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Development and Behaviour of Advanced Double Hull Sandwich Plate Systems: Experimental Investigation

by

Josef Linder

Dipl.-Ing. (FH), Fachhochschule Karlsruhe, Germany

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

MASTER OF ENGINEERING*

DEPARTMENT OF CIVIL AND ENVIRONMENTAL ENGINEERING

CARLETON UNIVERSITY

OTTAWA, CANADA

August 1995

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**Behaviour of Advanced Double Hull Sandwich Plate Systems:
Experimental Investigation**

submitted by

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

Dr. E. W. Wright, External Examiner

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August, 1995

ABSTRACT

Environmental concerns resulting from a number of recent catastrophic oil spills have led to the introduction of legislation that requires the use of double hull technology for oil tankers and the development of new design technologies, such as the advanced unidirectional double hull concept (DHUC). The advanced unidirectional double hull sandwich plate system (UDHSPS)-toughskin approach has a similar structural system as the DHUC, but replaces the outer hull (single skin) with a steel-polyurethane foam-steel sandwich plate and uses collapsible bulkheads. The system is designed to absorb energy from collision or groundings through plastic deformation of the structure. This structural system has improved collision performance, increased load carrying capacity and simplicity in design and construction.

The objectives of this study were to design and conduct a large scale steel-polyurethane foam-steel sandwich plate exploratory test to determine its behaviour and to generate the experimental data for future finite element analyses. The test specimen measured 5000 mm by 2600 mm in plan and was loaded monotonically to failure with a series of five hydraulic actuators, over a 2400 mm length along the longitudinal axis. The depth of the sandwich plate was 123.1 mm with a 100 mm thick core. The test

specimen was fixed to the horizontal reaction frame which represents the stiffness of the surrounding ship structure.

The sandwich plate had little flexural capacity and essentially behaved as a membrane acting in the transverse direction. At a total load of 8992 kN, concentrated local indentations caused shear tension failure of the outside hull plate under the corner of the central load plate. Load deflection curves and strain distributions are presented along with a discussion of the test results. Recommendations are made for future work and for improved structural performance of the sandwich plate.

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Special thanks go to my girlfriend Elke and the whole Schättgen family who showed continual hospitality and support throughout these two years of study.

t_f	Thickness of composite face.
δ	Total deflection.
δ_{pl}	Total deflection at forming of plastic hinges.
δ_{ult}	Total deflection at ultimate.
ϵ_f	Strain at fracture.
ϵ_{fi}	Strain at final reading.
ϵ_p	Strain at proportional limit.
ϵ_u	Ultimate strain.
ϵ_y	Yield strain.
ϵ_{st}	Strain at onset of strain hardening.
μs	Microstrain, mm/mm
ν	Poisson's ratio.
σ_f	Stress at fracture.
σ_{fi}	Stress at final reading.
σ_p	Proportional limit.
σ_{sy}	Static yield stress.
σ_u	Ultimate stress.
σ_y	Yield stress.

2.3.1	Introduction to the Advanced Double Hull Technology	10
2.3.2	The Compressive Behaviour of Advanced Double Hull Structure - A Review of Current Research at David Taylor, by Adamchak et al.	12
2.3.2.1	A Preliminary Look at the Relative Significance of Local Versus General Instability Failure of Double Hull Structure	12
2.3.2.2	Small Scale Experimental Analysis of Double Hull Sections Using Rigid Vinyl Modelling	13
2.3.2.3	The Application of Beam-on-Elastic-Foundation Analysis for High Aspect Ratio Stiffened Plates	14
2.3.3	Stranding Experiments on Double Hull Tanker Structures, by Rodd et al.	15
2.3.4	Tearing Resistance of Advanced Double Hulls, by Bracco and Wierzbicki	16
2.3.5	Applicable Standards for Double Hull Construction.	18
2.3.5.1	Introduction	18
2.3.5.2	Standards for the Double Hull Construction of Oil Tankers (TP 11710), Canadian Coast Guard Ship Safety Branch	19
2.3.5.3	Comparison of Double Hull Construction Requirements	21

2.4	Advanced Unidirectional Double Hull Sandwich Plate System-	
	Toughskin Approach	22
2.4.1	Introduction	22
2.4.2	Hull Girder Behaviour and Structural Integrity	23
2.4.3	Energy Absorption	24
2.4.4	Fracture Toughness	24
2.4.5	Construction and Reparability	25
2.5	Behaviour of Steel and Composite Sandwich Plate Elements	26
2.5.1	Tanker Structural Analysis for Minor Collisions, by McDermott et al.	26
2.5.2	Designing Tanker Structures for Collisions and Grounding Resistance, by Edinburg et al.	27
2.5.3	Behaviour for Transversely Loaded Continuous Steel- Concrete Composite Plates, by Kennedy and Cheng	29
2.6	Structural Plastics and Sandwich Construction	30
2.7	Summary	32

3 Experimental Program

3.1	General	47
3.2	Preliminary Analysis and Design	48
3.3	Test Set-Up	51
3.3.1	Loading Apparatus	51

3.3.2	Horizontal Reaction Frame	52
3.3.3	Instrumentation and Measurements	52
3.4	Installation, Alignment and Instrumentation of Test Specimen	53
3.5	Injection of Core Material	55
3.6	Test Procedure	57
3.7	Ancillary Tests	58
3.7.1	Steel Strength	58
3.7.2	Polyurethane Foam Strength	59
3.7.2.1	Compression Test	59
3.7.2.2	Tension Test	60
3.7.2.3	Shear Test	60
4	Test Results and Discussion	
4.1	Material Properties Steel Plate	76
4.2	Material Properties Polyurethane Foam	77
4.2.1	Compression Tests	77
4.2.2	Tension Tests	78
4.2.3	Shear Tests	78
4.3	Composite Steel-Polyurethane Foam-Steel Sandwich Plate Test	
Results		79
4.3.1	General Behaviour	79
4.3.2	Deflection Characteristics	81

4.3.3	Measured Strains	83
4.3.4	Comparison with IMASP Model	85
5	Summary, Conclusions, Recommendations, and Future Work	
5.1	Summary and Conclusions	127
5.2	Recommendations and Future Work	129
	References	132

List of Figures

1.1	Advanced Unidirectional Double Hull Sandwich Plate System	5
2.1	Conventional and Advanced Double Hull Design	44
2.2	Anticipated Instability Modes	45
2.3	Advanced Double Hull Deformation Model	45
2.4	Cargo Tank Boundary Lines	46
2.5	Tanker Cross-Sections	46
3.1	Fabrication Drawing of Test Specimen	64
3.2	Typical Sandwich Cross-Section	66
3.3	Cross-Section of Test Specimen in Test Apparatus	67
3.4	Load Application Points for Test Specimen	68
3.5	Plan Sectional View of Horizontal Reaction Frame	69
3.6	Location of Strain Gauges on Bottom Plate (Outside Hull Plate)	70
3.7	Location of Strain Gauges on Top Plate (Inside Hull Plate)	71
3.8	Location of LVDTs on Top Plate	72

3.9	Location of Injection Holes	73
3.10	Tension Coupon Definition	74
3.11	Polyurethane Foam Coupon Definition	75
4.1	Typical Stress-Strain Curves for Tension Tests of Steel Plates	92
4.2	Stress-Strain Curves for Compression Tests of Polyurethane Foam . .	93
4.3	Stress-Strain Curves for Extended Loading of Compression Test for Polyurethane Foam	94
4.4	Stress-Strain Curves for Tension Tests of Polyurethane Foam	95
4.5	Stress-Strain Curves for Shear Tests of Polyurethane Foam	96
4.6	Total Load Midpanel Deflection Curve illustrating Progression of Failure	97
4.7	Yielding Summary of Top Plate	98
4.8	Yielding Summary of Bottom Plate	99
4.9	Photographs of Deflected Shape	100
4.10	Measurements of Core Thickness and Final Deformed Shape of Transverse Direction	101
4.11	Representative Quadrant for Display of Test Results	102
4.12	LVDTs for Vertical Displacement Measurements on Top Plate	102
4.13	End View of Actuators and Deformed Plate	103
4.14	Total Load Deflection Curves for LVDT 18 and LVDT 31 located at Midpanel	103

4.15	Total Load Deflection Curves for LVDTs 19, 32, 17 and 31	104
4.16	Total Load Deflection Curves for LVDTs 14, 34, 16 and 34	105
4.17	Total Load Deflection Curves along Transverse Axis for LVDTs 18, 24, 5 and 1	106
4.18	Total Load Deflection Curves along 3/8l (Transverse Direction) for LVDTs 19, 12, 6 and 2	107
4.19	Total Load Deflection Curves along 1/4 (Transverse Direction) for LVDTs 14, 13, 7 and 3	108
4.20	Total Load Deflection Curves along 1/8 (Transverse Direction) for LVDTs 21, 8 and 4	109
4.21	Total Load Deflection Curves along Centreline of Longitudinal Direction for LVDTs 18, 19, 14 and 21	110
4.22	Displacement Values for the Specified Quadrant	111
4.23	Deflected Shape of Top Plate at Ultimate Load	113
4.24	Total Load Principal Strain Curves along the Centreline	114
4.25	Total Load Principal Strain Curves for Top and Bottom Plate along the Transverse Axis	116
4.26	Strain Values for Top Plate for the Specified Quadrant	118
4.27	Strain Values for Bottom Plate for the Specified Quadrant	122
4.28	Total Load Centre Span Deflection Curves for IMASP Model and Test Result	126

List of Tables

2.1	50 Major Oil Spills from Tankers and Combined Carriers	
	1960-1989	33
2.2	Oil Spills from Tankers and Combined Carriers 1990-1994	37
2.3	Summary of Analytical Predictions and Experimental Results	39
2.4	Test Results for Stranding Experiments	40
2.5	Test Results for Cutting Experiments	41
2.6	Characteristics for selected Polyurethane Materials	43
3.1	Analytical Results for Design of Test Specimen	61
3.2	Type of Measurement and Instrumentation	62
3.3	Measurement, Range and Accuracy	63
4.1	Chemical Composition and Mechanical Properties of the Structural	
	Steel Plate from the Mill Certificates	86
4.2	Characteristic Parameters for the Steel Plate determined from	
	Uniaxial Stress Strain Curves for the 12.7 mm Plates in the	

