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Development and Behaviour of Unidirectional Double Hull Sandwich Plate System: Analytical Investigation

**by
Angelo Ferro**

**A thesis submitted to the
Faculty of Graduate Studies and Research
in partial fulfilment of the requirements
for the degree of
Master of Engineering**

**Department of Civil and Environmental Engineering
Carleton University
Ottawa, Ontario
January, 1998**

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Department of Civil and Environmental Engineering

**The Undersigned recommend to the Faculty of Graduate Studies
and Research acceptance of the thesis**

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Sandwich Plate System:
Analytical Investigation**

**submitted by
Angelo Ferro, B.Eng.**

**in partial fulfilment of the requirements for
the degree of Master of Engineering**

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January, 1998

Abstract

Tanker accidents and oil spills continue to occur even with improvements in navigation systems, redundancy in propulsion systems and the use of highly skilled pilots. Existing legislation from around the world mandates the use of double hull technology in an attempt to mitigate or eliminate oil spills. Current double hull technology includes two main variants: conventional and advanced double hulls which provide a two metre collision gap between hulls, in which the structure can collapse and objects can penetrate without directly breaching the inner hull. It is generally recognized by the research community that these all-steel design alternatives will not achieve the objective of zero oil outflow due to insufficient energy absorption capacity related to premature rupture of the hull configuration, and due to propagation of cracks from the ruptured outer hull into the cargo tanks during accidental or extreme load events.

The Unidirectional Double Hull Sandwich Plate System (UDHSPS) was developed to provide equivalent structural performance to the Conventional Double Hull (CDH) product oil tanker for normal operational loads, improved resistance and energy absorption for accidental or extreme load events, and an improved structural configuration to minimize maintenance, corrosion, and fatigue problems.

This thesis presents a series of numerical analyses, which demonstrate the use of finite elements in modeling composite structural laminate plate behaviour for use in double hull construction. Based on the analysis it was concluded that the local and global behaviour of the UDHSPS was equivalent to a CDH design produced according to Lloyd's Rules and Regulations.

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Table of Contents

Abstract.....	i
Acknowledgements.....	ii
Table of Contents.....	iii
List of Figures.....	viii
List of Tables.....	xvi
List of Symbols.....	xx
1. Introduction.....	1
1.1 General.....	1
1.2 Design of All-Steel Conventional Double Hull Oil Tanker According to Lloyd's Rules and Regulations (1994).....	3
1.2.1 Overview of Ship Components.....	5
1.2.2 Longitudinal Strength Requirements	7
1.2.3 Shear Load Calculations.....	12
1.2.4 Baseline 40,000 DWT Oil Tanker Design Summary.....	13
1.3 State-of-the-Art Review.....	18
1.3.1 Tanker Structure Behaviour During Collision and Grounding (Daidola, 1995).....	22
1.3.2 Modeling of the Internal Mechanics in Ship Collisions (Paik and Pedersen, 1996).....	23
1.3.3 Observations on Conventional and Advanced Double Hull Grounding Experiments (Rodd, 1996).....	25
1.3.4 Numerical Simulation of Actual Collision and Grounding Accidents (Kuroiwa, 1996).....	27

1.3.5	Strength Analysis of a New Double Hull Structure for VLCC in Collision (Sano, Muragashi, Yoshikawa, Motoi, Murakami, 1996).....	30
1.3.6	Comparative Study on Collision Resistance of Side Structure (Kitamura, 1996).....	34
1.3.7	Collision Simulation of a Double-Hulled Structure with Uni-Directional Girder System (Mizukami, Tanigawa, Nagahama, Kamei, 1996).....	38
1.4	The Unidirectional Double Hull Sandwich Plate System (UDHSPS).	43
1.5	Objectives and Scope.....	48
1.6	Thesis Outline	49
2.	Elastomer Behaviour and Finite Element Analysis (FEA).....	51
2.1	General.....	51
2.2	Polyurethane Elastomers: Chemistry and Properties.....	51
2.3	Elastomer Specification and Related Information.....	54
2.4	Finite Element Modeling Theory.....	60
2.4.1	Full Newton-Raphson (NROPT) Solution Option and Convergence.....	60
2.4.2	Non-linear Geometry (NLGEOM) Option.....	62
2.4.3	Stress Stiffening (SSTIFF) Option.....	62
2.5	Finite Element Modeling of Polyurethane Elastomer Behaviour.....	63
2.5.1	Mooney-Rivlin Constitutive Law.....	64
2.5.2	Polyurethane Elastomer Core Material Characteristics and Mooney-Rivlin Constants.....	65
2.5.3	Modified Strain Energy Density.....	68

2.5.4	HYPER58 3-D 8-Node Mixed U-P Hyperelastic Solid Element.....	68
2.6	Analytical Verification of Tension Coupon Test.....	69
2.6.1	Problem Description.....	69
2.6.2	Modeling Notes and Analysis Assumptions.....	70
2.6.3	Results.....	72
2.7	Analytical Verification of Compression Disk Test.....	73
2.7.1	Problem Description.....	73
2.7.2	Modeling Notes and Analysis Assumptions.....	74
2.7.3	Results.....	76
3.	Finite Element Analysis and Verification of Composite Structural Laminate Plates and Double Hull Substructures.....	78
3.1	General.....	78
3.2	Characteristic Buckling Modes for Sandwich Plates.....	80
3.3	Element and Material Constitutive Model Selection for the Composite Structural Laminate and Steel Plates.....	83
3.4	Parametric Study of Mesh Density and Core Characteristics of Composite Structural Laminate Plates (Sandwich Panels) in Buckling Analysis.....	85
3.4.1	Problem Description.....	85
3.4.2	Modeling Notes and Analysis Assumptions.....	85
3.4.3	Results and Discussion.....	89
3.5	Stiffened Steel Plate Buckling Analysis.....	94
3.5.1	Problem Description.....	94

3.5.2	Modeling Notes and Analysis Assumptions.....	94
3.5.3	Results and Discussion.....	96
3.6	Equivalent Composite Structural Laminate Capacity	98
3.6.1	Problem Description.....	98
3.6.2	Modeling Notes and Analysis Assumptions.....	99
3.6.3	Results and Discussion.....	101
3.7	Analysis of Large-scale Experiment.....	104
3.7.1	Problem Description.....	104
3.7.2	Modeling Notes and Analysis Assumptions.....	110
3.7.3	Results and Discussion.....	115
3.7.4	Summary.....	149
4.	Midship Section Equivalency for Normal Operating Loads.....	160
4.1	Equivalency by Rationally Based Design.....	160
4.2	Preliminary Finite Element Analyses.....	161
4.2.1	Modeling Stiffened Plates.....	163
4.2.2	Modeling Composite Structural Laminate Plates for the UDHSPS.....	169
4.3	CDH Substructure Hull Module Analysis.....	172
4.3.1	Problem Description.....	172
4.3.2	Modeling Notes and Analysis Assumptions.....	172
4.3.3	Results and Discussion.....	179
4.4	UDHSPS Substructure Hull Module Analysis.....	184
4.4.1	Problem Description.....	184

4.4.2	Modeling Notes and Analysis Assumptions.....	184
4.4.3	Results and Discussion.....	189
5.	Summary, Conclusions, Recommendations, and Future Work.....	195
5.1	Summary.....	195
5.2	Conclusions.....	200
5.3	Recommendations and Future Work.....	201
	References.....	204
A.	Design Computations for the Midship of a 40,000 DWT Conventional Double Hull Oil Tanker.....	207
A.1	General.....	207
A.2	Strake Scantlings.....	207
A.2.1	Hull Envelope Plating.....	210
A.3	Transverse Elements.....	216
A.3.1	Plane Transverse Oiltight Bulkhead Scantlings.....	218
A.3.2	Vertically Corrugated Transverse Bulkhead.....	220
A.3.3	Double Plate Transverse Bulkhead.....	223
A.3.4	Deck Transverse Girders.....	225
A.4	Stiffener Scantlings.....	226
A.5	Design Summary.....	238

List of Figures

1.1	Conventional Double Hull (CDH) Product Oil Tanker Midship Section, Cross-section.....	6
1.2	Conventional Double Hull (CDH) Configuration with Corrugated Transverse Bulkhead.....	7
1.3	Design Wave Shear Force at any Position along the Oil Tanker Length.....	13
1.4	Hull Envelope Strake Scantlings and Identification Numbers.....	17
1.5	Hull Envelope Stiffener Scantlings and Group Identification Numbers.....	17
1.6	Double Plate Bulkhead and Transverse Framing Scantlings.....	18
1.7	Conventional and Advanced Double Hull Designs (after Rodd, 1996).....	21
1.8	Load Transmission Mechanism (after Daidola, 1995).....	21
1.9	Typical Model Installation.....	25
1.10	Conventional and Proposed Web Stiffener Detail (after Sano et al., 1996).....	31
1.11	Side Structure and Bow Model (after Sano et al., 1996).....	33
1.12	Section of VLCC with a New Double Side Design (after Ohtsubo et al., 1997).....	35
1.13	Simulation Model for Side Collision (after Kitamura, 1996).....	36
1.14	Comparison Energy Absorption Capacity (after Kitamura, 1996).....	38
1.15	Double Hull Structure with Unidirectional Girder System (after Mizukami et al., 1996).....	39
1.16	Collision Conditions (after Mizukami et al., 1996).....	41

1.17	Finite Element Mesh of the Collision Zone of the Struck Ship (after Mizukami et al., 1996).....	43
1.18	UDHSPS Design Concept.....	45
2.1	Basic Polyurethane Elastomer Chemical Structure.....	52
2.2	Tensile and Compressive Stress-Strain Curves for Polyurethane Elastomers.....	56
2.3(a)	Schematic of the RIM System.....	59
2.3(b)	Typical RIM Equipment (after Elastogran GmbH, 1995).....	59
2.4	Mooney-Rivlin and Experimental Stress-Strain Curves for Elastomers A and B.....	66
2.5	Mooney-Rivlin Stress-Strain Curves.....	67
2.6	Hyperelastic Mixed U-P Solid Element Type.....	68
2.7	Typical Tension Coupon.....	69
2.8	Problem Description and Representative Finite Element Model for the Tension Coupon.....	71
2.9	Comparison of Load-Elongation Curve for Tension Test.....	72
2.10	Coupon Deformed Shape.....	73
2.11	Typical Compression Disk.....	74
2.12	Problem Description and Representative Finite Element Model for Compression Disk.....	75
2.13	Load-Deflection Results Comparison for Compression Test.....	77
2.14	Compression Disk Deformed Shape.....	77
3.1	Application of Structural Laminate Construction in Product Oil Tankers.....	79
3.2	Geometric Properties.....	81

3.3	Characteristic Buckling Modes.....	81
3.4	Problem Description and Representative Finite Element Models for Sandwich Plate Buckling Parametric Study.....	87
3.5	Mesh Densities.....	88
3.6(a)	Out-of-plane Displacements (in mm) Obtained with Mesh Density [3] and Elastomer (A) Core.....	91
3.6(b)	Axial Load vs. Axial Shortening for a 2,000 mm by 3,600 mm Composite Plate with Elastomer (A) Core.....	91
3.7(a)	Out-of-plane Displacements (in mm) Obtained with Mesh Density [3] and Elastomer (B) Core.....	92
3.7(b)	Axial Load vs. Axial Shortening for a 2,000 mm by 3,600 mm Composite Plate with Elastomer (B) Core.....	92
3.8(a)	Out-of-plane Displacements (in mm) Obtained with Mesh Density [3] and Elastomer (C) Core.....	93
3.8(b)	Axial Load vs. Axial Shortening for a 2,000 mm by 3,600 mm Composite Plate with Elastomer (C) Core.....	93
3.9	Problem Description and Representative Finite Element Model for Stiffened Panel Analysis.....	95
3.10	Load vs. Axial Shortening for Stiffened Plates 1 to 5.....	97
3.11	Deformed Shape and Longitudinal Stresses (in MPa) at the Buckling Load for Plate 1.....	98
3.12	Problem Description and Representative Finite Element Models for Composite Structural Laminate Plate Buckling Equivalency Study.....	100
3.13	P/PY vs. Average Steel Plate Thickness, t_{avg}	103
3.14	Buckling Load vs. Average Steel Plate Thickness, t_{avg}	103
3.15	Large-scale Experiment Specimen.....	106
3.16	Test Specimen, Horizontal Reaction Frame, Knife-Edge-Roller Assemblies, and Actuators.....	107

3.17	Problem Description and Representative Finite Element Model for the Large-scale Test Specimen.....	108
3.18	LVDT Locations on Test Specimen.....	109
3.19(a)	Typical Deformed Shape Taken Through a Load Point for the Test.....	114
3.19(b)	Schematic of the Displacement Load Application.....	114
3.20	Total Applied Load vs. Transverse Displacement for Centreline LVDT.....	115
3.21(a)	Von Mises Element Stresses (in MPa) at First Yield of Outer Hull in Flexure, 697 kN.....	117
3.21(b)	Von Mises Element Stresses (in MPa) at First Yield of Outer Hull in Flexure, 697 kN.....	118
3.22(a)	Von Mises Element Stresses (in MPa) at Inelastic Buckling of Transverse Floor, 2925 kN.....	119
3.22(b)	Von Mises Element Stresses (in MPa) at Inelastic Buckling of Transverse Floor, 2925 kN.....	120
3.23(a)	Von Mises Element Stresses (in MPa) after Inelastic Buckling of Longitudinal Girders, 7389 kN.....	121
3.23(b)	Von Mises Element Stresses (in MPa) after Inelastic Buckling of Longitudinal Girders, 7389 kN.....	122
3.24(a)	Von Mises Element Stresses (in MPa) at Shear-tension Failure of Outer Plate, 8197 kN.....	123
3.24(b)	Von Mises Element Stresses (in MPa) at Shear-tension Failure of Outer Plate, 8197 kN.....	124
3.25(a)	Von Mises Element Stresses (in MPa) at Ultimate, 9762 kN.....	125
3.25(b)	Von Mises Element Stresses at Ultimate, 9762 kN.....	126
3.26	Total Applied Load vs. Transverse Displacement for LVDT 1.....	128
3.27	Total Applied Load vs. Transverse Displacement for LVDT 2.....	129
3.28	Total Applied Load vs. Transverse Displacement for LVDT 3.....	129

3.29	Total Applied Load vs. Transverse Displacement for LVDT 4.....	130
3.30	Displaced Shape for Various Displacement Load Models.....	132
3.31(a)	Total Applied Load vs. Transverse Displacement for LVDT 5.....	133
3.31(b)	Analytical Model for Spacer.....	133
3.32	Total Applied Load vs. Transverse Displacement for LVDT 6.....	134
3.33	Total Applied Load vs. Transverse Displacement for LVDT 7.....	134
3.34	Total Applied Load vs. Transverse Displacement for LVDT 8.....	135
3.35	Total Applied Load vs. Transverse Displacement for LVDT 9.....	135
3.36	Total Applied Load vs. Transverse Displacement for LVDT's 10 and 14.....	136
3.37(a)	Total Applied Load vs. Transverse Displacement for LVDT's 11 and 15.....	138
3.37(b)	System Behaviour at Longitudinal/Transverse Junction.....	138
3.37(c)	Compression of Core.....	138
3.38	Total Applied Load vs. Transverse Displacement for LVDT's 12 and 16.....	139
3.39	Total Applied Load vs. Transverse Displacement for LVDT's 13 and 17.....	139
3.40	Total Applied Load vs. Transverse Displacement for LVDT's 18 and 22.....	141
3.41	Total Applied Load vs. Transverse Displacement for LVDT's 19 and 23.....	141
3.42	Total Applied Load vs. Transverse Displacement for LVDT 24.....	142
3.43	Total Applied Load vs. Transverse Displacement for LVDT 21.....	142
3.44(a)	Total Applied Load vs. Transverse Displacement for LVDT 26.....	143
3.44(b)	Deformed Shape due to the Initial Load Step	143

3.45(a) Total Applied Load vs. Transverse Displacement for LVDT 27 and 30.....	145
3.45(b) Transverse Floor Buckling Verification.....	145
3.46 Total Applied Load vs. Transverse Displacement for LVDT's 28 and 31.....	146
3.47 Total Applied Load vs. Transverse Displacement for LVDT's 29 and 32.....	146
3.48 Total Applied Load vs. Transverse Displacement for LVDT 34.....	148
3.49 Total Applied Load vs. Transverse Displacement for LVDT's 33 and 35.....	148
3.50 Element Stresses (in MPa) and Their Progression in the X-Direction for the Outer Plate of Outer Hull.....	150
3.51 Element Stresses (in MPa) and Their Progression in the Y-Direction for the Outer Plate of Outer Hull.....	151
3.52 Element Stresses (in MPa) and Their Progression in the X-Direction for the Inner Plate of Outer Hull.....	152
3.53 Element Stresses (in MPa) and Their Progression in the Y-Direction for the Inner Plate of Outer Hull.....	153
3.54 Element Stresses (in MPa) and Their Progression in the Z-Direction for the Transverse Floor.....	154
3.55 Element Stresses (in MPa) and Their Progression in the Y-Direction for the Transverse Floor.....	155
3.56 Element Stresses (in MPa) and Their Progression in the Z-Direction for the Longitudinal Girder.....	156
3.57 Element Stresses (in MPa) and Their Progression in the X-Direction for the Longitudinal Girder.....	157
3.58 Element Stresses (in MPa) and Their Progression in the X-Direction for the Inner Hull.....	158
3.59 Element Stresses (in MPa) and Their Progression in the Y-Direction for the Inner Hull.....	159

4.1	Levels of Structural Analysis in Rationally-Based Design (after Hughes, 1988).....	162
4.2	Boundary Restraints for Doubly Symmetric Hull Module.....	163
4.3	Applied Stresses for a Stiffened Plate	166
4.4	Axial Load vs. Axial Shortening for Stiffened and Orthotropic Plates.....	169
4.5	Fine and Coarse Mesh Composite Structural Laminate Plate Models.....	170
4.6	Axial Load vs. Axial Shortening for 14,000 mm Long Composite Structural Laminate Plates.....	171
4.7	Problem Definition and Representative Finite Element Model for the CDH Product Oil Tanker Substructure Hull Module Analysis.....	173
4.8	Summary of Strake Locations.....	176
4.9	Hydrostatic Pressure Application.....	177
4.10	Neutral Axis Along Section a-a.....	178
4.11	Shear Force Application.....	179
4.12	CDH Element Longitudinal Stresses (in MPa) for Hogging Condition.....	181
4.13	CDH Element Longitudinal Stresses (in MPa) for Sagging Condition.....	182
4.14	Location of Correction Factor, r.....	183
4.15	Problem Definition and Representative Finite Element Model for the UDHSPS-3GS Product Oil Tanker Substructure Hull Module Analysis.....	186
4.16	Problem Description and Representative Finite Element Model for the UDHSPS-2GS Product Oil Tanker Substructure Hull Module Analysis.....	187
4.17	UDHSPS-3GS Hull Envelope Modification and Transverse Framing Location.....	188
4.18	UDHSPS-3GS Element Longitudinal Stresses (in MPa) for Hogging Condition.....	190

4.19	UDHSPS-3GS Element Longitudinal Stresses (in MPa) For Sagging Condition.....	191
4.20	UDHSPS-2GS Element Longitudinal Stresses (in MPa) for Hogging Condition.....	92
4.21	UDHSPS-2GS Element Longitudinal Stresses (in MPa) for Sagging Condition.....	193
5.1	Summary of Thesis Components.....	199
5.2	Collision Analysis Model.....	203
A.1	Strake Locations and Numbers.....	210
A.2	Plane Transverse Bulkhead.....	217
A.3	Corrugated Bulkhead Supported by Stool Structure.....	217
A.4	Double Plate Bulkhead.....	218
A.5	Cross-section of Corrugated Bulkhead.....	220
A.6	Corrugated Transverse Oiltight Bulkhead.....	222
A.7	Typical Deck Transverse Girder Cross-Section.....	226
A.8	Stiffener Groups Configuration.....	228
A.9	Geometric Properties of Stiffened Plate.....	236
A.10	Hull Envelope Strake and Stiffener Scantlings.....	238
A.11	Plane Bulkhead and Transverse Framing Scantlings.....	239
A.12	Corrugated Bulkhead and Transverse Framing Scantlings.....	240
A.13	Double Plate Bulkhead and Transverse Framing Scantlings.....	240

List of Tables

1.1	Total Losses in 1995 (after Ohtsubo et al., 1997).....	1
1.2	40,000 DWT UDHSPS Oil Tanker Principle Dimension Specifications.....	4
1.3(a)	Section Geometric Properties.....	11
1.3(b)	Permissible, Actual, and Design Stresses.....	11
1.4	Summary of Actual and Rule Strake Scantlings.....	14
1.4	Continued. Summary of Actual and Rule Strake Scantlings.....	15
1.5	Summary of Stiffener Dimensions and Modulus Values.....	16
1.6	Responses of CDH and ADH Structural Elements to Accidental or Extreme Load Events.....	24
1.7	Grounding Experiment Results.....	26
1.8(a)	Collision Simulation Details, Kuroiwa, (1996).....	28
1.8(b)	Grounding Simulation Details, Kuroiwa, (1996).....	29
1.9	Collision Simulation Details, Sano et al. (1996).....	32
1.10	Penetration, Reaction Force, and Absorbed Strain Energy.....	33
1.11	Collision Simulation Details, Kitamura (1996).....	36
1.12	Rupture Opening Initiation and Absorbed Energy (after Kitamura, 1996).....	37
1.13	Crashworthiness Index (after Kitamura, 1996).....	37
1.14	Collision Simulation Details – Bow Configuration Investigation (Mizukami et al., 1996).....	40
1.14	Continued. Collision Simulation Details – Bow Configuration Investigation (Mizukami et al., 1996).....	41

1.15	Collision Simulation Details – Hydrodynamic Response (Mizukami et al., 1996).....	42
1.16	State-of-the-art Summary.....	46
1.17	UDHSPS Design Alternative.....	47
1.17	Continued. UDHSPS Design Alternative.....	48
2.1	Intermediates and Related Properties	53
2.2	Thermoplastic Polyurethane Elastomer Characteristic Properties (after ASM International, 1988).....	53
2.3	General Specifications for Elastomer Properties.....	55
2.4	Polyurethane Elastomer RIM Formulation.....	65
2.5	Mooney-Rivlin Constants for Elastomers (A), (B), and (C).....	67
3.1	Characteristic Local Buckling Modes	82
3.2	Characteristic Global Buckling Modes.....	83
3.3	SHELL91 and SHELL93 Element Description.....	84
3.4	True Stress-Strain Data Input for Steel Faces of the Sandwich Panel (after Toscano, 1995).....	86
3.5	Summary of Buckling Solutions for Composite Structural Laminate Plates.....	90
3.6	Summary of Stiffened Panel Buckling Results for a Plate 3,000 mm Wide by 3,600 mm Long.....	97
3.7	Buckling Capacity of Composite Structural Laminate Plates with Width of 3,000 mm.....	102
3.8	Buckling Capacity of Composite Structural Laminate Plates with Width of 2,000 mm.....	102
3.9	True Stress-Strain Data for Steel Plates.....	111

3.10	Key Behavioural Events of the UDHSPS during Transverse Loading.....	116
4.1	SHELL63 Element Description.....	167
4.2	Orthotropic Properties for Stiffened Plate.....	168
4.3	Summary of Material Properties Data.....	174
4.3	Continued. Summary of Material Properties Data.....	175
4.4	Ratio of the Moment Applied to Nodes along the Neutral Axis.....	178
4.5	Longitudinal Stresses in Five Stiffened Panels.....	183
4.6(a)	Ratio of Moment Applied to Nodes along the Neutral Axis For UDHSPS-3GS Substructure Hull Module.....	188
4.6(b)	Ratio of Moment Applied to Nodes along the Neutral Axis For UDHSPS-2GS Substructure Hull Module.....	189
4.7	Average Longitudinal Element Stresses.....	194
4.8	Midship Displacement and Rotation at Section a-a.....	194
A.1	Applicable Rules Required for CDH Tanker Design.....	208
A.1	Continued. Applicable Rules Required for CDH Tanker Design.....	209
A.2	Minimum Hull Envelope Plating Thickness - LR P4 Ch.9 T9.4.1.....	211
A.3	Taper Requirements for Hull Envelope - LR P4 Ch.3 T3.2.1.....	213
A.4	Elastic Critical Buckling Stress of Plating - LR P3, Ch.4 T4.7.2.....	214
A.5	Inner Hull and Longitudinal Oiltight Bulkhead Scantlings - LR P4, Ch.9, Table 9.6.1 -.....	215
A.6(a)	Plane Transverse Oiltight Bulkhead Scantlings - LR P4, Chapter 9, Table 9.7.1 -.....	219
A.6(b)	Vertically Corrugated Transverse Oiltight Bulkhead Scantlings - LR P4, Chapter 9, Table 9.7.1 -.....	221

A.7	Deep Tank Bulkhead Scantlings – LR P4, Ch.1, Table 1.9.1.....	222
A.8	Symmetrical corrugations bulkhead (additional requirements) – LR P4, Ch.1, Table 1.9.2 –.....	223
A.9(a)	Double Plate Bulkhead Scantlings – LR P4, Chapter 1, Table 1.9.1.....	224
A.9(b)	Double Plate Bulkhead Scantlings – LR P4, Chapter 1, Table 1.9.1.....	224
A.9(c)	Double Plate Bulkhead Scantlings – LR P4, Chapter 1, Table 1.9.1.....	224
A.10	Deck Transverse Scantling Requirements.....	225
A.11	List of Applicable Rules for the Design of Stiffeners.....	227
A.12	F_1 Values.....	229
A.13	F_2 Values.....	230
A.14	Deck, Side and Bottom Longitudinals – LR P4, Chapter 9, Section 5.3.1.	231
A.15	Side and Bottom Longitudinals – LR P4, Chapter 1, Table 1.6.1.....	233
A.16	Inner Hull and Longitudinal Oiltight Bulkhead Scantlings – LR P4, Chapter 9, Table 9.6.1.....	235
A.17	Summary of Stiffener Modulus Calculations.....	237

List of Symbols

2GS	2 metre girder spacing
3GS	3 metre girder spacing
A	Cross-sectional area
ADH	Advanced (or unidirectional) double hull
a_{kl}	Mooney-Rivlin constants
ASTM	American Society for Testing Materials
B	Moulded breadth
B	Breadth of stiffened plate
B	Buoyancy
C_b	Block coefficient
CDH	Convention double hull
C_{ij}	Components of the Cauchy-Green deformation tensor
D	Moulded depth
d	Depth of sandwich from centre of top face to centre of bottom face
DOF	Degree of freedom
DWT	Dead weight tonne
E	Modulus of elasticity

E_c	Modulus of elasticity of core material
E_c	Modulus of elasticity of core material in compression
E_f	Modulus of elasticity of facing material
E_{ij}	Components of the Lagrangian strain tensor
E_t	Modulus of elasticity of core material in tension
E_x	Modulus of elasticity in the X-direction for orthotropic plates
E_y	Modulus of elasticity in the Y-direction for orthotropic plates
E_z	Modulus of elasticity in the Z-direction (out-of-plane direction)
F_B	Local scantling reduction factor at bottom
F_D	Local scantling reduction factor at deck
FEA	Finite element analysis
G	Shear modulus
G_c	Shear modulus of core material
h	Load head
HPIM	High Pressure Impingement Mixing
I	Moment of inertia
I_i	Invariants of the right Cauchy-Green deformation tensor
k	Steel strength factor
L	Rule length

ℓ	Gauge length
l	Length of sandwich panel
L	Plate length
LIM	Liquid Injection Moulding
L_L	Length on load waterline
L_{pp}	Length between perpendiculars
LR.PASS	Lloyd's Register Design Software
LVDT	Linear Variable Differential Transformer
MDI	4,4' – diphenylmethane diisocyanate
M_s	Still water sagging/hogging moment
M_w	Vertical wave sagging/hogging moment
M_{wo}	Unfactored vertical wave sagging and hogging moment
M_z	Longitudinal bending moment
NLGEOM	Non-linear geometry analysis option
NROPT	Newton-Raphson analysis option
P	Maximum compressive load
P_Y	Yield load
Q	Shear force
Q_w	Design hull wave shear force
r	Ratio of total cross-sectional area to plate area of a stiffened panel
R_B	Bilge radius
RIM	Reaction Injection Moulding

ROTX	Rotation about the X-axis
ROTY	Rotation about the Y-axis
ROTZ	Rotation about the Z-axis
s	Composite structural laminate plate width
S	Spacing of primary members
s	Stiffener spacing
SID	Ship Initial Design
S_{ij}	Stress component
SSTIFF	Stress stiffening analysis option
T	Draught for scantlings
t	Strake scantling (plate thickness)
t₁	Thickness of top face of sandwich panel
t₂	Thickness of bottom face of sandwich panel
t_{avg}	Average steel plate thickness
t_c	Thickness of core layer of sandwich panel
T_g	Glass transition temperature
t_s	Steel plate thickness
UDHSPS	Unidirectional Double Hull Sandwich Plate System

u_i	Deflection at iteration i
u_{ref}	Reference value for deflection convergence
U_X	Translation in the X-direction
U_Y	Translation in the Y-direction
U_Z	Translation in the Z-direction
VLCC	Very Large Crude Carrier
W	Strain energy density function
W	Weight
Z_B	Section modulus to bottom
Z_D	Section modulus to deck
Z_{min}	Minimum required section modulus

δ_p	Stiffened plate length change
δ_s	Stiffener length change
ϵ_{eng}	Engineering strain
ϵ_{ln}	Logarithmic (true) strain
ϵ_R	Convergence tolerance
ϵ_x	Strain in the X-direction
ϵ_y	Strain in the Y-direction
γ	Shear strain
κ	Bulk modulus
ν	Poisson's ratio
θ_X	Rotation about the X-axis
θ_Y	Rotation about the Y-axis
θ_Z	Rotation about the Z-axis
σ	Elastic buckling stress
σ_B	Maximum hull bending stress at bottom
σ_c	Maximum compressive hull vertical bending stress
σ_{cr}	Shear crimping stress
σ_D	Maximum hull vertical bending stress at deck

σ_{eng}	Engineering stress
σ_p	Average tensile stress in the stiffened plate
σ_s	Compressive stress in the stiffeners
σ_{true}	True stress
σ_{wra}	Critical buckling stress for antisymmetric face wrinkling failure
σ_{wrs}	Critical buckling stress for symmetric face wrinkling failure
σ_x	In-plane stress for orthotropic plate in the X-direction
σ_y	In-plane stress for orthotropic plate in the Y-direction
τ	Shear stress

Chapter 1

Introduction

1.1 General

Accidents continue to occur even with improvements in navigation systems (automation, global positioning satellite system technology), redundancy of propulsion systems, and the use of highly skilled pilots. The reasons for this are that traffic on the navigation routes is increasing, mechanical breakdowns occur, and the oceans/seas can be a hostile environment. The increase in accidents is illustrated by casualty statistics reported by Lloyd's Register of Shipping for 1995 (given in Table 1.1), where 188 ships greater than 100 GT were declared a total loss for that year. Twenty nine percent of these ships were lost due to collision, grounding, or contact. These ships represent 62% of the gross tonnage lost during that year. Thus, it is incumbent upon the naval architecture community to develop better structural systems to mitigate or eliminate oil outflow in accidental or extreme load events such as collisions or groundings.

Table 1.1 Total Losses in 1995 (after Ohtsubo et al., 1997)

Cause of casualty	Number of total losses	Gross Tonnage
Foundered	99	192070
Missing	2	3175
Fire/explosion	27	55693
Collision	17	134310
Grounding	31	312839
Contact	7	116456
Other	5	90271
Total	188	908814

There are a number of recognized structural problems or deficiencies inherent in existing and proposed all-steel double hull designs which make it virtually impossible to obtain the zero oil outflow objective. They are described as follows:

- Rodd (1996), demonstrated through large-scale tests that regardless of the variation in all-steel double hull tanker design, at least one cargo tank would be breached due to the propagation of cracks from the ruptured outer hull through the longitudinal girders and transverse elements to the inner hull.
- Kitamura (1996), has demonstrated by numerical collision simulations (using transient finite element analyses which include weld failures, material strain rate effects, and ductile rupture of steel plates) that simplified side structures composed of flexible hulls and stringers alone, have increased energy absorption capability over conventional stiffened plate all-steel hull structures.
- Conventional Double Hull (CDH) product oil tankers are susceptible to fatigue problems and increased corrosion due to the structural complexity and large number of connections between longitudinal stiffeners and transverse members, Daidola (1995). The accelerated deterioration of the structure (decreased service life) in combination with fatigue cracks may lead to a breach of the inner hull.
- Stiffened plates are prone to premature rupture due to the exhaustion of the ductility of the steel at locations of concentrated strains (i.e., sharp curvature due to bending about the hull plate, stiffener junction), Motoi et al. (1995) and Rodd (1996).
- The advantages accrued in implementing Advanced (or unidirectional) Double Hull (ADH) configurations may be limited to tankers in the range of 40,000 DWT to

50,000 DWT due to the limited buckling capacity of the unstiffened hull plates, Roseman et al. (1995). This implies that this technology cannot currently be applied to large tankers and that these must be constructed using CDH technology. With CDH tanker structures the transverse floors are more susceptible to inner hull breach from crack propagation, from rupture of the outer hull during accidental or extreme load events and from crack propagation of fatigue cracks at fatigue prone details.

The underlying objective is to resolve the inherent structural problems associated with all-steel tanker design by developing a better structural system. To accomplish this, an understanding of the behaviour and analysis of oil tankers subject to these extreme load events (i.e. mechanisms of grounding and collision failures) and the construction, sizing, detailing, and design criteria of conventional double hull design is required.

