5	55.6	6.2
3	Mat 600	0.6	G	0.167	0.3	8267	111.2	12.3	0.5	55.6	6.2
2	Mat 450	0.6	G	0.167	0.3	8267	111.2	12.3	0.5	55.6	6.2
1 outer	Mat 300	0.6	G	0.167	0.3	8267	111.2	12.3	0.5	55.6	6.2
Total		2.55		0.167	0.300	8267	111.2		0.5	55.6	
	Sum Col				Average	Average					
37	38	39	40	41	42	43	44	45	46	47	48
Ply	t/w	Thickness		Dist zgi				(E/I)	z_{crit} from SMI		Allowable
Nº	1/phi	ti	E-ti	from inside	E-ti-zgi	E-ti-zgi ²	E-ti-t ₁ ² /12	from inside	ZNA	ENA/(Ei-z _i crit)	Bending Mt
	1	mm	N/mm	mm	N	Nmm	Nmm ²	Nmm ²	mm	mm ² /mm	Nmm/mm
		(27)/(38)	(31)/(39)	Calc	(40)/(41)	(41)/(42)	(31)/(39) ² /12	(43)/(44)	calc	calc	(47)/(35)
5 inner	2.339	0.702	5801	0.35	2035	714	238	952	-2.98	5.93	330
4	2.339	1.053	8701	1.23	10685	13121	803	13924	-2.28	7.75	431
3	2.339	1.403	11602	2.46	28494	69979	1904	71883	-1.23	14.40	800
2	2.339	1.403	11602	3.86	44776	172806	1904	174710	1.58	-11.20	-622
1 outer	2.339	1.403	11602	5.26	61058	321333	1904	323237	2.98	-5.93	-330
total	2.34	5.96	49308	2.98	147048	577953		584706	146177		
				Z_{max} in				E_{lamin}	E_{lamin}		
49	50	51	52	53	54	55	56	57	58	59	60
Ply	ϵ	Stress	Compl	Type of							
Nº	Mdb/ESMI	in ply i	factor Cf	load	Location of layer	Z calc-ZNA from NA	First mt Q	Shear flow	τ_i	Shear compl	
	%	of i	od/oi	Tens vs Comp		$\Sigma E \cdot t_i(z_i - z_{NA})$	N	q	shear	factor td/ti	
	1	N/mm2	1			mm	Nmm	N/mm	N/mm2	C_t	
	(16)/(31)/(47)	(16)/(47)	(35)/(51)				calc	$F(56)/E_{lamin}$	(57)/1	(36)/(58)	
5 inner	-0.26%	-21.8	-2.55	Comp	4-5 interface	2.28	15264.49	0.3	0.34	18.31	of -> max outer plies
4	-0.20%	-16.7	-3.34	Comp	3-4 interface	1.23	30528.97	0.7	0.67	9.15	
3	-0.11%	-9.0	-6.20	Comp	2-3 interface	-0.18	36634.76	0.8	0.81	7.63	
2	0.14%	11.5	4.82	Tens	1-2 interface	-1.58	26458.44	0.6	0.58	10.56	
1 outer	0.26%	21.8	2.55	Tens	bottom of 1	-2.98	0.00	0.0	0.00		$\tau=0$ top/bottom
Allow, bending mt Mdb = Mdb req x Cf (Nmm/mm)				330	Allowable shear force F = F d req x Cf shear (N/mm)			24.59	7.63	Min Cf on shear	
Analysis by simplified method of Table A.5											
b	kc	P	k2b	odf (col35)	K_{SH}	odf simplified Table C,5	tp req 2018	W, req	W, ratio		
mm	1.00	kN/m2	1		simplif	N/mm2	mm	kg/m2			
252	0.685	36.21	0.492	55.6	0.9	52.7	3.18	1.36	1.88		
$tp (table A5) = b \cdot kc \cdot (P \cdot k2b / (1000 \cdot odf)) \cdot 0.5$											

Figure B.1: Analytical tool spreadsheet performing the scantling determination of the panel D7 from the Pescador 500 vessel.

Tables B.3, B.4 and B.5 present the results obtained from the analytical methods, the Abaqus software and the SolidWorks software, respectively, for the Pescador 500 vessel.

Table B.3: Comparison between the results obtained analytically using the Enhanced and Simplified methods for the Pescador 500 vessel.