	Longitudinal Direction	87
4.3	Characteristic Parameters for the Steel Plate determined from	
	Uniaxial Stress Strain Curves for the 12.7 mm Plates in the	
	Transverse Direction	88
4.4	Characteristic Parameters for Polyurethane Foam determined from	
	Uniaxial Compression Test Data	89
4.5	Characteristic Parameters for Polyurethane Foam determined from	
	Uniaxial Tension Test Data	89
4.6	Characteristic Parameters for Polyurethane Foam determined from	
	Uniaxial Shear Test Data	90
4.7	Progression of Failure	91

List of Symbols

B	Effective width of composite plate; width of ship.
D	Flexural stiffness of composite plate.
DWT	Deadweight in tons.
E	Modulus of Elasticity.
E_c	Elastic modulus of composite core.
E_f	Elastic modulus of composite face.
E_{st}	Strain hardening modulus.
H	Overall depth of composite plate.
HAZ	Heat affected zones.
l	Plate span.
mt	Metric tons.
P	Total applied load.
P_{pt}	Total applied load at forming of plastic hinges.
P_{ult}	Total applied load at ultimate.
SBT's	Segregated ballast tanks.
t_c	Thickness of composite core.

t_f	Thickness of composite face.
δ	Total deflection.
δ_{pl}	Total deflection at forming of plastic hinges.
δ_{ult}	Total deflection at ultimate.
ϵ_f	Strain at fracture.
ϵ_{fi}	Strain at final reading.
ϵ_p	Strain at proportional limit.
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μs	Microstrain, mm/mm
ν	Poisson's ratio.
σ_f	Stress at fracture.
σ_{fi}	Stress at final reading.
σ_p	Proportional limit.
σ_{sy}	Static yield stress.
σ_u	Ultimate stress.
σ_y	Yield stress.

Chapter 1

Introduction

1.1 General

With the introduction of the *Standards for the Double Hull Construction of Oil Tankers* (TP 11710) in 1993, the Canadian Coast Guard now requires double hull construction or approved equivalent alternative arrangements for all new oil tankers. Existing oil tankers without a double hull construction will have to be retired from service according to a specific schedule. The legislation of these standards is an attempt to reduce the risk of catastrophic oil tanker casualties, oil spills and the ensuing adverse impact on environment, and wildlife through the use of recently developed double hull technology.

Naval architects, structural engineers and maritime related government sectors from around the world are exploring operational alternatives and new tank vessel designs in order to improve survivability and to enhance the performance of oil tankers under

accident loads (groundings and collisions). This research led to the development of the Advanced Double Hull Concept (ADH), which is also called Double Hull Unidirectional Concept (DHUC). The hull is a twin skin structure which is separated by unidirectional longitudinal web-girders forming a cellular structure. This design concept yields a larger ship hull section modulus, increased strength and stiffness, giving the double hull improved ability to withstand damages due to collisions, groundings and seaway loadings.

A natural extension is the development of an Advanced Unidirectional Double Hull Sandwich Plate System (UDHSPS), that exploits the simplicity of the DHUC technology and incorporates the ultimate load carrying capacity and performance of a composite system. The proposed system will be comprised of the same principal elements as the DHUC but replaces the outer hull by a steel-polyurethane foam-steel sandwich plate and incorporates collapsible bulkheads to absorb the energy of groundings and collisions through plastic deformation. A typical cross-section is shown in Figure 1.1. The increased load carrying capacity and ductility of the sandwich plate will allow a reduction in weight (reduction of longitudinal stiffeners), simplified construction, and the optimization of protection spaces (increased cargo space), making oil tankers more efficient and cost effective.

1.2 Objectives and Scope

The first step in the development of the UDHSPS design concept is to conduct a feasibility study. The objectives of this study are:

1. to conduct a state-of-the-art literature review and to present the proposed concept ;
2. to select an appropriate structural plastic or foam material for the core;
3. to design and construct a sandwich plate;
4. to conduct a large scale test: (a) to determine the behaviour and ductility of the sandwich plate; (b) to obtain experimental data for future analytical comparisons; (c) to make recommendations for improved structural performances based on the observations made from this test;

The large scale sandwich test plate measured 5000 mm by 2600 mm in plan, and 123.1 mm in total depth with two steel plates of 11.56 mm thickness and a polyurethane foam core with a depth of 100 mm. The steel section was fully welded to the horizontal reaction frame which represents the stiffness of the surrounding ship structure and provides the in-plane restraint necessary to develop membrane action. The polyurethane foam core was injected into the cavity by a procedure that was established by trial and error on test cells. Instrumentation was installed. The sandwich plate was transversely loaded with five point loads each with an area of 200 mm by 300 mm, along the longitudinal axis. Load was applied quasi-statically and monotonically to failure. The test was conducted under laboratory conditions.

Ancillary tests were conducted in accordance with ASTM standards on steel tension coupons at Carleton and on polyurethane foam samples by DOW CHEMICAL to establish the corresponding material properties.

The sandwich plate test results are presented, discussed and the load deflection

curve is compared with the load deflection curve obtained from a computer program that predicts the elastic and inelastic membrane behaviour of steel plates.

1.3 Outline

A state-of the-art literature review, presented in Chapter 2, includes a summary of 50 major oil spills from oil tankers and combined carriers which occurred between 1965 and 1989, additional oil spills which occurred between 1990 and 1994, a review of papers describing the behaviour of advanced double hulls, a review of legislations regarding double hull oil tankers, a description of the structural behaviour of the UDHSPS concept, a review of papers describing the behaviour of steel and composite sandwich elements under transverse loads, and a review of structural plastics and dense foams for use as the core material.

Chapter 3 describes the experimental program, the test panel design, test set-up, instrumentation, experimental method and the corresponding ancillary tests.

The test results are presented and discussed in Chapter 4. The general behaviour, load-deflection curves, strain distributions, and failure modes of the sandwich plate are presented along with the material properties of the steel and polyurethane foam. Comparisons are made with the results of a computer program that predicts the load deflection curves for membrane behaviour of continuous steel plates.

A summary, conclusions, recommendations, and a brief discussion of possible future work are presented in Chapter 5.

Figure 1.1 Advanced Unidirectional Double Hull Sandwich Plate System
(after MIL Systems Engineering, 1991)

Chapter 2

Literature Review and Proposed Concept

2.1 General

Problems in existing oil tanker designs and related areas are described in this state-of-the-art literature review.

The first section gives an overview of the 50 major oil spills from tankers and combined carriers that occurred between 1960 and 1989, (National Research Council, United States, 1991) and data from oil spills that occurred between 1990 and 1994 (FactsOnFile, 1990-1994). This data, along with the causes and analyses of oil spills, will demonstrate the need for further research in developing better oil tanker designs that will minimize loss of oil products (spills), avoid devastating environmental damage and loss of tanker vessels.

Section 2.3 gives an introduction to what has been termed the Advanced Double Hull technology (Okamoto et al., 1985). This concept was developed by naval architects as an answer to improve surface ship survivability and behaviour of ships due to operational and accidental loads. Research conducted to examine the behaviour of double hull tanker structures subjected to compressive loads, stranding, and tearing are reviewed. Legislation which phases in and makes this technology mandatory for new tank vessels is presented.

Section 2.4 explains the principles and advantages of the Advanced Unidirectional Double Hull Sandwich Plate System - Toughskin Approach. Further papers review the behaviour of steel and composite sandwich elements and give analytical procedures of the membrane response action of continuous beam and plate elements subjected to transverse loading. Finally, use of structural plastics and dense foams for the core material of the steel-structural plastic-steel toughskin sandwich plate is briefly introduced.

2.2 Oil Tanker Failures

As a result of recent well-publicized oil tanker failures and spills, the ensuing environmental damages (perceived or otherwise), growing public awareness and demand for risk free technology (intolerance), new national and international standards (MARPOL 73/78; Oil Pollution Act of 1990, 1990; Standards for Double Hull Construction of Oil Tankers, 1993) have been legislated. This has in turn led to further research and development of design rules and new concepts in ship designs.

A review of oil tanker failures has been conducted to understand the various failure modes and to determine the most vulnerable sections of the oil tanker hull. Table 2.1 gives data of the 50 major oil spills from tankers and combined carriers that occurred between 1960 and 1989, (National Research Council, U.S., 1991). Listed is the name of the tanker, its dead weight (size) in tons (DWT), an estimate of the spillage in metric tons (mt), the approximate location, date and cause of the spill. These are ranked in order from the maximum to least spillage. Some additional spills since 1990 are listed in Table 2.2.

The causes of tanker failures are classified as either grounding or collision/ramming or fire/explosion or structural/machinery/other. Definitions of each cause after National Research Council, U.S., 1991), follows.

Groundings

The tank vessel is reported in contact with the sea or a bottom obstacle, struck object on the sea floor, or struck or touched the bottom; includes ships reported "hard and fast" for an appreciable period of time (wrecked/stranded).

Collision/Ramming

The tank vessel struck or was struck by another vessel on the water surface, or struck a stationary object, not another ship.

Fire/Explosion

The fire and/or explosion is the initiating event reported, except where the first event is

a hull/machinery failure leading to the fire/explosion. Therefore casualties involving fires or explosions after collisions or groundings are categorized under "collision" or "grounding", respectively.

Structural/Machinery/Other:

Hull/machinery damage, missing, and miscellaneous non-classified reasons. This category also combines ships sunk due to either weather or breakup related to causes not covered by other casualty categories.

The data in Table 2.1 illustrates that oil tanker failures are going to occur and that there is significant capital loss (spilled oil, damaged or loss of vessel, restoration, and clean-up) and damage to the environment.