1.2 Design of All-Steel Conventional Double Hull Oil Tanker According to Lloyd's Rules and Regulations (1994)

Current design practice is important since it provides a baseline CDH oil tanker structure, which can be used to establish equivalency for a tanker that has been designed with a new structural system. Part 4, Chapter 9, Section 1.2.12 of Lloyd's Register (1994) states the following:

“Alternative arrangements which are proposed as being equivalent to the Rules will receive individual consideration, taking into account any relevant National Authority requirements.”

This clause provides the grounds for developing new designs based on alternative design philosophies.

In this section, the design of an all-steel 40,000 DWT CDH oil tanker, completed according to Lloyd's Rules and Regulations (LR) for oil tankers, is described. Note, other researchers have also used this tanker size for comparative design studies involving new tanker structures (one such tanker is the MARAD PD-328 Double Hull Product Tanker, U.S. Department of Transportation (1991)). The design specifications for the baseline oil tanker are given in Table 1.2. Further explanation and definition of the terms in the table can be found in LR Part 3, Chapter 1.6.

Table 1.2 40,000 DWT UDHSPS Oil Tanker Principle Dimension Specifications

Principal Dimensions Specification (Units SI)	
Deadweight	40,000 t
Length Between Perpendiculars, L_{pp} (LR P.3 Ch.1, 6.1.7)	180.0 m
Length on Load Waterline, L_L (LR P.3 Ch.1, 6.1.8)	180.0 m
Rule length, L ($=97\% \times 180$ m) (LR P.3 Ch.1, 6.1.1)	174.6 m
Moulded Breadth, B (LR P.3 Ch.1, 6.1.3)	32.0 m
Moulded Depth, D (LR P.3 Ch.1, 6.1.4)	20.0 m
Draught for Scantlings, T (LR P.3 Ch.1, 6.1.5)	12.0 m
Rule Block Coefficient, C_b (LR P.3 Ch.1, 6.1.6)	0.85
Displacement	58414.170 t
Service Speed	16.0 knots

It is assumed that,

- the baseline CDH tanker operates under unrestricted sea going service, hence the service factor is 1.0 as per LR Part 3, Chapter 4, Section 5; and

- the tanker is constructed using typical mild steel for marine applications with a yield stress of 235 MPa. According to LR Part 3, Chapter 2, Section 1, the material strength factor k is equal to 1.0.

The design was completed using LR.PASS SID-4 ship initial design software developed by Lloyd's Register. The program gives the strake and stiffener scantlings of a defined ship structure. A detailed set of these calculations is given in Appendix A.

1.2.1 Overview of Ship Components

A general description of the structural elements for a 40,000 DWT CDH product oil tanker designed according to Lloyd's Rules and Regulations (1994) is given in this section. A typical midship cross-section is shown in Figure 1.1. The elements are identified by numbers on the cross-section and listed in the corresponding table. The circles indicate the strake seams that represent butt welds, which join full plate widths (strakes). The distance between strake seams is the strake width and must be no greater than 3,000 mm.

The width of the keel is 1,800 mm. The bilge and gunwale radii are both 2,000 mm and the sheerstrake depth is 2,000 mm. The deck has a knuckle at its centre, elevated at 1,500 mm with respect to the side. The bottom structure is 24,000 mm wide and is comprised of both an inner and outer shell separated by nine, 2,000 mm deep, bottom girders (also referred to as longitudinal girders), spaced at 3,000 mm on centre. The longitudinal bulkheads or inner hull runs between the inner bottom and the deck structure and is located 2,000 mm from the side shell. Longitudinal stringers, 10 mm thick, span

2,000 mm between the side shell and longitudinal bulkheads and provide 3,000 mm high corridors within this part of the structure. The strakes span 3,600 mm between transverse floors and are stiffened with longitudinal stiffeners at 600 mm on centre, as illustrated in Figure 1.2.

Item No.	Item
1	Deck
2	Sheerstrake and gunwale
3, 4, 5, and 6	Side shell
7	Bilge
8	Bottom shell
9	Keel
10	Inner bottom
11	Bottom centreline girder
12	Side girders
13	Longitudinal bulkhead

Figure 1.1 Conventional Double Hull (CDH) Product Oil Tanker Midship Section, Cross-section

There are seven cargo tanks along the ship length, each of which are separated by transverse bulkheads located every 18,000 mm. Each cargo tank is bound on either side by the longitudinal bulkheads (inner hull), on the bottom by the inner bottom and on top by the deck. The space between the inner and outer hulls along the bottom and side structure is used for ballast tanks.

Figure 1.2 Conventional Double Hull (CDH) Configuration with Corrugated Transverse Bulkhead

1.2.2 Longitudinal Strength Requirements

The most important load effects in the analysis and design of ship structures are the design hogging and sagging bending moments along the length of the vessel which are determined by superimposing the wave action and still water effects on the structure.

Hughes, (1988). This is reflected in all classification society rules for determining scantlings. Lloyd's Register developed a program LR.PASS Ship Initial Design (SID-4) to determine the Rule minimum scantlings for longitudinal elements, which meet both the local and longitudinal (global) strength requirements according to Lloyd's Rules and Regulations (1994). The program generates a midship section based on the design parameters and structural form given in Table 1.1 and Figures 1.1 and 1.2. An outline of the iterative design process used in the program follows:

STEP 1 The required vertical wave sagging and hogging moment resistances at midship are calculated according to LR Part 3, Chapter 4, Section 5.2, given as Equation 2.1, to be $M_w = -1,554,500$ kN.m and $M_w = 1,472,400$ kN.m, respectively.

$$M_w = f_1 f_2 M_{wo} \quad (2.1)$$

where, $f_1 = 1.0$ for unrestricted sea-going service
 $f_2 = -1.1$ for sagging moment
 $f_2 = \frac{1.9C_b}{(C_b + 0.7)} = 1.042$ for hogging moment given a block coefficient, C_b , of 0.85

$M_{wo} =$ Unfactored vertical wave sagging and hogging moment, as given by

$$M_{wo} = 0.1C_1C_2L^2B(C_b+0.7), \text{ and has a value of } 1,413,100 \text{ kN.m} \quad (2.2)$$

in which, $C_1 = 10.75 - \left(\frac{300-L}{100} \right)^{1.5}$, and has a value of 9.346
 $C_2 = 1.0$ at midships
 $L =$ Rule length.

STEP 2 The minimum permissible still water bending moment, $\overline{M}_{\min}$, is calculated according to LR Part 3, Chapter 2, Section 5.4, 5.5, and 5.6., in which the requirement is that $\overline{M}_{\min}$ is the lesser of the permissible still water sagging and hogging bending moment resistances, according to Section 5.5, which are given by Equation 2.3(a) and 2.3(b)

$$|\overline{M}_s| = F_D \sigma Z_D \times 10^3 - |M_w| \quad (2.3a)$$

$$|\overline{M}_h| = F_B \sigma Z_B \times 10^3 - |M_w| \quad (2.3b)$$

where, σ = The permissible combined (still water plus wave) stresses for hull vertical bending. It is equal to $175/k_L$. For this case, the higher tensile steel factor, k_L , taken for mild steel is 1.0, thus the value for σ is 175 MPa, as defined in Section 5.6.

F_D, F_B = Local scantling reduction factors above and below the mid-depth of the tanker, respectively. The scantling reduction is a ratio of the maximum hull vertical bending stresses at the deck, σ_D and the keel, σ_B to σ , as given by Section 5.7.2. F_D and F_B have an initial value of 1.0 and are refined in the design process. The subscripts D and B indicate deck and bottom.

where, the maximum hull vertical bending stresses σ_B and σ_D are given in Section 5.7.1 as:

$$\sigma_B = \frac{|\overline{M}_s + M_w|}{Z_B} \times 10^{-3} \text{ MPa} \quad (2.4a)$$

$$\sigma_D = \frac{|\overline{M}_s + M_w|}{Z_D} \times 10^{-3} \text{ MPa} \quad (2.4b)$$

Z_D, Z_B = The initial value for Z_D and Z_B is set equal to the minimum hull midship section modulus $Z_{\min} = 14.131 \text{ m}^3$, as defined in Section 5.4 and expressed in Equation 2.5. These are subsequently redefined in the design process.

$$Z_{\min} = f_1 k_L C_1 L^2 B (C_b + 0.7) \times 10^{-6}, \text{ and has a value of } 14.131 \text{ m}^3 \quad (2.5)$$

Based on Equations 2.3 (a) and (b) the permissible still water sagging and hogging bending moment resistances were calculated to be 918,500 kN.m and 1,000,600 kN.m, respectively. The former value governs.

STEP 3 The design still water bending moment, M_s , is the maximum moment due to hogging and sagging calculated from the loading conditions given in Section 5.3.3. For oil tankers these include,

- homogeneous loading conditions (excluding dry and clean ballast tanks) and ballast or part loaded conditions;
- any specified non-uniform distribution of loading; and
- mid-voyage conditions relating to tank cleaning or other operations where these differ significantly from the ballast condition.

STEP 4 Scantlings are computed according to Lloyd's Rules and Regulations (1994) which are expressed as a function of F_B and F_D . Refer to Appendix A for details and example calculations. For the first iteration, the values of F_B and F_D are set equal to one.

STEP 5 The section moduli, Z_D and Z_B , and the corresponding permissible still water hogging and sagging bending moment resistances at the midship section are calculated.

STEP 6 The moment resistance, $\overline{M}_s$, must be greater than or equal to the loading conditions, M_s , according to Section 5.3.1. The most economical scantlings are achieved when these two values are equal. LR.PASS SID-4 contains an algorithm that reduces F_D

and F_B such that the ratio of $\overline{M}_s/M_s$ asymptotically approaches a value of one, but never less than one.

STEP 7 Repeat steps (4) to (6) until the ratio of $\overline{M}_s/M_s$ is close to one, but not less than one.

STEP 8 If one of the values F_D and F_B is less than 0.67 for the plating or 0.75 for longitudinal stiffeners then that value is set to these minimum values and the scantlings are recalculated.

The results of the final iteration are summarized in Table 1.3(a) and 1.3(b).

Table 1.3(a) Section Geometric Properties

Total cross sectional area	42,267.169 cm ²
Lever arm to deck	12.399 m
Lever arm to keel	7.601 m
Section modulus to deck, Z_D	19.237 m ³
Section modulus to keel, Z_B	31.382 m ³
Vertical Moment of Inertia	238.525 m ⁴
Horizontal Moment of Inertia	575.746 m ⁴
First Moment of Area above Neutral Axis	14.345 m ³
Horizontal Modulus	35.984 m ³
Minimum of Moment of Inertia	74.020 m ⁴
Rule Modulus, Z_{min}	14.131 m ³

Table 1.3(b) Permissible, Actual, and Design Stresses

	Deck	Keel
Permissible Stress	175 MPa	175 MPa
Actual Stress [†] (Hogging, Sagging)	128.554 MPa	78.803 MPa
Stress Ratio [‡] (Hogging, Sagging)	0.735	0.450
Scantling Reduction Factor, F_D, F_B [*]	0.738	0.670
Design Stress, σ_D, σ_B	129.063 MPa	117.250 MPa

Note: [†] The actual stress is based on the loads (Equations 2.4(a) and 2.4(b)).

[‡] The stress ratio is defined as the actual stress divided by the permissible stress.

^{*} The minimum scantling reduction factor was used.

1.2.3 Shear Load Calculations

The design hull wave shear force, Q_w , at any position along the ship is given by LR Part 3, Chapter 4, Section 6.3 and expressed by Equation 2.6 as,

$$Q_w = K_1 K_2 Q_{w0} \text{ {kN}} \quad (2.6)$$

$$\text{where, } Q_{w0} = 0.3C_1 LB(C_B + 0.7) \text{ {kN}} \quad (2.7)$$

$$\text{and, } C_1 = 10.75 - \left(\frac{300 - L}{100} \right)^{1.5} \text{ and has a value of 9.346} \quad (2.8)$$

and value K_1 is to be taken as follows:

(a) Positive shear force envelope coefficient

$$\begin{aligned} K_1 &= 0 \text{ at aft end of } L \\ &= \frac{1.589C_B}{(C_B + 0.7)} \text{ between } 0.2L \text{ and } 0.3L \text{ from aft} \\ &= 0.7 \text{ between } 0.4L \text{ and } 0.6L \text{ from aft} \\ &= 1.0 \text{ between } 0.7L \text{ and } 0.85L \text{ from aft} \\ &= 0 \text{ at forward end of } L \end{aligned}$$

(b) Negative shear force envelope coefficient

$$\begin{aligned} K_1 &= 0 \text{ at aft end of } L \\ &= -0.92 \text{ between } 0.2L \text{ and } 0.3L \text{ from aft} \\ &= -0.7 \text{ between } 0.4L \text{ and } 0.6L \text{ from aft} \\ &= \frac{-1.727C_B}{(C_B + 0.7)} \text{ between } 0.7L \text{ and } 0.85L \text{ from aft} \\ &= 0 \text{ at forward end of } L \end{aligned}$$

Intermediate values are to be determined by linear interpolation.

$$K_2 = 1.0 \text{ for unrestricted sea-going service conditions}$$

The results of these computations giving the design wave shear force along the length are shown graphically in Figure 1.3. The central portion of the tanker consisting of 3 cargo tanks measuring 53,000 mm long is indicated. The maximum shear force of 20,795 kN and 20,125 kN were used in conjunction with the hogging and sagging

moments for the hull module analysis in Section 4.3, and 4.4. Note, the design still water shear forces were small in comparison to the design wave shear force.

Figure 1.3 Design Wave Shear Force at any Position along the Oil Tanker Length

1.2.4 Baseline 40,000 DWT Oil Tanker Design Summary

This section summarizes the design computations of Appendix A and presents the strake and stiffener scantlings in Table 1.4 and Table 1.5, respectively. Also listed are the corresponding rule thicknesses, dimensions and governing sections from Lloyd's Rules and Regulations (1994). The locations and identification numbers for the strakes for the hull envelope and for stiffener groups are shown in Figures 1.4 and 1.5, respectively. For double hull oil tankers, the transverse bulkheads are either a stiffened plate or a corrugated plate supported by stools. In addition, a double plate bulkhead was designed for the CDH

oil product tanker to maintain a similar structural layout to the UDHSPS design presented in Section 1.4. This transverse bulkhead design alternative is illustrated in Figure 1.6, and the others are presented in Appendix A.

Table 1.4 Summary of Actual and Rule Strake Scantlings

Item No.	Identity	Strake No.	Actual Thickness (mm)	Rule Thickness (mm)	Governing Design Criteria
1	Strength Deck	1.1	12.00	12.00	P3 Ch.4 T4.7.2(a)
		1.2	12.00	11.80	P3 Ch.4 T4.7.2(a)
		1.3	11.50	11.60	P3 Ch.4 T4.7.2(a)
		1.4	11.50	11.40	P3 Ch.4 T4.7.2(a)
		1.5	11.50	11.30	P3 Ch.4 T4.7.2(a)
2	Sheerstrake	2.6	12.50	12.26	P3 Ch.3 T3.2.1 (End t.)
4	Side Shell	4.4	12.50	12.26	P3 Ch.3 T3.2.1 (End t.)
		4.5	12.50	12.26	P3 Ch.3 T3.2.1 (End t.)
5		5.2	12.50	12.26	P3 Ch.3 T3.2.1 (End t.)
		5.3	12.50	12.26	P3 Ch.3 T3.2.1 (End t.)
6		6.1	12.50	12.39	P4 Ch.9 T9.4.1(5/6) (Pressure)
7	Bilge	7.1	13.00	12.99	P4 Ch.9 T9.4.1(7/8) (Pressure)
		7.2	13.00	12.99	P4 Ch.9 T9.4.1(7/8) (Pressure)
8	Bottom Shell	8.2	13.00	12.99	P4 Ch.9 T9.4.1(7/8) (Pressure)
		8.3	13.00	12.99	P4 Ch.9 T9.4.1(7/8) (Pressure)
		8.4	13.00	12.99	P4 Ch.9 T9.4.1(7/8) (Pressure)
		8.5	13.00	12.99	P4 Ch.9 T9.4.1(7/8) (Pressure)
		8.6	13.00	12.99	P4 Ch.9 T9.4.1(7/8) (Pressure)
9	Keel	9.1	15.00	15.08	P4 Ch.9 T9.4.1(9) (Pressure)
10	Inner Bottom	10.1	14.00	13.91	P4 Ch.9 T9.6.1(1)(b)
		10.2	14.00	13.91	P4 Ch.9 T9.6.1(1)(b)
		10.3	14.00	13.91	P4 Ch.9 T9.6.1(1)(b)
		10.4	14.00	13.91	P4 Ch.9 T9.6.1(1)(b)
11	Centre Girder	11.1	14.00	13.85	P4 Ch.9 9.3.4(a)
12	Side Girders	12.1	12.00	12.00	P4 Ch.9 9.3.4(b)
		12.2	12.00	12.00	P4 Ch.9 9.3.4(b)
		12.3	12.00	12.00	P4 Ch.9 9.3.4(b)
		12.4	12.00	12.00	P4 Ch.9 9.3.4(b)

Table 1.4 Continued. Summary of Actual and Rule Strake Scantlings

Item No.	Identity	Strake No.	Actual Thickness (mm)	Rule Thickness (mm)	Governing Design Criteria
13	Longitudinal Bulkhead	13.1	12.50	12.63	P4 Ch.9 T9.6.1(b)
		13.2	12.50	12.27	P4 Ch.9 T9.6.1(c)
		13.3	11.50	11.71	P4 Ch.9 T9.6.1(c)
		13.4	11.00	11.12	P4 Ch.9 T9.6.1(c)
		13.5	10.50	10.49	P4 Ch.9 T9.6.1(c)
		13.6	10.00	10.12	P4 Ch.9 10.2.1 (Comp. Min.)
		13.7	10.00	10.12	P4 Ch.9 10.2.1 (Comp. Min.)
14	Floors	N/A	12.00	12.00	P4 Ch.9 9.3.4(b)
15	Deck Transverse	15.1	13.00	N/A	P4 Ch.10 2.8
		15.2	25.00	N/A	P4 Ch.10 2.8
16	Corrugated Transverse Bulkhead	16.1	15.00	14.83	P4 Ch.1 Table 1.9.1
		16.2	14.00	13.65	P4 Ch.1 Table 1.9.1
		16.3	12.50	12.34	P4 Ch.1 Table 1.9.1
		16.4	11.00	10.81	P4 Ch.1 Table 1.9.1
17	Plane Transverse Bulkhead	17.1	12.50	12.36	P4 Ch.9 T9.7.1 (a)
		17.2	12.00	11.81	P4 Ch.9 T9.7.1 (a)
		17.3	11.50	11.36	P4 Ch.9 T9.7.1 (a)
		17.4	10.50	10.39	P4 Ch.9 T9.7.1 (a)
		17.5	10.00	10.12	P4 Ch.9 10.2.1 (Comp. Min.)
		17.6	10.00	10.12	P4 Ch.9 10.2.1 (Comp. Min.)
		17.7	10.00	10.12	P4 Ch.9 10.2.1 (Comp. Min.)
		17.8	10.00	10.12	P4 Ch.9 10.2.1 (Comp. Min.)
18	Double Plate Bulkhead	18.1	10.00	10.12	P4 Ch.9 10.2.1 (Comp. Min.)
		18.2	10.00	10.12	P4 Ch.9 10.2.1 (Comp. Min.)
		18.3	10.00	10.12	P4 Ch.9 10.2.1 (Comp. Min.)
		18.4	10.50	10.28	P4 Ch.1 T1.9.1
		18.5	11.50	11.32	P4 Ch.1 T1.9.1
		18.6	12.50	12.25	P4 Ch.1 T1.9.1
		18.7	12.50	12.25	P4 Ch.1 T1.9.1
		18.8	12.50	12.25	P4 Ch.1 T1.9.1

Table 1.5 Summary of Stiffener Dimensions and Modulus Values

Location	Group No.	Sizes (mm)	Section modulus (cm ³)	Rule modulus (cm ³)	Governing Design Criteria
Strength Deck	SD1	250x19	361.4	345.6	P4 Ch.9 5.3.1(b) (90% Head)
	SD2	250x23	426.1	401.4	P4 Ch.9 5.3.1(b) (90% Head)
Sheerstrake	SS6	300x20	534.1	461.5	P4 Ch.9 5.3.1(b) (90% Head)
Side Shell	SS1	400x25	1094.8	987.0	P4 Ch.9 5.3.1(a) (Mean Head)
	SS2	400x25	1094.8	901.8	P4 Ch.9 5.3.1(a) (Mean Head)
	SS3	350x23	799.6	755.5	P4 Ch.9 5.3.1(a) (Mean Head)
	SS4	300x25	648.3	619.0	P4 Ch.9 5.3.1(b) (Mean Head)
	SS5	300x21	557.4	539.7	P4 Ch.9 5.3.1(b) (90% Head)
Bilge	B1	400x25	1102.6	1023.4	P4 Ch.9 5.4.1
Bottom Shell	BS1	400x25	1117.4	1033.9	P4 Ch.1 T1.6.1(3b) (Mean Head)
	BS2	400x25	1102.6	1033.9	P4 Ch.1 T1.6.1(3b) (Mean Head)
	BS3	400x25	1102.6	1033.9	P4 Ch.1 T1.6.1(3b) (Mean Head)
	BS4	400x25	1102.6	1033.9	P4 Ch.1 T1.6.1(3b) (Mean Head)
	BS5	400x25	1102.6	1033.9	P4 Ch.1 T1.6.1(3b) (Mean Head)
Inner Bottom	IB1	400x27	1192.9	1123.5	P4 Ch.9 T9.6.1(2) (Mean Head)
	IB2	400x27	1192.9	1123.5	P4 Ch.9 T9.6.1(2) (Mean Head)
	IB3	400x27	1192.9	1123.5	P4 Ch.9 T9.6.1(2) (Mean Head)
	IB4	400x27	1192.9	1123.5	P4 Ch.9 T9.6.1(2) (Mean Head)
Longitudinal Bulkhead	LB1	400x25	1094.8	1071.2	P4 Ch.9 T9.6.1(2) (Mean Head)
	LB2	400x25	1094.8	940.6	P4 Ch.9 T9.6.1(2) (Mean Head)
	LB3	350x23	788.0	783.0	P4 Ch.9 T9.6.1(2) (Mean Head)
	LB4	300x25	633.8	627.1	P4 Ch.9 T9.6.1(2) (Mean Head)
	LB5	300x20	519.0	513.1	P4 Ch.9 T9.6.1(2) (90% Head)
	LB6	300x20	514.8	436.9	P4 Ch.9 T9.6.1(2) (90% Head)
	LB7	250x21	384.4	359.6	P4 Ch.9 T9.6.1(2) (90% Head)
Plane Transverse Bulkhead	TB1	350x25	858.6	826.1	P4 Ch.9 T9.7.1(2)
	TB2	350x23	793.9	753.8	P4 Ch.9 T9.7.1(2)
	TB3	350x20	699.1	669.3	P4 Ch.9 T9.7.1(2)
	TB4	300x23	585.4	560.8	P4 Ch.9 T9.7.1(2)
	TB5	250x27	477.4	452.3	P4 Ch.9 T9.7.1(2)
	TB6	250x21	384.5	343.7	P4 Ch.9 T9.7.1(2)
	TB7	250x14	268.6	247.2	P4 Ch.9 T9.7.1(2)
	TB8	225x13	206.5	190.6	P4 Ch.9 T9.7.1(2)
Double Plate Bulkhead	DPB1	200x23	277.0	275.8	P4 Ch.1 T1.9.1(2)
	DPB2	200x23	277.0	275.8	P4 Ch.1 T1.9.1(2)
	DPB3	200x23	277.0	275.8	P4 Ch.1 T1.9.1(2)
	DPB4	250x22	404.0	393.1	P4 Ch.1 T1.9.1(2)
	DPB5	300x20	527.0	511.0	P4 Ch.1 T1.9.1(2)
	DPB6	300x25	648.0	629.0	P4 Ch.1 T1.9.1(2)
	DPB7	300x25	648.0	629.0	P4 Ch.1 T1.9.1(2)

Figure 1.6 Double Plate Bulkhead and Transverse Framing Scantlings

1.3 State-of-the-Art Review

An understanding of the energy absorption behaviour of current oil tanker structures subject to collisions or grounding is essential to develop an alternative design, which will maximize the energy absorption capacity while providing equivalent performance for normal operating loads. These hull structure designs may be characterized as either a conventional double hull (CDH) or as an advanced double hull (ADH), both of which are illustrated in Figure 1.7. Papers by Daidola (1995), Paik and Pedersen (1996), Rodd (1996), Kuroiwa (1996), Sano et al. (1996), and Mizukami et al. (1996) examine the dynamic behaviour and response of CDH and ADH tanker designs

subject to extreme load events. These papers were reviewed to establish the answers, if any, to the following questions:

- What is the extent of damage that can be tolerated by a ship structure while meeting the objective of zero outflow of oil?
- What are the sources of the problems related to extreme load events?
 - What are the mechanisms for energy absorption and which of these maximize the energy absorption capacity of the hull?
 - What are the failure modes that curtail the energy absorption capacity of the structure and can these failure modes be avoided?
- What is the state-of-the-art for predicting the extent of damage?
- Are there any technologies that satisfy the requirement of zero outflow of oil?
- What are the relevant design criteria for extreme load events for double hull oil tankers?
- What is the ranking for those criteria for classification societies, insurance companies, ship owners, and ship builders?
- Are there any improvements that can be made to reduce fabrication, maintenance and inspection costs, and to improve operational behaviour?

In summary, the CDH design is based on traditional naval architecture and consists of a complex series of longitudinally and transversely framed elements that separate the inner and outer hulls by the International Maritime Organization (IMO) MARPOL 73/78 requirement of 2 metres. With respect to meeting the objective of zero oil outflow, the structural form offers a collision gap in which the structure can collapse and objects can

penetrate without directly breaching the inner hull. There are very few developments in maximizing the energy absorption capacity of the structure or in improving the cost effectiveness of the design (as-built fabrication, maintenance, and operational costs).

The ADH structure offers a simplified structural layout that incorporates only longitudinal framing and eliminates all transverse frames except for those supporting the transverse bulkhead. The longitudinal girder spacing is reduced to maintain the buckling capacity of the hull plates. The transverse bulkheads and side shell stringers must be made stronger and stiffer than the CDH design to carry the hydrostatic forces transverse to the length of the ship that are normally carried by the floors. Refer to the load transfer mechanism illustrated in Figure 1.8, Daidola, (1995). Similarly, the transverse bulkhead must carry the vertical loads due to the deck, normally transmitted by the transverse webs. The energy absorption capacity has proven to be similar to that of the CDH. The removal of transverse floors and longitudinal stiffeners reduces the level of complexity and lessens the number of fatigue prone details and the number of elements in which cracks can propagate from a ruptured outer hull to the inner hull. The ADH design offers some cost advantages in modularization and simplicity in fabrication. The relatively unobstructed surfaces simplify and improve the coating process, thereby minimizing corrosion problems. On the other hand, the narrow longitudinal girder spacing impedes inspection and makes maintenance more difficult. Detailed reviews follow.

Figure 1.7 Conventional and Advanced Double Hull Designs (after Rodd, 1996)

Figure 1.8 Load Transmission Mechanism (after Daidola, 1995)

1.3.1 Tanker Structure Behaviour During Collision and Grounding (Daidola, 1995)

The primary objective in structural design for normal operating loads is to provide sufficient resistance to maintain structural integrity and to incorporate the appropriate fatigue details. Other criteria that affect the design (strake scantlings, detailing, and layout) are level of complexity (related coating application, welding, and fatigue strength) structural weight, fabrication, and maintenance. For accidental or extreme load events, the primary objectives are to maintain vessel integrity (resistance to fire, explosion, collision, and grounding damage), to minimize or prevent oil outflow, and to provide sufficient residual strength after damage to permit salvage and rescue operations.

Daidola classifies a collision event as a contact problem between the side shell structure or bow of one ship and a struck rigid object, or of the same for two ships. Collision events are characterized by the collision intensity (fn.(speed, displacement, bow shape, draught, relative position of the struck ship centreline to the vector of the velocity of the striking object/ship)), the condition of struck tanker (fn.(displacement, draught, speed, orientation)), the environmental conditions (fn.(wind, wave, current)), and the structural resistance to collision.

A grounding event is a contact problem between the bottom structure and the sea floor (obstruction) which deflects the bottom inward and may penetrate or rupture the bottom structure as the tanker moves forward. These are characterized by the obstruction (fn.(depth, shape, size, geological characteristics)) and the tanker (fn. (draught, displacement, speed, hydrodynamic stability, heave and pitch action, structural resistance

to bottom deflection, rupture, raking damage)). It may be noted that grounding events include stranding of the tanker once it comes to rest on the obstruction.

A summary of the structural responses for the different elements of CDH and ADH tankers when subjected to collision and grounding events, according to Daidola, are given in Table 1.6. Strakes, supporting system, and details of a double hull tanker must be designed to be flexible, to prevent crack propagation and to allow plastic deformation, which maximizes the energy absorption capacity of the system for accidental or extreme load events. It is understood that the steel must have sufficient ductility (greater than 10%) and tearing resistance (greater than 2.1 J/mm^2).

1.3.2 Modeling of the Internal Mechanics in Ship Collisions (Paik and Pedersen, 1996)

Paik and Pedersen (1996) developed a design-oriented method for analysis of structural damage due to right angle ship collisions with a rigid bow. The energy absorption capacity was evaluated for a variety of collision speeds, scantlings, and arrangements. It was found that an increase in strength of the side stringers and transverse webs would cause an increase in the out-of-plane deformation of the inner hull plate and premature rupture.

Table 1.6 Responses of CDH and ADH Structural Elements to Accidental or Extreme Load Events

Structural Elements	Structural Response
Brackets	<ul style="list-style-type: none"> • These are rigid supports (hard spots) in which fractures are initiated.
Decks	<ul style="list-style-type: none"> • Deck failure generally occurs by transverse rupturing resulting from longitudinal membrane tension. • To maximize the energy absorption capacity, the deck must be designed to rip free of the webs and bulkhead during accidental or extreme load events.
Side and bottom stiffened [†] strakes	<ul style="list-style-type: none"> • Membrane tension of the strake is the most significant source of energy absorption capacity. • Membrane tension of the strakes between web frames and longitudinal girders can only fully develop when the stiffeners buckle, trip or rupture. • Depending on stiffener type and detailing the strake may rupture locally and limit the energy absorption capacity. • The propagation of cracks from a ruptured outer hull to the inner hull must be prevented to obtain membrane tension of the inner hull, which is in contact with the penetrating object, without it being breached.
Web frames, longitudinal girders, and transverse floors [†]	<ul style="list-style-type: none"> • Widely spaced, flexible web frames that distort plastically in bending and shear will allow the strakes to develop membrane action prior to rupture, which in turn will maximize the energy absorption capacity of the system. • The lack of transverse floors in the ADH design may increase the length of the raking damage for grounding events. • Daidola hypothesized that for ADH to be effective in absorbing energy due to collision or grounding, the inner and outer shells must distort in unison under membrane tension.
Transverse bulkheads	<ul style="list-style-type: none"> • Transverse bulkheads are generally very stiff and do not buckle yield or rupture. • For collisions, these elements limit the extent of longitudinal side shell distortion and the corresponding energy absorption (which has a minimum value when the collisions occur at or adjacent to it). • A flexible ductile bulkhead with proper detailing will maximize the energy absorption capacity.
Outboard bilge	<ul style="list-style-type: none"> • Bilges tend to be rigid and during collision events tend to rupture, exposing sharp elements which cut into the bow of the striking vessel, exposing more sharp elements which in turn will puncture the inner hull of the struck tanker. • Again, a properly designed bilge that allows for significant plastic distortion prior to rupture will maximize the energy absorption capacity of the system.

Note: [†] For CDH oil tankers only.

1.3.3 Observations on Conventional and Advanced Double Hull Grounding Experiments (Rodd, 1996)

Rodd (1996) conducted a series of one-fifth-scale oil tanker grounding experiments for both the CDH and ADH product oil tanker designs in the range of 30,000 to 40,000 DWT. The model, illustrated in Figure 1.9, represents a typical two cargo tank length bottom hull section, which was mounted to a railway car that traveled down inclined tracks to impact a pinnacle shaped mechanical rock. The vertical rock penetration above the keel and the energy dissipation characteristics were evaluated for each design. The tanker design designation, a general description of the framing, and a summary of the grounding experiment results are given for each tanker design in Table 1.7. The dissipated energy estimates were calculated as the area underneath the fore/aft rock force versus displacement curves over the shortest stopping distance of 5.4 m.

Figure 1.9 Typical Model Installation

Table 1.7 Grounding Experiment Results

Tanker designation and framing description	Summary
ADH/PB – relatively wide spacing of longitudinal girders	<ul style="list-style-type: none"> • Initial impact velocity was 12 knots. • The rock passed completely through the model and 94% of the kinetic energy was expended. • 2.66×10^6 J of energy was dissipated.
ADH/PD328 – tighter longitudinal girders spacing by 0.74 [†] times and slightly thicker inner shell plating by 1.13 [†] times.	<ul style="list-style-type: none"> • Initial impact velocity was 14 knots and the model came to rest on the rock near the aft bulkhead. • This design dissipated 5.34×10^6 J of energy and was the most resistant to rupture damage of the inner hull as the rock penetration engaged a large array of densely packed longitudinal components, which were plastically deformed.
ADH/PD328V – same as ADH/PD328 except the longitudinal girders are stiffened vertically and the web plating thickness was increased by 1.13 [‡] times.	<ul style="list-style-type: none"> • 6.03×10^6 J of kinetic energy was dissipated as the model came to rest on the rock. • Vertical stiffeners were added and the web plating thickness increased to increase the buckling resistance of the longitudinal girders in an attempt to promote inner shell lift that postpones its rupture. • The damage was more severe as the web geometry was maintained and the rock simply caused those webs in contact to be rotated outwardly as if they were hinged at the inner shell weld. The vertical stiffeners pushed through the inner shell plating and resulted in numerous punctures and tearing of the inner shell. • Although the energy dissipation was greater than the previous model, stiffening on the longitudinal webs was not recommended since the rupture initiation was more severe.
CONV/PD328 – CDH	<ul style="list-style-type: none"> • Initial velocity was 14 knots. • 76% of the total kinetic energy was dissipated as the rock passed through the structure. Typically transverse frames would trigger an inner shell rupture, but in this design, these frames were torn away without causing rupture of the first cargo tank. • 3.25×10^6 J of energy was dissipated.