ID	Enhanced Method		Simplified Method	
	σ_i [MPa]	CF_{Enh}	$W_{f,req}$ [kg/m ²]	CF_{Simp}
A1	17.6	3.16	1.13	2.26
A2	24.7	2.25	1.72	1.48
B3	31.7	1.75	1.60	1.59
B4	30.7	1.82	1.90	1.34
C5	37.5	1.48	1.84	1.38
C6	38.7	1.44	2.08	1.23
D7	21.8	2.55	1.36	1.88
D8	44.9	1.24	2.29	1.12
E9	32.4	1.75	1.89	1.35

Table B.4: Stress results in 1- and 2-directions and compliance factors, obtained with Abaqus software, for Pescador 500.

ID	P [kN/m ²]	$\sigma_{max,S11}$ [MPa]	$\sigma_{equi,1}$ [MPa]	$CF_{Abq,S11}$ [–]	$\sigma_{max,S22}$ [MPa]	$\sigma_{equi,2}$ [MPa]	$CF_{Abq,S22}$ [–]
A1	18.5	-0.4	8.3	6.72	0.7	13.5	4.12
A2	19.0	-0.5	9.5	5.85	-0.7	12.6	4.42
B3	22.0	-0.4	9.3	6.01	0.7	15.6	3.58
B4	24.0	-0.7	16.8	3.32	-1.0	24.4	2.28
C5	27.9	-0.4	11.4	4.87	-0.5	13.8	4.02
C6	29.3	-0.8	23.3	2.39	-1.3	37.3	1.49
D7	36.2	-0.5	18.8	2.95	-1.0	36.5	1.52
D8	30.9	-0.8	25.8	2.16	-1.3	41.0	1.36
E9	34.5	0.8	26.7	2.08	-1.1	39.0	1.42

Table B.5: Von Mises stress results and compliance factors, obtained with SolidWorks software for Pescador 500 vessel.

ID	P [kN/m ²]	$\sigma_{max,vM}$ [MPa]	$\sigma_{equiv,vM}$ [MPa]	$CF_{SW,vM}$ [–]
A1	18.5	0.3	5.8	9.59
A2	19.0	0.7	13.7	4.07
B3	22.0	0.3	7.3	7.64
B4	24.0	1.5	36.2	1.53
C5	27.9	0.3	9.4	5.94
C6	29.3	1.5	43.4	1.28
D7	36.2	0.6	21.9	2.53
D8	30.9	1.0	30.6	1.82
E9	34.5	0.7	23.1	2.40

Appendix C

Evolution 5.0 Scantling Data

Table C.1 shows the geometrical properties of the panels considered for the Evolution 5.0 vessel. This data is retrieved from the CAD model of the vessel.

Table C.1: Geometrical properties and curvature parameters of panels from the Evolution 5.0 vessel.

ID	Length [mm]	Width [mm]	Long. Pos. [m]	Long. Curv. [mm]	Trans. Curv. [mm]	k_C [1]	k_{2b} [1]
A1	656	442	0.374	0	0	1.209	0.448
B2	527	439	1.310	0	0	1.209	0.380
C3	879	444	2.014	0	0	1.209	0.492
ABC4	2254	439	1.186	0	0	1.209	0.500
D5	1243	195	3.048	22	4	1.141	0.500
D6	1245	412	3.056	20	15	0.776	0.500
E7	347	528	3.999	33	4	1.007	0.454
F8	675	420	4.631	149	29	0.500	0.466

Table C.2 shows the design stresses calculated for the application of the "Rovimat" material in the Evolution 5.0 vessel.

Table C.2: Design stresses calculated for the application of the "Rovimat" material in the Evolution 5.0 vessel.

Composite layers	$\sigma_{d_{\text{ENH}}}$ [MPa]	$\sigma_{d_{\text{FEM}}}$ [MPa]
Layer 1 - RVM	117.3	123.5
Layer 2 - CSM	85.3	89.8
Layer 3 - CSM	85.3	89.8
Layer 4 - CSM	85.3	89.8
Layer 5 - CSM	85.3	89.8
"Rovimat" Material	92.8	97.7

Figure C.1 shows the calculating table from the developed analytical tool that performs the scantling calculations, applied to a panel from the Evolution 5.0 vessel.