The data show no discernible pattern between the frequency, attributed causes, cargo losses and locations. The industry can minimize losses:

(1) by avoiding ship accidents through improved education of crew members, improved and enforced maintenance and inspection, and improved piloting and navigational technology such as GPS (Global Positioning Satellite System);

(2) by avoiding shipping-routes through ecologically sensitive areas such as the Shetland Islands, Scotland, where the *Braer* lost about 98,000 mt in a grounding in 1993, (FactsOnFile, 1993);

(3) by prevention of fires and explosions which require the removal of at least one of the three elements apparent on oil tankers (hydrocarbons, oxygen and the ignition

source, i.e. machinery hall or improper cleaning of cargo tanks). One method is the Inert Gas System (IGS). Combined with a positive pressure system in the cargo tanks, it prevents oxygen from entering the tanks, and

(4) by the development of new technologies and design standards by naval architects and structural engineers to improve the grounding and collision resistance of tankers which, in turn, will minimize oil outflow. Currently tankers are designed to assure their integrity during normal operating conditions and not extreme load events like collision and grounding. Developments include: oil outflow management, advanced double hull technology, various cargo tank arrangements and composite toughskin hulls.

2.3 Advanced Double Hull Technology and Applicable Design Standards

2.3.1 Introduction to the Advanced Double Hull Technology

Military interests in surface ship survivability and the need to enhance the structural performance of oil tankers subjected to extreme load events has led to the development of what has been termed the Advanced Double Hull Concept (ADH) or "Double Hull" Unidirectional Concept (DHUC), (Beach, 1991 and Okamoto et al., 1985). Hitachi-Zosen, Japan, with its EPOCH Mark II series, was among the first companies in the oil tanker industry who applied this new structural concept, (Okamoto et al., 1985).

The basic concept replaces the conventional hull design (a single skin structure

with a grillage system of stiffening and supporting members) with a twin skin structure, separated by unidirectional longitudinal web girders, as shown in Figure 2.1. The inherent strength of this cellular structure allows for the elimination of transverse frames and the longitudinal stiffeners. This cellular section, Figure 2.1(b), yields a larger section modulus than the conventional hull, enabling the ship to withstand higher global bending moments from seaway loading. Transverse bulkheads may also use this cellular construction.

The simplification achieved through the absence of transverse frames has the following additional benefits:

- (1) lower fabrication costs result from automated welding and simplicity of the structural arrangement;
- (2) smooth obstacle free storage tank, which leads to easier cleaning, less operational oil loss and reduced maintenance costs;
- (3) the unidirectional system also allows for dimensioning of structural components with the use of the finite element method, rather than traditional naval architecture design rules;
- (4) through increased longitudinal span large global deformations are more likely to happen, for the inner and outer hull act as an integral or sandwich construction rather than as independent plates.
- (5) the improved fatigue behaviour resulting fewer fatigue susceptible details, such as cutouts and transversely loaded welded stiffened sections.

2.3.2 The Compressive Behaviour of Advanced Double Hull Structure, - A Review of Current Research at David Taylor by Adamchak et al., (1993)

2.3.2.1 A Preliminary Look at the Relative Significance of Local Versus General Instability Failure of Double Hull Structure

One early assumption for the design of advanced double hulls (ADH) was that failure due to compressive instability would occur due to a series of local buckling of plate elements rather than overall instability of the cellular hull. The objective was to verify this assumption for typical ADH design through the use of finite element analyses (FEA). The modelling was derived from an existing cruiser class model. The investigated compartment lengths were 11.0 m and 14.6 m, and the hull spacings (distance between inner and outer hull) were 304.8 mm and 685.8 mm, respectively. For simplification, the concentrated compressive forces were made equal in magnitude at each node whereas in actual seaway these forces would vary. The bulkhead edge was fixed against vertical and transverse translation. Plate thicknesses were derived from previous experiments at *David Taylor Model Basin, Carderock Division, Naval Surface Warfare Center, Bethesda* and varied from 6.4 mm to 7.9 mm. Based on realistic values of hogging and sagging design moments, the analyses demonstrated that the primary mode of failure for an ADH structure was dictated by local plate buckling rather than by overall plate instability.

2.3.2.2 Small Scale Experimental Analysis of Double Hull Sections Using Rigid Vinyl Modelling

To complement the analytical study described in the former subsection, two rigid vinyl models were tested and compared with classical buckling solutions by Bleich (1958) and Bleich and Ramsey (1965) for critical buckling stress and local web plating instability and by column theory for global panel instability. **Model 1** and **Model 2** were designed to demonstrate local elastic web plating instability and global buckling for an advanced double hull panel, respectively. As illustrated in Figure 2.2, **Model 1** was a 1 to 9 scale (geometrically similar) to the double hull sheer strake of a cruiser. **Model 2** was 1 to 54 scale of a very thick-walled double hull panel. The critical buckling load was determined from load-deflection data, strain gauge data, and usual observations. Table 2.3 lists the test loads and the appropriate predicted buckling loads for both models. Bleich's multi-cell prediction for local web plating instability gives a test to predicted ratio of 1.11. The authors suggest that the instability pattern would be the same for a full scale ship as long as the behaviour remained elastic. Subsequent to buckling, Model 1 had exactly half the original axial stiffness. Again, for typical double hull construction, localized web plating instability governs the capacity.

2.3.2.3 The Application of Beam-on-Elastic-Foundation Analysis Methods for High Aspect Ratio Stiffened Plates

The elimination of transverse web frames in the advanced double hull concept increases the spans for longitudinal frames to a range of about 6.1-12.2 m from 2.44 m which is common naval architectural design. To keep strain values, permanent set, and buckling loads at acceptable levels the longitudinal stiffener scantlings had to be increased in size, which led to a corresponding increase in longitudinal frame weight. Side support of high aspect ratio stiffened plating is provided by the stiffened plate which behaves like a beam-on-elastic-foundation and analyzed as such to optimize spacing and sizing of the longitudinal frames for minimum weight. An approximate value for the spring constant was derived from an analysis of an equivalent single span beam loaded by uniformly spaced concentrated forces that are deflected through unit displacement. The MAESTRO-N version of the MAESTRO ship structural design program was then used to analyze three different models representing variations of a basic frigate-type hull that were designed using the beam-on-elastic-foundation analogy.

High compressive in-plane loads eliminated the foundation stiffness through plate buckling. The resulting buckling loads and bending moments were indistinguishable from those calculated according to current practices, hence a decrease in structural weight could only be achieved in panels having small compressive in-plane loading. The authors believe that side edge support might lead to an increased postbuckling strength.

2.3.3 Stranding Experiments on Double Hull Tanker Structures, by Rodd et al., (1994)

The crashworthiness in stranding and grounding accidents was examined using 1 to 4.25 scale models of tanker bottom structures of a 30,000 DWT conventional double bottom tanker with transverse and longitudinal framing and of a corresponding advanced double hull design with unidirectional longitudinal webs. In addition, a 1 to 4.8 scale model, representing a 40,000 DWT variant of an advanced double hull design incorporating curved plates was also tested. The test specimen for both the conventional double bottom and advanced double hull measured 2540 mm by 4267 mm. The curved plate model measured 2540 mm by 3048 mm. Mild steel was used for all three models. The hulls were constructed of 3.97 mm thick plates and the longitudinal webs and transverse webs of the conventional hull design were 3.18 mm thick. The test specimen was fixed against rotation and displacement. This represents the boundary restraint provided by transverse bulkheads.

Each model was impacted on a large fabricated "rock", with a geometry which consisted of a 90 degree cone with a rounded tip and a cone base of 183 mm. The radius of the rounded tip was 229 mm which corresponds to a 965 mm radius in full scale. The main reason for selecting this rock geometry was to engage the transverse and longitudinal web structure in the distortion process before rupture of the inner hull occurred.

The rock was attached to the load head of a 12 million pound testing machine and

was applied to the three models to indent each structure vertically as to cause rupture of the outer hull, and subsequently of the inner hull.

The depth of the hull designs, the ultimate load carrying capacity and the rock penetration values for each model are given in Table 2.4. Rupture of the inner and outer hull occurred for both the conventional and advanced double hull design at approximately the same load levels and penetration values, whereas the values for the curved plate design were considerably lower. According to the authors, the lower load and penetration level for the curved plate may not be less safe, as the crashworthiness of bottom structures could be defined as the vertical penetration without inner hull rupture and not as the ultimate load carrying capacity. Based on these definitions of failure all three designs are considered to be equivalent.

2.3.4 Tearing Resistance of Advanced Double Hulls, by Bracco and Wierzbicki, (1994)

Experimental and analytical studies were performed to examine the tearing resistance of advanced double hull (ADH) small scale models subjected to a wedge that is pushed in-plane through the specimen.

Six experiments with six different wedge geometries were conducted. All ADH models had the same geometry, which comprised of seven cells, with cell dimensions of 41.66 mm by 41.66 mm. The plate thickness of 0.749 mm resulted in a b/t ratio of 55.7. The wedges were fabricated from solid blocks of mild steel and were hardened to a depth

of 0.762 mm. Wedge geometries comprised of semi-angles of 30 degree and 45 degree and are listed in Table 2.5. The tip of the wedges were either sharp or cylindrical.

Fixed boundary conditions were provided along the edges of each ADH model. The wedges were attached to a load cell. Tests were terminated when the wedge cut through both plates. The primary failure mode and the test procedure are shown schematically in Figure 2.3. A description of the failure modes with the corresponding experimental mean cutting forces determined experimentally are given in Table 2.5. In the concertina failure mode (wrinkling of plate perpendicular to the direction of the load), failure occurred in what appeared to be the heat affected zone due to the welding process. The tests showed that the longitudinal and attached plating buckled and folded out of plane due to shear forces when the wedge cut into the plate. Tearing of the longitudinal plating did not occur.

An analytical study was conducted to find a closed-form solution for the cutting force. Basic assumptions included: steady-state incremental deformations; an equal thickness for the hull plating and the longitudinal girders; equal height of longitudinal girders to their spacings; and concertina tearing failure mode of the plate under the wedge tip in the central cell associated with deformation of the plating in the adjacent cells.

The results of the closed-form solution for each test were compared with the corresponding experimental mean force and error and are given in Table 2.5.

2.3.5 Applicable Standards for Double Hull Construction

2.3.5.1 Introduction

International regulations for the prevention of operational pollution have existed since 1954 as the International Convention for the Prevention of Pollution of the Sea by Oil (OILPOL), (Ryan, 1994). Amendments to these conventions published in 1971 limit the size of cargo tanks to reduce oil outflow in case of an accident. These amendments are raised due to the grounding of the *Torrey Canyon* in 1967 (124,000 mt oil spill).