Note: [†] fraction increase or decrease relative to ADH/PB

[‡] fraction increase or decrease relative to ADH/PD328

Rupture initiation and breach of the inner hull was similar for each tanker design and occurred when the penetrating rock, which was in contact with the inner shell causing it to be in inelastic membrane action, passed through the next full depth continuous transverse member beneath the inner hull. Rodd (1996) suggests that the most crashworthy design is the ADH/PD328 with tightly spaced longitudinal girders. It is anticipated that grounding of a full scale tanker of this type at maximum speed with the worst case rock pinnacle shape would damage the bow and rupture one cargo tank.

1.3.4 Numerical Simulation of Actual Collision and Grounding Accidents (Kuroiwa, 1996)

According to Paik and Perdersen (1996), the analysis of a ship-ship collision is usually separated into the external mechanics (or the outer collision dynamics), and the internal mechanics (or the inner collision dynamics). The external collision dynamics involve the simulation of the rigid body motion of the colliding ships including surrounding water effects. The inner collision dynamics involve the evaluation of the force-indentation responses of the striking ship (deformation of the bow structure) and the struck ship (deformation of the side structure) during the collision.

A numerical scheme was developed by Kuroiwa to simulate the structural damage, hydrodynamic force, and ship motion during a collision. The structural damage of ships was simulated using LS-DYNA 3D, a finite element method with explicit integration. The contact force causing structural failure was calculated and passed to a ship motion program (MCOL) to obtain the position and velocity of the colliding objects that would produce that structural damage. To study the accuracy and applicability of this

numerical method, Kuroiwa (1996) presented the numerical simulation of a known collision (Malacca Strait, September 20, 1992) and a grounding accident (Buffalo Reef off of Singapore, January 6, 1975) of oil tankers, as described in Table 1.8(a) and 1.8(b), respectively.

Table 1.8(a) Collision Simulation Details, Kuroiwa, (1996)

Ship Types
<ul style="list-style-type: none"> • Struck oil tanker was a 100,000 DWT oil tanker traveling at 12 knots under slack loading conditions. • The colliding ship was a fully loaded 22,600 DWT container ship traveling at 22 knots.
Element Type
<ul style="list-style-type: none"> • The hull structure of the damaged part of the struck oil tanker and the bow structure of the colliding ship were modeled. • Inelastic material behaviour was included and the strain rate effect on the yield point was taken into account. • Plate rupture and weld failure was simulated. • Finite element sizes range from 400 mm to 800 mm with thickness values ranging from 8.5 mm and 23.5 mm. • The struck ship was modeled with 70,000 elements and the colliding ship was modeled with 13,000 elements. • The remaining parts of the colliding vessels were modeled as rigid bodies, which will move according to the ship motion program.
Failure Behaviour
<ul style="list-style-type: none"> • Time zero occurs at the initial point of contact. The maximum contact force 125 MN occurred at time 0.8 sec when the side shell of the struck ship was ruptured. After this, the force decreases to a value of 60 MN and then increases to a value of 120 MN at 1.9 sec when the longitudinal bulkhead was ruptured. • The entire collision was run for 4.0 sec.
Miscellaneous Information
<ul style="list-style-type: none"> • Collision angle was approximately at right angles. • Side structure and longitudinal bulkhead were ruptured. • 10,000 tons of cargo oil was spilled. • CPU Time for an EWS Dec Alpha 8200 was 160 hours.

Table 1.8(b) Grounding Simulation Details, Kuroiwa, (1996)

Ship Type
<ul style="list-style-type: none"> The grounded ship was a 240,000 DWT single hull structure fully loaded oil tanker traveling at 12 knots.
Element Type
<ul style="list-style-type: none"> The lower half of the hull structure was modeled similar to the contact portion of the struck ship in the collision analysis. The other portion of the vessel was modeled as a rigid body, which can depict the ship's rigid body movement. The rock was modeled as a rigid body and was located such that its height was 3 m above the baseline of the ship. A total of 58,000 elements were used.
Failure Behaviour
<ul style="list-style-type: none"> Contact forces in the longitudinal direction have two peak values of 60 MN before time 4 sec (of the 15 sec grounding analysis). These correspond to the rupture of the bulbous bow and subsequent rupture of collision bulkhead located 21 m aft of the bow. After time 4 sec, the rock penetrated into the cargo tanks with peak contact forces of 35 MN as the bottom transverses moved over the rock. When the tanker hull, between the transverses, moved over the rock, the contact force decreased to a value of 16 MN. The contact force in the vertical direction reached its maximum at a value of 80 MN, when the shell plate of the bulbous bow was ruptured. It then gradually decreased to about 35 MN. The vertical contact force moves the ship upward. Penetration depth, which is the relative height from the baseline of the ship to the top of the rock, changed from the initial 3 m to 1.5 m in the first 5 sec. This difference is attributed to 0.4 m of heave motion and 1.1 m of pitch motion. The penetration depth subsequently increased to approximately 2.25 m after 8 sec.
Miscellaneous Information
<ul style="list-style-type: none"> The analysis neglected ingress of sea water and oil outflow during the course of the numerical simulation. The effect of shallow water was taken into account with a constant draft. The grounding simulation required 450 hours of CPU time since grounding events take longer. Contact forces were repetitive for rupture and damage of similar structural sections. The damage extended from the bow to the middle of the third cargo tank, a distance of approximately 180 m.

Kuroiwa concluded that numerical simulations provide a practical method of analyzing full-scale collision and grounding accidents since the extent of damage of the

ship structures obtained by the simulations corresponded with the structural damage surveyed after the actual accident or extreme load events.

For the grounding load event, the deformation and rupture of the transverse members of the bottom structure contributed 35% of the total energy absorbed for single bottom Very Large Crude Carrier (VLCC). This may be as high as 60% for double bottom VLCC, since the bottom shell tends to be thinner and the floors thicker.

For the collision load event, the progressive speed (forward motion) of the struck ship caused the bulbous bow of the colliding ship to bend laterally, increasing the energy absorbed by the ship structure by approximately 7% over that of a collision with a stationary vessel. Seventy-six percent of the initial kinetic energy of the colliding ship (1200 MJ) was absorbed as strain energy. Seventy two percent of that was attributed to the damaged struck ship and the remainder was due to the damage of the colliding ship. The remaining 24% of the initial kinetic energy was dissipated to kinetic energy related to the lateral motion of the struck ships and wave damping. Since the deck structure of the colliding ship was weak compared to that of the struck ship, and the bulbous bow was rigid compared to the bottom structure, the most effective way to increase the energy absorption capacity of the struck ship would be to increase the lateral stiffness of the deck.

1.3.5 Strength Analysis of a New Double Hull Structure for VLCC in Collision (Sano, Muragashi, Yoshikawa, Motoi, Murakami, 1996)

Sano et al. (1996) presented a comparison of the collision strength, based on numerical simulations, of a conventional double hull VLCC oil tanker structure with a variant that arranges the stiffeners on the transverse web parallel to the side shell plate as

illustrated in Figure 1.10. The proposed modification to these details eliminates the welds between web stiffeners and longitudinal stiffener faceplates, reduces the number of structural members, decrease the possibility of fatigue cracks in regions of high tensile stress, and simplifies construction. These fatigue cracks may initiate early rupture of the side shell structure in the case of collision events. The collision analysis details are given in Table 1.9. The collision was simulated using appropriate finite element techniques (few details were given) with the mesh for the side structure and bulk carrier illustrated in Figure 1.11.

**Figure 1.10 Conventional and Proposed Web Stiffener Detail
(after Sano et al., 1996)**

Sano et al. (1996) presented the results of these simulations in the form of deformed shapes for given levels of penetration, reaction force versus penetration and strain energy versus penetration. The characteristic reaction force and strain energy absorption versus penetration curves were similar for both designs. The peak reaction forces and strain energy values are summarized in Table 1.10 for rupture of the side shell plate and inner hull plate for both structures.

Table 1.9 Collision Simulation Details, Sano et al. (1996)

Ship Types
<ul style="list-style-type: none"> • Struck oil tanker was a 280,000 DWT double hull oil tanker. • Colliding ship was a 190,000 DWT bulk carrier.
Collision Details
<ul style="list-style-type: none"> • Bulk carrier (striking ship) had a speed of 10 knots (5180 mm/sec). • A collision perpendicular to the centre of the side structure between transverse bulkhead of the oil tanker (worst case) was considered.
Element Type
<ul style="list-style-type: none"> • The side shell plate was modeled as shell elements, which fail upon reaching a critical strain, predefined as 18% nominal strain. When this strain value is reached, the element is considered to be ruptured and can no longer carry load.
Boundary Conditions
<ul style="list-style-type: none"> • The upper end of the model was considered to be a plane of symmetry and is constrained accordingly. • All degrees of freedom were restrained at the bulkheads. • The plane of nodes connected to the bilge was fixed in the direction of collision and set to be free vertically.
Miscellaneous Information
<ul style="list-style-type: none"> • The analysis neglected ship motion as well as ingress of sea water and oil outflow during the course of the numerical simulation. • The horizontal girders located at the centre of the tank were omitted. • Only the lower half of the striking bow was modeled and it was assumed to be rigid. • The bow part was given a mass equal to half the mass of a whole bulk carrier, a value of 95,000 tons.

Based on the simulations, Sano et al., (1996) demonstrated that the incorporation of flexible transverse webs that buckle prior to side shell plate rupture (as a result of providing appropriate detailing) increases both the penetration and strain energy

absorption of the system. Buckling of transverse webs allows a larger portion of the hull plate to become engaged and to absorb energy in membrane action in a collision event. Consistent with this design, the inner hull contact area and its energy absorption capacity were increased prior to rupture. The side shell plate absorbed approximately 70% of the total strain energy through membrane action. The second largest contributor to strain energy absorption was the transverse web frame at the striking position.

Sano et al. (1996) also presented a simplified method for evaluating the collision strength that gave reasonable correlation for side shell plate and inner hull plate rupture.

Figure 1.11 Side Structure and Bow Model (after Sano et al., 1996)

Table 1.10 Penetration, Reaction Force, and Absorbed Strain Energy

Structure	Ruptured Plate	Penetration, m	Reaction Force, MN	Strain Energy, MJ
VLCC with new structural details	Side Shell	3.9	40.6	92
	Inner Hull	6.2	46.1	171
Conventional VLCC	Side Shell	3.1	42.3	72
	Inner Hull	5.4	45.1	148

1.3.6 Comparative Study on Collision Resistance of Side Structure (Kitamura, 1996)

Kitamura (1996) conducted a series of finite element transient analyses, using LS-DYNA3D, to determine the “crashworthiness” of a variety of different side shell structures for VLCC that were subjected to collision. Crashworthiness was expressed by the structure’s capacity to absorb energy. Side shell structure types include:

- Standard VLCC (Standard VLCC - Type A) was used as the base ship structure and was assigned a crashworthiness index value of 100.
- VLCC constructed entirely of 350 Class HT steel (YP350 Class HT – Type B) instead of the mild steel and YP315 HT steel used in the standard VLCC.
- VLCC with additional side stringers (Add. S. Stringers – Type C), where the number was increased from three to six.
- VLCC with top side tank (With Add. S.Stringers – Type D) and an additional stringer at the bottom of the tank.
- VLCC with a strut in the cargo oil side tank (Strut in C.O.S.Tk – Type E), arranged to laterally support the side structure at mid-depth by the longitudinal bulkhead.
- VLCC with unidirectional girders (Unidirectional Gird. – Type F). All transverse webs were deleted with the exception of those used in the upper and lower part of the double side, which were required for structural continuity. Side shells (outer hull) and the longitudinal bulkhead were stiffened with side stringers instead of longitudinal stiffeners, which normally would span between transverse webs. A transverse oiltight bulkhead and a swash bulkhead supported each side stringer.
- VLCC with a corrugated side shell double side design (New Double Side – Type G). A double skinned frame panel, as illustrated in Figure 1.12, replaced the conventional longitudinally stiffened inner and outer shells. The frame panels consisted of outer and inner plates connected by longitudinal web plates, which were supported by conventional transverse main webs spaced at 4,000 mm to 5,000 mm. Typically, the width of a frame panel was usually 300 mm and the plate thicknesses were in the range of 10 to 14 mm. A top side tank was included with an additional stringer at the bottom of the tank.

**Figure 1.12 Section of VLCC with a New Double Side Design
(after Ohtsubo et al., 1997)**

The characteristics of the collision and details relating to the simulation are given in Table 1.11. The corresponding finite element mesh for the colliding ship and the struck ship are illustrated in Figure 1.13. The results of the simulation were presented in the form of illustrations that show the deformed shape at different load levels, contact force versus penetration graphs, and absorbed energy versus penetration graphs. A summary of the peak energy absorption values and penetration associated with rupture of the side shell and longitudinal bulkhead are given in Table 1.12 and illustrated in Figure 1.14. The crashworthy index for the different side shell designs are given in Table 1.13 and are expressed as a function of the maximum absorbed energy associated with rupture of the longitudinal bulkhead of a given design to the standard VLCC.

Table 1.11 Collision Simulation Details, Kitamura (1996)

Ship Type
<ul style="list-style-type: none"> • Struck oil tanker was a fully loaded 300,000 DWT double hull oil tanker class VLCC. • Colliding ship was a 150,000 DWT Suez Max class, in normal ballast condition, having raked stem of blunt form with a regular bulbous bow.
Finite Element Analysis
<ul style="list-style-type: none"> • The behaviour and failure of fillet welds, the effects of strain rate on material properties and ductile rupture of steel plates was modeled. • The total number of finite elements plus bar elements of struck ship models ranged from 32,000 to 61,000. • The total number of finite elements plus bar elements of the colliding ship model was approximately 15,000. • Midship tank structure of $\frac{1}{2}$ tank length of struck vessel was modeled. • A swash bulkhead at the mid length of the cargo oil side tank was also included.
Miscellaneous Information
<ul style="list-style-type: none"> • The bulbous bow of the colliding ship was positioned in line with the transverse web at the mid-point between a transverse oiltight bulkhead and a swash bulkhead. • Collision speed was 10 m/s (19 knots). • Friction coefficient for steel-steel contact was assumed to be a constant value of 0.3. • The analysis neglected the rigid body motion of the ships.

Figure 1.13 Simulation Model for Side Collision (after Kitamura, 1996)

The standard VLCC side structure tended to be stiff and contained the damage to a smaller contact area. Rupture of the outer shell was initiated at an earlier load level with less total energy absorption as a result of a large number of stiff supporting elements, which caused high localized strain concentrations.

Reduction in the global rigidity of the side structure delayed the time of maximum peak contact force prior to rupture, as illustrated by the simulation of the unidirectional double hull (Type F).

**Table 1.12 Rupture Opening Initiation and Absorbed Energy
(after Kitamura, 1996)**

Type	Side Shell		Longitudinal Bulkhead	
	Rupture Initiation	Absorbed Energy	Rupture Initiation	Absorbed Energy
	(m)	(MJ)	(m)	(MJ)
A. Standard VLCC	1.750	73.6	5.000	230.4
B. YP350 Class HT	2.125	106.0	5.250	286.8
C. Add. S. Stringers	1.750	76.7	4.875	248.6
D. Top Side Tank	1.875	84.0	5.625	283.1
E. Strut in C.O.S. Tk	1.750	76.7	5.000	228.8
F. Unidirectional Gird.	2.000	83.5	5.000	287.3
G. New Double Side	2.000	131.5	5.375	404.5

Table 1.13 Crashworthiness Index (after Kitamura, 1996)

Type	Crashworthiness Index
A. Standard VLCC	100
B. YP350 Class HT	124
C. Add. S. Stringers	108
D. Top Side Tank	123
E. Strut in C.O.S. Tk	99
F. Unidirectional Gird.	125
G. New Double Side	176

The proposed corrugated frame panel design given by curve G in Figure 1.14 gave a crashworthiness index of 176. However, as the author recognized, the design may or may not be practical, as issues related to production engineering, fabrication, maintenance, inspection, and cost were not addressed. Although not specifically stated by the author, the best design concept is that which allows membrane action to develop over a large contact area and will include collapse of side stringers prior to rupture. This maximizes the energy absorption capacity.

Figure 1.14 Comparison Energy Absorption Capacity (after Kitamura, 1996)

1.3.7 Collision Simulation of a Double-Hulled Structure with Uni-Directional Girder System (Mizukami, Tanigawa, Nagahama, Kamei, 1996)

Mizukami et al. (1996) conducted numerical collision simulations on a CDH tanker and a series of ADH tankers (unidirectional girder system illustrated in Figure 1.15) to

determine the collision resistance for different bow configurations and contact points, including the hydrodynamic response. A summary of the collision details and the parameters investigated for each case are given in Tables 1.14 and 1.15.

**Figure 1.15 Double Hull Structure with Unidirectional Girder System
(after Mizukami et al., 1996)**

The numerical simulations, using the MSC/DYTRAN (1993) finite element program, were able to model the internal structural dynamic behaviour including girder buckling, outer and inner hull plate rupture, bulkhead rupture, membrane action, and the external hydrodynamics of the ship reacting with the surrounding water. Mizukami et al. (1996) presented deflected shapes, contact force versus time, contact force versus penetration, and strain energy versus penetration curves for a representative sample of the cases investigated. Tables 1.14 and 1.15 also give a description of the general behaviour and the collision resistance of each variant case with the corresponding reference case.

Table 1.14 Collision Simulation Details – Bow Configuration Investigation
(Mizukami et al., 1996)

Ship Size				
<ul style="list-style-type: none"> • Double hull 40,000 DWT product oil tanker. • The bows of the colliding vessel were modeled as rigid cylinders that strike the ship side structure at the centre of a cargo hold at right angles. 				
Framing System of Struck Ship and Collision Data				
Case	Frame System	Bow Radius	Height	Failure Strain
G0 - 1	Unidirectional	1 m	Low	10 %
G0 - 2	Unidirectional	1 m	Low	5 %
G0 - 3	Unidirectional	1 m	Low	15 %
G1	Unidirectional	1 m	High	10 %
G2	Unidirectional	3 m	Low	10 %
T	Conventional	1 m	Low	10 %
Material Behaviour and Scantlings				
<ul style="list-style-type: none"> • Elastic-plastic steel with a yield stress of 300 MPa and a modulus of elasticity of 200,000 MPa. • The side structure of the reference case G0 was given different failure strain values ranging from 5% to 10% to reflect the reduced ductility associated with strain concentrations of structural discontinuities. • The effects of weld joints were neglected. • The outer hull strake thickness was 13 mm. The inner hull strake thickness and the longitudinal girder thicknesses were 14 mm and 9 mm, respectively. 				
Boundary Conditions				
<ul style="list-style-type: none"> • A quarter part of the ship side and bow was modeled assuming similar construction for the deck and bottom structure and longitudinal symmetry. • Fixed along the transverse bulkhead and the centreline. • Probable contact surfaces were defined on the outer and inner hull, and the longitudinal girders. • Bow penetration was at a constant velocity of 10 m/s (19 knots). 				
Behaviour of Reference Case, G0 – 1				
<ul style="list-style-type: none"> • The first peak contact force occurred at 0.02 seconds, when the longitudinal stringers buckled. • The contact force increased due to the development of elastic membrane action in the outer hull with bow penetration. • Rupture of the outer hull was initiated at 0.065 seconds around the upper and lower corners of the striking cylindrical bow shape. These cracks continued to propagate and another rupture developed in the outer hull plate along the bulkhead at 0.15 seconds. • Contact and rupture of the inner hull followed the complete rupture of the outer hull along the contact point, between the bow and the plane of transverse symmetry, at 0.2 seconds. 				

Table 1.14 Continued. Collision Simulation Details – Bow Configuration Investigation (Mizukami et al., 1996)

Behaviour Comparisons with Case G0 – 1
<ul style="list-style-type: none"> • Case G0 – 2, G0 – 3: Increasing the failure strain delayed the initiation of rupture in both the outer and inner plates and increased the amount of penetration and energy absorption capacity of the structure prior to rupture. The converse was true when decreasing the failure strain values. • Case G1: Since the deck and bottom structures were engaged in high bow height collision, an increase in energy absorption resulted. Rupture initiation occurred along the transverse bulkhead. • Case G2: Due to the large bow radius, rupture of the outer hull at the collision point was preceded by rupture along the transverse bulkhead. • Case T: Rupture of the outer hull was localized and plastic deformation was restricted to the contact area between transverse frames. The corresponding energy absorption was small relative to the other cases considered.

(a) Unidirectional framing, ADH

(b) Conventional double hull

Figure 1.16 Collision Conditions (after Mizukami et al., 1996)

Table 1.15 Collision Simulation Details – Hydrodynamic Response
(Mizukami et al., 1996)

Ship Type
<ul style="list-style-type: none"> • The struck ship was a 60,000 DWT double hull tanker, 180 m long, 32 m wide and 20 m deep, with a draught height of 10 m. • The striking ship, a 20,000 DWT ship 140 m long, 20 m wide, and 13 m deep, was modeled with a bow radius of 1 m.
Load Cases
<ul style="list-style-type: none"> • Case 1: To obtain the basic contact force-penetration curve and energy absorption-penetration curve, the struck ship was fixed and the analysis was conducted neglecting the influence of fluid. • Case 2: To obtain the relationship between the ships' motion and both the contact force-penetration curves and energy absorption-penetration curves, the resistance to the collision was based on the struck ships inertia alone (no fluid). • Case 3: Fluid was included to determine its effect on the ships' motion and collision resistance.
Finite Element Modeling
<ul style="list-style-type: none"> • The struck oil tanker was modeled as a rigid body outside the collision zone. • The striking ship was modeled as a rigid body. • Fluid Eulerian elements were used to model the water around the struck tanker. Void Eulerian elements were used above the water level to allow for vertical movement of the water (waves). • The struck ship was modeled with approximately 12,000 elements with a refined mesh for the collision zone, refer to Figure 1.17. The failure strain for these elements was 10%.
Boundary Conditions
<ul style="list-style-type: none"> • An arbitrary Lagrange-Euler (ALE) boundary was defined between the struck ship and surrounding fluid to account for the fluid-structure interaction. • The striking ship was supported by springs corresponding to its buoyancy, so that the hydraulic force of the fluid would not reduce the striking speed.
Failure Behaviour
<ul style="list-style-type: none"> • Case 1: as per Case G0 – 1. • Case 2: The load-penetration curve was similar to Case 1 until penetration of 1.2 m where the velocity of the struck ship became equal to the striking ship (bounced off each other). The inner hull was not ruptured. • Case 3: Based on the dynamic analysis, it was determined that increasing the mass of the struck ship by approximately 20% would produce the same effect as including the surrounding fluid.
Miscellaneous Information
<ul style="list-style-type: none"> • The initial velocity of the striking ship was 2.5 m/s (5 knots). • The gravity field was defined for the struck ship and fluid, to equilibrate the weight of the struck ship and the buoyancy force.

**Figure 1.17 Finite Element Mesh of the Collision Zone of the Struck Ship
(after Mizukami et al., 1996)**

For the bow configuration investigation, the maximum energy absorption was for case G1 with decreasing capacity for cases G2, G0-1, and T. The energy absorption capacity of the system was reduced if the deformation was localized by the supporting structure. Note, the severe change in geometry at the edges of the cylindrical shape bow, used in this study, may initiate hull plate rupture sooner than it would for a bulbous bow, limiting its collision resistance.

1.4 The Unidirectional Double Hull Sandwich Plate System (UDHSPS)

Based on an understanding of current design philosophies as expressed by classification societies, an understanding of limitations of existing technology, and an understanding of failure mechanisms related to accidental or extreme load events, a new

structural system was developed entitled the Unidirectional Double Hull Sandwich Plate System (UDHSPS), illustrated in Figure 1.18. This design takes advantage of the structural system offered by the advanced (unidirectional) double hull technology and enhances the system performance by incorporating a structural composite laminate for inner and outer hulls, transverse bulkheads, and deck girders, with the appropriate detailing. The objectives of the UDHSPS design are,

- to provide equivalent structure for normal operating loads;
- to provide superior energy absorption capacity during accidental or extreme load events by maximizing plastic distortion and deformation without rupture of the inner hull;
- to eliminate oil outflow due to crack propagation;
- to provide the residual strength required to maintain hull integrity during stranding, rescue, and repair operations; and
- to be a constructible and viable design alternative.

A summary of structural performance and problems for normal operating loads and extreme load events for existing double hull technology is given in Table 1.16 as related to the objectives of the UDHSPS design. Summaries of the UDHSPS characteristics that achieve the design objectives are listed in Table 1.17. This table also has additional comments regarding cost effectiveness, recycling, fire resistance and repair.

(a) Isometric view of UDHSPS oil tanker structure

(b) Side structure

Figure 1.18 UDHSPS Design Concept

Table 1.16 State-of-the-art Summary

Structural Equivalency for Normal Operating Loads
<ul style="list-style-type: none"> • Narrow girder spacing of ADH is dictated by the buckling of hull plates due to in-plane loads from hogging and sagging moments. This restricts ship sizes from 40,000 to 50,000 DWT, Roseman et al. (1995).
Maximizing energy absorption (Pollution control)
<ul style="list-style-type: none"> • Side shell structure absorbs the most energy in collisions through membrane action, Sano et al. (1996) and Kitamura (1996). • Must engage (deform) larger surface of shell plating to allow for greater energy absorption, Mizukami et al. (1996) and Sano et al. (1996).
Eliminate Oil Outflow due to Crack Propagation (Pollution Control)
<ul style="list-style-type: none"> • Connections between transverse members and stiffeners are the potential location for stress concentration and is a source of fatigue crack initiation, Daidola (1995). • With CDH designs, the presence of longitudinal stiffeners and transverse floors limit the development of membrane action to a single plate, as rupture will occur at or near the supporting elements, Dailoda (1995). • An increase in strength of the side stringers and transverse webs causes an increase in out-of-plane deformation of the inner shell plates and premature rupture, Paik and Pedersen (1996). • For grounding events, cracks propagate from the ruptured outer hull through the transverse floor and longitudinals breaching the cargo tanks of both the CDH and ADH all-steel designs, Rodd (1996).
Structural Complexity
<ul style="list-style-type: none"> • CDH is an extremely complex structure, which does not lend itself to mass production, Daidola (1995).
Other Considerations
<ul style="list-style-type: none"> • Kitamura (1996) examined the use of nonferrous metals, CFRP, and honeycomb sandwich structures for use in alternate double hull structures. It was determined that these were not suitable due to limited elongation (energy absorption), limited constructibility (difficult to form, cut, and join), poor resistance to the corrosive environment (salty or oily water), poor fire resistance and excessive cost.

Table 1.17 UDHSPS Design Alternative

Structural Equivalency for Normal Operating Loads
<ul style="list-style-type: none"> • The increased buckling capacity of the sandwich plate system allows for wider girder spaces. • The increased buckling capacity of the sandwich plate system also provides the transverse in-plane capacity to transmit the hydrostatic forces, which normally would be transmitted through the floors of the CDH.
Maximizing Energy Absorption (Pollution Control)
<ul style="list-style-type: none"> • This design concept offers two structural laminate layers of protection for the cargo tank with enhanced energy absorption capability. • Maximizes energy absorption by promoting plastic membrane action prior to rupture and by providing collapsible longitudinal girders and side shell stringers to engage a larger area of vessel to resist load events. • The steel is reallocated to the side shell entirely (no stiffeners) so as to maximize the amount of steel acting in membrane action. • The elastomer core absorbs a portion of the energy due to impact as it transmits the force to the inner face of the outer hull. It has tensile strengths and elongation greater than 20 MPa and 20%, respectively, and contributes to carrying membrane forces.
Eliminate Oil Outflow due to Crack Propagation (Pollution Control)
<ul style="list-style-type: none"> • Numerous fatigue crack prone connections that lead to premature rupture of the shell are eliminated since the stiffeners and transverse floors are removed. • A design philosophy has been developed to size the stringers to allow collapse prior to the inner or outer hull rupture. • The elastomer of the composite structural laminate for the inner hull in combination with fatigue crack arrest details forms an effective crack arrest layer for the cargo tanks.
Structural Complexity
<ul style="list-style-type: none"> • Stiffeners and transverse frames are eliminated thereby reducing complexity. • Reduced complexity (removal of stiffeners) provides flat, unobstructed surfaces which results in: a better application of protective coatings (less corrosion), ease of access, improves the inspection process, and removal of a large fatigue and corrosion prone details that require maintenance. • Substantial cost savings in welding, detailing, labour, and materials related to the elimination of the longitudinal stiffeners.

Table 1.17 Continued. UDHSPS Design Alternative

Recycling, Fire Resistance, Repair, other Considerations
<ul style="list-style-type: none"> • The elastomer is a thermo-recyclable material, BASF (1997). • Fire retardants can be added to the elastomer to improve its fire resistance. Smoke suppressants can also be added to the elastomer. • The structural laminate construction will degrade locally in the region of the fire and, as in all-steel construction, would have to be repaired. • The elastomer chars like wood when exposed to open flame and the off-gases are comparable to wood. • Sections would be cold cut and a local strip of elastomer around the repair region would be shot blasted away. The replacement procedure would be similar to its construction. • The elastomer is fuel and oil resistant. • The elastomer is hydrophobic.
Cost effectiveness (Weight, Fabrication, and Cargo Heating)
<ul style="list-style-type: none"> • The UDHSPS reduces the weight of the tanker hull section by 12.8% over that of an equivalent CDH design, which will result in lower operating costs, Bennett (1998). • Reduced fuel consumption (decreased operating cost) due to a lighter tanker weight. • Fabrication and production engineering issues have been addressed in the details of the design. • The cost of the UDHSPS vessel, according to Bennett (1998), is 11.6 % less than the CDH vessel designed in Section 1.2. • The elastomer is an insulator. As a result, the cargo heating costs (operating cost) will be reduced.

1.5 Objectives and Scope

Experimental testing, numerical verification and simulation of the elastomer behaviour, composite structural laminate construction, and the implementation of UDHSPS technology in double hull ship design is required to demonstrate its validity. In doing so, the provisions in Lloyd's Rules and Regulations (1994) and similar provisions for other classification societies for alternative and innovative designs may also be satisfied. The objectives and scope of this study compliment those by Bennett, (1998) and are:

- To verify the application of existing finite element technology in modeling the behaviour of polyurethane elastomer in both tension and compression.
- To determine the geometry and material characteristics required of the composite structural laminate to provide normal operating performance equivalent to that of typical conventional double hull panels. As part of establishing equivalency, an inelastic buckling analysis of typical all-steel stiffened panels and a parametric study which evaluates the buckling characteristics of the composite panel as a function of mesh density and structural properties of the elastomer are required.
- To verify the application of existing finite element technology to model the composite structural laminate construction through comparison with the test results of a large-scale double hull section subject to transverse loads. This analysis also provides a complete picture of structural behaviour of this construction, which will verify the proposed design philosophy, Bennett (1998).
- To apply the same finite element technology in the analysis of hull modules for both CDH and UDHSPS designs, subject to specified design loads, to establish equivalent performance.

The numerical simulations presented and verified within this thesis make it possible to simulate the structural behaviour of the UDHSPS design for load combinations and design configurations that are not easily tested.

1.6 Thesis Outline

A state-of-the-art review of double hull oil tanker technology as related to current design provisions and to performance in accidental or extreme load events has been

presented in Chapter 1. A brief description of a unidirectional double hull sandwich plate system (UDHSPS) that was developed to give equivalent in-plane operational performance and improved resistance to collision and groundings has also been presented. Chapter 2 briefly describes the behaviour of the polyurethane elastomer and the finite element constitutive model used to simulate its behaviour. Chapter 3 presents buckling solutions (obtained from nonlinear finite element analyses) for both stiffened panels and composite structural laminate panels. These solutions are used to establish structural equivalency under normal operating loads. Verification of the application of finite element modeling of this type of construction is demonstrated through a comparison of the results from the finite element analysis with those obtained from a large-scale test. The finite element modeling techniques developed in the previous chapters were used to compare the behaviour of a conventional double hull structure module to two equivalent unidirectional double hull sandwich plate structure modules under typical design loads. A complete description of the global analysis is given in Chapter 4. A summary, conclusions, and recommendations, along with a discussion of possible future work are presented in Chapter 5.

Chapter 2

Elastomer Behaviour and Finite Element Analysis (FEA)

2.1 General

**“Replacement of metals by plastics is not new but is presently taking place at an increasing pace due, in part, to the advantageous cost to weight and stiffness ratios of plastics and a faster development of automated fabrication processes.”
Hepburn (1982)**

Polyurethane elastomers have strength, ductility, bond, and stiffness properties that make them ideally suited for a composite structural laminate panel application in which the elastomer is used as the core and steel as the faceplates. The basic components of polyurethane elastomers, their functions, the injection process typical for this application, the behaviour and relevant properties, and elastomer specifications are described. Existing finite element analysis technology is used to model the behaviour of the elastomer in both tension and compression. The solution options, verification of the Mooney-Rivlin constitutive law (that models the hyperelastic behaviour of the elastomer), and comparisons with the tension coupon and the compression disk test data are presented.