PRELIMINARY CALCULATIONS											
1	2	3	4	5	6	7	8	9	10	11	12
Design Pressure	Panel short dim	Panel long dim	Curvature Factor	Geometric aspect ratio	El/Elb From Laminate	Effective aspect ratio	Bending Moment and shear force factors (Table A.2)				
P	b	l	k_c	$A_r=U/b$			along b	along l	along b	along l	
kN/m^2	mm	mm	1	1	1	1	k_{2b}	k_{2l}	k_{SHb}	k_{SHl}	
19.52	412	1245.2	0.962	3.02	1	3.02	0.500	0.337	0.520	0.460	
13	14	15	16	17	18	19	20	21	22	23	24
Shear force (corrected)	along b	along l	Bending mt along b (ini.-correct)	Bending mt along b (ini.-correct)							
			Table A.4								
$F_{2b}=k_c \cdot k_{SH} \cdot P \cdot b/1000$		$M_{2b}=1/6 \cdot k_{2b} \cdot P \cdot b^2 \cdot 10^{-3}$	$M_{2b,cor}=1/6 \cdot k_{2b} \cdot P \cdot (E/E_2)^{0.5} \cdot b^2 \cdot 10^{-3}$								
F_{2b}	F_{2b}	M_{2b}	$M_{2b,cor}$	M_{2b}	$M_{2b,cor}$	k_{SH} (table 15)	k_{SH} (table 16)		k_{SH} and k_{SH} used for mech properties of laminate		
N/mm	N/mm	Nmm/mm	Nmm/mm	Nmm/mm	Nmm/mm	1	1		k_{SH} used for mech properties of core		
4.02	3.56	-276.17	-265.74	-186.14	-179.11	0.95	0.95		tables 15 and 16		
PANEL LAMINATE ANALYSIS											
25	26	27	28	29	30	31	32	33	34	35	36
Ply	Definition			Fibre				Interlam			
Nº		dry mass	type * G,C,A	Vol frac ϕ Table C,7	Mass frac ψ Table C,2	El N/mm ²	z of, u flexural c > NA, t < N/mm ²	t_u	σ_{2b}/σ_{2l}	of d N/mm ² (32)/(34)	Interlam t d N/mm ² (33)/(34)
		kg/m ² input					σ or $\tau \cdot k_{SH} \cdot k_{SH}$	t_u	τ_u/t_u interlam Table 17		
5 inner	oviMat 500/45	0.95	G	0.309	0.488	14149	235.0	11.8	0.5	117.5	5.9
4	Mat 450	0.45	G	0.225	0.382	10071	170.6	14.3	0.5	85.3	7.2
3	Mat 600	0.6	G	0.225	0.382	10071	170.6	14.3	0.5	85.3	7.2
2	Mat 600	0.6	G	0.225	0.382	10071	170.6	14.3	0.5	85.3	7.2
1 outer	Mat 600	0.6	G	0.225	0.382	10071	170.6	14.3	0.5	85.3	7.2
Total		3.20		0.246	0.409	11030	185.7		0.5	92.9	
		Sum Col		Average	Average						
37	38	39	40	41	42	43	44	45	46	47	48
Ply	t/w	Thickness		Dist zgi				(Elji)	z_{crit} from	SMI	Allowable
Nº	1/phi	ti	El-ti	from inside	El-ti-zgi	El-ti-zgi ²	El-ti-t ₁ ² /12	from inside	z_{crit} ZNA	ENA/(El-z crit)	Bending Mt
	1	mm	N/mm	mm	N	Nmm	Nmm	Nmm ²	mm	mm ² /mm	Nmm/mm
		(27)/(38)	(31)/(39)	Calc	(40)/(41)	(41)/(42)	(31)/(39) ² /12	(43)/(44)	calc	calc	(47)/(35)
5 inner	1.26	1.201	16 992	0.60	10 203	6 127	2042	8 169	-2.38	3.84	451
4	1.74	0.781	7 868	1.59	12 522	19 930	400	20 331	-1.18	10.87	927
3	1.74	1.042	10 491	2.50	26 258	65 726	949	66 674	0.64	-20.08	-1 713
2	1.74	1.042	10 491	3.54	37 186	131 814	949	132 762	1.68	-7.64	-652
1 outer	1.74	1.042	10 491	4.59	48 114	220 668	949	221 616	2.72	-4.72	-403
total	1.60	5.11	56 332	2.38	134 264	444 264		449 553	129445		
				$Z_{NA, in}$				E_{lamin}	E_{lamin}		
49	50	51	52	53	54	55	56	57	58	59	60
Ply	ϵ	Stress	Compl	Type of			Shear stress analysis				
Nº	Mdb/ESMi	in ply i	factor Cf	load	Location of layer	Z calc-ZNA from NA	First mt Q $\Sigma E_i t_i (z_i - z_{NA})$	Shear flow	τ_i	Shear compl	
	%	of i	od/oi	Tens vs Comp		mm	Nmm	q	shear	factor td/ti	
	1	N/mm2	1				calc	F(56)/ E_{lamin}	(57)/1	(36)/(58)	
	(16)/(31)/(47)	(16)/(47)	(35)/(51)								
5 inner	-0.49%	-69.2	-1.70	Comp	4-5 interface	1.18	30302.02	0.9	0.94	6.27	of -> max outer plies
4	-0.24%	-24.5	-3.49	Comp	3-4 interface	0.40	36535.31	1.1	1.14	6.31	
3	0.13%	13.2	6.44	Tens	2-3 interface	-0.64	35284.61	1.1	1.10	6.53	
2	0.35%	34.8	2.45	Tens	1-2 interface	-1.68	23106.17	0.7	0.72	9.97	
1 outer	0.56%	56.3	1.51	Tens	bottom of 1	-2.72	0.00	0.0	0.00		$\tau=0$ top/bottom
Allow, bending mt Mdb = Mdb req x Cf (Nmm/mm)				403			Allowable shear force F = F d req x Cf shear (N/mm)		25.25	6.27	Min Cf on shear
Analysis by simplified method of Table A.5											
b	kc	P	k2b	odf (col35)	K_{SH}	odf simplified Table C,5	tp req 2018	W, req	W, ratio		
mm	1.00	kN/m2	1		simplif	N/mm2	mm	kg/m2			
412	0.962	19.52	0.500	92.9	0.9	88.0	4.18	2.62	1.22		