For further discussion and development of regulations to reduce operational pollution due to oil and other dangerous cargoes like chemicals and noxious substances, the *International Maritime Organization (IMO)* set up the International Convention for the Prevention of Pollution from Ships (MARPOL) in 1973. Due to difficulties this convention was not established. It took a number of more catastrophic tanker accidents in the mid-seventies for the *IMO* to convene the MARPOL Convention (1978) and to adopt a protocol, known as MARPOL 73/78. Essentially, this required segregated ballast tanks (SBT's) to provide cargo tank boundary protection in lieu of mandatory double bottoms. This requirement was not implemented by the United States.

Regulations for the protection spaces of cargo tanks, which would protect cargo tanks throughout the entire length of the ship, are based on a survey by (Card 1975) in which bottom damage of 30 casualties, which occurred in U.S. waters from 1969 to 1973, were examined. It was concluded from the depth of penetration data that 27 out of the 30

tankers would not have leaked if they had been fitted with B/15 double bottoms, and 29 out of 30 tank vessels would not have leaked if fitted with a 2 m double bottom. Standards for double hull technology adopted this spacing between hulls. Rupture of the outer hull will occur in the case of a grounding or collision and the probability of penetrating the inner tank, causing oil outflow, is minimized. Energy of the impact would be absorbed through the rupture of the outer hull. Space is required between the hulls to allow room for the plastic deformation to occur and to protect the inner hull from protruding rocks or ship bows if rupture occurs. Generally the 2 m spacing is considered a minimum distance in which longitudinal stiffeners can be placed and sufficient space is left (minimum of 750 mm) for the inspection and maintenance personnel.

2.3.5.2 Standards for the Double Hull Construction of Oil Tankers (TP 11710), Canadian Coast Guard Ship Safety Branch (1993)

The *Canadian Coast Guard (CCG)* established new standards effective on July 6 1993, which will govern the design of oil tankers and their operation in Canadian waters and fishing zones. The requirements consist of the International Convention for the Prevention of Pollution from Ships (MARPOL 73/78), specifically regulations 13F and 13G and related amendments to Annex I of MARPOL 73/78 and the U.S. Oil Pollution Act of 1990 (OPA 90). The former regulations apply to new oil tankers, existing crude oil tankers of 20,000 tonnes deadweight (DWT) or greater, and existing product oil carriers of 30,000 DWT or greater. The latter regulations apply to existing crude oil

tankers less than 20,000 DWT and existing product carriers less than 30,000 DWT.

The standards require protection of the cargo tanks throughout its entire length by either ballast tanks or spaces. These protection spaces for wing tanks should be extended either for the full depth of the ship's side or from the top of the double bottom to the uppermost deck, disregarding a rounded gunwale where fitted, and arranged such that the cargo tanks are located inboard of the moulded line of the side shell plating. The width of the space must be equal to or greater than the distance w , illustrated in Figure 2.4, which is measured at any cross-section at right angles to the side shell. The distance w is defined by Equation 2.1 as,

$$w = 0.5 + \frac{\text{DWT}}{20,000} \text{ (m), or 2.0 m, whichever is less} \quad (2.1)$$

The minimum value of w is 1.0 m.

The depth of protection spaces for double bottom tanks should be the distance h between the bottom of the cargo tanks and the moulded line of the bottom shell plating at any cross-section as illustrated in Figure 2.4. The distance h is defined by Equation 2.2 as,

$$h = B/15 \text{ (m), or 2.0 m, whichever is less} \quad (2.2)$$

where, B = width of the ship. The minimum value for h is 1.0 m.

The depth or width of the protection spaces for the turn of the bilge area or the locations without a clearly defined turn of the bilge where h and w are different will be defined by the distance w at levels exceeding $1.5 h$ above the baseline as illustrated in Figure 2.4.

The standards also recommend an additional segregated ballast capacity anywhere

within the ship to reduce longitudinal hull girder bending stresses for oil tankers of 20,000 DWT and above, and product carriers of 30,000 DWT and above, and give a schedule by which single-hulled tankers have to be retrofitted to double hulls.

2.3.5.3 Comparison of Double Hull Construction Regulations

This section will explain and briefly compare the differences between national and international regulations governing the operation of existing double hull tankers and the replacement of existing single hulled tankers with double hulled tankers or with some design alternative.

As a result of the grounding of the *Exxon Valdez* in 1989, which was built in full compliance with MARPOL 73/78, the *U.S. Coast Guard (USCG)* and *IMO* initiated studies to examine the effectiveness of double hull constructions and alternative design theories, (Ryan, 1994). The independent studies revealed that double hulled tankers have superior oil outflow performance in low energy groundings compared to most of the common design concepts, like single hulled tankers, but concluded that design alternatives like the mid-deck tanker, Figure 2.5, have a more favourable oil outflow performance in more serious casualties. Since low energy collisions and groundings represent the majority of the oil tanker failures (Table 2.1), the double hull construction remained as the only design concept for oil tankers in U.S. waters. All new oil tankers built after January 1, 1994 will have to be double hulled and existing single hulled tankers calling at U.S. ports have to be either retired or converted to double hulls by 2015 at the latest. International

regulations require double hulled tankers or equivalent design alternatives. Under these rules, the last year in which a single hulled tanker can call at any port in the world where the government has acceded to MARPOL 73/78 will be 2002 in the case of a tanker with no segregated ballast tanks (SBT's), or 2025 in the case of tankers with such spaces. In adopting parts of both the *IMO* and *U.S.* regulations the Canadian standards follow these schedules. Given the economical ties in the oil business between Canada and the United States, e.g. shared waterways in the Alaskan crude oil trade, the CCG attempted to bridge the differences between the two regulations to avoid negative impacts for the Canadian tanker industry.

2.4 Advanced Unidirectional Double Hull Sandwich Plate System - Toughskin Approach

2.4.1 Introduction

Based on the Double Hull Unidirectional Concept (DHUC), the Advanced Unidirectional Double Hull Sandwich Plate System (UDHSPS) - toughskin approach, is a structural system that is being developed, that exploits the simplicity of the DHUC technology and incorporates the ultimate load carrying capacity and performance of a composite system. The concept is illustrated in Figure 1.1 and features a double hull bottom and appropriate naval architecture details. It has the same principal elements as the DHUC, an outer hull, longitudinal web stiffeners, and inner hull, but replaces the

outer hull (steel skin) by a sandwich plate composed of two thin steel plates that sandwich a structural plastic or dense foam core (Kennedy, 1994). This core will have some natural adhesion to the plates but shear connectors may have to be provided to ensure that the three components perform as a single unit. The hull side is constructed of the sandwich plate alone.

2.4.2 Hull Girder Behaviour and Structural Integrity

The UDHS^{PS} has a higher percentage of hull material oriented in the longitudinal direction than a traditional hull design. This feature, given the same structural weight, leads to a stronger hull girder under longitudinal bending. The core of the sandwich panels will carry a significant portion of the in-plane loads due to longitudinal bending. As a result the primary bending stresses will be low and buckling of the steel plates and fatigue related failures will not occur. At a local level the sandwich construction offers a number of advantages over conventional alternatives. This can be illustrated in terms of a nominal load-deflection curve for a sandwich plate, shown in Figure 2.6. In this figure the three distinct regions of load versus deflection behaviour are illustrated. The load carrying capacity at the end of the flexural response, which corresponds with the service and operational load level, is dependant on the thickness of the core and the shear bond capacity between the steel plate and the core material. The ultimate load capacity, which corresponds with the extreme or accidental load level, is dependant on the thickness of the steel plate. The foam core of the sandwich system will spread and

distribute loads over a larger area, lessening the load intensities and the chances of shear tension failure. The sandwich plate will retain its axial stiffness along the length of the ship even at large transverse deflections. The core will stiffen and strengthen the attached steel plates, preventing premature buckling.

2.4.3 Energy Absorption

The UDHSPS has a large capacity to absorb energy through large deformations before the plate ruptures. Extreme or accidental loads, induced through an impact such as a grounding or collision normal to the surface, will be carried by membrane action. Depending upon the contact area resulting from the collision, one or more panels may develop membrane action as longitudinal web frames collapse. As illustrated in Figure 2.6, the panels will be able to deflect inelastically to values of approximately 20 % of their span (panel short axis) prior to rupture. Additional energy will be absorbed by densification of the core material. The ability to predict the load versus deflection characteristics up to rupture has been developed, (Kennedy, 1987 and Bond, 1995) and will be introduced in Chapter 3.

2.4.4 Fracture Toughness

Modern oil tankers demand a high level of toughness from their hull materials. The outer hull steel is subjected to low temperatures and thus is the critical component

with respect to fracture toughness. As the plate thicknesses increases it becomes increasingly difficult to maintain toughness in the plate at the plate manufacturing level and during fabrication due to welding heat inputs. Thus thin outer hull plate is sought. With the UDHSPS the outer hull plating is thin, with high toughness levels, as the total thickness of the outer and inner hull plates (in the sandwich alone as in the side shell or in the full double bottom area with essentially three plates) can be distributed arbitrarily. The temperature gradient through the thickness of the sandwich core exposes the inner plates to temperatures much less than the climatic temperatures, lessening fracture toughness and susceptibility of the inner plate.

2.4.5 Construction and Reparability

Normal fabrication procedures can be followed in the construction of a UDHSPS vessel. Dense foam is injected into the cavity of the steel plates to form the core. In the double bottom and shear strake box girder regions, the separation between the inner hull plate and the outer sandwich plates will be such to allow access for fabrication and welding. Damaged sections can be cut free by cutting disks (cold process). The dense foam could then be removed through shotblasting. The section would be replaced as per normal construction practices.

The following sections review research conducted to predict the behaviour of steel and composite elements and on the application of structural plastics or polyurethane foams for the core material of the composite sandwich plate.

2.5 Behaviour of Steel and Composite Sandwich Plate Elements

2.5.1 Tanker Structural Analysis for Minor Collisions, by McDermott et al., (1974)

Based on single stiffened hull plates module tests and inspection of collision damaged ships, a plastic analysis procedure was developed and applied to a limited number of idealized ship collisions. The model was developed with the purpose of protecting the cargo of single and double hulled vessels during a low energy (minor) collision, the damage phase in which the cargo tank remains still intact.