2.2 Polyurethane Elastomers: Chemistry and Properties

Polyurethane elastomers are engineered plastics in which the chemistry (polymer chains) can be modified to obtain physical characteristics, which make them useful for a

variety of applications. The term polyurethane is derived from the urethane group, which is formed in the polyaddition reaction of an isocyanate with compounds containing at least two hydroxyl groups (a polyol and a chain extender). These are considered the basic building blocks. For a 'one shot' reaction sequence, the basic building blocks are simultaneously mixed to produce the polyurethane elastomer structure (a linear block copolymer), illustrated in Figure 2.1. By modifying the building blocks, the chemical structure can vary its chemical properties (related to segmented flexibility, chain entanglement, interchain attraction, and chemical crosslinking) to produce the desired mechanical properties. A general description of the intermediates (blocks) is given in Table 2.1. Typical ranges of properties for elongation, tensile behaviour, flexural modulus, tear strength, hardness, glass transition temperature, and Bayshore rebound are given in Table 2.2.

Figure 2.1 Basic Polyurethane Elastomer Chemical Structure

Table 2.1 Intermediates and Related Properties

Isocyanate (MDI) Rigid Block	
<ul style="list-style-type: none"> • The rigid blocks of polyurethane elastomers are formed by the reaction of 4,4'-diphenylmethane diisocyanate (MDI) with a glycol (1,4-butane diol) or a diamine chain extender. • MDI is one of the most commonly used isocyanates in commercial elastomer manufacturing and has lower volatility than other isocyanates. • MDI has a high concentration of polar groups, which form hydrogen bonds that have strong interchain attractive forces producing a rigid segment. • In general, an increase in the MDI content increases the glass transition temperature, the flexural modulus, the tensile strength, the hardness, the coefficient of thermal expansion, and decreases the elongation. 	
Polyol Flexible Block	
<ul style="list-style-type: none"> • Typical polyols are polyesterols or polyetherols (most commonly used) with molecular weights of about 2,000 and carry terminal -OH groups. • Polyetherol is an amorphous material with molecular weight between 600 to 3,000. It is a low melting point solid and has glass transition temperatures below room temperature. • Generally affect properties related to flexibility, elongation, and glass transition temperature performance. • Increasing the molecular weight of this flexible block will result in an increase in elongation and a decrease in glass transition temperature, modulus of elasticity, and strength. 	
Flexible (1,4-butane diol) or Rigid (diamine) Chain Extenders	
<ul style="list-style-type: none"> • Diamines provide urea linkages with strong hydrogen bonds making a diamine extended material harder and stronger than a diol extended material. 	

Table 2.2 Thermoplastic Polyurethane Elastomer Characteristic Properties
(after ASM International, 1988)

Material Characteristic	Range
Elongation, %	100 – 800
Tensile Strength, MPa	20 – 55
Tensile Stress at 100% Elongation, MPa	2 – 35
Flexural Modulus, MPa	< 20 – 1035
Tear Strength, N/mm	25 – 260
Hardness, Shore	70A – 80 D
Glass Transition Temperature, °C	-70 – 15
Bayshore rebound, %	0 – 60

2.3 Elastomer Specification and Related Information

The use of a polyurethane elastomer as a core material for a structural composite laminate for a product oil tanker requires certain performance characteristics related to normal operating conditions, accidental or extreme load events, and fabrication as listed in Table 2.3.

The following presents the characteristic behaviour and limitations of the elastomer, as given by the tensile and compressive behaviour, energy absorption characteristics, effect of temperature on properties, hydrolytic stability, chemical resistance, and processing.

Tensile and Compressive Behaviour

Ranges of characteristic tension and compression stress-strain curves for a variety of polyurethane elastomers of different chemical composition are illustrated in Figure 2.2. Also given are the corresponding modulus of elasticity in tension, E_t , and compression, E_c . The curves are nonlinear and exhibit linear elastic behaviour up to strains of approximately $\pm 25,000 \mu\text{s}$, which is similar to the strain hardening value for steel. A normal operating strain range would be $\pm 1,000 \mu\text{s}$, which is well below the elastic limit. Poisson's ratio is approximately 0.5.

Table 2.3 General Specifications for Elastomer Properties

Normal Operating Conditions	
Condition	Description
Normal Operating Loads	<ul style="list-style-type: none"> The predominant load case is for a combination of hydrostatic and in-plane compressive loads due to the hogging and sagging moments as specified by the appropriate classification society. The elastomer material should have a modulus of elasticity in compression greater than 275 MPa, sufficient bond, and fatigue strength to provide composite sandwich behaviour throughout the life of the vessel.
Ambient Temperature	<ul style="list-style-type: none"> The normal operating temperature range as specified by Lloyd's Rules and Regulations (1994) Part 3, Chapter 3 is from -45°C to an unspecified upper temperature limit.
Corrosion	<ul style="list-style-type: none"> Corrosion may expose the elastomer to seawater and the oil products carried by the tanker. It should have hydrolytic stability and sufficient chemical resistance to these fluids.
Modification and Repair	<ul style="list-style-type: none"> Elastomer must provide sufficient resistance to heat induced by welding which may be, for example, as high as 260°C for a 5 mm fillet weld and 338°C for a 8 mm fillet weld on a 10 mm plate.
Accidental or Extreme Load Events	
Condition	Description
Fire and Explosion	<ul style="list-style-type: none"> The composite structural laminate must meet the requirements of SOLAS (1974) and subsequent amendments.
Collision and Grounding	<ul style="list-style-type: none"> The elastomer must provide sufficient rebound resilience, tear strength, compressive strength, tensile strength and elongation to maximize the energy absorption of the structural composite system. The compressive and tensile strengths should be greater than 20 MPa. The elongations should be equal to or greater than those of the steel.
Fabrication	
Condition	Description
Reaction Injection Moulding (RIM)	<ul style="list-style-type: none"> The material must be cast under shipyard conditions (temperatures and humidity). Pumping equipment should have the capacity to pump between 0.5 m³ and 1.0 m³ of elastomer in 3 to 5 min. Pumping equipment should have the ability to inject the elastomer at any angle with a reach of 20 m. The pumping process should provide quick release ports that mitigate overflow problems and a monitoring system to ensure that the cavities are properly filled.

Figure 2.2 Tensile and Compressive Stress-Strain Curves for Polyurethane Elastomers

Effect of Temperature on Properties

The maximum upper operating temperature is 80°C. Above this value, there is a non-reversible destructive chemical change in the structure of the polyurethane elastomer, which produces gradual permanent reduction in all properties. Polyurethane elastomers stiffen at low temperatures and exhibit a significant increase in hardness, tensile strength, tear strength, and torsional stiffness. This corresponds to a decrease in elongation and rebound resilience. The material becomes brittle at a temperature defined as the glass transition temperature, T_g , which occurs in the range between -60°C and -80°C. Note, there is no degradation of the chemical properties with decreasing temperatures and the effect on the properties is reversible when the temperature of the elastomer increases. The elastomer is a good insulator and may have large thermal gradients through its thickness. For this application, this implies a variation in properties through the thickness.

Hydrolytic Stability and Chemical Resistance

The -COOC- and the -COC- groups in the main chain structures of polyurethane elastomers will undergo hydrolytic attack in which the rate of degradation of properties is a function of temperature, pH value, and duration of contact. Hydrolytic degradation of polyurethane elastomers is characterized by the development of deep random cracks on the surface. Acidic or alkaline solutions, with pH values outside the range of 5.5 and 8, will accelerate the hydrolytic attack. Otherwise, the solution will behave much like water. Seawater has pH values of 8.24 ± 0.05 or 7.61 ± 0.09 depending upon whether it is warm or cold. Note, polyurethane elastomers formed with polyether polyols have more hydrolytic stability than those formed with polyester polyols.

The fluid and chemical resistance of polyurethane elastomers is dependent on the chemical groups present in the material with which they are in contact. In certain fluids, polyurethane elastomers swell, but when allowed to dry they return to their original dimensions. At temperatures below 50°C, polyurethane elastomers are considered to be one of the most oil resistant plastics available.

Energy Absorption Characteristics

The energy absorption of an elastomer can be characterized by its rebound resilience, tear strength, maximum tensile and compressive strength, and the corresponding elongation. These may all be altered by ambient temperature effects. Rebound resilience gives a measure of energy loss related to the hysteresis behaviour of the elastomer under fluctuating dynamic loads that is converted to heat. For components in flooring that may be subjected to normal operation vibration loads, it is anticipated that the composite structural laminate will offer substantial damping due to the rebound resilience of the elastomer core.

With respect to anticipated accidental or extreme load events, the dissipation of energy through heat loss will be negligible compared to the strain energy absorbed by the elastomer as it undergoes large deformations.

RIM Processing

Reaction injection moulding (RIM) may also be referred to as liquid injection moulding (LIM), or sometimes high pressure impingement mixing (HPIM). It is a variation of a 'one shot' process in which the polyol, chain extender, and catalyst are premixed and then reacted with the isocyanate through a mixing head and injected under

pressure into a mould, where it cures to produce the finished polyurethane product. For this application, the steel plate is the mould for the structural laminate.

Figure 2.3(a) Schematic of the RIM System

Figure 2.3(b) Typical RIM Equipment (after Elastogran GmbH, 1995)

A schematic diagram of the process and an illustration of typical equipment are given in Figure 2.3(a) and 2.3(b), respectively. The equipment illustrated in 2.3(b) shows a jib-arm-supported mixing head that could easily be modified to incorporate current

robotic technology to allow some automation of the casting process in a shipyard. Based on the existing equipment, the volume and time limits given in the specification can easily be achieved with a single unit.

2.4 Finite Element Modeling Theory

The finite element modeling of the polyurethane elastomer, composite structural laminate plates, stiffened steel plates, conventional double hull midship section (modeled with orthotropic plates), and UDHSPS midship section (modeled with sandwich elements) was conducted using ANSYS finite element program version 5.3 on a SUN SPARCstation 20 and a Pentium 133MHz/200MHz personal computer. ANSYS is a well-established finite element analysis (FEA) program and has the capability to conduct a full nonlinear analysis (geometric and material), including buckling and post-buckling behaviour, for the material, structural elements, and structures cited above. A brief overview of the solution option, convergence criteria, nonlinear geometry, and stress stiffening options that are applicable to all the analyses carried out in this study, is presented in this section.

2.4.1 Full Newton-Raphson (NROPT) Solution Option and Convergence

The standard Newton-Raphson method was used to achieve convergence of the system of nonlinear equilibrium equations. For a general system, these equations can be expressed as:

$$\underbrace{\{F^s\} - \{F_i^s\}}_{\text{Residual or Out-of-balance load vector}} = [K_i^T] \{\Delta u_i\} \quad (2.1)$$

Residual or Out-of-balance load vector
(the amount the system is out of equilibrium)

In which,
$$\{u_{i+1}\} = \{u_i\} + \{\Delta u_i\} \quad (2.2)$$

where,

- $\{F^a\}$ = Applied load vector
- $\{F_i^r\}$ = Vector of restoring force corresponding to the element internal forces
- $[K_i^T]$ = Jacobian matrix or tangent stiffness matrix
- i = Subscript representing the current iteration
- $\{u_i\}$ = Displacement configuration for the current iteration
- $\{u_{i+1}\}$ = Displacement configuration for the next iteration
- $\{\Delta u_i\}$ = Degree of freedom (DOF) increment vector

The iterative solution procedure is as follows:

- STEP 1** Given $\{u_0\}$ from the previous converged solution (taken as a zero vector for the first load step), compute the updated tangent matrix $[K_i^T]$ and the restoring forces $\{F_i^r\}$ from configuration $\{u_i\}$.
- STEP 2** Calculate $\{\Delta u_i\}$ from Equation 2.1.
- STEP 3** Add $\{\Delta u_i\}$ and $\{u_i\}$ to obtain an approximation of the next displaced configuration $\{u_{i+1}\}$ using Equation 2.2.
- STEP 4** Repeat steps 1 to 3 until convergence (equilibrium) is obtained where $\{F_i^r\}$ is within a predefined tolerance value of $\{F^a\}$.

The stiffness matrix is updated for each iteration and the solution obtained corresponds to load level $\{F^a\}$. Tolerance value, ϵ_R , was set between 0.001 and 0.05 depending upon the structure being analyzed. Convergence is achieved when:

$$\|\{R\}\| < \epsilon_R R_{ref} \text{ (out-of-balance convergence)} \quad (2.3a)$$

where, $\{R\} = \{F^a\} - \{F_i^r\}$ (residual vector as in equation (2.1))
 ϵ_R = Tolerance value
 R_{ref} = R_{ref} is the out-of-balance reference value and defaults to $\|\{F^a\}\|$. For structural DOFs, if $\|\{F^a\}\|$ falls below 1.0, then R_{ref} assumes a value of 1.0.

The terms in the absolute values are vector norms, which are a scalar measure of the magnitude of the vector. For all analyses, the default Euclidian norm was used, and is

defined as the square root of the sum of the squares of the vector terms, such that for

$\|\{R\}\|$, the vector norm is $\left(\sum R_i^2\right)^{\frac{1}{2}}$.

2.4.2 Non-linear Geometry (NLGEOM) Option

The nonlinear geometry option accounts for large strains (distortions and significant element shape change) which affects the local element stiffness and large deflections (rotations) which transforms the element's local stiffness into global components. It allows surface loads to rotate with the element. This option can also be applied to cases where the strains are small and rotations are large. All the problems investigated as part of this thesis involved either large strains or large displacements, or both.

2.4.3 Stress Stiffening (SSTIFF) Option

Stress stiffening (also referred to as geometric stiffening, incremental stiffening, initial stress stiffening, or differential stiffening) addresses the out-of-plane stiffening or weakening of a structure due to in-plane stresses and is most pronounced for a thin, highly stressed structure, with a relatively small bending stiffness as compared to its axial stiffness. For example, membranes stretched by in-plane tensile loads have substantial lateral stiffness.

2.5 Finite Element Modeling of Polyurethane Elastomer Behaviour

The polyurethane elastomer core is a nearly incompressible non-linear elastic material. ANSYS uses the Mooney-Rivlin constitutive law to characterize this hyperelastic behaviour. The elastic potential function W (or strain energy density function) for hyperelastic materials is a scalar function of one of the strain deformation tensors, whose derivative with respect to a strain component determines the corresponding stress component and is expressed as:

$$S_{ij} = \frac{\partial W}{\partial E_{ij}} \equiv 2 \frac{\partial W}{\partial C_{ij}} \quad (2.4)$$

where, S_{ij} = Stress component
 W = Strain energy function per unit undeformed volume
 E_{ij} = Components of the Lagrangian strain tensor
 C_{ij} = Components of the right Cauchy-Green deformation tensor

The Lagrangian strain may be expressed as follows:

$$E_{ij} = \frac{1}{2}(C_{ij} - \delta_{ij}) \quad (2.5)$$

where, $\delta_{ij} = \begin{cases} 1.0 & i = j \\ 0.0 & i \neq j \end{cases}$

The deformation tensor C_{ij} is comprised of the products of the deformation gradients f_{ij} .

$$C_{ij} = f_{ik} f_{kj} = \text{component of the Cauchy-Green deformation tensor} \quad (2.6)$$

where, $f_{ij} = \frac{\partial x_i}{\partial X_j}$
 X_i = undeformed position of a point in direction i
 x_i = $X_i + u_i$ = deformed position of a point in direction i
 u_i = Displacement of a point in direction i

2.5.1 Mooney-Rivlin Constitutive Law

The strain energy density function for the hyperelastic elements can be expressed by the following polynomial.

$$W = \sum_{k+l=1}^N a_{kl} (\bar{I}_1 - 3)^k (\bar{I}_2 - 3)^l + 1/2 \kappa (\bar{I}_3 - 1)^2 \quad (2.7)$$

where, a_{kl} = Mooney-Rivlin material constants
 $\bar{I}_i$ = Reduced strain invariants in the i^{th} direction given by,

$$\bar{I}_1 = I_1 I_3^{-1/3} \quad (2.8a)$$

$$\bar{I}_2 = I_2 I_3^{-2/3} \quad (2.8b)$$

$$\bar{I}_3 = I_3^{1/2} \quad (2.8c)$$

where I_1 , I_2 , and I_3 are defined as,

$$I_1 = C_{ii} \quad (2.9a)$$

$$I_2 = \frac{1}{2} (I_1^2 - C_{ij} C_{ij}) \quad (2.9b)$$

$$I_3 = \text{volume change ratio} = \det C_{ij} \quad (2.9c)$$

and, the bulk modulus is given by,

$$\kappa = \frac{2(a_{10} + a_{01})}{(1 - 2\nu)} \quad (2.10)$$

where, ν = Poisson's ratio and has a value of 0.4999.

A nine-parameter cubic Mooney-Rivlin relationship was used for all the analyses involving the polyurethane elastomer. The strain energy density function for this relationship was obtained by setting $N = 3$ in Equation 2.7, which gives,

$$\begin{aligned} W = & a_{10}(\bar{I}_1 - 3) + a_{01}(\bar{I}_2 - 3) + a_{20}(\bar{I}_1 - 3)^2 + a_{11}(\bar{I}_1 - 3)(\bar{I}_2 - 3) \\ & + a_{02}(\bar{I}_2 - 3)^2 + a_{30}(\bar{I}_1 - 3)^3 + a_{21}(\bar{I}_1 - 3)^2(\bar{I}_2 - 3) \\ & + a_{12}(\bar{I}_1 - 3)(\bar{I}_2 - 3)^2 + a_{03}(\bar{I}_2 - 3)^3 + \frac{1}{2} \kappa (\bar{I}_3 - 1)^2 \end{aligned} \quad (2.11)$$

For small strains, $2(a_{10} + a_{01})$ represents the shear modulus, G , and $6(a_{10} + a_{01})$ represents the modulus of elasticity, E .

2.5.2 Polyurethane Elastomer Core Material Characteristics and Mooney-Rivlin Constants

Stress-strain curves were calculated from tension coupon and compression disk test data for a low strength, high elongation elastomer (Type A), and for the large-scale test elastomer (Type B) used in Bennett (1998), and are illustrated in Figure 2.4(a). The RIM formulation for the large-scale test elastomer is given in Table 2.4.

Table 2.4 Polyurethane Elastomer RIM Formulation

Component No.	Formulation	Percentage of Stream Weight	Mixture Ratio
1	Stream 1 – Elastocast 70559R Resin		
2	Polyol (polyetherol)	> 95%	0.8
3	1,4-Butane diol	< 5%	
	Drying agent	< 1%	
4	Stream 2		
	Diphenylmethane diisocyanate (MDI)	100%	1

The Mooney-Rivlin constants (a_k) were derived internally by ANSYS as curve fits from the tension and compression stress-strain data and are listed in Table 2.5. Note, a third set of stress-strain data for planar tension or planar compression could be used in combination with the other stress-strain data to determine the Mooney-Rivlin constants. Tests for these stress-states are very difficult to perform, and are not necessarily required as the Mooney-Rivlin constants can be derived adequately from the two stress-states described. The Mooney-Rivlin curves for elastomers A and B are illustrated along with the experimental curves in Figure 2.4. The modulus of elasticity in tension is greater than

that in compression for elastomers A and B. The ANSYS curve fit assigns the same modulus for both stress-states and is the only significant variance with respect to the test curves, as illustrated in Figure 2.5.

Mooney-Rivlin constants were developed for a proposed stronger and stiffer elastomer (Type C) that would give a structural laminate plate equivalent in-plane strength to a typical stiffened hull plate, as described Section 3.5. The constants are given in Table 2.5 and the corresponding stress-strain curve is plotted along with the Mooney-Rivlin curves for elastomer A and B in Figure 2.5.

Figure 2.4 Mooney-Rivlin and Experimental Stress-Strain Curves for Elastomers A and B

Table 2.5 Mooney-Rivlin Constants for Elastomers (A), (B), and (C)

		Elastomer (A)	Elastomer (B)	Elastomer (C)
Mooney-Rivlin Parameters	a_{10}	-1.0300076912	26.3234427789	76.2739906957
	a_{01}	4.3319485103	-8.8924665569	-30.8518380306
	a_{20}	51.8279083638	-1767.7325337845	-28832.4515295927
	a_{11}	-103.2455386321	3496.2437904760	57466.6840102031
	a_{02}	50.3263502391	-1743.5912460026	-28703.5153680227
	a_{30}	1.1005011568	-81.8333684499	-1238.6887681943
	a_{21}	-5.7413883867	311.6216948946	4659.7637656580
	a_{12}	-2.8222757348	44.1160141286	1068.7443593990
	a_{03}	0.3948049581	-5.6052858900	-163.1402641275
Modulus of Elasticity	E	19.81 MPa	104.59 MPa	272.53 MPa
Shear Modulus	G	6.60 MPa	34.86 MPa	90.84 MPa

Figure 2.5 Mooney-Rivlin Stress-Strain Curves

2.5.3 Modified Strain Energy Density

Strain energy density for hyperelastic elements in ANSYS is modified to include hydrostatic pressure as an explicit solution variable and can be expressed as,

$$W + Q = W - \frac{1}{2\kappa}(p - \bar{p})^2 \quad (2.12)$$

where,

- Q = energy augmentation due to volume constraint condition
- κ = bulk modulus
- p = pressure obtained from W
- $\bar{p}$ = separately interpolated pressure

Note, it uses separate interpolations for the displacements and hydrostatic pressure and is referred to as the U-P formulation. This modification enforces the incompressibility constraint and reduces the possibility of the displacement locking which can occur for nearly incompressible materials at large strains.

2.5.4 HYPER58 3-D 8-Node Mixed U-P Hyperelastic Solid Element

The hyperelastic mixed U-P solid (HYPER58), developed for nearly incompressible materials with large displacements and strains, was used to model the elastomer for the numerical simulations of the tension coupon and compression disk material tests. HYPER58 element is a three-dimensional, eight node element with three degrees of freedom at each node and is shown in Figure 2.6.

HYPER58
8 nodes, 3-D space
DOF UX, UY, UZ

Figure 2.6 Hyperelastic Mixed U-P Solid Element Type

2.6 Analytical Verification of Tension Coupon Test

Objective:	To assess the validity of the Mooney-Rivlin constitutive law by comparing the results of a numerical simulation with the data from a tension coupon test.
Analysis Type:	Nonlinear static analysis
Element Type:	Hyperelastic solid (HYPER58)

2.6.1 Problem Description

Tension coupons were tested in accordance with ASTM D412-83 (1984), "The Standard Test Method for Rubber Properties in Tension", to obtain the stress-strain characteristics in uniaxial tension. A typical coupon is illustrated in Figure 2.7 and the corresponding load-elongation curve is given in Figure 2.9, Bennett (1998).

Note, the coordinate axes system is shown. The positive Z-axis is out of the page.

Figure 2.7 Typical Tension Coupon

2.6.2 Modeling Notes and Analysis Assumptions

Element Type and Material Characteristics

The polyurethane elastomer was modeled with HYPER58 solid elements and elastomer (B) Mooney-Rivlin constants.

Geometry, Boundary Conditions and Finite Element Mesh

Due to the uniform state of stress and strain within the gauge length of the tension coupon and due to symmetry, only the quarter volume shaded in Figure 2.7 was modeled. The corresponding volume with the appropriate boundary conditions is illustrated in Figure 2.8. Nodes on the Y-Z plane of transverse symmetry were restrained in the X-direction. Nodes on the X-Z plane of longitudinal symmetry were restrained in the Y-direction. A single row of nodes, at the centre of these planes of symmetry, were restrained to prevent out-of-plane movement.

The volume was subdivided with a fine mesh (4 elements across the width and depth, 16 elements along the length, for a total of 256 elements) and a coarse mesh (2 elements across the width and depth, 3 elements along the length, for a total of 12 elements) to determine the effect of mesh density on the solution. The solution results were identical.

Loading

Loading was applied to the volume through a prescribed displacement in the Y-direction to the master node on the top face. The remaining nodes on that face were coupled to it in the Y-direction, as illustrated in Figure 2.8. The reaction force was retrieved from the nodal data for the master node in the coupled set on the bottom face.

Due to instabilities in the material constitutive law for hyperelastic elements, the load should be applied in step sizes (load increments) small enough such that the resulting strain increments do not exceed 100%. The load was applied in a 10 mm displacement with 20 substeps, such that the tensile strain for any given substep was 3.9%. Due to symmetry, this displacement is half of the total displacement within the gauge length, hence, for comparison purposes, the displacement values were multiplied by two.

Loading <ul style="list-style-type: none"> • Total Displacement = 10 mm • Number of substeps = 20 	Model Properties (coarse mesh) <ul style="list-style-type: none"> • Number of Nodes = 36 • Number of Elements = 12
Material Properties <ul style="list-style-type: none"> • Elastomer (B) Mooney-Rivlin Constants • $\nu = 0.4999$ 	Analysis Options <ul style="list-style-type: none"> • NROPT, FULL • NLGEOM, ON • SSTIFF, ON • ϵ_R = Force Convergence Tolerance = 0.1%

Figure 2.8 Problem Description and Representative Finite Element Model for the Tension Coupon

2.6.3 Results

The load-deflection curve obtained from the numerical simulation is illustrated along with the tension coupon load-deflection data in Figure 2.9. The undeformed and deformed elastomer volume at a Y-displacement of 10 mm is illustrated in Figure 2.10. The variance in the initial part of the curve is directly related to the initial curve fit done by ANSYS in determining the Mooney-Rivlin constants, as previously illustrated in Figure 2.4.

Figure 2.9 Comparison of Load-Elongation Curve for Tension Test.

Figure 2.10 Coupon Deformed Shape

2.7 Analytical Verification of Compression Disk Test

Objective:	To assess the validity of the Mooney-Rivlin constitutive law by comparing the results of a numerical simulation with the data from a compression disk test.
Analysis Type:	Nonlinear static analysis
Element Type:	Hyperelastic solid (HYPER58)

2.7.1 Problem Description

Compression disks were tested in accordance with ASTM D575-83 (1984), "The Standard Test Method for Rubber Properties in Compression" to obtain the stress-strain characteristics in uniaxial compression. A typical disk prior to loading and at ultimate are

shown in Figure 2.11 and the corresponding load deflection curve is given in Figure 2.13, Bennett (1998). The disks were compressed to about 70% of the original thickness.

Figure 2.11 Typical Compression Disk

2.7.2 Modeling Notes and Analysis Assumptions

Element Type and Material Characteristics

The polyurethane elastomer was modeled with HYPER58 solid elements and elastomer (B) Mooney-Rivlin constants.

Geometry, Boundary Conditions and Finite Element Mesh

Due to symmetry, only a quarter volume section of the compression disk shaded in Figure 2.11, was modeled. The nodes on the plane [A] and [B] were fixed in the Y- and X-directions, respectively. The nodes lying in the centre plane [B] were restrained to prevent any rigid body translation in the Z-direction. It was assumed that the contact surface between the compression disk and the load platens were frictionless since the

elastomer surfaces were waxy and the restrained expansion indicated by the curved surface around the perimeter of the disk was very localized.

The element size was selected to provide an even number of elements through the thickness, a sufficient number of elements to model the curved surface of the disk, and to give aspect ratios close to one, leading to a total of 162 elements, as illustrated in Figure 2.12.

Figure 2.12 Problem Description and Representative Finite Element Model for Compression Disk

Loading

Loading was applied to the volume through prescribed displacements in the Z-direction to master nodes on the top [C] and bottom [D] faces to compress the disk. The corresponding nodes on those faces were coupled to the master nodes in the Z-direction. The reaction force was retrieved from the nodal data for the master nodes on either face. A total displacement of 9 mm was applied, half to each face [C and D] in opposite directions in 25 substeps such that the compressive strain for any given substep was 2.8%.

2.7.3 Results

The load-deflection curve obtained from the numerical simulation is illustrated along with the compression disk load-deflection data in Figure 2.13. The undeformed and deformed elastomer volume at a total Z-displacement of 9 mm is illustrated in Figure 2.14. Consistent with the frictionless surface assumed in the analysis, the perimeter surface is straight.

The undulations illustrated for the ANSYS curve in Figure 2.13 reflects the curve fit given by ANSYS in determining the Mooney-Rivlin constants from the compression data. The analytical response gives a close fit to the experimental curve.

Figure 2.13 Load-Deflection Results Comparison for Compression Test

Figure 2.14 Compression Disk Deformed Shape

Chapter 3

Finite Element Analysis and Verification of Composite Structural Laminate Plates and Double Hull Substructures

3.1 General

The emphasis of this chapter is to verify the application of existing finite element technology in modeling the composite structural laminate construction for product oil tankers. To accomplish this, the following series of finite element analyses on the relevant ship components, illustrated in Figure 3.1, was conducted:

- to obtain consistent buckling loads that are independent of the mesh density and to ensure global buckling, a parametric study was conducted on a continuous composite structural laminate plate subjected to in-plane loads, in which the mesh density and core material properties were varied. Details are given in Section 3.4.
- to establish structural equivalency (in-plane strength including the influence of hydrostatic forces), typical stiffened steel plates from key locations around the hull of a CDH tanker and composite structural laminate plates with varying core and plate thicknesses were analyzed. Details are given in Section 3.5 and 3.6.

- to calibrate the use of finite elements to predict the behaviour of a substructure of a UDHSPS product oil tanker subjected to transverse loading, a comparison was made of the results from a numerical simulation with the data of the corresponding large-scale test, Bennett (1998). Details are given in Section 3.7.

Figure 3.1 Application of Structural Laminate Construction in Product Oil Tankers

3.2 Characteristic Buckling Modes for Sandwich Plates

Oil tankers must resist the hogging and sagging moments due to wave motion. The composite structural laminate plates between longitudinal girders must provide in-plane compressive resistance greater than the compressive forces corresponding to these moments.

Depending upon the material properties of the structural laminate (E_c , E_f , and G_c) and the geometry (t_1 , t_2 , t_c , and d) as illustrated in Figure 3.2, there are five different characteristic buckling modes that can be classified as either local or global buckling. These are listed in Figure 3.3, with theoretical elastic buckling solutions and buckled configurations given in Table 3.1 and 3.2, for local and global buckling modes, Dreher (1992) and Plantema (1966). Note, the formulae apply to sandwich plates with edges parallel to the direction of loading that are free to displace and rotate. The highlighted local and global buckling modes in Figure 3.3 are the likely failure modes for composite structural laminate construction with an elastomer core.

In summary, the maximum buckling capacity of a composite structural laminate plate will occur if local buckling modes are prevented. Local buckling may either be symmetric or antisymmetric about the central plane of the core as illustrated in Table 3.1 and is characterized by the formulation of short waves along the face material. To prevent local buckling or face instability the core must have sufficient stiffness, E_c and shear stiffness, G_c . Dreher (1992) and others note that composite structural laminates subject to global buckling exhibit significant post-buckling capacity.

Figure 3.2 Geometric Properties

Figure 3.3 Characteristic Buckling Modes

Table 3.1 Characteristic Local Buckling Modes

Symmetric Face Wrinkling
<ul style="list-style-type: none"> • The critical buckling stress is independent of the shear modulus of the core • The critical buckling stress is based on the buckling of a column face plate attached to an elastic foundation (core) and is given by, $\sigma_{wr} = \phi \sqrt{\frac{t_f}{t_c} \cdot E_f \cdot E_c}$ <p>where, $\phi = \left(\frac{9(1 - v_c)^2}{2(1 + v_c)(3 - 4v_c)^2(1 - v_f^2)} \right)$</p> <p>$E_f$ = Modulus of elasticity for the face material E_c = Modulus of elasticity for the core material v_c, v_f = Poisson's ratio for the core and face.</p> <p>Note $\phi = 0.82$ for steel facing and incompressible core.</p>
Antisymmetric Face Wrinkling
<ul style="list-style-type: none"> • The shear modulus is included in the formulation as shear deformations occur in the core. • The critical buckling stress is given by, $\sigma_{wr} = \phi \sqrt[3]{E_f \cdot E_c \cdot G_c}$ <p>where, G_c = Shear modulus for the core material.</p> <ul style="list-style-type: none"> • The critical buckling stress due to antisymmetric wrinkling is always less than that for symmetric wrinkling, Allen (1969).
Face Dimpling	
N/A	<ul style="list-style-type: none"> • Only applicable to honeycomb sandwich construction.

Table 3.2 Characteristic Global Buckling Modes

Global buckling without shear strain
<ul style="list-style-type: none"> The elastic buckling stress is given by, $\sigma = \frac{\pi \mu^2}{\lambda} \cdot \frac{E_f}{1 - \nu_{xy} \nu_{yx}}$ <p>where, λ = Slenderness ratio = $\frac{l(t_1 + t_2)}{d\sqrt{t_1 t_2}}$</p> <p>$\nu_{xy}, \nu_{yx}$ = Poisson's ratio in the plane for the x and y direction</p> <p>μ = 2.0, 1.0, 1.43, 0.5 for fixed-fixed, pinned-pinned, fixed-pinned, fixed-free boundary conditions, respectively.</p>
Global buckling with shear strain
<ul style="list-style-type: none"> The elastic buckling stress includes the shear crimping stress, σ_{cr}, which is a function of the stress stiffness of the core and is given by, $\frac{1}{\sigma_b} = \frac{1}{\sigma} + \frac{1}{\sigma_{cr}}$ <p>where, $\sigma_{cr} = \frac{G_c \cdot d^2}{t_c(t_1 + t_2)}$</p>

3.3 Element and Material Constitutive Model Selection for the Composite Structural Laminate and Steel Plates

ANSYS had only one element, SHELL91 that was suitable for modeling the composite structural laminate plate. This element allows for shear deformation of the core and takes into account both geometric and material nonlinearities including stress

stiffening. To match midside nodes and functionality, the SHELL93 element was selected to model the steel plates. Both elements are quadratic with midside nodes that cannot be removed and are described in Table 3.3.