Figure C.1: Analytical tool spreadsheet performing the scantling determination of the panel D6 from the Evolution 5.0 vessel.

Tables C.3, C.4 and C.5 present the results obtained from the analytical methods, the Abaqus software, and the SolidWorks software, respectively, for the Evolution 5.0 vessel.

Table C.3: Comparison between the results obtained analytically using the Enhanced and Simplified methods for the Evolution 5.0 vessel.

ID	Enhanced Method		Simplified Method	
	σ_i [MPa]	CF_{Enh}	$W_{f,req}$ [kg/m ²]	CF_{Simp}
A1	54.7	1.56	2.89	1.11
B2	59.5	1.43	3.02	1.06
C3	76.2	1.12	3.41	0.94
ABC4	49.4	1.73	2.75	1.16
D5	17.6	4.85	1.54	2.07
D6	56.3	1.51	2.62	1.22
E7	51.7	1.65	2.56	1.25
F8	24.3	3.50	1.24	2.58

Table C.4: Stress results in 1- and 2-directions and compliance factors, obtained with Abaqus software.

ID	P [kN/m ²]	$\sigma_{max,S11}$ [MPa]	$\sigma_{equi,1}$ [MPa]	$CF_{Abq,S11}$ [1]	$\sigma_{max,S22}$ [MPa]	$\sigma_{equi,2}$ [MPa]	$CF_{Abq,S22}$ [1]
A1	14.68	0.7	10.1	9.7	1.1	16.7	5.9
B2	19.05	-0.5	9.6	10.2	0.9	17.2	5.7
C3	18.44	-1.1	20.9	4.7	-1.7	30.5	3.2
ABC4	12.03	-2.4	29.1	3.4	-3.1	37.1	2.6
D5	24.43	-0.5	12.9	7.6	0.8	18.9	5.2
D6	19.52	-1.2	23.2	4.2	-1.9	36.4	2.7
E7	26.59	0.5	12.6	7.7	0.4	10.1	9.7
F8	23.32	-1.5	34.8	2.8	1.2	28.2	3.5

Table C.5: Von Mises stress results and compliance factors, obtained with SolidWorks software for Evolution 5.0 vessel.

ID	P [kN/m ²]	$\sigma_{max,vM}$ [MPa]	$\sigma_{equi,vM}$ [MPa]	$CF_{SW,vM}$ [1]
A1	14.68	1.0	14.0	7.0
B2	19.05	0.9	17.4	5.6
C3	18.44	2.1	38.9	2.5
ABC4	12.03	2.2	26.8	3.6
D5	24.43	0.6	14.0	7.0
D6	19.52	1.7	33.9	2.9
E7	26.59	0.6	14.8	6.6
F8	23.32	1.4	31.9	3.1