For colliding ships the bow of the striking ship can be considered as infinitely stiff, hence, the collision energy is assumed to be absorbed by the struck ship through plastic deformation. Initially, the plastic analysis included:

- (i) longitudinal plastic bending of the stiffened hull plating;
- (ii) plastic membrane tension in the stiffened hull plating and the deck;
- (iii) yielding or buckling of the web framing (transverse bulkheads).

The plastic energy due to bending was relatively small, and was subsequently neglected in the calculation of the potential plastic energy available. The plastic energy absorbed by the hull in membrane action accounted for 69 to 94 percent of the total energy absorbed. It was concluded that the plastic energy absorbed during the hull membrane tension phase could be maximized by maximizing the extent of the damage (area over which membrane action occurred), the tensile strength, and ductility of the

steel.

An experimental study consisting of ten 1 to 5 scale tests was conducted to verify the analytical model. The L/d ratio for all specimens was 151.5. Six specimens consisted of an angle shape longitudinal stiffener welded to a hull plate while the others used the hull plate alone. The concentrated loads were gradually applied through a loading nose at various locations along the length. As predicted, the test specimens had little bending capacity. The slope of the load-deflection curve increased significantly as plastic action occurred. The stiffeners tripped at midspan during the membrane tension phase and eventually rupture occurred in the plate at either the support or around the load point.

Based on this test data, an analytical model was developed to describe the load-deflection curve of the stiffened plate. A mathematical model that relates the bend angle to the maximum strain under sharp load was also developed.

2.5.2 Designing Tanker Structures for Collisions and Grounding Resistance, Edinberg et al., (1990)

Based on the work introduced by McDermott et al., (1974), a Low Energy Collision Analysis Method (LECAM) was developed. Low energy collisions are defined as collisions in which rupture of the cargo tank will not occur, whereas high energy collisions involve massive tearing and possible cargo spillage or oil outflow (oil tankers).

Structures such as nuclear powered vessels must be designed to resist all strikes, collisions or groundings, (Woisin, 1979). These vessels tend to be so stiff that the energy

resulting from a collision is essentially absorbed by the striking object, e.g. the ship's bow. To withstand such an impact the struck ship must be very massive and heavy and would prove to be impractical for tanker design. In contrast, tankers must be collision absorbing structures and can best absorb energy by plastic deformation (large deformation) and membrane tension action. Generally for collisions, membrane tension action develops as a result of right-angle striking by another vessel. For grounding, the striking is parallel to the shell plate. This leads to considerable tearing, shearing, and crushing of the plating. Membrane tension action can only be developed if rupture does not occur. Once a ship has come to rest, it may be stranded, and exposed to wave action and pounding against rock. These additional loads must be accounted for.

LECAM is a static analysis that calculates the total energy absorbed for each individual structural element and for the vessel as a whole. The analysis assumes that the damage length does not extend beyond adjacent transverse bulkheads. Studies on different tanker structures involved in collisions and grounding show that from the four following types of energy absorption mechanisms; elastic vibration of the struck ship, local elastic deformation of the structure of the struck ship, plastic deformations of the structure of the struck ship, and rigid body motion of the struck ship, the latter two account for up to 90 % of the total kinetic energy of the collision. Based on observations of actual collision damage and laboratory tests, the energy absorption due to mechanisms such as tearing of the steel plate is neglected, as it is relatively small. The method is fast, as compared to a finite element analysis, and can be readily used:

- (i) to evaluate designs to determine the collision resistance,

- (ii) to establish design criteria and guidelines (typical collisions and groundings would be evaluated and compared with the spillage protection provided),
- (iii) to optimize the arrangement of structural components to minimize cost, and
- (iv) to establish operating criteria (safe speeds within harbours or other confined areas).

2.5.3 Behaviour of Transversely Loaded Continuous Steel-Concrete Composite Plates, by Kennedy and Cheng (1987)

The authors conducted both an experimental and an analytical study to determine the behaviour of transversely loaded continuous steel-concrete-steel composite plates for use in offshore structures to ward off ice.

The experimental program consisted of five one-span continuous composite plate elements without mechanical shear connectors and one comprised of two steel plates, one lying on top of the other. The clear distance between support bulkheads was 2850 mm. The section depth, span to depth ratios between 15 to 25, and steel plate thicknesses between 3.18 mm and 6.35 mm were varied to determine their influence on flexural and membrane behaviour. Ancillary tests were conducted to determine material properties and static and dynamic coefficients of friction between the steel and concrete.

Load was applied through knife-edge-roller assemblies at four evenly spaced points along the clear span.

The behaviour of composite steel-concrete-steel plates (without mechanical shear connectors) can be subdivided into three distinct regions: flexural, flexural membrane and

membrane (elastic and inelastic) action. The flexural behaviour was dominated by progressive cracking and crushing of the concrete with the formation of plastic hinges at the supports, and then at midspan. Membrane action developed with increasing deflection and without loss of flexural capacity. Full membrane action was realized when the load-deflection curve exhibited an increased and constant slope. The deflection at centre span was approximately 30% of the final deflection at failure. With increased load, the concrete core broke down into a series of blocks with a fanlike crack pattern at either end where significant curvature occurred. Cracks were the most severe at the supports. Shear tension failure of the steel plates occurred at one or both supports. The deflection at centre span and the deflected shape were essentially the same for all tests. The ultimate load was found to be a direct function of the total plate thickness and steel strength.

Analytical models were developed to predict and verify the flexural and membrane behaviour. Failure criteria were developed to predict the ultimate strength.

Specific aspects of applying flexible composite sandwich plate systems to marine structures are discussed. These include: proportioning of the section to obtain the required flexural and ultimate capacity, determining of boundary stiffnesses, energy absorption capacity, construction, fracture toughness, punching failure and fatigue characteristics.

2.6 Structural Plastics and Sandwich Construction

Steel-structural plastic-steel plate sandwich construction yields an ideal outer hull for oil tankers and other ocean going vessels. This construction gives strong weldable

tough wearing surfaces, a core that bonds to the steel without special mechanical interconnectors, simplicity in design and construction, reduced fabrication costs (less cutting, fitting, and welding of stiffeners), increased in-plane strength and stability, comparable flexural strength, superior impact resistance and energy absorption capacity for both low energy and high energy collisions. The following general characteristics of plastics must be reviewed in the selection of a core material:

- (1) tensile and compressive strength (generally all plastics exhibit non-linear behaviour)
- (2) ductility (elastomeric vs. rigid)
- (3) variation of material properties with time (creep, reduction in strength, degradation in hostile environments)
- (4) isotropic or anisotropic behaviour
- (5) hardness and density
- (6) constructability
- (7) fire resistance
- (8) thermal characteristics

After contacting numerous chemical companies, two possible polyurethane materials were considered to serve as the core material, one being a rigid foam and the other, an elastomer. The rigid foam, consisting of *XUS 15315.00L Experimental Polyol* and *Papi 27 Polymeric MDI*, was developed by Dow Chemical and the *WUC 5079R / WUT 3253*, which is an elastomer was developed by the BASF Corporation. The general characteristics of these are given in Table 2.6. Based on availability and constructability the rigid foam was selected for the core material in this study.

2.7 Summary

During the last couple of decades, governments, industry and researchers throughout the world have worked to define and improve marine regulations, to improve navigation systems, to develop cargo tank variations that will minimize oil outflows, and to improve the structural integrity of a tanker vessel through new design concepts (DHUC) to prevent oil tanker accidents, loss, and the corresponding environmental damage. Notwithstanding these efforts, statistical data indicates that there are at least two major spills per year. This may diminish as older vessels are retired and newer vessels with double bottoms and complete double hulls are commissioned.

The next step in development of impact resistant oil tankers is the focus of this thesis and pertains to the development of an advanced unidirectional double hull sandwich plate system. This system, shown in Figure 1.1, features a composite outer hull (steel-structural plastic-steel sandwich) which has simplified geometry, comparable flexural and in-plane load carrying characteristics and the capacity to absorb large impact loads (kinetic energy) from groundings and collisions through global plastification of the hull.

The following chapters will describe the analysis and experimental program conducted to examine the behaviour and feasibility of a steel-polyurethane foam-steel sandwich plate.

Table 2.1 50 Major Oil Spills from Tankers and Combined Carriers 1960-1989
(after National Research Council (U.S.)(1991))

Rank	Name	Tanker Size (DWT)	Spillage (mt x 10 ³)	Location	Date	Cause				
						CO ¹	GRN ²	F/E ³	SH ⁴	N/K ⁵
1	ATLANTIC EMPRESS	292,666	257	West Indies	1979	X				
2	CASTILLO DE BELLVER	271,540	239	South Africa	1983			X		
3	AMOCO CADIZ	237,439	221	France, Atlantic	1978				X	
4	ODYSSEY	138,392	132	Mid Atlantic	1988			X		
5	TORREY CANYON	121,000	124	U.K., Channel	1967		X			
6	SEA STAR	120,300	123	Gulf of Oman	1972	X				
7	HAWAIIAN PATRIOT	101,038	101	Hawaiian Islands	1977				X	
8	INDEPENDENTA	152,408	95	Turkey, Bosphorus	1979	X				
9	URQUIOLA	111,225	91	Spain, North Coast	1976		X			
10	IRENES SERENADE	105,460	82	Greece	1980			X		
11	KHARK 5	284,632	76	Morocco, Medit.	1989			X		
12	NOVA	239,435	68	Gulf of Iran	1985	X				
13	WAFRA	68,600	62	South Africa	1971		X			

Table 2.1 (continued)

Rank	Name	Tanker Size (DWT)	Spillage (mt x 10 ³)	Location	Date	Cause				
						CO ¹	GRN ²	F/E ³	SH ⁴	N/K ⁵
14	EPIC COLOCOTRONIS	64,000	58	West Indies	1975		X			
15	SINCLAIR PETROLORE	56,000	57	Brazil	1960			X		
16	YUYO MARU NO 10	52,836	52	Japan	1974	X				
17	ASSIMI	59,032	50	Oman	1983			X		
18	ANDROS PATRIA	122,173	48	Spain, North Coast	1978			X		
19	WORLD GLORY	45,000	46	South Africa	1968				X	
20	BRITISH AMBASSADOR	44,929	46	Japan	1976				X	
21	METULA	206,719	45	Chile	1975		X			
22	PERICLES G.C.	59,096	44	Qatar	1983			X		
23	MANDOIL II	45,000	41	USA, West Coast	1969	X				
24	JAKOB MAERSK	88,000	41	Portugal	1975		X			
25	BURMAH AGATE	62,663	41	USA, Gulf	1979	X				
26	J. ANTONIO LAVALLEJA	131,663	38	Algeria	1970		X			
27	NAPIER	38,561	37	Chile	1973		X			
28	EXXON VALDEZ	214,861	36	USA, Alaska	1989		X			