Table 3.3 SHELL91 and SHELL93 Element Description

SHELL91 16-Layer Structural Shell Element – Composite Structural Laminate
<p>8-nodes, 3-D space</p> <p>DOF: UX, UY, UZ, ROTX, ROTY, ROTZ</p>	<ul style="list-style-type: none"> • Suitable for modeling thick sandwich structures consisting of three layers. • Element definitions include layer thickness (may be variable), layer materials, and direction angles for orthotropic materials. • Different orthotropic or isotropic nonlinear material properties can be assigned to each layer. • The core thickness must be greater than or equal to 5/7 of the total thickness of the sandwich. • The core material carries transverse shear. • Shear deformations of the core are included. • The plane containing the 8 nodes can be located at the centre of the top, middle, or bottom surface of the sandwich. • Stores output for the centre of the top and bottom layers.
SHELL93 Structural Shell Element – Steel Plate
<p>8-nodes, 3-D space</p> <p>DOF: UX, UY, UZ, ROTX, ROTY, ROTZ</p>	<ul style="list-style-type: none"> • Geometric and material non-linear capabilities including stress stiffening, large deflection, and large strain. • Element definition includes the thickness at each corner and orthotropic (if applicable) nonlinear material properties.

The nonlinear material behaviour of the elastomer was modeled with Mooney-Rivlin curves, as described in Chapter 2, and the steel with the multilinear isotropic

hardening material model, which uses the Von Mises yield criterion and the associative flow rule.

3.4 Parametric Study of Mesh Density and Core Characteristics of Composite Structural Laminate Plates (Sandwich Panels) in Buckling Analysis

Objective: To obtain consistent buckling loads that are independent of mesh density for a composite structural laminate subject to in-plane loads. To obtain core stiffness that ensures global buckling.

Analysis Type: Nonlinear static analysis

Element Type: 16-Layer Structural Shell (SHELL91)

3.4.1 Problem Description

The composite structural laminate plate is 2,000 mm by 3,600 mm in plan, and represents a typical panel size for a product oil tanker, where its dimensions are defined by a section of hull plate supported by the major structural elements. The boundaries for the hull panel are continuous and provide rotational restraint. The hydrostatic load is constant and equal to that at the base of the ship for a 12,000 mm pressure head. The in-plane loads are varied and are a function of the loads induced by the hogging and sagging moments. A summary is given in Figure 3.4.

3.4.2 Modeling Notes and Analysis Assumptions

Element Type and Material Characteristics

The structural composite laminate was modeled with SHELL91 elements with constant thickness for the steel and the elastomer of 10 mm and 50 mm, respectively. The elastomer material was defined by the Mooney-Rivlin parameters for all three elastomers (A), (B), and (C), as described in Section 2.4.2. The multilinear true stress-strain

characteristics for the steel are given in Table 3.4. True stress-strain data is the required input for large strain analyses.

**Table 3.4 True Stress-Strain Data Input for Steel Faces of the Sandwich Panel
(after Toscano, 1995)**

True Stress (MPa)	True Strain, $\times 10^{-4}$ (mm/mm)
320.5	1545
322.6	4988
326.2	9950
352.9	19803
376.0	29559
395.2	39221
414.8	48790
438.7	67659
464.2	95310
492.2	139760
505.3	161270
E = 207430 MPa	
$\nu = 0.3$	

Geometry, Boundary Conditions and Finite Element Mesh

The boundary conditions are illustrated in Figure 3.4. All the edges of the composite structural laminate plate were restrained against translation in the Y- and Z- directions and rotation about all axes. The X-displacements were fixed along the centreline to prevent rigid body motion of the panel in that direction. Note, the entire panel was modeled because typical boundary conditions, related to modeling symmetric structures (enforced zero rotation along the line of symmetry), would produce only symmetric buckling solutions.

Buckling solutions for mesh densities [1], [2], and [3], illustrated in Figure 3.5, were computed for elastomer types (A), (B), and (C). An additional solution was produced for mesh density [4] with elastomer type (B) core.

<p>Material Properties</p> <ul style="list-style-type: none"> Steel properties are defined in Table 3.4 $E = 207430 \text{ MPa}$ $\nu = 0.3$ Mesh density [1], [2], and [3] were each analyzed with elastomers (A), (B), and (C). Mesh density [4] was analyzed with elastomer (B). 	<p>Model Properties</p> <p>Mesh Density [1]</p> <ul style="list-style-type: none"> Number of Nodes = 65 Number of Elements = 16 <p>Mesh Density [2]</p> <ul style="list-style-type: none"> Number of Nodes = 121 Number of Elements = 32 <p>Mesh Density [3]</p> <ul style="list-style-type: none"> Number of Nodes = 381 Number of Elements = 112 <p>Mesh Density [4]</p> <ul style="list-style-type: none"> Number of Nodes = 1433 Number of Elements = 448
<p>Loading</p> <p>Two load conditions:</p> <ul style="list-style-type: none"> Initial hydrostatic pressure of 0.118 MPa applied to entire area to simulate a pressure head of 12 m. Compressive displacement load. 	<p>Analysis Options</p> <ul style="list-style-type: none"> NROPT,FULL NLGEOM, ON SSTIFF, ON Force Convergence Tolerance = 5%

Figure 3.4 Problem Description and Representative Finite Element Models for Sandwich Plate Buckling Parametric Study

Figure 3.5 Mesh Densities

Loading

The load was applied in two stages, with the first being hydrostatic pressure of 0.118 MPa applied transversely to the plate, which is equivalent to a 12 m pressure head at the keel. The in-plane load to cause buckling was applied in equal displacements to the master nodes located on edges A and B. The remaining nodes along the edges were coupled to the displacements of the master nodes in the X-direction. Displacement controlled loading provides a stable post-buckling response and provides uniform loading across the edge.

3.4.3 Results and Discussion

The results of this parametric study are summarized in Table 3.5, in which the element size, aspect ratio, buckling load, transverse deflection, ratio of buckling load to yield load of the steel (P/P_Y), and percentage difference in buckling load as mesh density is increased, are given for each material type. Figure 3.6, 3.7, and 3.8 illustrate a typical displaced shape at ultimate for mesh density [3] and the load – axial shortening curve for each mesh density, for elastomers (A), (B), and (C), respectively.

Composite structural laminate plates with weak cores, elastomer types (A) and (B), fail by antisymmetric wrinkling of the steel plates. Increasing the mesh density produced a larger number of waves and a lower buckling load. The maximum out-of-plane displacements on the wrinkled face occurred along element lines. A consistent load could not be achieved with increased mesh density. Note, the transverse behaviour was unaffected by mesh density as the transverse displacements were similar for all cases.

The effectiveness of the steel in carrying the in-plane loads is given by the ratio of P/P_Y . As indicated by the values given in Tables 3.5 (a), (b), and (c), the global buckling of the composite structural laminate plate is the preferred failure mode, with the steel achieving a minimum of 85% of its yield capacity prior to buckling for mesh density [3]. Based on this limited parametric study for the stated thicknesses of the plate and core, a minimum modulus of elasticity of the core required to obtain global buckling is 273 MPa. For global buckling, convergence of the buckling load was essentially achieved with mesh density [3], with the in-plane response being essentially identical for all mesh densities up to buckling.

Table 3.5 Summary of Buckling Solutions for Composite Structural Laminate Plates

Mesh Density	Element Length	Element Width	Aspect Ratio	Buckling Load, P	Transverse Deflection	P/P _y	Percent Difference in P
	mm	mm	-	kN	mm	-	%
1	900	500	1.80	6902.40	26.57	0.54	
2	450	500	1.11	4974.59	26.49	0.39	38.75
3	257	250	1.03	2827.88	26.49	0.22	75.91

(a) Type (A) Elastomer Core

Mesh Density	Element Length	Element Width	Aspect Ratio	Buckling Load, P	Transverse Deflection	P/P _y	Percent Difference in P
	mm	mm	-	kN	mm	-	%
1	900	500	1.80	9224.02	20.25	0.72	
2	450	500	1.11	7281.35	20.28	0.57	26.68
3	257	250	1.03	5060.45	20.29	0.40	43.21
4	129	125	1.03	4605.96	20.17	0.36	10.39

(b) Type (B) Elastomer Core

Mesh Density	Element Length	Element Width	Aspect Ratio	Buckling Load, P	Transverse Deflection	P/P _y	Percent Difference in P
	mm	mm	-	kN	mm	-	%
1	900	500	1.80	12388.7	11.90	0.97	
2	450	500	1.11	11378.6	11.92	0.89	8.88
3	257	250	1.03	10896.4	11.92	0.85	4.43

(c) Type (C) Elastomer Core

Figure 3.6(a) Out-of-plane Displacements (in mm) Obtained with Mesh Density [3] and Elastomer (A) Core

Figure 3.6(b) Axial Load vs. Axial Shortening for a 2,000 mm by 3,600 mm Composite Plate with Elastomer (A) Core

Figure 3.7(a) Out-of-plane Displacements (in mm) Obtained with Mesh Density [3] and Elastomer (B) Core

Figure 3.7(b) Axial Load vs. Axial Shortening for a 2,000 mm by 3,600 mm Composite Plate with Elastomer (B) Core

Figure 3.8(a) Out-of-plane Displacements (in mm) Obtained with Mesh Density [3] and Elastomer (C) Core

Figure 3.8(b) Axial Load vs. Axial Shortening for a 2,000 mm by 3,600 mm Composite Plate with Elastomer (C) Core

3.5 Stiffened Steel Plate Buckling Analysis

Objective:	To establish structural equivalency of the composite structural laminate plates with typical stiffened plates from key locations around the hull of a CDH product oil tanker, a series of inelastic buckling analyses were conducted on these stiffened steel plates to determine the behaviour and buckling loads for comparison.
Analysis Type:	Nonlinear static analysis
Element Type:	8-node Structural Shell (SHELL93)

3.5.1 Problem Description

Typical stiffened steel plates, measuring 3,000 mm by 3,600 mm in plan, from the bottom hull, side shell, and deck of a CDH oil tanker, as illustrated in Figure 3.9, were analyzed to determine the buckling capacity, when subjected to both hydrostatic and in-plane loads due to hogging and sagging moments. The strake and stiffener scantlings for these are taken from the design of a 40,000 DWT tanker (refer to Section 1.2.4) as listed in Table 3.6. The boundaries are continuous and provide rotational restraint.

3.5.2 Modeling Notes and Analysis Assumptions

Element Type and Material Characteristics

The steel was modeled with SHELL93 elements and the material was characterized by the true stress-strain data given in Table 3.4.

<p>Material Properties</p> <ul style="list-style-type: none"> Nonlinear material properties for steel are defined in Table 3.4 Elastic properties $E = 207430 \text{ MPa}$ $\nu = 0.3$ 	<p>Model Properties</p> <ul style="list-style-type: none"> Number of Nodes = 1325 Number of Elements = 414
<p>Loading</p> <p>Two load conditions:</p> <ul style="list-style-type: none"> Initial hydrostatic pressures are applied to the unstiffened surface to simulate the appropriate pressure head in zero substeps. Compressive displacement load of 5 mm was applied to either end in 100 substeps. The next 20 mm were applied in 50 substeps. 	<p>Analysis Options</p> <ul style="list-style-type: none"> NROPT,FULL NLGEOM, ON SSTIFF, ON Force Convergence Tolerance = 5%

Figure 3.9 Problem Description and Representative Finite Element Model for Stiffened Panel Analysis

Geometry, Boundary Conditions and Finite Element Mesh

The boundary conditions are illustrated in Figure 3.9. All of the edges were restrained against translation in the Y- and Z-direction and rotation about all axes. The X-displacements were fixed along the centreline to prevent rigid body motion in that direction. The stiffened plate was subdivided into 18 elements along its length and 15 elements along its width, with the stiffener height divided into two elements. This discretization gave aspect ratios close to one for all elements.

Loading

The loading was applied in two stages, with the first being the hydrostatic pressure corresponding to the location of the stiffened plate. Note that the density of the crude oil cargo is approximately 85% that of water. The in-plane load to cause buckling was applied in equal displacements to the master nodes on edges A and B. The remaining nodes along these edges were coupled to the displacement of the master node in the X-direction.

3.5.3 Results and Discussion

The initial out-of-plane deflections corresponding to the applied hydrostatic pressure, the ultimate buckling load, yield load, and ratio of buckling load to yield are summarized in Table 3.6 for each stiffened plate. The load – axial shortening curves for each stiffened plate are illustrated in Figure 3.10. The deformed shape and longitudinal stress distribution of the buckling load for plate 1 are illustrated in Figure 3.11.

All plates exhibited linear behaviour up to the buckling load. Buckling was characterized by a combination of stiffener and hull plate buckling about the midlength of the stiffened plate. The plates exhibited post-buckling strength varying from 45% to 70%

of the buckling load. Plates 1, 2, and 3 are relatively stocky and are able to obtain the yield strength of the entire section.

Table 3.6 Summary of Stiffened Panel Buckling Results for a Plate 3,000 mm Wide by 3,600 mm Long

Plate No.	Plate Thickness	Stiffener Spacings		Hydrostatic Pressure	Initial Out-of-Plane Deflection	Ultimate Buckling Load, P	Yield Load, P_Y	P/ P_Y
		Height	Thickness					
-	mm	mm	mm	MPa	mm	kN	kN	-
1	13	400	25	0.11772	1.64	25600	25318	1.01
2	14	400	27	0.14592	1.67	27690	26279	1.05
3	12.5	400	25	0.07848	1.14	25321	24837	1.02
4	11.5	350	23	0.10423	2.05	21115	23876	0.88
5	12	250	19	0.00981	0.30	18206	24356	0.75

Figure 3.10 Load vs. Axial Shortening for Stiffened Plates 1 to 5

Figure 3.11 Deformed Shape and Longitudinal Stresses (in MPa) at the Buckling Load for Plate 1

3.6 Equivalent Composite Structural Laminate Capacity

Objective: A series of inelastic buckling analyses was conducted on composite structural laminate plates to obtain behaviour and strengths that are comparable to typical stiffened plates from a CDH product oil tanker.

Analysis Type: Nonlinear static analysis

Element Type: 16-Layer Structural Shell (SHELL91)

3.6.1 Problem Description

Composite structural laminate plates, measuring 3,000 mm by 3,600 mm or 2,000 mm by 3,600 mm in plan, with varying steel plate and polyurethane elastomer core thicknesses, were analyzed to determine the buckling capacity when subjected to both hydrostatic and in-plane loads due to hogging and sagging moments. The 2,000 mm and 3,000 mm widths represent probable longitudinal girder spacing for the UDHSPS oil

tanker. The boundary conditions are considered to be the same as the stiffened steel plates, that is, continuous with rotational restraint.

3.6.2 Modeling Notes and Analysis Assumptions

Element Type and Material Characteristics

The composite structural laminate plate was modeled with SHELL91 elements. The elastomer material was defined by the Mooney-Rivlin parameters for elastomer C and the material properties for the steel were characterized by the true stress-strain data given in Table 3.4.

Geometry, Boundary Conditions and Finite Element Mesh

The steel plate and the elastomer core thicknesses, given in Tables 3.7 and 3.8, were selected to represent realistic sizes that can be produced and that must meet the sandwich thickness limitation specified in ANSYS for the SHELL91 element.

The boundary conditions are identical to those of the stiffened steel plates given in the previous section.

The mesh densities for plates A and B, defined in Figure 3.12, are consistent with mesh density [2] as given in Section 3.4. For this preliminary parametric study, it is believed that this mesh density will give sufficiently accurate buckling loads for global buckling behaviour.

Loading

As before, the hydrostatic pressures were applied followed by the application of displacements to cause buckling.

(a) Mesh for plate (A)

(b) Mesh for plate (B)

(c) Variables used to describe the dimensions of the different sandwich components.

Material Properties <ul style="list-style-type: none"> Steel properties as per Table 3.4 <ul style="list-style-type: none"> $E = 207430 \text{ MPa}$ $\nu = 0.3$ Core material is elastomer (C). 	Model Properties <p>Plate (A)</p> <ul style="list-style-type: none"> Number of Nodes = 173 Number of Elements = 48 <p>Plate (B)</p> <ul style="list-style-type: none"> Number of Nodes = 121 Number of Elements = 32
Loading <p>Two load conditions:</p> <ul style="list-style-type: none"> Initial hydrostatic pressure of 0.11772 MPa applied to entire area to simulate a pressure head of 12 m, in 10 substeps. Compressive displacement load of 100 mm at either end, in 300 substeps. 	Analysis Options <ul style="list-style-type: none"> NROPT, FULL NLGEOM, ON SSTIFF, ON Force Convergence Tolerance = 5%

Figure 3.12 Problem Description and Representative Finite Element Models for Composite Structural Laminate Plate Buckling Equivalency Study

3.6.3 Results and Discussion

The initial transverse deflection due to hydrostatic forces, buckling load, yield load based on steel plates alone, and P/P_Y are given for varying steel plate and elastomer core thicknesses for plate widths of 3,000 mm and 2,000 mm in Table 3.7 and Table 3.8, respectively.

The effectiveness of steel usage is illustrated in Figure 3.13, where P/P_Y is plotted against the average steel thicknesses for varying elastomer core thickness and composite structural laminate plate widths. Values of P/P_Y greater than 1.0 are possible due to the contribution of the core strength. Figure 3.14 illustrates the buckling load versus the average steel plate thickness for varying elastomer core thicknesses and widths of composite structural laminate plates. Note, the average thickness for these plates is equal to the sum of the steel plate thicknesses. Also shown are the buckling loads for the stiffened steel plate thickness, which is calculated by dividing the total cross-sectional area of steel by the plate width. These values can only be directly compared to the corresponding value of the composite structural laminate plates with a width of 3,000 mm.

To establish equivalency, initial transverse deflections due to hydrostatic pressures, in-plane stiffness, and buckling loads were compared. Although the initial transverse deflections due to hydrostatic forces were greater for composite structural laminate plates, the overall in-plane stiffness given by the axial shortening curves were similar to those of the stiffened steel plates. The buckling capacity for the composite structural laminate plate, with a core thickness of 70 mm and a width of 3,000 mm, is comparable to those for the stiffened plates from around the CDH structure. The buckling loads for the composite

structural laminate plates, for widths of 3,000 mm and 2,000 mm, are proportional to their widths. It is anticipated that with increased core stiffness and strengths, the buckling capacity for composite structural laminate plates, with a 50 mm core thickness, could be increased to match stiffened plate panels.

Table 3.7 Buckling Capacity of Composite Structural Laminate Plates with Width of 3,000 mm

Steel Thickness	Elastomer Core Thickness	Initial Transverse Deflection	Ultimate Buckling Load, P	Yield Load, P_Y	P/P_Y
mm	mm	mm	kN	kN	-
8	50	23.9	13596	15383	0.884
10	50	22.5	16208	19229	0.843
8	60	20.2	14301	15383	0.930
10	60	19.0	18154	19229	0.944
12	60	18.3	19616	23075	0.850
8	70	17.2	15070	15383	0.980
10	70	16.4	18941	19229	0.985
12	70	15.8	22229	23075	0.963
14	70	15.4	23084	26920	0.867

Table 3.8 Buckling Capacity of Composite Structural Laminate Plates with Width of 2,000 mm

Steel Thickness	Elastomer Core Thickness	Initial Transverse Deflection	Ultimate Buckling Load, P	Yield Load, P_Y	P/P_Y
mm	mm	mm	kN	kN	-
8	50	12.4	10614	10255	1.035
10	50	11.9	11379	12819	0.888
8	60	10.4	11370	10255	1.109
10	60	10.0	13000	12819	1.014
12	60	9.8	13788	15383	0.896
8	70	8.9	11190	10255	1.091
10	70	8.7	14298	12819	1.115
12	70	8.5	15633	15383	1.016
14	70	8.4	16300	17947	0.910

Figure 3.13 P/P_Y vs. Average Steel Plate Thickness, t_{avg}

Figure 3.14 Buckling Load vs. Average Steel Plate Thickness, t_{avg}

3.7 Analysis of Large-scale Experiment

Objective:	To calibrate the use of finite element analysis to predict the behaviour of a substructure of a UDHSPS product oil tanker subjected to transverse loads, a comparison of the experimental data obtained in the large-scale test conducted by Bennett (1998) was made with the corresponding results from numerical simulations.
Analysis Type:	Nonlinear static analysis
Element Types:	8-Node Structural Shell (SHELL93) 16-Layer Structural Shell (SHELL91)

3.7.1 Problem Description

The test specimen was a large-scale double hull typical bottom substructure of a 40,000 DWT UDHSPS product oil tanker and is illustrated in 3.15(a). Figure 3.15(b) illustrates an exploded view showing the components which include the outer and inner plate of the outer hull, spacers (required for fabrication), a stiffened outer perimeter with cap plates, transverse floors, longitudinal girders and inner hull plate with access holes for measurements. The substructure measures 2,600 mm by 5,000 mm in plan, with a depth between hulls of 700 mm and a girder spacing of 900 mm. The substructure was welded into a 40 ton horizontal reaction frame to provide strength and stiffness restraint for the test specimen, which would be similar to the surrounding tanker structure. A cross-sectional view and plan view of the test specimen in the horizontal reaction frame is shown in Figure 3.16.

The load was applied monotonically to failure via four knife-edge roller plate assemblies by a series of 500 ton actuators. The assemblies ensure known direction and point of application of load. The load remains vertical (perpendicular to the original orientation of the plate throughout the test). The knife-edges allowed rotation and the roller assemblies allowed freedom of longitudinal displacement with respect to the actuators. The 75 mm thick knife-edge load plates, in contact with the outer plate of the outer hull, measured 200 mm by 300 mm and were located along the plane of transverse symmetry between longitudinal girders. One was centered over the transverse floor and the others were uniformly spaced at 550 mm on centre as illustrated in Figure 3.17.

The numerical simulation results were compared to test results that characterize the behaviour of the composite structural laminate outer hull and its interaction with the longitudinal girders, transverse floor, and inner hull. Comparisons were made with all total applied load displacement curves for linear variable differential transformers (LVDT's) measuring the displacement of each knife-edge plate, the transverse displacement of the inner and outer hulls, and the out-of-plane displacements for the transverse floor and longitudinal girders. The locations of these measurements and the corresponding LVDT identification numbers are given in Figure 3.18.

Figure 3.15 Large-scale Experiment Specimen

(a) Cross-sectional View

(b) Plan View

Figure 3.16 Test Specimen, Horizontal Reaction Frame, Knife-Edge-Roller Assemblies, and Actuators

Material Properties <ul style="list-style-type: none"> • Nonlinear material properties for steel are given in Table 3.9 • Elastomer (B) Mooney-Rivlin constants 	Model Properties <ul style="list-style-type: none"> • Number of Nodes = 761 • Number of Elements = 2400
Loading Two load conditions: <ul style="list-style-type: none"> • Pressures were applied to the transverse floor plate and longitudinal girder plates to generate the initial imperfections. • The transverse load was applied by displacements at the load points using the corresponding measured data as input. 	Analysis Options <ul style="list-style-type: none"> • NROPT, FULL • NLGEOM, ON • SSTIFF, ON • Force Convergence Tolerance = 5%

Figure 3.17 Problem Description and Representative Finite Element Model for the Large-scale Test Specimen

Figure 3.18 LVDT Locations on Test Specimen

3.7.2 Modeling Notes and Analysis Assumptions

Element Type and Material Characteristics

The composite structural laminate (outer hull), illustrated in blue in Figure 3.17, was modeled with the SHELL91 element with its nodes located within the inner plate to properly model the connection between the inner plate of the outer hull and the longitudinal girders and transverse floor. The longitudinal girders, transverse floor, and inner hull were modeled with SHELL93 elements and are illustrated in purple in Figure 3.17.

The elastomer material was defined by the Mooney-Rivlin parameters for elastomer B. The engineering stress-strain data for the steel plates, determined by tension coupons tested by Bennett (1998), were converted to true stresses and strains using Equations 3.1 (a) and (b). True stress-strain data is the required input for large strain analyses.

$$\sigma_{\text{true}} = \sigma_{\text{eng}} (1 + \epsilon_{\text{eng}}) \quad (3.1a)$$

$$\epsilon_{\text{ln}} = \ln(1 + \epsilon_{\text{eng}}) \quad (3.1b)$$

The values for each plate are tabulated along with the corresponding modulus of elasticity and Poisson's ratio in Table 3.9.

Table 3.9 True Stress-Strain Data for Steel Plates

Inner Hull		Longitudinal Girders, Transverse Floors, and Spacers		Inner Plate of Outer Hull		Outer Plate of Outer Hull	
True Stress	True Strain, $\times 10^{-6}$	True Stress	True Strain, $\times 10^{-6}$	True Stress	True Strain, $\times 10^{-6}$	True Stress	True Strain, $\times 10^{-6}$
MPa	mm/mm	MPa	mm/mm	MPa	mm/mm	MPa	mm/mm
388.038	1790	376.679	1770	388.052	1830	393.292	1870
400.543	19180	392.520	20540	405.678	20700	414.403	21180
404.966	21320	403.346	27200	411.938	23700	424.304	27060
408.150	21920	440.715	36910	417.864	25140	445.892	31910
414.429	23200	460.150	41720	422.387	26390	463.417	36720
420.741	24610	474.638	46520	425.661	27260	475.832	41520
445.756	30940	484.103	51290	427.453	27930	490.990	46290
469.130	38390	505.938	60770	429.929	29410	508.329	55770
475.969	40840	524.746	70160	475.034	39030	529.275	65160
482.652	43410	541.279	79460	516.589	62670	543.391	74460
489.424	46190	553.333	88680	538.346	71970	556.649	83680
520.437	62980	564.440	97810	556.122	85780	569.020	92810
556.154	86180	577.812	111360	567.220	99370	582.740	106360
591.745	122220	589.789	124720	581.939	112800	593.777	119720
614.078	152720	599.187	137910	593.272	126040	602.380	132910
621.247	165510	621.338	176460	605.498	143430	616.268	154510
588.572	221140	486.482	229640	549.178	197950	570.120	200440
E = 216340 MPa		E = 212350 MPa		E = 212490 MPa		E = 210200 MPa	
$\nu = 0.286$		$\nu = 0.284$		$\nu = 0.279$		$\nu = 0.287$	

Geometry, Boundary Conditions and Finite Element Mesh

The outer hull is a composite structural laminate of two 7.94 mm thick steel plates separated by a 50 mm polyurethane elastomer core. The composite structural laminate outer hull is extended the full plan width and length under the assumption that the difference in behaviour, by replacing the stiffened perimeter with the laminate, will be negligible. Longitudinal and transverse spacers were 19.05 mm and 6.35 mm thick, respectively. The longitudinal girders and transverse floor were both 12.7 mm thick. The inner hull was 7.94 mm thick. Holes, measuring 200 mm by 300 mm with 75 mm

radii corners, were made at the locations shown on the model in Figure 3.17 to provide access between hulls for measurement and observations.

Due to transverse symmetry, only half the structure was modeled. The nodes on the plane of symmetry were fixed against translation in the Y-direction and fixed against rotation about the X- and Z-axes. All degrees of freedom for the nodes on the remaining edges were completely fixed, representing the welded connection between the test specimen and the reaction frame.

Loading

Initial imperfections were applied as surface pressures of 0.06 MPa in the positive X-direction on the transverse floor and 0.225 MPa in the negative Y-direction on the longitudinal girder. These surface loads produced sinusoidal out-of-plane deformations with a maximum at midheight of 1.0 mm and 6.5 mm for the transverse floor and longitudinal girders, respectively. These values are not unreasonable and are within the range of normal fabrication practices and limits. The modeled imperfections provide the same effect on the buckling behaviour as those related to out-of-plumb alignment of the supporting elements and the mechanics of load transfer from the composite structural laminate plate to these elements. The magnitudes of the surface loads were determined by trial and error, to buckle the elements at the same load levels observed in the test.

The transverse load was applied monotonically to failure by displacements at the four load points previously described, using the corresponding measured data as input. Note, due to a premature weld failure along the perimeter of the structure at a load of 8197 kN, displacement input data in the form of displacement increments for each

actuator had to be derived as the analytical model had not yet failed. This displacement data had the same proportions as those prior to failure.

The displacement model for each load point requires the definition of master nodes, slave nodes and constraint equations, such that it gives the same behavioural effects as the test load, which was applied as a pressure on the outer plate of the outer hull. A typical deformed shape taken through a load point for the test is shown in Figure 3.19(a). The deformed shape was characterized by a constant displacement of the outer plate across the width of the load plate, a variable thickness of the core, and a smooth inner plate. The deflections of the outer plate were constrained to the plane in contact with the load plate, which rotated about the knife-edge. The displacement model, illustrated in Figure 3.19(b), shows the pattern of nodes overlaying the load plate in which the master nodes are indicated by solid squares or circles, slave nodes by open circles, and the dashed lines indicate the dependencies of the slave nodes to the master nodes applied through constraint equations. For the loaded area centered underneath the transverse floor, two master nodes (indicated by the solid squares) were displaced. All other loaded areas were displaced through the centrally located master node. To simulate the rigidity of the load plate and the constrained displacements about the knife-edge, constraint equations for the Z-direction and rotation about the Y-axis were introduced about the centreline of the master nodes along the knife-edge. The deformed configuration consistent with this displacement load model, illustrated in Figure 3.19(a), results in a constant elastomer core thickness and a deformed inner plate that is similar to that of the test. Note, the through-thickness compression and deformation of the core was not accounted for. It is anticipated that due to the variance in deformation of the outer

plate and core, there will be discrepancies in displacements measured on the outer plate and in the stresses in the region around the load plates. It will be demonstrated, however, that the global behaviour is well modeled.

Figure 3.19(a) Typical Deformed Shape Taken Through a Load Point for the Test

Figure 3.19(b) Schematic of the Displacement Load Application

3.7.3 Results and Discussion

The general behaviour of the large-scale test specimen can be characterized by the total applied load – transverse centreline displacement curve for actuator number 3, as given in Figure 3.20. Key behavioural events are indicated on the load-displacement curve and are briefly described in Table 3.10. A detailed description of the experimental results is given in Bennett (1998).

Figure 3.20 Total Applied Load vs. Transverse Displacement for Centreline LVDT

Table 3.10 Key Behavioural Events of the UDHSPS during Transverse Loading

Point	Total Applied Load (kN)	Event Description
a	718 (697)	First yield of outer hull in flexure between longitudinal girders.
b	1020	Change in flexural strain distribution in longitudinal girders and localized yielding of the inner plate of the outer hull/transverse floor web junction.
c	1346	Spread of transverse floor yielding.
d	2957 (2925)	Inelastic buckling of transverse floor.
e	3783	Stable transverse floor web/outer hull plate post-buckled configuration.
f	5718	Outer plate of outside hull butt-weld failure.
g	5881	Corresponding load level after repair.
h	7388 (7777)	Inelastic buckling of longitudinal girders.
i	7485	Development of inelastic membrane action across the entire width of the outer hull.
j	8003	Fracture of inner cover plate fillet weld along the north side of horizontal reaction frame.
k	8201 (8197)	Shear-tension failure of outer plate of outside hull along north side of horizontal reaction frame at the edge of the composite plate.
l	8466	Termination of large-scale test.
m	(9762)	The ultimate load obtained from the numerical simulation
n	(8720)	Last load value before the analysis was terminated

Note: All load values within parentheses are the finite element loads for the behavioural events.

The behaviour and the progression of yielding, development of membrane action, occurrence of inelastic buckling and material failure exhibited in this test are also captured in Figures 3.21 to 3.25, which illustrate the Von Mises element stresses and deformed shape from the numerical simulation. For clarity, the stresses and displaced shapes are shown in an isometric view and planar views of the inner plate of the outside hull, inner hull plate, transverse floor and longitudinal girder.

Figure 3.21(a) Von Mises Element Stresses (in MPa) at First Yield of Outer Hull in Flexure, 697 kN

Inner Plate of Outer Hull

Transverse Floor

Inner Hull

Longitudinal Girder

Figure 3.21(b) Von Mises Element Stresses (in MPa) at First Yield of Outer Hull in Flexure, 697 kN

Figure 3.22(a) Von Mises Element Stresses (in MPa) at Inelastic Buckling of Transverse Floor, 2925 kN

Inner Plate of Outer Hull

Transverse Floor

Inner Hull

Longitudinal Girder

Figure 3.22(b) Von Mises Element Stresses (in MPa) at Inelastic Buckling of Transverse Floor, 2925 kN

Figure 3.23(a) Von Mises Element Stresses (in MPa) after Inelastic Buckling of Longitudinal Girders, 7389 kN

Inner Plate of Outer Hull

Transverse Floor

Inner Hull

Longitudinal Girder

Figure 3.23(b) Von Mises Element Stresses (in MPa) after Inelastic Buckling of Longitudinal Girders, 7389 kN

Figure 3.24(a) Von Mises Element Stresses (in MPa) at Shear-tension Failure of Outer Plate, 8197 kN

Inner Plate of Outer Hull

Transverse Floor

Inner Hull

Longitudinal Girder

Figure 3.24(b) Von Mises Element Stresses (in MPa) at Shear-tension Failure of Outer Plate, 8197 kN

Figure 3.25(a) Von Mises Element Stresses (in MPa) at Ultimate, 9762 kN

Transverse Floor

Longitudinal Girder

Inner Plate of Outer Hull

Inner Hull

Figure 3.25(b) Von Mises Element Stresses (in MPa) at Ultimate, 9762 kN

**Total Applied Load – Transverse Displacement Results:
Actuators, LVDT's 1, 2, 3, and 4.**

The total applied load – transverse displacement curves for actuator LVDT's 1, 2, 3, and 4 and the corresponding curves from the numerical simulation are illustrated in Figures 3.26, 3.27, 3.28, and 3.29, respectively. The finite element model was calibrated against these curves to capture the flexural behaviour of the composite structural laminate plate and inelastic buckling of both the transverse floor and longitudinal girders.

A series of simulations was conducted (results not shown), varying the displacement boundary conditions at the load points, which includes displacing all 7 master nodes, two adjacent master nodes (indicated by the solid squares in Figure 3.19(b)), or a single master node. The applied displacements were equal to the displacements measured in the test. The displaced shapes corresponding to the experiment and two of the displacement load models are illustrated in Figure 3.30. It may be noted, as seen in Figure 3.30(c), that the shear span is less for the 7 master nodes model, which yields an increased flexural capacity when compared to that obtained from the test. Decreasing the length of the hull plate between the longitudinal girders and the load plate also results in a stiffer response for any given displacement. The single master node displacement model more closely replicates the experiment in displaced shape, flexural capacity, and in total load-displacement response. The load point under the transverse floor was more closely modeled with the two master nodes displacement model as it yielded the correct buckling load.