Table 2.1 (continued)

Rank	Name	Tanker Size (DWT)	Spillage (mt x 10 ³)	Location	Date	Cause				
						CO ¹	GRN ²	F/E ³	SH ⁴	N/K ⁵
29	CORINTHOS	56,882	36	USA, Delaware R.	1975	X				
30	TRADER	35,000	36	Greece	1972				X	
31	ST. PETER	34,730	33	Ecuador	1976			X		
32	GINO	48,760	32	France, Atlantic	1979	X				
33	GOLDEN DRAKE	30,004	32	Bermuda	1972			X		
34	IONNIS ANGELICOUSSIS	68,106	32	Angola	1979			X		
35	CHRYSSI	29,653	32 ⁶	Bermuda	1970				X	
36	IRENES CHALLENGE	34,884	31	Pacific Ocean	1977				X	
37	ARGO MERCHANT	28,691	28	USA, East Coast	1976		X			
38	HEIMVARD	55,000	31	Japan	1965	X				
39	PEGASUS	-	25 ⁶	USA, East Coast	1968					X
40	PACOEAN	30,016	31	North West Pacific	1969				X	
41	TEXACO OKLAHOMA	35,072	29	USA, East Coast	1971				X	
42	SCORPIO	42,000	31	Mexico, East Coast	1976		X			
43	ELLEN CONWAY	47,566	31	Algeria	1975		X			

Table 2.1 (continued)

Rank	Name	Tanker Size (DWT)	Spillage (mt x 10 ³)	Location	Date	Cause				
						CO ¹	GRN ²	F/E ³	S/H ⁴	N/K ⁵
44	CARIBBEAN SEA	30,661	30	East Pacific Ocean	1976				X	
45	CRETAN STAR	30,372	27	India, West Coast	1975					X
46	GRAND ZENITH	29,930	26	South Africa	1975				X	
47	ATHENIAN VENTURE	31,016	26	Canada, Nfld.	1988			X		
48	VENCIL	330,954	26	South Africa	1977	X				
49	ARAGON	238,959	24	Madeira	1989				X	
50	OCEAN EAGLE	18,824	21	Puerto Rico	1968		X			
Total, Number Accidents:						11	13	12	12	2
Total, Spillage lost (mt x 10 ³):						792	693	851	653	52

Table 2.2 Oil Spills from Tankers and Combined Carriers 1990-1994,
(FactsonFile (1990-1994))

Rank	Name	Tanker Size (DWT)	Spillage (mt x 10 ³)	Location	Date	Cause				
						CO ¹	GRN ²	F/E ³	S/H ⁴	N/K ⁵
1	HAVEN	-	150	Gulf of Genoa, Italy	1991			X		
2	MEGA BORG	-	145	Gulf of Mexico, Texas	1990			X		
3	BRAER	-	98	Shetland Islands, Scotland	1993		X			
4	AEGAN SEA	-	81	La Coruna, Spain	1992		X			
5	AGIP ABRUZZO	-	70	Livorino Harbor, Italy	1991	X				
6	MAERSK NAVIGATOR	-	25	Strait of Malacca	1993	X				
7	KIRKI	-	11	Indian Ocean, Western Australia	1991			X		
8	MORRIS J. BERMAN	-	3	San Juan, Puerto Rico	1994	X				
9	SHINOUSA	-	2	Galveston, Texas	1990	X				
10	AMERICAN TRADER	-	2	Huntington Beach, California	1990				X	

Table 2.2 (continued)

Rank	Name	Tanker Size (DWT)	Spillage (mt x 10 ³)	Location	Date	Cause				
						CO ¹	GRN ²	F/E ³	S/H ⁴	N/K ⁵
11	B.T. NAUTILIUS	-	2	New York		X				
Total, Number Accidents:						4	3	3	1	-
Total, Spillage lost (mt x 10 ³):						100	181	306	2	-

¹CO - Collision

²GRN - Grounding/Stranding

³F/E - Fire/Explosion

⁴S/H - Structural/Machinery/Others

⁵N/K - Not known

⁶Spillage data questionable

Table 2.3 **Summary of Analytic Predictions and Experimental Results**
 (after Adamchak et al. (1993))

Model	Test (lbs)	Predicted (lbs)	tested/predicted ratio
Model 1			
Double Hull Local	200	180	1.11
Web Plating Instability			
Model 2			
Double Hull Global	370	250 Fixed-Pinned	1.48
Panel Instability		500 Fixed-Fixed	0.74

Table 2.4 **Test Results for Stranding Experiments**
(after Rodd et al. (1994))

	Rupture of outer hull		Rupture of inner hull		Double hull height (mm)
	Load (kN)	Penetration (mm)	Load (kN)	Penetration (mm)	
Conventional Hull Design	823	157.5	3336	756.9	470
Advanced Double Hull Design	756	157.5	3203	772.2	470
Curved Plate Design	529	203.8	1196	663.0	457

Note: Due to different scaling factors the penetration values for the curved plate model were multiplied by a factor of 1.13 and the load values by a factor of 1.28 to be comparable with the two other test data.

Table 2.5 Test Results for Cutting Experiments
(after Bracco and Wierzbicki (1994))

Test No.	Wedge geometry (semi-angle)	Primary Failure Modes	Experimental Mean Force F_m (N)	Closed-form Solution F (N)	Error (%)
1	45°, sharp tip	Cutting of both plates under the wedge tip	22,792	24,142	+5.9
2	45°, tip radius= 9.525 mm	Concertina tearing of both plates under the wedge tip	23,299	24,142	+3.6
3	30°, sharp tip	Cutting of one plate and concertina tearing of the other plate under the wedge tip	19,128	19,136	+0.04
4	30°, tip radius= 9.525 mm	Concertina tearing of both plates under the wedge tip	18,814	19,136	+1.7
5	cylindrical	Cutting of one plate and concertina tearing of the other plate under the wedge tip. There was fracture of the plate being cut running ahead of the wedge face causing drop-offs of the force level. The fracture would run the side and stop once it encountered a longitudinal. Once the wedge caught up to the crack tip, the force level would again increase and cutting recommence.	27,434	24,142	-12.0

Table 2.5 (continued)

Test No.	Wedge geometry (semi-angle)	Primary Failure Modes	Experimental Mean Force F_m (N)	Closed-form Solution F (N)	Error (%)
6	19.05 mm, tip radius=9.525 mm	Concertina tearing only due to the wedge width being narrower than the spacing between longitudinals. This test was used to get a force level for concertina tearing only.	-	-	-

Note: The error was calculated by $\text{error}(\%) = (100) * (F - F_m) / F_m$

Table 2.6 Characteristics for selected Polyurethane Materials

Characteristics		WUC 50792R / WUC 3253T elastomer, BASF Corp.	Rigid foam Dow Chemical		
Specific Gravity		resin: 1.00 g/cm ³ at 25° C Iso: 1.15 g/cm ³ at 25° C	free rise: 0.054 g/cm ³ molded: 0.059 g/cm ³		
		-	0.032 g/cm ³	0.16 g/cm ³	0.4 g/cm ³
Strength	Compression (kPa)	-	246	2531	8789
	Tensile (kPa)	40261	316	3796	11591
	Shear (kPa)	-	197	1758	3164
	Hardness, ASTM D2240, (Shore A)	92-93	-		
Elongation, ASTM D412, (%)		400	-		
Graves, ASTM D1004, Die C Tear, (pli)		680	-		
Brashore Rebound, ASTM D2632, (%)		31	-		
Viscosity		Resin: 405 cps at 30° C Iso: 1950 cps at 25° C	1100 cps at 21° C		
Component Temperatures for processing		Resin: 30 - 43.3° C Iso: 30 - 43.3° C	24-38° C

(a) Conventional Hull

(b) Double Hull

Figure 2.1 Conventional and Advanced Double Hull Design
(after Beach, 1991)

(a) Model 1

(b) Model 2

Figure 2.2 Anticipated Instability Modes
(after Adamchak et al., 1993)

Figure 2.3 Advanced Double Hull Deformation Model
(after Bracco and Wierzbicki, 1994)

Figure 2.4 Cargo Tank Boundary Lines
(after Canadian Coast Guard, 1993)

Figure 2.5 Tanker Cross-Sections
(after Ryan, 1994)

Figure 2.6 Load vs. Deflection Behaviour of Sandwich Plate

Chapter 3

Experimental Program

3.1 General

An experimental program was undertaken to examine and verify the behaviour of a typical sandwich plate used for the outside hull in the UDHSPS system, as illustrated in Figure 1.1. The test plate measured 5000 mm by 2600 mm in plan and is composed of two steel plates with thicknesses of 12.70 mm and a polyurethane foam core with a depth of 100 mm. Shop drawings of the test specimen are shown in Figures 3.1 (a) and (b). Surrounding the plate is a heavily stiffened border (marks the perimeter of the test plate) which transmits the forces from the plate to the horizontal reaction frame and permits welding and cutting of the test plate to and from the frame without igniting the foam core. Seven 6 mm internal cell plates (stitch-welded to the outer plate) form eight cells. These cell plates were installed to control the spread and rise of the polyurethane foam as it was injected. The supporting horizontal reaction frame represents the stiffness

of the surrounding ship structure and provides the in-plane restraint to the horizontal tensile forces developed through membrane action. This boundary condition is the same as that applied by other researchers in this area. Transverse load was applied to the test specimen, monotonically to failure, through five evenly spaced point loads over a length of 2400 mm centred about midpanel.

The steel section illustrated in Figure 3.1 (a) and (b) was fabricated by the CANAM Group, Boucherville, Quebec. The polyurethane foam core material, technical data, material testing of the core material and in kind support was provided by the DOW CHEMICAL Company, Sarnia, Ontario. A VALLEY INSULATION, Ottawa, was retained to inject the polyurethane foam into place.

3.2 Preliminary Analysis and Design

Based on Ratzlaff (1985, 1986), Kennedy (1987), and Bond (1995) the central portion of the composite plate with an aspect ratio of approximately 1.92 could be analyzed as a wide beam spanning in the transverse direction a distance of 2600 mm and with a width equal to an effective length of the applied load of 2750 mm. A simple plastic hinge analysis was conducted to determine the possible flexural capacity of a composite plate with full shear connection between the plates and the core, (Benjamin, 1969; Allan 1969). The cross-section shown in Figure 3.2 has a flexural stiffness equal to

$$D = \frac{E_f B (H^3 - t_c^3)}{12} + \frac{E_c B t_c^3}{12}, \quad (3.1)$$

where D = flexural stiffness of composite plate;

E_f = elastic modulus of composite face;

E_c = elastic modulus of composite core;

B = effective width of composite plate;

H = overall depth of composite plate;

t_c = depth of composite core;

t_f = thickness of composite face.

The second term is relatively small and can be neglected. For example, the ratio of the second term to the first for the composite section of this thesis is 9.6×10^{-6} . The corresponding deflection to the development of the first plastic hinge for a continuous beam with a single point load at midspan is

$$\delta = \frac{Pl^3}{192D}, \quad (3.2)$$

where δ = total deflection;

P = total applied load;

l = plate span.

The membrane behaviour of the same section was predicted using the computer program IMASP, "Inelastic Membrane Analysis of Steel Plates", (Bond, 1995). The program predicts the load deflection behaviour and failure of sandwich plates for sections of constant depth, varying in-plane boundary stiffnesses, varying material properties

including heat affected zones and any arrangement of transverse loads.

Based on these analyses, the load and deflection for the development of the first plastic hinge and failure were predicted for plates of varying core thicknesses, depths, and plate thicknesses. The results are summarized in Table 3.1. Based on these results, the capacity of the reaction frame, the displacement and load capacity of the actuators, and the availability of the plate material, two 12.70 mm plates with a core thickness of 100 mm were selected for the test specimen.

As described in Chapter 2, a two component polyurethane foam, XUS 15315.00L Experimental Polyol and Papi 27 Polymeric MDI (DOW CHEMICAL), was selected for the core material. Discussions with a local firm (A VALLEY INSULATION) with experience in non-structural sandwich panel construction led to the decision to subdivide the cavity into eight 550 mm wide cells spanning in the transverse direction to obtain full coverage with the injected foam. The foam was to be injected after the test specimen was welded into place to prevent burning and off-gases which may result from the heat of the welding process. A stiffened edge with cover plates for the top and a continuous bottom plate were provided to facilitate the welding and subsequent removal of the test specimen to and from the reaction frame, so that the foam material would not be burned. In addition a series of 1/2" x 2 1/8" Nelson shear studs, three per compartment, were welded at the ends and along the sides of the end cells to provide in-plane continuity of the core material with the reaction frame. The fabrication drawings for the test specimen are shown in Figure 3.1 (a) and (b).

3.3 Test set-up

The test set-up consists of a horizontal reaction frame, high strength steel tie-down rods, a stiffened plate girder, hydraulic actuators, electronic instrumentation and a data acquisition system as shown in Figure 3.3 by the schematic diagram of a cross-sectional view of the test specimen in the test apparatus. The data acquisition system, instrumentation and hydraulic pump have not been illustrated.

3.3.1 Loading Apparatus

The transverse loads were applied to the test specimen through a series of five hydraulic actuators centred along the long axis shown in Figure 3.4. The numbers indicate the number associated with the hydraulic actuator. On top of each hydraulic actuator sits a knife-edge-roller assembly (refer to Figure 3.3) which allows rotation and lateral displacement along the length with respect to the actuator, while maintaining the vertical orientation of the load. Copper plates were placed on top of the knife edge load plate to soften the edge to reduce the possibility of local shearing and tearing at low load levels. The hydraulic actuators (3 - 4500kN (no's 1,2,3) and 2 - 1800 kN (no's 4,5)) are activated through an air on hydraulic oil system, consisting of an air driven pump, a hydraulic oil reservoir, manifolds and solenoid valves. The valves are controlled by the data acquisition program. Reactions from the actuators are transmitted through a stiffened

plate girder to the strong floor of the laboratory.

3.3.2 Horizontal Reaction Frame

Figure 3.5 shows a plan sectional view of the test specimen welded into the 40 t horizontal reaction frame. The frame represents the stiffness of the surrounding ship structure, providing rotational and translational restraint along its edges and in-plane restraint to the horizontal tensile forces developed through membrane action in the test plate.

3.3.3 Instrumentation and Measurements

Applied loads, displacements and strains were measured and reported throughout the test using pressure transducers, linear variable displacement, and electrical resistance gauges, respectively. Pairs of strain gauge transformers were placed orthogonally at each location on the outside of the bottom (outside hull plate) and top plate (inside hull plate) shown in Figures 3.6 and 3.7, respectively. Even gauge numbers are associated with gauges oriented parallel to the long direction of the plate.

Type EP-08-250BG-120 micro-measurement high elongation strain gauges, with a resistance of 120 ($\pm 0.15\%$) ohms, a gauge length of 8 mm and a gauge factor of 2.055 $\pm 0.5\%$ at 24° C were used. The gauges were bonded to the steel plates with A-12 ADHESIVE, a two component adhesive which allows the elongation of the strain gauges

for up to 200,000 μ s.

The transverse displacements were measured on the top plate using 30 linear variable differential transformers (LVDTs). The locations and numbers of the LVDTs are shown in Figure 3.8. The LVDTs were held in position by a supportive framework of angles. Five additional LVDTs were installed at the hydraulic jacks in order to measure their extension during the loading process.

Pressure transducers, mounted in the pressure line at the actuator, were used to monitor the applied loads.

After each load step the strains, deflections and loads were recorded using a Hewlett Packard data acquisition system and the control program LABCON. The type of measurement, the instrumentation used, purpose, location and frequency of measurements range and accuracy are given in Tables 3.2 and 3.3.

3.4 Installation, Alignment and Instrumentation of Test Specimen

The installation, alignment and instrumentation of the test specimen consisted of the following steps:

1. the plate was fitted to the horizontal reaction frame;
2. strain gauges were installed on the bottom side of the test specimen;
3. the panel was lifted into position and welded into place;
4. four 3/8" diameter holes per cell, as shown in Figure 3.9, were drilled in the top plate to accommodate the injection of the core material;

5. the core material was injected through each hole, starting with the hole in the lower right hand corner of Figure 3.9 and progressing along each cell and across the plate as indicated by the arrows. The number adjacent to each hole represents the number of pump strokes of injected material, which in turn is equivalent to 0.003 m^3 volume of material per stroke;
6. the hydraulic actuators were connected to the hydraulic pump system;
7. the remaining strain gauges were installed on the top plate;
8. the framework of angles to support the LVDTs was installed and attached to the horizontal reaction frame;
9. the LVDTs were installed;
10. all strain gauge wires, LVDT cables and pressure transducers were connected to the data acquisition system using strain gauge bridge and T-bridge cards;
11. the test set up and data files were generated using LABCON;
12. the power supply was adjusted and the hydraulic actuators extended to bring the knife-edge-rollers into contact with the test panel; Note, the hydraulic actuators and knife-edge-rollers were already assembled and positioned;
13. the instrumentation was calibrated and malfunctioning instrumentation was fixed;
14. a small preload was applied to confirm the functionality of the test

set-up;

15. the specimen was unloaded and the initial (zero) readings were recorded;

3.5 Injection of Core Material

The polyurethane core material is a free rising foam that was injected into the cavity of each cell of the steel shell of the test plate. For a density of 0.0472 g/cm^3 each stroke of the machine pumps 0.003 m^3 of material, so that 43 strokes would fill the 0.129 m^3 cavity, as compared to an average of 46.5 strokes which was injected. The viscosity of the resin is a logarithmic function of temperature and was 1100 cps at 21 degrees C, 900 cps at 27 degrees C, and 600 cps at 32 degrees C. To ensure that the cells would be filled with a quality foam (no discontinuities, air bubbles, poor mix, or voids) a test box consisting of two cells, with the same dimensions as the cell of the test plate was built of wood. The wooden cells were lined with polyethylene sheets and injected with foam. The cells were opened and the quality of coverage and foam were evaluated and the process revised until an acceptable core material was produced. This trial and error procedure was required as a process had not yet been devised to inject foam into a cavity oriented horizontally.

A summary of the trial and error process follows:

(a) trial type 1:

Injection of the foam material through a 3/8" diameter hose attached to the nozzle. The hose ran the full length of the cell and was inserted through one end. The hose was slowly withdrawn from the cell as material was injected. The quantity was established on achieving a certain density and knowing the volume of material being injected per stroke of the pump. This process led to a poor quality foam with significant voids (poor coverage) as the material mixed in the nozzle began to react and cure as it was pushed through the 2.50 m long hose. Subsequent trials revealed that any hose length led to a poor quality mix. The material must be injected into the cell directly from the nozzle. The holes at the ends of the cells were sealed and four new holes were drilled in the top plate.

(b) trial type 2:

Injection of the foam was accomplished by injecting equal quantities, approximately 12 strokes, through each hole. The temperature of the resin played a major role in achieving a good quality foam. The viscosity of the resin varies logarithmically with respect to temperature with increasing temperature leading to a decrease in viscosity. According to the available product information, the resin could be mixed and injected at temperatures as low as 24° C. With the equipment available, the resin had to be heated to 34° C to get a good mix. Below this temperature, the material was too viscous, and the compressor in use was not able to keep up the air pressure needed to pump the material uniformly. Good coverage and quality foam was achieved by injecting 14, 12, 12, and 12

strokes of material into each hole. The last two samples from the trial cells were used for the material tests on the foam. Figure 3.9 shows the pattern selected to inject the foam into the eight cells of the test plate. The numbers adjacent to the holes represent the number of strokes of the pump injecting material.

3.6 Test Procedure

The set-up of three sets of load actuators, the 2-1800 kN exterior actuators, the 2-4500 kN interior actuators and the central 4500 kN actuator was designed for individual control in load and displacement for each set. Load was applied monotonically in total load increments of approximately 100 kN up to a total load of 7000 kN. Load beyond this was applied through the 4500 kN actuators only. The 1800 kN actuators were kept at a constant load of 1400 kN.