A second series of simulations was conducted (results not shown), varying the magnitude of imperfections on the longitudinal girders, to give buckling loads and a post-buckling response similar to those of the test.

As part of the development of this analytical model the mesh density was refined, increasing the number of elements through the depth of the girders from 6 to 8 and quadrupling the number of elements in the hull plates. The total applied load-displacement responses were not affected.

The ultimate load of the analysis, indicated by point (m), represents a material failure in the outer hull at the load points. The unloading portion of the curve from point (m) to (n) is consistent with the stress-strain characteristics for the steel in the outer hull.

Figure 3.26 Total Applied Load vs. Transverse Displacement for LVDT 1

Figure 3.27 Total Applied Load vs. Transverse Displacement for LVDT 2

Figure 3.28 Total Applied Load vs. Transverse Displacement for LVDT 3

Figure 3.29 Total Applied Load vs. Transverse Displacement for LVDT 4

**Total Applied Load – Transverse Displacement Results:
Outer Hull Along Plane of transverse symmetry LVDT's 5, 6, 7, 8, and 9**

The total applied load – transverse displacement curves for LVDT 5 and the corresponding curves from the numerical simulations for models with and without spacers are illustrated in Figure 3.31(a). Without spacers, the total load displacement curve started to deviate from the experimental curve after buckling of the transverse floor. This discrepancy, for any given load level, is the result of increasing sag in the outer hull plate between the transverse floor and the reaction frame. Since the measurements were taken from the outside of the hull plate, the increased sag translates to a smaller measured displacement for the same load level. Obviously, spacers that were present in the test increased the stiffness of the hull plate in this region and decreased sag.

Although a new model incorporating both the transverse and longitudinal spacers yielded a total applied load-displacement curve that more closely matched the experimental curve, some discrepancies still existed. In the experiment, the spacer was welded to both the inner and outer plates of the outer hull, while in the analysis, the spacer was connected to the inner plate only, as illustrated in Figure 3.31(b). The analytical model does not explicitly provide shear connection to the bottom plate and is consistent with the discrepancies seen in the load-displacement curves. A portion of this discrepancy may also be attributed to through thickness core compression. No further attempts were made to improve the model beyond this point as the cause for the discrepancy is known and its effect is local. Note that the analytical results presented in this section are all based on the model with spacers.

The total applied load – transverse displacement curves for the remaining LVDT's 6, 7, 8, and 9, on the plane of longitudinal symmetry, between, and adjacent to load points, and the corresponding curves from the numerical simulation are illustrated in 3.32, 3.33, 3.34, and 3.35, respectively. The pairs of curves exhibit similar characteristics to that of LVDT 5, with the magnitude of the discrepancies being different depending on the location and variations in the hull plate stiffness between the experiment and the finite element model at these locations.

(a) Experimental

(b) Single Master Node Displacement Model

(c) 7 Master Nodes Displacement Model

Figure 3.30 Displaced Shape for Various Displacement Load Models

Figure 3.31(a) Total Applied Load vs. Transverse Displacement for LVDT 5

Figure 3.31(b) Analytical Model for Spacer

Figure 3.32 Total Applied Load vs. Transverse Displacement for LVDT 6

Figure 3.33 Total Applied Load vs. Transverse Displacement LVDT 7

Figure 3.34 Total Applied Load vs. Transverse Displacement for LVDT 8

Figure 3.35 Total Applied Load vs. Transverse Displacement for LVDT 9

**Total Applied Load – Transverse Displacement Results:
Outer Hull Along Longitudinal Girder LVDT pairs 10/14, 11/15, 12/16, and 13/17,**

The total applied load – transverse displacement curves for LVDT 10/14 and the corresponding curve for the numerical simulation are illustrated in Figure 3.36. Even though the initial stiffness was the same, the characteristic curves are completely different. It is apparent that, the finite element model did not capture the load-displacement behaviour at these points.

Figure 3.36 Total Applied Load vs. Transverse Displacement for LVDT's 10 and 14

LVDT pairs 11/15, 12/16, and 13/17, measured the transverse displacement of the outer plate of the outer hull along longitudinal girders 1 and 2. The total applied load – transverse displacement curves and the corresponding curves from the numerical

simulation are illustrated for these pairs in Figure 3.37(a), 3.38, and 3.39. In all cases, the shape of the characteristic curves are the same and the change in slope is dominated by the buckling of the longitudinal girders. The variation in displacements for LVDT's 11, 12, and 13 with respect to LVDT's 15, 16, and 17, indicate that longitudinal girder 2 has a stiffer response, which may be attributed to imperfections in the longitudinal girders or alignment of the test specimen within the horizontal reaction frame. The variations with respect to the finite element analysis are of the same order of magnitude and are attributed to the soft outer hull response due to the single node displacement load model, as illustrated in Figure 3.37(b). Note that a portion of the increased displacements of the experiment over that of the finite element analysis may be attributed to through thickness compression of the elastomer core beneath the longitudinal girders, as illustrated in Figure 3.37(c).

Figure 3.37(a) Total Applied Load vs. Transverse Displacement for LVDT's 11 and 15

Figure 3.37(b) System Behaviour at Longitudinal/Transverse Junction

Figure 3.37(c) Compression of Core

Figure 3.38 Total Applied Load vs. Transverse Displacement for LVDT's 12 and 16

Figure 3.39 Total Applied Load vs. Transverse Displacement for LVDT's 13 and 17

**Total Applied Load – Lateral Displacement Results:
Inner Hull LVDT's 18/22, 19/23, 21, 24, and 26**

The total applied load – lateral displacement curves for the inner hull LVDT's 18/22, 19/23, 21, 24, and 26 and the corresponding curves from the numerical simulation are illustrated in 3.40, 3.41, 3.42, 3.43, and 3.44(a), respectively. There are three sets of characteristic curves, one for the inner hull–transverse floor–longitudinal girder junction, the inner hull–longitudinal girder junction, and the inner hull along the plane of transverse symmetry. The steeper slope in the numerical prediction for the first two sets of curves indicate that there is less load being transferred into the inner hull plate than that of the experiment. This is consistent with previous observations. For these cases the experimental curves show a change in slope at point (d), buckling of the transverse floor, while the numerical curve was unaffected.

The curves for LVDT's 18/22 illustrated in Figure 3.40 show a relaxation in deflection (and elastic rebound) at point (h) due to buckling of the longitudinal girders. Once a stable equilibrium position was achieved for the buckled girders, incremental load was transmitted once again into the inside hull and is indicated by the positive slope on the load-displacement curve.

The load-displacement curves for both the experiment and numerical simulation, Figures 3.40 to 3.44, exhibit similar characteristics with post-buckling rebound of the inner hull, which is a function of inelastic creep of the plate girders. This time dependent event manifests itself as a zigzag pattern in the experimental data with horizontal displacements without load increment related to the extended periods of no loading. Positive sloped portions occurred during loading. The numerical simulation, based on static analysis, does not exhibit the time dependent loading effects.

Figure 3.40 Total Applied Load vs. Transverse Displacement for LVDT's 18 and 22

Figure 3.41 Total Applied Load vs. Transverse Displacement for LVDT's 19 and 23

Figure 3.42 Total Applied Load vs. Transverse Displacement for LVDT 24

Figure 3.43 Total Applied Load vs. Transverse Displacement for LVDT 21

The total applied load-displacement curve from the numerical simulation for LVDT 26, illustrated in Figure 3.44(a), shows an initial offset of 5.4 mm. Figure 3.44(b), illustrates the displaced shape consistent with the initial load step required to create the imperfections and the corresponding negative displacement at the LVDT 26 location. To eliminate this fictitious displacement, a linear correction was applied. The resulting curve was essentially identical to that of the experiment with a small variation at the end of the test where only the transverse floor was loaded.

Figure 3.44(a) Total Applied Load vs. Transverse Displacement for LVDT 26

Figure 3.44(b) Deformed Shape due to the Initial Load Step

**Total Applied Load – Lateral Displacement Results:
Longitudinal Girder LVDT Pairs 27/30, 28/31, and 29/32**

The total applied load – lateral displacement curves for the longitudinal girders LVDT pairs 27/30, 28/31, and 29/32 and the corresponding curves for the numerical simulation are illustrated in Figure 3.45(a), 3.46, 3.47. As before, a linear correction was applied to the curves for the numerical simulation to eliminate the displacement from the initial imperfection load step. The curves are essentially identical and are characteristic of inelastic web girder buckling problems. The limitations of the measurement system to effectively monitor large displacements without manual adjustments lead to the discontinuity in the displacements as represented by the vertical and horizontal jumps in the experimental curves. Thus, displacement predictions by the numerical simulation in the post-buckling region would be larger than those in the experiment. This effect is illustrated in Figure 3.45(b). The numerical simulation best represents the lateral displacements in this region and beyond.

Figure 3.45(a) Total Applied Load vs. Lateral Displacement for LVDT's 27 and 30

Figure 3.45(b) Transverse Floor Buckling Verification

Figure 3.46 Total Applied Load vs. Lateral Displacement for LVDT's 28 and 31

Figure 3.47 Total Applied Load vs. Lateral Displacement for LVDT's 29 and 32

**Total Applied Load – Lateral Displacement Results:
Transverse Floor LVDT's 34 and 33/35**

The total applied load – lateral displacement curves for the transverse floor, LVDT's 34 and 33/35 and the corresponding curves for the numerical simulations are illustrated in Figures 3.48 and 3.49. The load-displacement curves, LVDT 34, for the central panel of the transverse floor are similar and characteristic of inelastic web buckling. The numerical simulation at point (d) predicted buckling. Variations between the two beyond this point are a function of the large lateral displacements of the web and limitations of the measuring system, as previously described. In reference to Figure 3.45(b), it can be seen that displacement prediction by the numerical simulation would be larger than those present in the experiment.

The characteristic load – lateral displacement curves for the outer panels of the transverse floor, given in Figure 3.49, are similar. As anticipated the out-of-plane displacements are relatively small, as the initial imperfections were small and buckling had not yet occurred.

Figure 3.48 Total Applied Load vs. Lateral Displacement for LVDT 34

Figure 3.49 Total Applied Load vs. Lateral Displacement for LVDT's 33 and 35

3.7.4 Summary

Based on the comparisons of the displacement data with the results of the numerical simulation, it can be concluded that the finite element model presented was successful in modeling the complex behaviour of the UDHSPS double hull substructure for the given loading.

Application of the finite element model for design, at load levels that exceed those associated with normal operating conditions, is beyond the scope of this thesis. It does however, provide both qualitative and quantitative information on how the load was carried and transmitted throughout the structure as load was increased. For example, the development and spread of membrane action in the outer hull as the longitudinal girders buckled was effectively captured by the in-plane element stresses in the Y-direction as shown in Figures 3.51 and 3.53. For completeness, an entire set of the in-plane element stresses for the directions aligned within the planes of the members are given for the outer and inner plate of the outer hull, transverse floor, the longitudinal girder and the inner hull in Figures 3.50 to 3.59. For each structural component, the element stresses are illustrated at load levels associated with first yield of the outer hull, transverse floor buckling, longitudinal girder buckling, and ultimate.

(a) At First Yield, 697 kN

(b) At Transverse Floor Buckling, 2925 kN

(c) At Longitudinal Girder Buckling, 7777 kN

(d) At Ultimate, 9762 kN

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 SMX =528.489
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 161.26
 272.853
 386.447
 499.04
 611.633

Figure 3.50 Element Stresses (in MPa) and Their Progression in the X-Direction for the Outer Plate of Outer Hull

(a) At First Yield, 697 kN

(b) At Transverse Floor Buckling, 2925 kN

(c) At Longitudinal Girder Buckling, 7777 kN

(d) At Ultimate, 9762 kN

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 270.037
 371.978
 473.918
 575.859
 677.799

Figure 3.51 Element Stresses (in MPa) and Their Progression in the Y-Direction for the Outer Plate of Outer Hull

(c) At First Yield, 697 kN

(d) At Transverse Floor Buckling, 2925 kN

(c) At Longitudinal Girder Buckling, 7777 kN

(d) At Ultimate, 9762 kN

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	-43.206
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	195.794
	315.294
	434.794
	554.294
	673.794

Figure 3.52 Element Stresses (in MPa) and Their Progression in the X-Direction for the Inner Plate of Outer Hull

(a) At First Yield, 697 kN

(b) At Transverse Floor Buckling, 2925 kN

(c) At Longitudinal Girder Buckling, 7777 kN

(d) At Ultimate, 9762 kN

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 3.468
 140.54
 277.611
 414.682
 551.753
 688.824

Figure 3.53 Element Stresses (in MPa) and Their Progression in the Y-Direction for the Inner Plate of Outer Hull

Plane of
transverse
symmetry

(a) At First Yield, 697 kN

(b) At Transverse Floor Buckling, 2925 kN

(c) At Longitudinal Girder Buckling, 7777 kN

(d) At Ultimate, 9762 kN

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182.955
309.895
436.836
563.776
690.716

```

Figure 3.54 Element Stresses (in MPa) and Their Progression in the Z-Direction for the Transverse Floor

(a) At First Yield, 697 kN

(b) At Transverse Floor Buckling, 2925 kN

(c) At Longitudinal Girder Buckling, 7777 kN

(d) At Ultimate, 9762 kN

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 SMX =256.183
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 -133.603
 2.468
 140.54
 277.611
 414.682
 551.753
 688.824

Figure 3.55 Element Stresses (in MPa) and Their Progression in the Y-Direction for the Transverse Floor

Figure 3.56 Element Stresses (in MPa) and Their Progression in the Z-Direction for the Longitudinal Girder

Figure 3.57 Element Stresses (in MPa) and Their Progression in the X-Direction for the Longitudinal Girder

(a) At First Yield, 697 kN

(b) At Transverse Floor Buckling, 2925 kN

(c) At Longitudinal Girder Buckling, 7777 kN

(d) At Ultimate, 9762 kN

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 -43.206
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 195.794
 315.294
 434.794
 554.294
 672.794

Figure 3.58 Element Stresses (in MPa) and Their Progression in the X-Direction for the Inner Hull

(a) At First Yield, 697 kN

(b) At Transverse Floor Buckling, 2925 kN

(c) At Longitudinal Girder Buckling, 7777 kN

(d) At Ultimate, 9762 kN

ANSYS 5.2
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 SMX =256.183
 -544.816
 -407.745
 -270.674
 -133.603
 3.468
 140.54
 277.611
 414.682
 551.753
 688.824

Figure 3.59 Element Stresses (in MPa) and Their Progression in the Y-Direction for the Inner Hull

Chapter 4

Midship Section Equivalency for Normal Operating Loads

4.1 Equivalency by Rationally Based Design

Classification rules and regulations do not have rule based equations for determining the scantlings for UDHSPS oil tankers, however, provisions allow the use of alternative designs if these can be proven equivalent to those produced by the rules and regulations. To establish equivalency, the rationally based design philosophy for ship structures as described by Hughes (1988) was used.

Since ship structures are complex and would require a large number of elements to model them, these structures are subdivided into substructure hull modules, as illustrated in Figure 4.1, and analyzed for the appropriate internal forces and boundary conditions. The length of the hull module must be sufficient to allow any form of overall hull girder failure to occur within the hull module and is generally a function of ship type. For this particular case, of a 40,000 DWT product oil tanker, the hull module should extend at least 3 cargo tanks with boundaries located at the transverse bulkheads, Hughes (1988). Due to both longitudinal and transverse (port-starboard) symmetry, only a quarter section of the three cargo holds need be modeled, with the appropriate boundary conditions as

illustrated in Figure 4.2. The design loads (shears, hogging/sagging moments, hydrostatic pressures) acting on the hull modules are representative of those from a hull girder analysis and were defined by Lloyd's Rules and Regulations (1994). Note that torsional loads are not considered as their effects are relatively small and would not be applicable to this symmetric model.

For this particular case, substructure hull modules for the baseline 40,000 DWT CDH product oil tanker and UDHSPS designs with either a 2 m or 3 m longitudinal girder spacing were modeled and analyzed. In conjunction with the results of the stiffened plate and composite plate buckling analysis presented in Section 3.5 and 3.6, and the results of the substructure hull analysis, structural equivalency is discussed.

4.2 Preliminary Finite Element Analyses

The substructure hull modules for the CDH and UDHSPS oil tankers are modeled with SHELL63 elastic orthotropic plate elements, and 16-layer structural shell (SHELL91) and 8-node structural shell (SHELL93) elements, respectively. Each element is relatively large and represents stiffened panels or composite structural laminate plates with plan dimensions consistent with the framing system. This section describes the preliminary finite element analysis required to demonstrate equivalent elastic behaviour of the plates modeled, with either the coarse or fine mesh.

Figure 4.1 Levels of Structural Analysis in Rationally-Based Design
 (after Hughes, 1988)

Figure 4.2 Boundary Restraints for a Doubly Symmetric Hull Module

4.2.1 Modeling Stiffened Plates

The CDH substructure hull module consists of a three dimensional framework of stiffened steel plates, which can be modeled with orthotropic plates, Hughes (1988). The orthotropic plates are unstiffened plates with different moduli of elasticity in the two principle directions that give equivalent axial stiffness for the stiffened plate. The derivation of the orthotropic modeling technique according to Hughes (1988), assumes that,

- The stiffeners are uniformly distributed over the plate; and
- The in-plane stress σ_x for the orthotropic plate is an average of the plate stress σ_p and the stiffener stress σ_s , illustrated in Figure 4.3.

An applied in-plane tensile stress for the orthotropic plate in the X-direction, σ_x , (parallel to the stiffeners), Figure 4.3(a) produces a strain ϵ_x equal to σ_x/rE , where r is the ratio of total cross-sectional area to plate area Bt . The corresponding transverse strain due to Poisson's effect is ϵ_y and is equal to $-\nu\sigma_x/rE$.

Due to an applied in-plane tensile stress, σ_y , Figure 4.3(b), the plate tries to contract in the X-direction due to the Poisson effect, but is restrained by the stiffeners. The compatibility of the plate and stiffener deflections places the stiffeners in compression and the plates in tension in the X-direction. For equilibrium, the force in the plate must be equal to the force in the stiffeners,

$$\sigma_p Bt = \sigma_s (r - 1) Bt \quad (4.1)$$

where, σ_p = Average tensile stress in the plate
 B = Plate width
 t = Plate thickness
 σ_s = Compressive stress in the stiffener
 r = Ratio of total cross-sectional area to plate area Bt

The corresponding changes in length L for the plate and stiffeners are,

$$\delta_p = \frac{L}{E} (\sigma_p - \nu \sigma_y) \quad (4.2a)$$

$$\delta_s = \frac{L}{E} \sigma_s \quad (4.2b)$$

where, δ_p = Change in length of the plate
 L = Plate Length
 E = Modulus of Elasticity
 ν = Poisson's Ratio
 σ_y = Stress in the direction perpendicular to the stiffeners
 δ_s = Change in length of the stiffeners

Substituting Equation 4.1, 4.2(a) and 4.2(b) into the compatibility equation, $\delta_p = \delta_s$, and rearranging the terms gives,

$$\frac{\sigma_p}{\sigma_y} = \frac{v(r-1)}{r} \quad (4.3a)$$

$$\frac{\sigma_s}{\sigma_y} = \frac{-v}{r} \quad (4.3b)$$

The corresponding strains in the X- and Y-directions, parallel and perpendicular to the stiffeners, respectively, are given by,

$$\frac{\epsilon_x}{\left(\frac{-v\sigma_y}{E}\right)} = \frac{1}{r} \quad (4.4a)$$

$$\frac{\epsilon_y}{\left(\frac{\sigma_y}{E}\right)} = 1 - \frac{v^2(r-1)}{r} \quad (4.4b)$$

For both the stiffened and orthotropic plates a uniform shear stress τ produces a shear strain, γ equal to τ/G . The elastic stress-strain relationships for the orthotropic plate are summarized by Equation 4.5.

$$\begin{Bmatrix} \epsilon_x \\ \epsilon_y \\ \gamma \end{Bmatrix} = \frac{1}{rE} \begin{bmatrix} 1 & -v & 0 \\ -v & r - v^2(r-1) & 0 \\ 0 & 0 & 2r(1+v) \end{bmatrix} \begin{Bmatrix} \sigma_x \\ \sigma_y \\ \tau \end{Bmatrix} \quad (4.5)$$

or equivalently $\epsilon = D^{-1}\sigma$

The moduli of elasticity in the X- and Y-directions are,

$$E_x = rE \quad (4.6a)$$

$$E_y = \frac{r}{r - \nu^2(r-1)} E \quad (4.6b)$$

Note, the stresses in the X- and Y-direction, given by the finite element analysis based on the use of orthotropic plates must be modified by the corresponding ratios of the elastic moduli. For example, parallel to the stiffeners, the stresses must be multiplied by $1/r$.

It is paramount to the global response of the substructure hull module that the orthotropic plates correctly model the in-plane behaviour. The local out-of-plane behaviour (bending of stiffeners and plating due to hydrostatic forces) is not well modeled. This is of no consequence as it has little or no effect on the global response provided the plates remain elastic and do not buckle.

Figure 4.3 Applied Stresses for a Stiffened Plate

The 40,000 DWT CDH product oil tanker is composed of a large number of stiffened plates, with plan dimensions of 3,600 mm by 3,000 mm, that are supported by transverse floors or webs and by longitudinal girders or side stringers. To simplify the substructure hull module analysis, these stiffened plates were modeled with a single orthotropic plate. To determine the effect of mesh density and orthotropic plate modeling (modification of the elastic moduli) analyses were conducted and compared with the results from an inelastic buckling analysis of one typical stiffened plate, plate 1 in Section 3.5.

The orthotropic steel plates were modeled with the elastic shell element, SHELL63, described in Table 4.1.

Table 4.1 SHELL63 Element Description

SHELL63 – Elastic Shell Element
<p>4 nodes, 3-D space</p> <p>DOF: UX, UY, UZ ROTX, ROTY, ROTZ</p>	<ul style="list-style-type: none"> • Capable of modeling bending and membrane action due to in-plane and normal loads. • Geometric nonlinear capabilities, including stress stiffening and large deflections. • Element definition includes the thickness at each corner and orthotropic or isotropic elastic material properties.

The geometric properties for the stiffened and orthotropic plates are given in Table 3.6 and 4.2, respectively. The elastic modulus used in the computation of the orthotropic material properties, listed in Table 4.2, are based on an E value of

207,430 MPa and a Poisson's ratio of 0.3. The boundary conditions, loading, and analysis were similar to those for stiffened plate studied in Section 3.5.

Table 4.2 Orthotropic Properties for Stiffened Plate

Plate Thickness	Stiffener Scantlings		Total Area of Stiffened Plate	Plate Cross-sectional Area	Area Ratio	Orthotropic Material Data		
T	d-t	w	A	Bt	r	E_x	E_y	E_z
13	400	25	79000	39000	2.026	420180	217096	207430

The axial load – shortening curves, given in Figure 4.4, characterize the results for the nonlinear buckling analysis for stiffened plate 1 and for both the elastic analysis on a single element and a four element orthotropic plate model. The latter analysis included the application of hydrostatic pressure equal to 0.118 MPa. As illustrated, by the slope of the curves, the elastic response for the single plate element closely matches that of the nonlinear analysis. As long as the compressive stresses in the substructure hull module analysis are below those associated with buckling (refer to Table 3.6), then the single orthotropic plate model should give results that are representative of the global behaviour of the stiffened plate model. Note that the 4-element orthotropic plate model gave results similar to those for the single element model. The stiffness was slightly reduced if hydrostatic forces were included.

Figure 4.4 Axial Load vs. Axial Shortening for Stiffened and Orthotropic Plates

4.2.2 Modeling Composite Structural Laminate Plates for the UDHSPS

The 40,000 DWT UDHSPS product oil tanker is composed of a number of composite structural laminate plates with plan dimensions of 14,000 mm (spacing between transverse bulkhead stools) by 2,000 mm or 3,000 mm, depending on the spacing between longitudinal girders. To simplify the analysis, the number of elements required for this typical component was reduced to a minimum of 4 elements along the length, as illustrated in Figure 4.5. To determine the effect of this coarse mesh in modeling the in-plane compressive behaviour of the composite structural laminate plates comparative analyses were conducted using mesh densities similar to those that gave stable solutions in Sections 3.4 and 3.6.

Figure 4.5 Fine and Coarse Mesh Composite Structural Laminate Plate Models

The composite structural laminate plates were modeled with SHELL91 elements, with constant thickness for the steel and the elastomer core of 10 mm and 50 mm, respectively. The elastomer material was defined by the Mooney-Rivlin parameters for elastomer C and the material properties for the steel were characterized by the true stress-strain data given in Table 3.4. The boundary conditions, loading and analysis were similar to those for the composite structural laminate plates studied in Sections 3.4 and 3.6.

The results are characterized by the axial load – axial shortening curves illustrated in Figure 4.6 for both mesh densities and for both longitudinal girder spacings. The buckling loads and the corresponding stresses (based on the area of the steel plate alone) are also shown. The coarse mesh models accurately predict the elastic behaviour and as anticipated, over predict the buckling loads. As long as the compressive stresses in the substructure hull module analysis are below 285 MPa and 295 MPa for the 3,000 mm and

2,000 mm longitudinal girder spacing, respectively, the coarse mesh model should give results that are representative of the global behaviour. Note that the buckling loads for the fine mesh models are approximately 5% greater than the corresponding models with plan dimensions equivalent to the CDH panels in Section 3.6.

Figure 4.6 Axial Load vs. Axial Shortening for 14,000 mm Long Composite Structural Laminate Plates

4.3 CDH Substructure Hull Module Analysis

Objective:	To determine longitudinal stresses for a midship section of a 40,000 DWT CDH product oil tanker subject to the design loads specified by Lloyd's Rules and Regulations (1994). These will be used to establish equivalency of the corresponding UDHSPS product oil tanker.
Analysis Type:	Elastic analysis
Element Type:	Elastic shell (SHELL63)

4.3.1 Problem Description

A substructure hull module, three cargo holds long, from the midship region of the conventional double hull product oil tanker designed according to Lloyd's Rules and Regulations (1994) in Section 1.2 and illustrated in Figure 4.7, was analyzed to determine the midship stresses when subjected to the corresponding specified design loads. The quarter section of the substructure hull module is 26,500 mm long, 16,000 mm wide, and 21,500 mm deep. The loading and boundary conditions are consistent with the symmetric model and those imposed by the surrounding tanker.

4.3.2 Modeling Notes and Analysis Assumptions

Element Type and Material Characteristics

The stiffened steel plates were modeled with orthotropic elastic shell (SHELL63) elements with the appropriate material properties aligned with the element coordinate system. These properties are listed in Table 4.3 for the stiffened plates identified by the strake numbers shown in Figure 4.8. The element thickness is equal to the corresponding plate thicknesses. The unstiffened steel plates were modeled with the same elastic shell and assigned isotropic properties.

Material Properties <ul style="list-style-type: none">All stiffened plates were modeled as orthotropic plates with the material properties listed in Table 4.3.	Model Properties <ul style="list-style-type: none">Number of Nodes = 534Number of Elements = 881
Loading <ul style="list-style-type: none">1st Load Case: Hydrostatic pressures2nd Load Case: Hogging/Sagging moment3rd Load Case: Shear Force	Analysis Options <ul style="list-style-type: none">NROFT, FULLNLGEOM, ONSSTIFF, OFFForce Convergence Tolerance = 5%

Figure 4.7 Problem Description and Representative Finite Element Model for the CDH Product Oil Tanker Substructure Hull Module Analysis

Table 4.3 Summary of Material Properties Data

Stake No	Suffener Group No.	Stake Scantling	Stiffener Scantling		Cross-sectional Area	Plate Area	Area Ratio	Orthotropic Material Data					
			d x t	W				A	Bt	I	E _x	E _y	MP ₀
		mm	mm	mm	mm ²	mm ²			mm ⁴	mm ²	MPa	MPa	MPa
1.1	SD1	12	250	19	59750	36000	1.660	344277	215126	207430			
1.2	SD1	12	250	19	59750	36000	1.660	344277	215126	207430			
1.3	SD1	11.5	250	19	58250	34500	1.688	350227	215332	207430			
1.4	SD1	11.5	250	19	58250	34500	1.688	350227	215332	207430			
1.5	SD2	11.5	250	23	40250	23000	1.750	363003	215752	207430			
2.6	SS6	12.5	300	20	63275	39275	1.611	334186	214762	207430			
2.6B	SS6	12.5	300	20	43000	25000	1.720	356780	215551	207430			
4.4	SS4	12.5	300	25	67500	37500	1.800	373375	216073	207430			
4.5	SS5	12.5	300	21	62700	37500	1.672	346824	215215	207430			
5.2	SS2	12.5	400	25	77500	37500	2.067	428690	217535	207430			
5.3	SS3	12.5	350	23	69700	37500	1.859	385544	216429	207430			
6.1	SS1	12.5	400	25	55000	25000	2.200	456347	218139	207430			
7.1	B1	13	400	25	80846	40846	1.979	410565	217098	207430			
7.2	B1	13	400	25	80846	40846	1.979	410565	217098	207430			
8.2	BS1	13.5	400	25	80500	40500	1.988	412300	217141	207430			
8.3	BS2	13	400	25	79000	39000	2.026	420180	217334	207430			
8.4	BS3	13	400	25	79000	39000	2.026	420180	217334	207430			
8.5	BS4	13	400	25	79000	39000	2.026	420180	217334	207430			
8.6	BS5	13	400	25	56000	26000	2.154	446773	217938	207430			
9.1	BS1	15	400	25	85000	45000	1.889	391813	216604	207430			

Table 4.3 Continued. Summary of Material Properties Data

Strake No.	Stiffener Group No.	Stake Scantling	Stiffener Scantling		Cross-sectional Area	Plate Area	Area Ratio	Orthotropic Material Data			
			d - t	w				E _x	E _y	E _z	
-	-	mm	mm	mm	mm ²	mm ²	-	MPa	MPa	MPa	
10.1	IB1	14	400	27	85200	42000	2.029	420787	217349	207430	
10.2	IB2	14	400	27	85200	42000	2.029	420787	217349	207430	
10.3	IB3	14	400	27	85200	42000	2.029	420787	217349	207430	
10.4	IB4	14	400	27	85200	42000	2.029	420787	217349	207430	
13.1	LB1	12.5	400	25	75350	35350	2.132	442147	217838	207430	
13.2	LB2	12.5	400	25	77500	37500	2.067	428690	217535	207430	
13.3	LB3	11.5	350	23	66700	34500	1.933	401032	216852	207430	
13.4	LB4	11	300	25	63000	33000	1.909	396004	216718	207430	
13.5	LB5	10.5	300	20	55500	31500	1.762	365473	215830	207430	
13.6	LB6	10	300	20	52280	28280	1.849	383468	216370	207430	
13.7	LB7	10	250	21	30500	20000	1.525	316331	214063	207430	
18.1	TB1	10	200	23	48400	30000	1.613	334654	214779	207430	
18.2	TB2	10	200	23	48400	30000	1.613	334654	214779	207430	
18.3	TB3	10.5	250	22	53500	31500	1.698	352302	215402	207430	
18.4	TB4	11.5	300	20	58500	34500	1.696	351730	215383	207430	
18.5	TB5	12.5	300	25	67500	37500	1.800	373375	216073	207430	
18.6	TB6	12.5	300	25	67500	37500	1.800	373375	216073	207430	
18.7	TB7	12.5	300	25	67500	37500	1.800	373375	216073	207430

Figure 4.8 Summary of Strake Locations

Geometry, Boundary Conditions and Finite Element Mesh

The overall geometry of the substructure hull module is illustrated in Figure 4.7. The longitudinal girder or stringer spacing and transverse floor or web spacing is 3,000 mm and 3,600 mm, respectively. The double plate transverse bulkhead was used for this model to give a configuration similar to the UDHSPS model in Section 4.4. Note that the gunwale and bilge areas were modeled with 5 elements located along the arc to capture the curved surface geometry. Local details representing joints and brackets were not modeled. The boundary conditions and finite element mesh are illustrated in Figure 4.7.

Loading

Load Case 1: Hydrostatic Pressure

The hydrostatic pressures from the seawater and crude oil cargo are illustrated on the midship section, shown in Figure 4.9. The maximum pressures are based on the density of seawater and crude oil of $1,025 \text{ kg/m}^3$ and 850 kg/m^3 ; and the draught line and height of oil of 12,000 and 17,500 mm, respectively. These surface levels do not coincide with element junction points and were adjusted accordingly, as shown in Figure 4.9.

Figure 4.9 Hydrostatic Pressure Application

Load Case 2: Hogging/Sagging Moment

The specified design hogging or sagging moments were applied to the nodes along the neutral axis at Section a-a, illustrated in Figure 4.10, in proportion to the ratio of the moment of inertia of the elements associated with that node to the moment of inertia of

the section. For example, the plates associated with node F are bold in Figure 4.10. The ratios of the moment of inertia for each node are listed in Table 4.4. Note, due to longitudinal symmetry, only half of the specified design moment was applied to the model.

Table 4.4 Ratio of the Moment Applied to Nodes along the Neutral Axis

Node	I	Ad^2	Moment of Inertia	Ratio $I/\Sigma I_i$
A	5.65511E+12	5.11603E+12	1.07711E+13	0.14474161
B	1.2358E+11	1.26632E+13	1.27868E+13	0.171828132
C	29205151656	1.29556E+13	1.29848E+13	0.1748864
D	8001488729	9.99673E+12	1.00047E+13	0.134442594
E	8001641359	1.04895E+13	1.04975E+13	0.141063809
F	8001523250	1.10443E+13	1.10523E+13	0.148519956
G	9334199880	6.30975E+12	6.31908E+12	0.084915259
			$\Sigma I_i = 7.44164E+13$	$\Sigma(I/\Sigma I_i) = 1.0$

$$M_z = -1472396.8 \times 10^6 \text{ N.mm (Hogging)}$$

$$M_z = 1554449.9 \times 10^6 \text{ N.mm (Sagging)}$$

Figure 4.10 Neutral Axis Along Section a-a

Load Case 3: Shear Force

The specified design shear force was applied to the nodes on the plane of longitudinal symmetry, which were coupled for the Y-displacements. The coupled nodes are associated with the elements highlighted by the bold lines in Figure 4.11. The shear force for hogging (20,125 kN) and sagging (20,795 kN) conditions were applied in the positive and negative Y-directions to the node indicated. Due to transverse symmetry, only half of the specified design shear force was applied.

Figure 4.11 Shear Force Application

4.3.3 Results and Discussion

The element longitudinal stresses for the load cases, hydrostatic force, moment, and shears, associated with hogging and sagging are illustrated on isometric, side shell, strength deck, and outer bottom views in Figure 4.12 and 4.13, respectively. These

stresses are applicable for the orthotropic plates and must be adjusted by a factor $1/r$ to obtain the in-plane longitudinal stress in the hull plates. The r -values are summarized for each strake in Figure 4.14. The element longitudinal stresses for both longitudinal edges of stiffened plates 1 to 5 (refer to Section 3.5) from this elastic analysis for both hogging and sagging are listed in Table 4.5. The corresponding adjusted hull plate stress values are also given. Note, the negative and positive signs indicate compressive and tensile stresses, respectively. Furthermore, although the magnitude of these stresses is relatively small, fatigue problems related to the construction details still occur.

Based on the stresses illustrated in Figure 4.12 and 4.13, and the sample set of adjusted stresses listed in Table 4.5, for typical stiffened plates around the hull, it can be concluded that all the stresses are elastic and are well within the range applicable to the use of orthotropic plate elements. The single orthotropic plate element gives approximately a 1.13 times stiffer elastic response than that of the stiffened plate with which it was compared, as illustrated in Figure 4.4. It is anticipated that this simplified substructure hull module will be slightly stiffer, thus under load, the deflections and stresses would be smaller than in a more detailed model. The midspan deflection and rotation at Section a-a for the hogging and sagging moments are 17.5 mm and 702×10^{-6} radians; and -18.5 mm and 776×10^{-6} radians, respectively.

Figure 4.12 CDH Element Longitudinal Stresses (in MPa) for Hogging Condition

Figure 4.13 CDH Element Longitudinal Stresses (in MPa) for Sagging Condition

Bold text/lines are plates that run longitudinally in the hull envelope.

Plain text/lines are plates in transverse bulkhead and stools.

Figure 4.14 Location of Correction Factor, r

Table 4.5 Longitudinal Stresses in Five Stiffened Panels

			Longitudinal Stresses for Hogging Condition, MPa		Longitudinal Stresses for Sagging Condition, MPa	
Plate No.	Correction Factor, r	Stress Category	Left/Top Edge	Right/Bottom Edge	Left/Top Edge	Right/Bottom Edge
1	2.029	FEA	-107.560	-99.597	126.160	119.015
		Adjusted	-53.011	-49.086	62.178	58.657
2	2.026	FEA	-56.723	-49.266	58.770	49.794
		Adjusted	-27.998	-24.317	29.008	24.577
3	2.067	FEA	-11.232	-66.412	40.551	90.921
		Adjusted	-5.434	-32.130	19.618	43.987
4	1.933	FEA	25.635	-17.505	-28.532	17.154
		Adjusted	13.262	-9.056	-14.760	8.874
5	1.660	FEA	167.46	167.735	-169.51	-191.63
		Adjusted	100.880	102.114	-102.114	-115.440

4.4 UDHSPS Substructure Hull Module Analysis

Objective:	To determine longitudinal stresses for two typical midship sections of a 40,000 DWT UDHSPS product oil tanker subject to design loads specified by Lloyd's Rules and Regulations (1994). These will be used to establish equivalency with the corresponding CDH product oil tanker.
Analysis Type:	Static analysis
Element Type:	Structural shell (SHELL93) 16-Layer structural shell (SHELL91)

4.4.1 Problem Description

Substructure hull modules from the midship region of two proposed UDHSPS product oil tanker configurations, illustrated in Figure 4.15 and 4.16, were analyzed to determine the midship stresses when subjected to the specified design loads given by Lloyd's Rules and Regulations (1994). The quarter section size, loading, and boundary conditions are identical to the CDH substructure hull module, described in Section 4.3.

4.4.2 Modeling Notes and Analysis Assumptions

Element Type and Material Characteristics

The structural composite laminate was modeled with SHELL91 elements with constant thickness for the steel and elastomer of 10 mm and 50 mm, respectively. The unstiffened steel plates were modeled with SHELL93 elements. The elastomer was defined with the Mooney-Rivlin parameters for elastomer C, and the material properties for the steel were characterized by the stress-strain data given in Table 3.4.

Geometry, Boundary Conditions and Finite Element Mesh

The first configuration, shown in Figure 4.15 and 4.17, is referred to as the UDHSPS-3GS (3GS indicates a 3,000 mm longitudinal girder spacing). It has no transverse floors or webs between transverse bulkheads and the hull, side shells, deck, deck girders, and double-walled transverse bulkheads (shown as white elements in Figure 4.15) are composed of composite structural laminate plates. The deck girders are extended from the cargo tank into the gunwale area within the double walled side shell. The longitudinal girders (12 mm thick) and side stringers (10 mm thick) are unstiffened steel plates (shown as gray elements in Figure 4.15) and span 14,400 mm between transverse frames. Additional side stringers and longitudinal girders were used to frame the bilge and gunwale areas, to compensate for the elimination of transverse frames.

The second configuration, shown in Figure 4.16, is referred to as the UDHSPS-2GS (2GS indicates a 2,000 mm longitudinal girder spacing). The geometry is essentially the same, with the exceptions that the deck girders now run longitudinally with a 2,000 mm spacing, and all longitudinal girders and side stringers are 10 mm thick.

The boundary conditions are identical to those used in the previous analysis. The finite element meshes for each configuration are illustrated in Figures 4.15 and 4.16.

Material Properties <ul style="list-style-type: none"> • White elements are composite structural laminate plates constructed of two 10 mm steel plates filled with 50 mm polyurethane elastomer core type C. • Gray elements are steel plates. • Non-linear material properties for the steel are defined in Table 3.4. 	Model Properties <ul style="list-style-type: none"> • Number of Nodes = 1732 • Number of Elements = 821
Loading <ul style="list-style-type: none"> • 1st Load Case: Hydrostatic pressures • 2nd Load Case: Hogging/Sagging moment • 3rd Load Case: Shear Force 	Analysis Options <ul style="list-style-type: none"> • NROPT, FULL • NLGEOM, ON • SSTIFF, ON • Force Convergence Tolerance = 5%

Figure 4.15 Problem Description and Representative Finite Element Model for the UDHSPS-3GS Product Oil Tanker Substructure Hull Module Analysis

Material Properties <ul style="list-style-type: none"> • White elements are composite structural laminate plates constructed of two 10 mm steel plates filled with 50 mm polyurethane elastomer core type C. • Gray elements are steel plates. • Non-linear material properties for the steel are defined in Table 3.4. 	Model Properties <ul style="list-style-type: none"> • Number of Nodes = 2299 • Number of Elements = 1083
Loading <ul style="list-style-type: none"> • 1st Load Case: Hydrostatic pressures • 2nd Load Case: Hogging/Sagging moment • 3rd Load Case: Shear Force 	Analysis Options <ul style="list-style-type: none"> • NROPT, FULL • NLGEOM, ON • SSTIFF, ON • Force Convergence Tolerance = 5%

Figure 4.16 Problem Description and Representative Finite Element Model for the UDHSPS-2GS Product Oil Tanker Substructure Hull Module Analysis

Figure 4.17 UDHSPS-3GS Hull Envelope Modification and Transverse Framing Location

Loading

The load was applied to the substructure hull modules in the manner previously described. The distribution of hogging and sagging moments was applied to the nodes along the neutral axis in proportion to the relative moments of inertia associated with any given node as listed in Table 4.6(a) and 4.6(b) for the two configurations.

Table 4.6(a) Ratio of Moment Applied to Nodes along the Neutral Axis For UDHSPS-3GS Substructure Hull Module

Node	Ratio $I/\Sigma I_i$
A	0.133132212
B	0.224843264
C	0.182946619
D	0.123693275
E	0.12823364
F	0.132899535
G	0.074251455
$\Sigma(I/\Sigma I_i) = 1.0$

Table 4.6(b) Ratio of Moment Applied to Nodes along the Neutral Axis For UDHSPS-2GS Substructure Hull Module

Node	Ratio $I_i/\Sigma I_i$
A	0.081665411
B	0.114154257
C	0.110956038
D	0.107821151
E	0.104749595
F	0.101741369
G	0.124669978
H	0.155036122
I	0.099206079
$\Sigma(I_i/\Sigma I_i) = 1.0$

4.4.3 Results and Discussion

The element longitudinal stresses for the load cases, hydrostatic force, moment, and shears, associated with hogging and sagging are illustrated on isometric, side shell, strength deck, and outer bottom views, for the UDHSPS-3GS and UDHSPS-2GS, in Figure 4.18 to 4.21. A summary of the average longitudinal stresses for the elements associated with plates 1 and 5 of the conventional double hull model (typical outer bottom and strength deck plates), for hogging and sagging moments are given in Table 4.7. The corresponding midship displacements and rotations at Section a-a are given in Table 4.8. Based on the magnitude of the stresses, illustrated in Figures 4.18 to 4.21, it can be concluded that all the stresses are elastic and are well within the range applicable to the use of a coarse mesh in modeling the in-plane behaviour of composite structural laminate plates.

Figure 4.18 UDHSPS-3GS Element Longitudinal Stresses (in MPa) for Hogging Condition

Figure 4.19 UDHSPS-3GS Element Longitudinal Stresses (in MPa) For Sagging Condition

Figure 4.20 UDHSPS-2GS Element Longitudinal Stresses (in MPa) for Hogging Condition

Figure 4.21 UDHSPS-2GS Element Longitudinal Stresses (in MPa) for Sagging Condition

Table 4.7 Average Longitudinal Element Stresses

Substructure Hull Module	Average Longitudinal Element Stresses for Hogging Moment Condition, MPa		Average Longitudinal Element Stresses for Sagging Moment Condition, MPa	
	Plate 1	Plate 5	Plate 1	Plate 5
CDH	-51.0	101.5	60.4	-109.3
UDHSPS-3GS	-52.0	96.4	81.4	-100.6
UDHSPS-2GS	-56.5	72.0	81.6	-73.1

Table 4.8 Midship Displacement and Rotation at Section a-a

Substructure Hull Module	Hogging Moment Condition		Sagging Moment Condition	
	Midship Deflection, mm	Rotation at Section a-a, $\times 10^6$ radians	Midship Deflection, mm	Rotation at Section a-a, $\times 10^6$ radians
CDH	17.5	-701.8	-18.5	776.0
UDHSPS-3GS	17.1	-807.2	-17.2	863.0
UDHSPS-2GS	14.1	-708.4	-15.3	718.5

Based on the limited information produced by the three hull module analyses, it can be concluded that the relative stiffnesses (midship deflections and rotations at Section a-a) of the substructure hull modules for the CDH and the two variants of the UDHSPS are similar. The magnitude and distribution of stresses are also similar. Essentially, the two types of substructure hull modules give equivalent global performance under normal operating loads.

Parametric studies have not yet been conducted on the UDHSPS substructure hull module; hence, it is difficult to make specific conclusions related to the variations between the two variants presented. Optimizing the design of the UDHSPS by varying plate sizes and sandwich core thicknesses may lead to improved performance over the CDH system for normal operating loads.

Chapter 5

Summary, Conclusions, Recommendations, and Future Work

5.1 Summary

Tanker accidents and oil spills continue to occur even with improvements in navigation systems (automation, global positioning satellite technology), redundancy of propulsion systems, and the use of highly skilled pilots. As a result, legislation was introduced around the world mandating the use of double hull technology in an attempt to mitigate or eliminate oil spills. Current double hull technology includes two main variants: a Conventional Double Hull (CDH) which is based on traditional naval architecture and is characterized by a complex series of longitudinally and transversely framed elements that separate the stiffened plates of the inner and outer hulls by a distance of two metres; and an Advanced Double Hull (ADH) which has a simplified structural layout that incorporates only longitudinal framing and transverse bulkheads. These structural forms only offer a collision gap in which the structure can collapse and objects can penetrate without directly breaching the inner hull.

The CDH and ADH all-steel double hull product oil tanker designs have a number of structural problems which make it virtually impossible to realize the objective of zero oil outflow for accidental or extreme load events. Experimental evidence from large-scale grounding tests have demonstrated that at least one cargo tank will be breached due to the propagation of cracks from the ruptured outer hull to the inner hull, through the

supporting elements, regardless of the all-steel double hull design alternative, Rodd (1996). Due to structural complexity and a large number of connections, the CDH product oil tankers are susceptible to fatigue and corrosion problems, Daidola (1995), which accelerate deterioration, decrease service life, and lead to breaching of the inner hull through the propagation of fatigue cracks. Both the conventional and advanced double hull alternatives are composed of stiffened plates, which are prone to rupture at locations of concentrated strains (related to sharp curvature due to bending about the hull plate-longitudinal stiffener, longitudinal girder, transverse web, or transverse floor junctions) for low energy impacts, Motoi et al. (1995). Numerical simulations have demonstrated that to increase the energy absorption capacity the structure must be simplified (removal of longitudinal stiffeners and transverse frames) and the plates dimensioned to provide flexible hulls and collapsible longitudinal framing elements that can obtain membrane action over multiple support elements prior to localized rupture, Kennedy (1987), Bond (1995), Kitamura (1996), and Sano et al. (1996). A limiting characteristic in achieving these simplified structures is the buckling capacity of the hull plates.

Clearly, there exists a need for the development of an innovative, practical, and cost effective double hull design which provides equivalent structural performance for normal operating loads, superior energy absorption capacity, and mitigates oil outflow during accidental or extreme load events. The Unidirectional Double Hull Sandwich Plate System (UDHSPS) was developed to meet these objectives. This system integrates composite structural laminate plates (steel-polyurethane elastomer-steel) into the hulls, deck, and transverse bulkheads of the ADH design concept. The UDHSPS design eliminates the longitudinal stiffeners of the inner and outer hull plates, providing a

simplified structural system with flat surfaces. The structural system improves hull integrity by maximizing the energy absorption potential of the plating during an extreme load event, while offering in-plane load capacity equivalent to stiffened panels during normal operating loads and sufficient residual strength for rescue operations. The design reduces the number of structural intersections characteristic of the CDH and ADH designs, thereby decreasing the number of fatigue and corrosion prone details. A portion of the steel, previously used for the longitudinal stiffeners, is reallocated to the hull plating to increase the energy absorption characteristics of the plating. Hard points at longitudinal girder-transverse floor and hull plate-stiffener intersections are diminished due to the polyurethane core, which dissipates the localized strains over a larger field. The core and appropriate construction and fatigue detailing provide an effective crack arrest barrier to guard against crack propagation into the inner hull. The flat unobstructed surfaces of this system lend themselves to better coating applications and less corrosion related problems. The elimination of stiffeners and the reduction of structural intersections give increased accessibility, which leads to ease of inspection and maintenance.

The numerical analyses presented in this thesis complement the experimental investigation conducted by Bennett (1998), and demonstrate the use of finite elements in modeling composite structural laminate plate behaviour for use in double hull construction (UDHSPS). The interrelationship of these analyses is illustrated in Figure 5.1 and is summarized as follows:

- Nonlinear analyses of tension coupon and compression disk tests on polyurethane elastomer material were conducted to verify the use of the Mooney-Rivlin constitutive law.

- A series of nonlinear buckling analyses on composite structural laminate plates with varying mesh densities and polyurethane elastomer material properties was conducted to obtain a global buckling mode of failure and consistent buckling solutions.
- A series of nonlinear buckling analyses of typical stiffened plates from a CDH product oil tanker was conducted to establish in-plane capacities and stiffnesses.
- A nonlinear analysis of a UDHSPS product oil tanker substructure, subjected to transverse loads, was conducted to confirm the success of finite element analysis in predicting the behaviour of the structure by making a comparison of the results from this simulation with those obtained by Bennett (1998) from experimental studies on the corresponding large-scale test specimen.
- Analyses were conducted to establish that coarsely meshed composite structural laminate and orthotropic plates could be used to give elastic in-plane behaviour comparable to that obtained from detailed models, so that simplified substructure hull modules could be developed.
- A linear elastic analysis was conducted on a CDH substructure hull module, designed according to Lloyd's Rules and Regulations (1994), to determine the global behaviour associated with the specified design loads.
- Nonlinear analyses were conducted on two different UDHSPS substructure hull modules, loaded with the same specified design loads, to establish global behaviour equivalent to that of the CDH product oil tanker.

Figure 5.1 Summary of Thesis Components

5.2 Conclusions

Based on the numerical investigations presented, it can be concluded that:

- The Mooney-Rivlin finite element constitutive model is effective in representing the behaviour of the elastomer in both tension and compression.
- The minimum modulus of elasticity of the polyurethane elastomer core required to obtain global buckling of the composite structural laminate plate was 273 MPa.
- Parametric studies on composite structural laminate plates in which the steel plate and core thicknesses were varied demonstrated that strengths and stiffnesses comparable to the stiffened plates from a CDH product oil tanker could be obtained.
- A comparison of the results of a nonlinear finite element analysis of a model of a UDHSPS product oil tanker, subject to transverse loads, with the corresponding test data demonstrated that the numerical simulation was able to accurately predict the complex behaviour (first yield, transverse floor buckling, longitudinal girder buckling, development of membrane action) and the interaction of composite structural laminate plates with the supporting structural elements.
- The in-plane elastic behaviour of a stiffened plate, with plan dimensions of 3,000 mm by 3,600 mm, can be represented by a single orthotropic plate element.
- The elastic in-plane behaviour of a composite structural laminate plate is well-represented with a very coarse mesh with one of the plan dimensions set to the spacing of the supporting elements and the other to give aspect ratios close to 1.0.

- A linear elastic analysis conducted on a CDH substructure hull module to determine the global behaviour associated with the specified design loads gave maximum midship displacements and compressive and tensile longitudinal stresses for hogging and sagging moments of 17.5 mm and –18.5 mm, and 102 MPa and –109 MPa, respectively. Under these loads, the stiffened plates behaved elastically and were within the range applicable to the use of orthotropic plate elements for this modeling purpose.
- The nonlinear analysis conducted on two different UDHSPS substructure hull modules to determine the global behaviour associated with the specified design loads gave maximum midship displacements and compressive and tensile longitudinal stresses for hogging and sagging moments that were equivalent to the corresponding values for the CDH product oil tanker. Note that the stresses were elastic and well within the range applicable to the use of a coarse mesh in modeling the in-plane behaviour.

Based on these conclusions it is expected that the UDHSPS concept could be applied to product oil tankers of all sizes.

5.3 Recommendations and Future Work

For the UDHSPS technology to be readily accepted by ship owners and ship builders, it must be approved by classification societies. These societies stipulate that experimental testing and numerical simulations must be conducted to prove that the UDHSPS design is structurally equivalent to structures designed according to the rules

and regulations. Design equations, design philosophies, and methods for evaluating the energy absorption capacity and residual strengths require further development. With respect to numerical simulations, the following series of analyses needs to be conducted:

- to determine the energy absorption capacity and the extent of structural damage for UDHSPS hull modules subject to collision and grounding load events. Models similar to those cited in the thesis and as illustrated in Figure 5.2 could be used. Note that the core material may be strain-rate sensitive, and like other materials, the strengths increase and the ductility decreases with increasing strain rates. Little or no data is available regarding this effect and the dynamic material properties need to be established, Vinson (1995).
- to evaluate the residual strength of a UDHSPS product oil tanker for post-accidental or extreme load events.
- to determine the fatigue capacity of the structural details required for the UDHSPS product oil tanker.
- to determine the structural behaviour of the product oil tanker for fire events and for thermal creep.
- to optimize the configuration, scantlings, and core material properties.

Figure 5.2 Collision Analysis Model

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Appendix A

Design Computations for the Midship of a 40,000 DWT Conventional Double Hull Oil Tanker

A.1 General

The calculations presented in this appendix serve as a verification of the computations from the LR.PASS program used to design the midship of the conventional double hull oil tanker structure. The results of the subsequent computations, longitudinal strength requirements, and shear force calculations are summarized in Chapter 1, Section 1.2.4.

A.2 Strake Scantlings

The following section will present the rules criteria used to determine the strake scantlings. This is step 4 of the iterative solution procedure of the LR.PASS software as described in Chapter 1, Section 1.2. Table A.1 summarizes the rule references that apply to the appropriate sections of the hull envelope. The strake locations are illustrated on the sectional views of the tanker shown in Figure 1.1.

Table A.1 Applicable Rules Required for CDH Tanker Design

Item No.	Item	Number of Strakes	Applicable Rules in Lloyd's Rules and Regulations (1994)
1	Deck	5	P4 Ch.9 T9.4.1(1) (Buckling) P4 Ch.9 10.2.1 (Comp. Min) P3 Ch.3 T3.2.1 (End t.) P3 Ch.4 T4.7.2(a)* P3 Ch.4 T4.7.2(c)*
2	Sheerstrake and gunwale	1	P4 Ch.9 T9.4.1(2) (Buckling) P4 Ch.9 T9.4.1(2) (Pressure) P4 Ch.9 10.2.1 (Comp. Min) P3 Ch.3 T3.2.1 (End t.) P3 Ch.4 T4.7.2(a)* P3 Ch.4 T4.7.2(c)*
3, 4, 5, and 6	Side shell	5	P4 Ch.9 T9.4.1(5/6) & N2 (Buckling) P4 Ch.9 T9.4.1(5/6) & N2 (Pressure) P4 Ch.9 T9.4.1(5/6) & N7 (Buckling) P4 Ch.9 T9.4.1(5/6) (Pressure) P4 Ch.9 10.2.1 (Comp. Min) P3 Ch.3 T3.2.1 (End t.) P3 Ch.4 T4.7.2(a)* P3 Ch.4 T4.7.2(c)*
7	Bilge	2	P4 Ch.9 T9.4.1(7/8) (Buckling) P4 Ch.9 T9.4.1(7/8) (Pressure) P4 Ch.9 10.2.1 (Comp. Min) P3 Ch.3 T3.2.1 (End t.)
8	Bottom shell	5	P4 Ch.9 T9.4.1(7/8) (Buckling) P4 Ch.9 T9.4.1(7/8) (Pressure) P4 Ch.9 10.2.1 (Comp. Min) P3 Ch.3 T3.2.1 (End t.) P3 Ch.4 T4.7.2(a)* P3 Ch.4 T4.7.2(c)*

* These rules were not verified since the iterative routines and assumptions used in the program replicating the critical buckling calculations were not known.

Table A.1 Continued. Applicable Rules Required for CDH Tanker Design

Item No.	Item	Number of Strakes	Applicable Rules in Lloyd's Rules and Regulations (1994)
10	Inner bottom	4	P4 Ch.9 6.2.1 (tm) P4 Ch.9 6.2.1 (tl) P4 Ch.9 T9.6.1(1) (b) P4 Ch.9 T9.6.1(1) (d) P4 Ch.9 10.2.1 (Comp. Min.) P4 Ch.1 T1.9.1 P3 Ch.4 T4.7.2(a)* P3 Ch.4 T4.7.2(c)*
11	Bottom centreline girder	1	P4 Ch.9 9.3.4(a)
12	Side girders	4	P4 Ch.9 9.3.4(b)
13	Longitudinal bulkhead	7	P4 Ch.9 6.2.1 (tm) P4 Ch.9 6.2.1 (tl) P4 Ch.9 T9.6.1(1) (b) P4 Ch.9 T9.6.1(1) (d) P4 Ch.9 10.2.1 (Comp. Min.) P3 Ch.4 T4.7.2(a)* P3 Ch.4 T4.7.2(c)*
14	Floors	-	P4 Ch.9 9.3.4 (b)
15	Deck transverses	-	P4 Ch.10 2.8
16	Vertically corrugated transverse bulkhead	8	P4 Ch.9 T9.7.1 P4 Ch.1 T1.9.2 P4 Ch.9 10.2.1 (Comp. Min.)
17	Plane transverse bulkhead	8	P4 Ch.9 T9.7.1 P4 Ch.9 10.2.1 (Comp. Min.)
18	Double Plate Bulkhead	8	P4 Ch.9 7.4.2 P4 Ch.9 T1.9.1 P4 Ch.9 T1.9.2 P4 Ch.9 10.2.1 (Comp. Min.)

- These rules were not verified since the iterative routines and assumptions used in the program replicating the critical buckling calculations were not known.

The strakes, illustrated in Figure A.1, are identified by numbers, with the item number (deck, sheerstrake, etc.) before the decimal and the assigned plate number after the decimal. Strake seams are indicated with a circle.

Figure A.1 Strake Locations and Numbers

A.2.1 Hull Envelope Plating

LR Table 9.4.1 of Part 4, Chapter 9.4 was used to determine the hull envelope plating minimum thickness for items 1 through 9, for tankers framed with both longitudinal and transverse stiffeners. Table A.2 lists the formulae for minimum hull envelope plating thickness with longitudinal stiffeners.

Table A.2 Minimum Hull Envelope Plating Thickness - LR P4 Ch.9 T9.4.1

Item No.	Item	Hull envelope plating - minimum thickness formulae	Strake No.	Plating - minimum thickness (mm)
1	Deck	$t = \frac{s}{J} + 2.0$	1.1	11.29
			1.2	11.29
			1.3	11.29
			1.4	11.29
			1.5	11.29
2	Sheerstrake and gunwale	the greater of the following: $t = \frac{s}{J} + 2.0 = 11.289 \text{ mm}$ $t = 0.0042s\sqrt{h_{T1}k} = 11.877 \text{ mm}$	2.6	11.88
3 and 4	Side shell above mid-depth	the greater of the following: $t = 0.001s(0.059L_1 + 7)\sqrt{\frac{F_D}{k_L}} = 10.401 \text{ mm}$ $t = 0.0042s\sqrt{h_{T1}k} = 11.877 \text{ mm}$	4.4	11.88
			4.5	11.88
5 and 6	Side shell below mid-depth	$t = 0.001s(0.059L_1 + 7)\sqrt{\frac{F_B}{k_L}} = 9.913 \text{ mm}$ But cannot be less than the following (Intermediate thickness by interpolation): (a) $t = 0.0042s\sqrt{h_{T1}k} = 11.877 \text{ mm}$ (b) $t = 0.0054s\sqrt{\frac{h_{T2}k}{2 - F_B}} = 12.438 \text{ mm}$ at upper turn of bilge (see Notes 1 and 2): Note 2: $t = \frac{s}{J} + 2.0 = 10.806 \text{ mm}$ $t = 0.0052s\sqrt{\frac{h_{T2}k}{1.8 - F_B}} = 12.994 \text{ mm}$	5.2	12.99
			5.3	12.99
7 and 8	Bilge and bottom shell	the greater of the following: $t = \frac{s}{J} + 2.0 = 10.806 \text{ mm}$ $t = 0.0052s\sqrt{\frac{h_{T2}k}{1.8 - F_B}} = 12.994 \text{ mm}$	7.1	12.99
			8.2	12.99
			8.3	12.99
			8.4	12.99
			8.5	12.99
			8.6	12.99
9	Keel	As for previous item, +2 mm, but need not exceed: $25\sqrt{k} = 25 \text{ mm}$	9.1	14.99

The scantlings calculated in Table A.2 use the following terms, which are defined as follows:

s = spacing, in mm, of longitudinals, but should not be taken less than:

$$\text{the lesser of } \begin{cases} s = 470 + \frac{L}{0.6} = 761 \text{ mm} \\ s = 700 \text{ mm (governs)} \end{cases}$$

C_w = a wave head, equivalent to $7.71 \times 10^{-2} L e^{-0.0044L}$ and has a value of 6.244 m

R_B = 2000 mm, bilge radius

T_1 = $T = 12.0$ m draught, which is greater than $0.05L = 9.0$ m

F_B = 0.67, scantling reduction factor for plates below the mid-depth of the tanker

F_D = 0.7375, scantling reduction factor for plates above the mid-depth of the tanker

For items 1 through 4 above mid-depth of the tanker,

σ = 175 MPa , permissible stress

σ_c = $F_D \times \sigma = 0.7375 \times 175.0 = 129.063$ MPa, maximum compressive hull vertical bending stress

σ_0 = 235.0 MPa, specified minimum yield stress

α = $\frac{\sigma_0}{\sigma_c}$ and has a value of 1.8208

$$J = 1720.5 \sqrt{\frac{1 - \frac{1}{\alpha}}{\sigma_0}} \text{ for } \alpha \leq 2 \text{ and has a value of } 75.355$$

For items 5 through 9 below mid-depth of the tanker,

σ = 175 MPa , permissible stress

σ_c = $F_B \times \sigma = 0.67 \times 175.0 = 117.25$ MPa, maximum compressive hull vertical bending stress

σ_0 = 235.0 MPa, specified minimum yield stress

α = $\frac{\sigma_0}{\sigma_c}$ and has a value of 2.004

h_{T1} = $T + C_w = 18.244$ m, but need not be taken greater than $1.36T = 16.32$ m

h_{T2} = $T + 0.5C_w = 15.122$ m, but need not be taken greater than $1.2T = 14.4$ m

$$J = 860.7 \sqrt{\frac{\alpha}{\sigma_0}} \text{ for } \alpha > 2 \text{ and has a value of } 79.487$$

Part 4, Chapter 9.10.2.1 of Lloyd's Rules and Regulations (1994) states that within the cargo tank region, including wing ballast tanks and cofferdams at the ends of or between cargo tanks, the thickness of the primary member webs and face plates, the hull envelope and bulkhead plating is to be not less than the greater of the following:

$$t = 2.15L^{0.3}, \text{ which has a value of } 10.117 \text{ mm (governs)} \quad (\text{A.1a})$$

$$t = 7.5 \text{ mm} \quad (\text{A.1b})$$

This applies to all the strakes associated with the items in Table A.1 excluding the transverse and longitudinal framing items (11,12,14, and 15).

The plating thickness for the strake scantlings of items 1 through 9 must be greater than the basic shell end thickness for taper given in LR Part 3, Chapter 3, Table 3.2.1. Since the design focuses solely on the midship section of the vessel between 0.3L and 0.7L, only that portion of the table is given in Table A.3.

Table A.3 Taper Requirements for Hull Envelope - LR P4 Ch.3 T3.2.1

Item	Requirement
Shell envelope plating	The thickness, in mm, is to be the following: (b) $t_t = (6.5 + 0.033L) \sqrt{\frac{ks_1}{s_b}}$ and has a value of 12.626 mm
Strength deck plating	The thickness, in mm, is to be the following: (b) $t_t = (5.5 + 0.02L) \sqrt{\frac{ks_1}{s_b}}$ and has a value of 8.992 mm

Note, $s_b = 3600$ mm, the standard frame spacing
 $s_1 = s = 700$, but is to be taken not less than s_b , therefore $s_1 = 3600$ mm

The hull buckling strength is checked for strake scantlings of items 1 to 10 and 13, and the scantling revised if required according to LR Part 3, Chapter 4, Table 4.7.2, of which only subsections (a) and (c) are applicable to the baseline ship and are reproduced in Table A.4. The computed values, given by LR.PASS, indicate that the resulting strake

scantlings calculated using these expressions govern only for the strength deck plating, thus only those relevant data will be presented.

Table A.4 Elastic Critical Buckling Stress of Plating - LR P3, Ch.4 T4.7.2

Mode	Elastic critical buckling stress, N/mm ²	Strake No.	Plate Thickness (mm)
(a) Compression of plating with longitudinal stiffeners	$\sigma_E = 3.6E \left(\frac{t_p}{s} \right)^2$ $t_p = s \sqrt{\frac{\sigma_E}{3.6E}}$	1.1	12.00
		1.2	11.80
		1.3	11.60
		1.4	11.40
		1.5	11.30
(c) Shear	$\tau_E = 3.6 \left[1.335 + \left(\frac{s}{1000S} \right)^2 \right] E \left(\frac{t_p}{s} \right)^2$ $t_p = s \sqrt{\frac{\tau_E}{3.6E \left[1.335 + \left(\frac{s}{1000S} \right)^2 \right]}}$	1.1	11.30
		1.2	11.30
		1.3	11.30
		1.4	11.30
		1.5	11.30

The scantlings calculated in Table A.4, use the following additional terms which are defined as follows:

- t_p = As-built thickness of plating less standard deduction d_t such that $t_p = t - d_t = 0.9t$
- d_t = 0.10t, standard deduction for corrosion
- E = 206 000 MPa, modulus of elasticity
- s = 600 mm, spacing of secondary members
- S = 3.6 m, spacing of primary members
- σ_E = Elastic critical buckling stress in compression, in MPa
- τ_E = Elastic critical buckling stress in shear, in MPa

The inner bottom (Item 10) and the longitudinal bulkhead (Item 13) are both designed according to LR Part 4, Chapter 9, Table 9.6.1. The appropriate sections of that table have been reproduced in Table A.5.

Table A.5 Inner Hull and Longitudinal Oiltight Bulkhead Scantlings
– LR P4, Ch.9, Table 9.6.1 –

Plating thicknesses of bulkhead		Load Height h (m)	h ₁	t ₀	Strake No.	Strake Scantlings (mm)
(a) Within 0.1D of the deck t = t ₀		2.833	8.996	8.998	13.7	9.34
(b) Within 0.1D of the bottom shell: $t = \frac{t_0}{\sqrt{2 - F_B}}$ (but not less than t ₁)	Longitudinal Bulkhead	18.833	20.833	13.693 t _m = 9.825 t ₁ = 12.630	13.1	12.63
	Inner Bottom	19.500	21.500	13.910 (= t _m = t ₁)	10.1	13.91
					10.2	13.91
					10.3	13.91
					10.4	13.91
(c) Elsewhere t = t ₁ (take the minimum value, t _m , and then determine value of t ₁)		16.500 13.500 10.500 7.500 4.833	19.415 17.255 15.095 12.935 10.725	13.219 12.462 11.656 10.790 9.825 (= t _m)	13.2 13.3 13.4 13.5 13.6	12.73 11.71 11.12 10.50 9.83
(d) But not less than t = 0.0009s(0.059L ₁ + 7)					13.7	9.34

The scantlings calculated in Table A.5 use the following additional terms, which are defined as follows:

$$t_o = 0.005s\sqrt{kh_1}$$

$$h_1 = \left(h + \frac{D_1}{8} \right)$$

h = Load height measured as in P4, Ch.9, Section 9.6.2

$D_1 = D = 20$ m, but is to be taken not less than 110 and need not be taken greater than 16. Thus $D_1 = 16$ m.

$$t_1 = t_o \left[0.84 + 0.16 \left(\frac{t_m}{t_o} \right)^2 \right]$$

t_m = minimum value of t_o within 0.4D each side of mid-depth of bulkhead

s = 600 mm, spacing of stiffeners

The strake scantling of plate 10.1 of the inner bottom must also be designed according to Part 4, Chapter 11, Table 1.9.1. The thickness of strake 10.1 is calculated according to Equation A.2.

$$t = 0.004sf\sqrt{\frac{\rho h_4 k}{1.025}} + 2.5 \text{ and has a value of 13.098 mm} \quad (\text{A.2})$$

where, $\rho = 1.025$, relative density (specific gravity) of liquid cargo

$$f = 1.1 - \frac{s}{2500S}, \text{ which has a value of 1.033, but must be greater than 1.0.}$$

Therefore, $f = 1.0$.

$h_4 = 19.50 \text{ m}$, load head

The thickness of floors (Item 14) and girders (Items 11 and 12) must conform to the requirements of LR Part 4, Chapter 9, Section 9.3.4 (a) and (b).

Centre girder:

$$t = (0.008d_{DB} + 1.0)\sqrt{k} \text{ and has a value of 13.85 mm} \quad (\text{A.3})$$

Floors and side girders:

$$t = (0.007d_{DB} + 1.0)\sqrt{k} \text{ which has a value of 12.24 mm, but need not exceed} \quad (\text{A.4a})$$

$$t = 12.0\sqrt{k} \text{ and has a value of 12.0 mm (governs)} \quad (\text{A.4b})$$

where, $d_{DB} = 28B + 205\sqrt{T} = 1606.14 \text{ mm}$, is the Rule depth of the centre girder.

A.3 Transverse Elements

The three different types of transverse bulkheads that were designed include a bulkhead consisting of a single stiffened plate, a vertically corrugated bulkhead supported by stools, and a double plate bulkhead supported by stools, as illustrated in Figure A.2. A.3, and A.4, respectively. The designs of these components are given in Section A.3.1, A.3.2, and A.3.3.

Figure A.2 Plane Transverse Bulkhead

Figure A.3 Corrugated Bulkhead Supported by Stool Structure

Figure A.4 Double Plate Bulkhead

A.3.1 Plane Transverse Oiltight Bulkhead Scantlings

The plane bulkhead consists of vertically stiffened plates separated by horizontal stringer plates. These are framed between the bottom hull and the deck, and separate the cargo holds. The plates of the plane bulkhead are referred to as items 17.1 to 17.8. The minimum stiffener modulus values are presented in this section. The stiffener scantlings are calculated subsequently in Section A.4. The single stiffened plate transverse bulkhead was designed according to LR P4 Chapter 9, Table 9.7.1, which is summarized in Table A.6(a).

Table A.6(a) Plane Transverse Oiltight Bulkhead Scantlings
- LR P4, Chapter 9, Table 9.7.1 -

Item	Vertically Stiffened					
	Plate No.	h	h_1	S_1	f	t (mm)
Plating thickness (mm) including corrugations $t = 0.0044sf\sqrt{kh_1}$	17.1	20.83	22.83	2.0	0.98	12.36
	17.2	18.83	20.83	2.0	0.98	11.81
	17.3	16.50	18.50	3.0	1.00	11.36
	17.4	13.50	15.50	3.0	1.00	10.39
	17.5	10.50	12.78	3.0	1.00	9.44
	17.6	7.50	10.62	3.0	1.00	8.60
	17.7	4.83	8.70	2.0	0.98	7.63
	17.8	2.33	6.90	2.0	0.98	6.79
But not less than		a	B	Thickness, t (mm)		
For the upper $\frac{3}{4}$ of the bulkhead $t = \frac{a}{\left(85 + 30\left(\frac{a}{b}\right)^2\right)\sqrt{k}}$	17.3	600	3000	6.96		
	17.4	600	3000	6.96		
	17.5	600	3000	6.96		
	17.6	600	3000	6.96		
	17.7	600	2000	6.84		
	17.8	600	2000	6.84		
For the lower $\frac{1}{4}$ of the bulkhead $t = \frac{a}{\left(73 + 27\left(\frac{a}{b}\right)^2\right)\sqrt{k}}$	17.1	600	2000	7.95		
	17.2	600	2000	7.95		
Stiffener modulus (cm ³)	Plate No.	h	h_2	Z		
$Z = 0.0067ks_1^2h_2$	17.1	20.83	22.83	826.11		
	17.2	18.83	20.83	753.75		
	17.3	16.50	18.50	669.33		
	17.4	13.50	15.50	560.79		
	17.5	10.50	12.50	452.25		
	17.6	7.50	9.50	343.71		
	17.7	4.83	6.83	247.23		
	17.8	2.33	5.27	190.61		

The scantlings calculated in Table A.6(a) use the following additional terms, which are defined as follows:

- a = The lesser dimension of an unstiffened plate panel
- b = The greater dimension of an unstiffened plate panel
- b_1 = 18 m for plane bulkhead
= 16.2 m for corrugated bulkhead
- $f = 1.1 - \frac{s}{2500S_1}$
- h = Load height
- $h_1 = h + \frac{D_1}{8}$ but not less than $0.72(h + Rb_1)$
- $h_2 = h + \frac{D_1}{8}$ but not less than $0.55(h + Rb_1)$
- s = 600 mm spacing of bulkhead stiffeners or 760 mm, which is the greater of the breadth of flange or web
- D_1 = 16 m
- S_1 = Spacing of primary members.

A.3.2 Vertically Corrugated Transverse Bulkhead

The corrugated bulkhead does not frame directly into the deck or bottom hull, but into stool plates and floors. The vertically corrugated bulkhead, referred to as item 16, is illustrated in Figure A.2. It is designed according to LR Part 4, Chapter 9, Section 7 of Lloyd's Rules and Regulations (1994). The angle of the corrugation was selected to be 70° , which is greater than the minimum required angle of 40° as in Figure A.5.

Item	Dimension
b	740 mm
c	760 mm
d	714 mm
p	1000 mm
θ	70°

Figure A.5 Cross-section of Corrugated Bulkhead

Table A.6(b) Vertically Corrugated Transverse Oiltight Bulkhead Scantlings
– LR P4, Chapter 9, Table 9.7.1 –

Item	Vertically Corrugated					
	Plate No.	h	h ₁	S ₁	f	t (mm)
Plating thickness (mm)	16.1	16.0	18.00	3.0	0.997	14.17
including	16.2	13.0	15.00	3.0	0.997	12.93
corrugations	16.3	10.0	12.00	3.0	0.997	11.57
$t = 0.0044sf\sqrt{kh_1}$	16.4	7.0	9.73	3.0	0.997	10.42
Corrugation Properties	Plate No.	h	h ₂	Z		I
Modulus (cm ³)	16.1	16.0	22.52	55131		217552
$Z = 0.0085ph_2l_e^2k$	16.2	13.0	19.52	47787		179904
Inertia (cm ⁴)	16.3	10.0	16.52	40443		152256
$I = 0.032ph_2l_e^3$	16.4	7.0	13.52	33099		124608

The scantlings calculated in Table A.6(b) use the following additional terms which are defined as follows:

$$R = 0.54 - \frac{L}{1270} = 0.4025$$

$$l_e = 12.0 \text{ m, effective span of corrugated bulkhead}$$

The scantlings of the corrugated bulkhead must also conform to the requirements of LR Part 4, Ch.1, Section 9.2.1 for deep tank bulkheads with the load head h_4 measured to the highest point of the tank, but must be greater than $0.44(h_4 + Rb_1)$. These calculations are presented in Table A.7 and A.8. The stool structure is designed according to Part 4, Ch.9, Section 7.4. The sloping stool plate thickness must be greater than the thickness of the corrugation flange at midspan. The stool must be reinforced with plate diaphragms or deep webs, and for the bottom stools the diaphragms should be aligned with double bottom side girders. Continuity must be maintained between the diaphragms and the webs of bulkhead corrugations. Figure A.6, illustrates the positioning of the bulkhead webs with respect to the diaphragms in the stool and longitudinal girders which

are spaced at 3,000 mm on centre. The circles indicate the plate seams. Since the span of the corrugations does not exceed 15 m, a diaphragm plate is not required.

Figure A.6 Corrugated Transverse Oiltight Bulkhead

Table A.7 Deep Tank Bulkhead Scantlings – LR P4, Ch.1, Table 1.9.1

		Strake No.	h_4	t
(1) Plating thickness for symmetrically corrugated bulkheads	$t = 0.004sf \sqrt{\frac{\rho h_4 k}{1.025}} + 2.5$	16.1	16.5	14.83
		16.2	13.5	13.65
		16.3	10.5	12.34
		16.4	7.5	10.81

The scantlings calculated in Table A.7 use the additional terms, which are defined as follows:

ρ = 1.025, relative density (specific gravity) of liquid cargo
 s = 760 mm, equivalent to c for corrugated bulkhead transverses

f = $1.1 - \frac{s}{2500S}$ and has a value of 0.9987

S = 3.0 m, spacing of member

h_4 = load head measured to the highest point of the tank, but must be greater than $0.44(h_4 + Rb_1)$.

Table A.8 Symmetrical corrugations bulkhead (additional requirements)
- LR P4, Ch.1, Table 1.9.2 -

Parameter	Requirement	Actual	
b/t	not to exceed $70\sqrt{k} = 70$ at top and bottom	16.1	49.9
		16.2	54.2
		16.3	60.0
		16.4	68.5
D	Must be greater than $39l_e$	468	
θ	not to be less than 40°	70°	

Note, l_e = 12.0 m, effective span of corrugated bulkhead
 d = depth, in mm, of symmetrical corrugation or double plate bulkhead

A.3.3 Double Plate Transverse Bulkhead

The double plate bulkhead consists of two vertically stiffened plates separated by stringers, which rest on stool plates (similar to the corrugated bulkhead), and separates each cargo hold. The plates of the bulkhead are referred to as items 18.1 to 18.8. According to LR Part 4, Chapter 9, Section 7.4.2, the scantlings of double plate bulkheads supported by stools must be determined in accordance with the requirements of Chapter 1, Section 9.2.1 for deep tank bulkheads with the load head h_u , measured to the highest point of the tank, and must be greater than $0.44(h_u + Rb_1)$. This section states that the double plate bulkhead design must conform to the requirements of Table 1.9.1 and Table 1.9.2. To reduce the complexity of the text, the calculations related to Table 1.9.2 were not presented. Since the stool plates adjacent to the double plate bulkhead have the same properties as the double plate bulkhead, these plates are not included in Table A.9.

Table A.9(a) Double Plate Bulkhead Scantlings – LR P4, Chapter 1, Table 1.9.1

Item and requirement	Deep tank bulkheads	Plate No.	h_e	t (Rule)	t
Plating thickness for double plate bulkheads.	$t = 0.004sf \sqrt{\frac{\rho h_e k}{1.025}} + 2.5 \text{ mm}$	18.6	16.5	12.249	12.5
		18.5	13.5	11.318	11.5
		18.4	10.5	10.277	10.5
		18.3	7.5	9.073	10.0

Table A.9(b) Double Plate Bulkhead Scantlings – LR P4, Chapter 1, Table 1.9.1

Item and requirement	Deep tank bulkheads	Group No.	h_e	Z (Rule)	Scant.	Z (act.)
Modulus of double plate bulkheads	$Z = \frac{\rho s k h_e l_e^2}{22\gamma(\omega_1 + \omega_2 + 2)} \text{ cm}^3$	DPB1	16	628.98	300x25	648
		DPB2	13	511.04	300x20	527
		DPB3	10	393.11	250x22	404
		DPB4	7	275.78	200x23	277

Table A.9(c) Double Plate Bulkhead Scantlings – LR P4, Chapter 1, Table 1.9.1

Item and requirement	Deep tank bulkheads	Plate No.	h_e	Z	I	t
Stringers	$Z = 11.7\rho k h_e S l_e^2 \text{ cm}^3$	1	17.5	5667	42498	10
		2	14.5	4695	35213	10
		3	11.5	3724	27928	10
		4	8.5	2752	20642	10
		5	5.5	1781	13357	10
Webs	$I = \frac{2.5}{k} l_e Z \text{ cm}^4$	1	16.5	5343	40070	10
		2	13.5	4371	32784	10
		3	10.5	3400	25499	10
		4	7.5	2429	18214	10

The scantlings calculated in Tables A.9 (a), (b), and (c) use the additional terms which are defined as follows:

s = Stiffener spacing = 600 mm

$f = 1.1 - \frac{s}{2500S}$, but not greater than 1.0. 1.0 governs.

$K = 1$

l_e = Effective length of stiffened members = 3.0 m

$\gamma = 1.6$ for flat bars

$\omega_1 = 1.0$

$\omega_2 = 1.0$

A.3.4 Deck Transverse Girders

The deck transverses are aligned with the junction of the vertical transverses and the longitudinal bulkheads. The scantlings are designed according to LR Part 4, Ch.10, Section 2.8 with a deck transverse girder depth of 2.0 m. The corresponding calculations are summarized in Table A.10.

Table A.10 Deck Transverse Scantling Requirements

Transverse Requirements	Formulae	Minimum geometric properties
Minimum Section Modulus (cm ³)	$Z = 53.75(0.0269sL + 0.8)(S_T + 1.83)k$	24585
Net Sectional Area of the web (cm ²)	$A = 0.12Q_x k$	163
Minimum Moment Inertia (cm ⁴)	$I = \frac{7.5}{k} S_T Z$	4425388

The scantlings calculated in Table A.10 used the additional terms, which are defined as follows:

$$\begin{aligned}
 S_T &= 24 \text{ m, span of transverses} \\
 s &= 3.6 \text{ m, spacing of deck transverses} \\
 Q_x &= 1355 \text{ kN, maximum shear force in deck transverse} \\
 Q_1 &= 10.05K_2sh_bS_T = 1355 \text{ kN, shear force at ends of deck transverse} \\
 Q_2 &= \left[\frac{1}{2K_2} - 1 \right] Q_1 = 0 \text{ kN, shear force at midspan of deck transverse} \\
 h_b &= \frac{L_1}{56} = 3.12 \text{ m, head} \\
 K_2 &= 0.5
 \end{aligned}$$

Based on a deck transverse depth of 2,000 mm, a 30 mm thick plate would have to be used to provide the required moment of inertia and section modulus. This is impractical. The required sectional properties were obtained by adding a flange (strake number 15.1) to the plate (strake 15.2) and is illustrated in Figure A.7.

Figure A.7 Typical Deck Transverse Girder Cross-Section

A.4 Stiffener Scantlings

Table A.10, tabulates the rules for the minimum required stiffener section modulus values for the stiffener groups associated with the items illustrated in Figure 1.1. The stiffener scantlings were calculated to give section moduli greater than those required.

According to LR Part 3, Chapter 3.2.7.1, the section modulus of each stiffener group of the same scantling, as illustrated in Figure A.8, must be greater than the following:

- the mean head within the group, or
- 90% of the maximum head within the group.

Table A.11 List of Applicable Rules for the Design of Stiffeners

Item No.	Item	Applicable Rules in Lloyd's Rules and Regulations (1994)
1	Deck	P4 Ch.9 5.3.1(a) (Mean Head) P4 Ch.9 5.3.1(b) (Mean Head) P4 Ch.9 5.3.1(a) (90% Head) P4 Ch.9 5.3.1(b) (90% Head) P3 Ch.4 T4.7.3 (Buckling)
2-6	Sheerstrake and Gunwale, Side Shell	P4 Ch.9 5.3.1(a) (Mean Head) P4 Ch.9 5.3.1(b) (Mean Head) P4 Ch.9 5.3.1(a) (90% Head) P4 Ch.9 5.3.1(b) (90% Head) P4 Ch.9 T1.6.1(3) (Mean Head) P4 Ch.9 T1.6.1(3) (90% Head) P3 Ch.4 T4.7.3 (Buckling)
8 and 9	Bottom Shell, and Keel	P4 Ch.9 5.3.1(a) (Mean Head) P4 Ch.9 5.3.1(b) (Mean Head) P4 Ch.9 5.3.1(a) (90% Head) P4 Ch.9 5.3.1(b) (90% Head) P4 Ch.9 T1.6.1(3) (Mean Head) P4 Ch.9 T1.6.1(3) (90% Head) P3 Ch.4 T4.7.3 (Buckling)
7	Bilge	P4 Ch.9 5.4.1 P3 Ch.4 T4.7.3 (Buckling)
10	Inner Bottom	P4 Ch.9 T9.6.1(2) (Mean Head) P4 Ch.9 T9.6.1(2) (90% Head) P4 Ch.1 T1.9.1(2) (Mean Head) P4 Ch.1 T1.9.1(2) (90% Head) P3 Ch.4 T4.7.3 (Buckling)
13	Longitudinal Bulkhead	P4 Ch.9 T9.6.1(2) (Mean Head) P4 Ch.9 T9.6.1(2) (90% Head) P3 Ch.4 T4.7.3 (Buckling)

- $s = 600$ mm, spacing of longitudinals
 $l_e = 3.6$ m, effective length of longitudinals
 $R = 0.54 - \frac{L}{1270} = 0.4025$
 $b_1 = 32$ m, breadth of ship, where wing ballast tanks port and starboard are interconnected
 $F_1 =$ Refer to Table A.11
 $F_2 =$ Refer to Table A.12
 $F_s = \frac{11}{k} \left[1 - \frac{2b_{f11}}{b_f} (1 - k) \right]$, fatigue factor which is valid at 0.6D above the base line
 $F_s = 1.0$, at upper deck at side and at the base line, intermediate values by linear interpolation
 $\frac{b_{f11}}{b_f} = 0.5$, for flat bar stiffeners

Table A.12 F_1 Values

Item	F_1
Deck longitudinals and side longitudinals above D/2	$\frac{Dc_1}{4D + 20h}$
Side longitudinals and bottom longitudinals below D/2	$\frac{Dc_1}{25D - 20h}$
NOTE: Minimum $F_1 = 0.12$	

where, $c_1 = \frac{60}{225 - 165F_D}$ at deck
 $= 1.0$ at D/2
 $= \frac{75}{225 - 150F_B}$ at base line of ship

Table A.13 F_2 Values

Item	F_2
Deck longitudinals and side longitudinals above $D/2$	$\frac{Dc_2}{D + 2.18h}$
Side longitudinals and bottom longitudinals below $D/2$	$\frac{Dc_2}{3.18D - 2.18h}$
NOTE: Minimum $F_2 = 0.73$	

$$\begin{aligned}
 \text{where, } c_2 &= \frac{165}{345 - 180F_D} \text{ at deck} \\
 &= 1.0 \text{ at } D/2 \\
 &= \frac{165}{345 - 180F_B} \text{ at base line of ship}
 \end{aligned}$$

Note that the section moduli of side longitudinals need not exceed those of bottom longitudinals having the same spacing and configuration. The results of the calculations for Equations A.5(a) and A.5(b) are given in Table A.14.

Table A.14 Deck, Side and Bottom Longitudinals – LR P4, Chapter 9, Section 5.3.1

Item	Group No.	H	h ₀	h ₁	h ₃	c ₁	c ₂	F ₁	F ₂	F ₃	(a) Z	(b) Z	Condition P3 Ch.3 2.7.1
Deck	SD1	0	1.283	3.283	10.94*	0.593	0.786	0.148	0.786	1.000	221.8	341.0	90% Head
	SD2	0	0	3.118	12.88	0.593	0.786	0.148	0.786	1.000	201.1	401.4	Both
Side Shell	SS1	17.20	17.20	19.20	30.08	0.760	0.846	0.12	0.73	1.023	1026.8**	870.9**	Mean Head
	SS2	14.50	14.50	16.50	27.38	0.850	0.904	0.12	0.73	1.046	901.8	792.7	Mean Head
	SS3	11.50	11.50	13.50	24.38	0.950	0.968	0.12	0.73	1.071	755.5	705.8	Mean Head
	SS4	8.50	8.50	10.50	21.38	0.939	0.968	0.12	0.73	1.096	601.3	619.0	Mean Head
	SS5	5.76	5.76	7.76	18.64	0.827	0.909	0.12	0.73	1.072	434.7	539.7	90% Head
	SS6	3.06	3.06	5.06	15.94	0.717	0.717	0.12	0.73	1.038	274.6	461.5	90% Head
Bottom Shell	BS1 - BS5	20.00	20.00	17.00	27.88	0.667	0.786	0.133	0.786	1.000	987.0	868.7	Mean Head

*Note 1: The value for $b_1 = 24$ m since the ballast tanks are non-existent at the cambered deck elevation.

**Note 2: This group of stiffeners will have a modulus equivalent to the bottom shell longitudinals, since the modulus of the side shell longitudinals need not exceed that of a bottom longitudinal having the same spacing and configuration.

The section moduli of bottom and side shell longitudinals must also satisfy the requirements of LR Part 4, Chapter 1, Table 1.6.1(3), in which the section moduli must be equal to or greater than:

$$Z = \gamma s k h_{T2} l_e^2 F_1, \text{ or} \quad (\text{A.6a})$$

$$Z = \gamma s k h_{T3} l_e^2 F_1 F_{\Delta} \quad (\text{A.6b})$$

where,

$$\gamma = 0.0021 l_{e1} + 0.046 = 0.0532$$

$$s = 600 \text{ mm}$$

$$l_{e1} = 3.6 \text{ m}$$

$$h_{T2} = (T + 0.5C_w) = 15.122 \text{ m, but need not be taken greater than } 1.2T = 14.4 \text{ m (governs)}$$

$$C_w = 7.71 \times 10^{-2} L e^{-0.0044L} = 6.2439 \text{ m, wave head}$$

$$F_1 = \frac{D_2 C_1}{4D_2 + 20h_s} \text{ for side longitudinals above } D_2/2$$

$$= \frac{D_2 C_1}{25D_2 - 20h_s} \text{ for side longitudinals below } D_2/2 \text{ and bottom}$$

$$\text{longitudinals}$$

$$= 0.14 \text{ minimum value}$$

$$D_2 = 20 \text{ m, but need not be greater than } 1.6T = 19.2 \text{ m (governs)}$$

$$C_1 = \frac{60}{225 - 165F_D} = 0.581226 \text{ at deck}$$

$$= 1.0 \text{ at } D_2/2$$

$$= \frac{75}{225 - 150F_B} = 0.60241 \text{ at base line}$$

$$h_{T3} = h_4 - 0.25T$$

$$h_4 = \text{load head}$$

$$F_{\Delta} = 0.5(1 + F_s \text{ at } 0.6D_2) = 1.05$$

$$F_s = \frac{L1}{k} \left[1 - \frac{2b_{f1}}{b_f} (1 - k) \right] = L1 \text{ at } 0.6D_2 (11.52 \text{ m}) \text{ above the base line}$$

$$= 1.0 \text{ at } D_2 \text{ and above, and } F_{\Delta} \text{ at the base line intermediate values by linear interpolation}$$

The results of the calculations for Equations A.6(a) and A.6(b) are given in Table A.15.

Table A.15 Side and Bottom Longitudinals – LR P4, Chapter 1, Table 1.6.1

Item	Group No.	h_4	h_s	h_{r3}	C_1	F_1	F_2	(a) Z	(b) Z	Condition P3 Ch.3 2.7.1
Side Shell	SS1	17.20	1.40	14.20	0.711	0.14	1.05	833.985	863.583	Mean Head
	SS2	14.50	3.80	11.50	0.812	0.14	1.05	833.985	699.392	Mean Head
	SS3	11.50	6.80	8.50	0.937	0.14	1.05	833.985	516.958	Mean Head
	SS4	8.50	9.80	5050	0.940	0.14	1.05	833.985	344.524	Mean Head
	SS5	5.76	12.80	2.76	0.831	0.14	1.05	833.985	167.894	90% Head
	SS6	3.06	15.80	0.06	0.724	0.14	1.05	833.985	3.703	90% Head
Bottom Shell	BS1 - BS5	20.00	0.80	17.00	0.6024	0.14	1.10	833.985	1033.855	Mean Head

The inner bottom and longitudinal bulkhead stiffener dimensions are designed according to LR Part 4, Chapter 9.6, Table 9.6.1(2). The appropriate sections of that table have been reproduced in Table A.16. The scantlings calculated in this table use the additional terms, which are defined as follows:

$$h_2 = \left(h + \frac{D_1}{8} \right), \text{ in metres, but in no case to be taken less than } \frac{L_1}{56} \text{ m or}$$

$(0.01L_1 + 0.7) \text{ m, whichever is the greater}$

$$h_3 = \text{Distance of longitudinal below deck at side, in metres, but is not to be less than 0}$$

$$h_4 = h + Rb_1$$

$$h_5 = h_2 \text{ but is to be greater than } 0.55h_4$$

$$D_1 = 16 \text{ m}$$

$$R = 0.54 - \frac{L}{1270} = 0.4025$$

$$\theta = \tan^{-1}(1/1) = 45^\circ$$

$$b_1 = 26 \text{ m, the greater horizontal distance from midpoint of span of the stiffener to the tank corner at top of tank, either side.}$$

$$F_1 = \frac{Dc_1}{4D + 20h_3} \text{ above } D/2$$

$$= \frac{Dc_1}{25D - 20h_3} \text{ below } D/2$$

$$= 0.12, \text{ at least}$$

$$c_1 = \frac{60}{225 - 165F_D} = 0.581226$$

$$= 1.0 \text{ at } D/2$$

$$= \frac{75}{225 - 150F_B} = 0.6024096 \text{ at base line of ship}$$

$$F_2 = \frac{Dc_2}{D + 2.18h_3} \text{ above } D/2$$

$$= \frac{Dc_2}{3.18D - 2.18h_3} \text{ below } D/2$$

$$= 0.73, \text{ at least}$$

$$c_2 = \frac{165}{345 - 180F_D} = 0.7777149$$

$$= 1.0 \text{ at } D/2$$

$$= \frac{165}{345 - 180F_B} = 0.735294 \text{ at base line of ship}$$

Table A.16 Inner Hull and Longitudinal Oiltight Bulkhead Scantlings – LR P4, Chapter 9, Table 9.6.1

Horizontally Stiffened:													
(i) $Z = 0.056kh_2sl_e^2F_1$ (ii) $Z = 0.0051kh_4sl_e^2F_2$ } whichever is the greater													
Item	Group No.	h	h ₂	h ₃	h ₄	G ₁	G ₂	F ₁	F ₂	Z (i)	Z (ii)	Condition	
Longitudinal Bulkhead	LB1	18.50	20.50	18.50	28.82	0.717	0.818	0.12	0.73	1071.2	834.3	Mean Head	
	LB2	16.00	18.00	16.00	26.47	0.800	0.871	0.12	0.73	940.6	766.2	Mean Head	
	LB3	13.00	15.00	13.00	23.47	0.900	0.936	0.12	0.73	783.8	679.3	Mean Head	
	LB4	10.00	12.00	10.00	20.46	1.000	1.000	0.12	0.73	627.1	592.5	Mean Head	
	LB5	7.26	9.26	7.26	17.73	0.888	0.941	0.12	0.73	483.9	513.2	90% Head	
	LB6	4.77	6.77	4.77	15.09	0.787	0.888	0.12	0.73	353.9	436.9	90% Head	
	LB7	2.76	4.76	2.76	12.42	0.705	0.845	0.12	0.73	248.7	359.6	90% Head	
Vertically Stiffened:													
$Z = 0.0067ksl_e^2h_5$													
Item	Group No.	h	h ₂	h ₃	h ₄	h ₅	Z	Condition					
Inner Bottom	IB1 - IB4	19.50	21.50	29.965	21.50	1120.1	Mean Head						

The bilge stiffeners are designed according to Part 4, Chapter 4, Section 5.4.1, which states that the scantlings of bilge longitudinals are to be graduated between those required for the bottom and lowest side longitudinals. The required section modulus for the bottom shell longitudinal and the lowest side longitudinal is 1034 cm³ and 987 cm³, respectively. Given that the bottom longitudinal is at an elevation of zero and the lowest side longitudinal is at an elevation of 2,200 mm, and using linear interpolation, the section modulus for the bilge longitudinals was calculated to be 1,023 cm³.

The stiffeners along the inner bottom must conform to the requirements of LR Part 4, Chapter 1, Table 1.9.1(2). Also, all stiffeners must meet the buckling requirements of LR Part 3, Chapter 4, Table 4.7.3. The modulus values obtained according to these sections do not govern and have been omitted from the text.

Once all the rule modulus values have been found, the actual stiffener scantlings are computed, given the following properties of the geometric section illustrated in Figure A.9.

$$A = bt + w(d - t)$$

$$y = \frac{1}{2} \left(\frac{bdt}{A} + d - t \right)$$

$$I = \frac{1}{12} \left[bt^3 + w(d - t)^3 + \frac{3bwtd^2(d - t)}{A} \right]$$

$$S = \frac{I}{y}$$

Figure A.9 Geometric Properties of Stiffened Plate

The stiffener moduli, stiffener scantlings, and section calculations are summarized in Table A.17. The required stiffener moduli specified by the rules are also given.

Table A.17 Summary of Stiffener Modulus Calculations

Group No.	b (mm)	d (mm)	t (mm)	w (mm)	y (mm)	A (mm ²)	I (cm ⁴)	Z (cm ³)	Z(Rule) (cm ³)
SD1	600	261.75	11.75	19	203.19	11800	7343	361.4	345.6
SD2	600	261.5	11.5	23	196.32	12650	8364	426.1	401.4
SS1	600	412.5	12.5	25	288.39	17500	31574	1094.8	987.0
SS2	600	412.5	12.5	25	288.39	17500	34574	1094.8	901.8
SS3	600	362.5	12.5	23	262.42	15550	20983	799.6	755.5
SS4	600	312.5	12.5	25	228.13	15000	14790	648.3	619.1
SS5	600	312.5	12.5	21	234.92	13800	13094	557.4	539.7
SS6	600	312.5	12.5	20	236.81	13500	12648	534.1	461.5
B1	600	413	13	25	290.49	17800	32030	1102.6	1023.4
BS1	552	415	15	25	293.99	18280	32851	1117.4	1033.9
BS2	600	413	13	25	290.49	17800	32030	1102.6	1033.9
BS3	600	413	13	25	290.49	17800	32030	1102.6	1033.9
BS4	600	413	13	25	290.49	17800	32030	1102.6	1033.9
BS5	600	413	13	25	290.49	17800	32030	1102.6	1033.9
IB1	600	414	14	27	290.56	19200	34660	1192.9	1123.5
IB2	600	414	14	27	290.56	19200	34660	1192.9	1123.5
IB3	600	414	14	27	290.56	19200	34660	1192.9	1123.5
IB4	600	414	14	27	290.56	19200	34660	1192.9	1123.5
LB1	600	412.5	12.5	25	288.39	17500	31574	1094.8	1071.2
LB2	600	412.5	12.5	25	288.39	17500	31574	1094.8	940.6
LB3	600	361.5	11.5	23	258.42	14950	20364	788.0	783.0
LB4	600	311	11	25	222.79	14100	14120	633.8	627.1
LB5	600	310.5	10.5	20	229.52	12300	11913	519.0	513.1
LB6	600	310	10	20	227.50	12000	11713	514.8	436.9
LB7	600	260	10	21	194.33	11250	7471	384.5	436.9
TB1 [‡]	600	362.5	12.5	25	258.65	16250	22209	858.6	826.1
TB2 [‡]	600	362	12	23	260.46	15250	20678	793.9	753.8
TB3 [‡]	600	361.5	11.5	20	264.72	13900	18506	699.1	669.3
TB4 [‡]	600	310.5	10.5	23	224.10	13200	13118	588.4	560.8
TB5 [‡]	600	260	10	27	186.18	12750	8889	477.4	452.3
TB6 [‡]	600	260	10	21	194.33	11250	7471	384.5	343.7
TB7 [‡]	600	260	10	14	207.11	9500	5564	268.6	247.2
TB8 [‡]	600	235	10	13	191.49	8925	3954	206.5	190.6

[‡] The b/t ratio for the plane transverse bulkhead stiffeners was taken to be less than $18\sqrt{k}$ as required by LR P4 Ch.9 Table 9.7.1

A.5 Design Summary

This section summarizes the scantlings determined from design calculations of sections A.1, A.2, A.3, and A.4. Figure A.10 illustrates the hull envelope plate thickness and the stiffener scantlings. Figures A.11, A.12, and A.13 illustrate the stiffened plate bulkhead, the corrugated bulkhead, and the double plate bulkhead, respectively.

Figure A.10 Hull Envelope Strake and Stiffener Scantlings

Figure A.11 Plane Bulkhead and Transverse Framing Scantlings

Figure A.12 Corrugated Bulkhead and Transverse Framing Scantlings

Figure A.13 Double Plate Bulkhead and Transverse Framing Scantlings