At the end of each load step measurements (strain gauges, LVDTs, pressure transducers) were taken electronically and recorded through the data acquisition system. Observations were made after every load step and any phenomenon, noises, deflected shape, and progression of failures were recorded. Photographs were taken periodically to complement these recordings.

After failure had occurred, the test panel was unloaded. The residual deformations were measured and recorded. A photographic record was made.

3.7 Ancillary Tests

3.7.1 Steel Strength

Ten tension coupons from both the longitudinal and transverse direction were taken from a 1000 mm x 500 mm piece of plate which was flame cut from the parent plate used as the test panel (refer to Figure 3.10). Tension coupons were cut and milled from this plate and tested according to ASTM E8M-88 (ASTM 1988) to determine the characteristics of the steel plate. The tension coupon numbers were stamped on each piece, as indicated in Figure 3.10, and were milled to the dimensions shown.

Ten measurements of width and thickness along the gauge length were taken to determine the cross-sectional dimensions. Single strain gauges, aligned along the length of the coupon, were mounted on both sides of 14 coupons, seven from each direction. The remaining six coupons, three from each direction, were instrumented with pairs of strain gauges on each side to measure both longitudinal and transverse strains. From this data, Poisson's ratio was calculated. Between 15 and 20 electronic readings were taken in the elastic region and where possible three static yield values were observed. Large strains (strains greater than 20,000 μs) were measured using dividers and a scale over the 200 mm gauge length. The tests were performed in accordance with ASTM Designation E8M-88 (ASTM, 1988) in a 1780 kN Tinius Olsen testing machine.

3.7.2 Polyurethane Foam Strength

Compression, tension and shear tests were conducted in accordance with ASTM standards at and by the Dow Chemical laboratories to determine the material characteristics of the polyurethane foam core. The polyurethane samples were taken from a 300 mm x 300 mm x 100 mm block cut from the foam core extracted from one of the compartments of the test injection box. The sample was injected in the same sequence, pattern and temperature as the foam for the test specimen.

3.7.2.1 Compression Test

Four specimens, with cross-sectional parameters of 127 mm by 101.6 mm and a thickness of 50.8 mm, shown in Figure 3.11 (a), were tested in accordance to Test D - Constant Deflection Compression Test ASTM D3574-86 (ASTM 1986). Three specimens were tested until the foam under the load point tore away from the rest of the sample to determine the compressive behaviour and strength of the polyurethane foam. The fourth specimen was tested beyond this definition to determine the ultimate compressive behaviour. The testing rate was 20 mm/min. and was applied transversely to the specimen through a flat disk load point with a diameter of 88.9 mm. Load and head displacement of the Instron testing machine were automatically recorded 20 times per second throughout the test.

3.7.2.2 Tension Test

Two specimens with a width of 12.70 mm and a thickness of 11.387 mm, shown in Figure 3.11 (b), were tested according to Test E-Tension, ASTM D3574-86 (ASTM 1986) to determine the tensile strength of the polyurethane foam. The specimens were subjected to a tension force at a speed of 5 mm/min.. Load and head displacement of the Instron testing machine were recorded automatically at 20 times per second throughout the test.

6.7.2.3 Shear Test

Three test specimens with a diameter of 25.15 mm and thicknesses of 4.85, 3.89, and 4.22 mm, shown in Figure 3.11 (c), were tested according to the Standard Method for Shear Strength of Plastics by Punch Tool ASTM D732-93 (ASTM 1993) to determine the shear strength of the polyurethane foam. The load was applied transversely through a punch tool with a diameter of 9.5 mm at a speed of 1.25 mm/min. until the moving portion of the specimen had completely cleared the stationary part.

Table 3.1 Analytical Results for Design of Test Specimen

core thickness (t_c) [mm]	plate thickness (t_p) [mm]	P_{pl} [kN]	δ_{pl} [mm]	P_{ult} [kN]	δ_{ult} [mm]	P_{ult} HAZ+weld [kN]	δ_{ult} HAZ+weld [mm]
50	6.35	1063	16.2	11400	602	4877	316
100	6.35	2005	8.6	11400	600	4905	314
50	9.53	1685	15.3	17200	605	7330	315
100	9.53	3100	8.4	17200	603	7364	314
0	11.11	397	60.0	20394	620	-	-
50	11.11	2114	14.7	19500	604	8157	309
100	11.11	3830	8.2	19500	602	8195	308

Table 3.2 Type of Measurement and Instrumentation

Instrumentation	Location/Description	Purpose	Frequency of Measurement
1. Applied Loads			
Pressure transducers	On the input pressure line of each actuator.	Calibrate with actuator to give the applied load per actuator.	Discrete readings taken at the end of each load increment.
2. Steel Strains			
Electrical Resistance Gauges	At locations indicated in Figures 3.6 and 3.7 for the bottom and top of plates; measure longitudinal and transverse strains.	Provides strain distributions along the surface of each plate.	Discrete readings taken at the end of each load increment.
3. Deflected Shape			
LVDTs	A series of calibrated LVDTs were mounted to measure transverse deflections as shown in Figures 3.8	Used to monitor the load deflection response of the sandwich plate.	Discrete readings taken at the end of each load increment.

Table 3.3 Measurement, Range and Accuracy

Measurement	Instrument	Specified Range	Accuracy
Applied Load	5 Calibrated Pressure Transducers	0-10000 psi	± 10 psi
Steel Strains	94-1/4 Bridge Micro-Measurement High Elongation Strain Gauges. Resistance $120.0 \pm 0.15\%$ ohms. Gauge length 8 mm. Gauge Factor $2.055 \pm 0.5\%$ at 24°C .	0-200000 μs	$\pm 0.5 \mu\text{s}$
Deflected Shape	LVDTs	$\pm 3.0''$	$\pm 0.006''$

Note: LVDTs were reset to keep the instruments within their operating range.

ASSEMBLY PLATE "Dc to 50 & 50"

SECTION C-C

SECTION E-E

SECTION D-D

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Figure 3.1(b) Fabrication Drawing of Test Specimen

Figure 3.2 Typical Sandwich Cross-Section

Figure 3.3 Cross-Section of Test Specimen in Test Apparatus

Figure 3.4 Load Application Points for Test Specimen

Figure 3.5 Plan Sectional View of Horizontal Reaction Frame

—275 mm x 299 mm grid, typ.

Figure 3.6 Location of Strain Gauges on Bottom Plate (Outside Hull Plate)

275 mm x 299 mm grid, typ.

Figure 3.7 Location of Strain Gauges on Top Plate (Inside Hull Plate)

2/5 mm x 299 mm grid typ.

Figure 3.8 Location of LVDTs on Top Plate

Figure 3.9 Location of Injection Holes

Figure 3.10 Tension Coupon Definition

(b) Tension Coupon Definition

Figure 3.11 Polyurethane Foam Test Specimen Definition

Chapter 4

Test Results and Discussion

4.1 Material Properties Steel Plate

The chemical and mechanical properties reported from the mill certificate for the 12.70 mm plate used for the construction of the test specimen are given in Table 4.1. Ten coupons from both the longitudinal (coincident with the rolling direction) and transverse directions were cut from a section of the parent plate and tested to determine the material properties. The material properties E , ν , σ_y , σ_{xy} , σ_u , σ_t , ϵ_y , ϵ_{xt} , ϵ_u , ϵ_t and E_{xt} for the longitudinal and transverse directions are given in Tables 4.2 and 4.3, respectively. Typical stress-strain curves for coupons no. 5 (longitudinal direction) and no. 13 (transverse direction) are shown in Figure 4.1. The mean thickness as determined from 10 measurements along the gauge length of the coupons was 11.56 mm.

2 of /de 2

PM-1 3 1/2" x 4" PHOTOGRAPHIC MICROCOPY TARGET
NBS 1010a ANSI/ISO #2 EQUIVALENT

PRECISIONSM RESOLUTION TARGETS

4.2 Material Properties Polyurethane Foam

Material test specimens for compression, tension, and shear tests were taken from a 300 x 300 x 100 mm block cut from a foam sample (with the correct consistency) cast in the test wooden forms. The density was 0.0472 g/cm³. A sample of this block was sent to DOW CHEMICAL laboratories for material testing according to ASTM standards. The test data was returned to Carleton on diskette along with two of the tested compression samples and photographs of the test set-up.

4.2.1 Compression Tests

The stress-strain curves for four compression test samples are given in Figure 4.2 for sample 1, 2, and 3 and Figure 4.3 for sample 4. Point *a* in both figures marks the onset of linear elastic response of the sample and testing machine. The non-linear response prior to point *a* attributed to mechanical slack in the loading components of the testing machine. Point *b* in Figure 4.2 represents the failure load which is defined as the load point at which the foam tears away from the rest of the sample around the circumference of the load point. This does not represent the ultimate compressive capacity of the foam. Sample 4 was tested beyond this definition up to the displacement limit of the testing machine as indicated by point *c* in Figure 4.3. Again, this is not its maximum compressive resistance as the stress-strain curve shows an ever increasing slope. The mean value of the proportional limit of 176.1 kPa may be used for

serviceability conditions but would be inappropriate for extreme loads such as collisions and groundings where the compressive capacity would be defined by point *c* or better. Characteristic parameters E , σ_p , σ_n , ϵ_p and ϵ_n for the polyurethane foam are given in Table 4.4.

4.2.2 Tension Tests

The stress-strain curves for two tension coupon tests are shown in Figure 4.4. Again, point *a* delineates the point of non-linear behaviour of the testing machine and sample which is attributed to mechanical slack. The behaviour up to failure is essentially linear elastic. As indicated by the curves and by the summary of material properties given in Table 4.5 there is substantial variability in E , σ_n and ϵ_u . Because of the low and variable tensile strength and possible decrease in strength due to voids, discontinuities and poor mixing, the design tensile strength should be taken as zero.

4.2.3 Shear Tests

The shear stress-strain curves for shear samples 1, 2, and 3 are shown in Figure 4.5. The punching tests were terminated with a separation of the foam disk under the load point under the sample. The relevant material properties σ_p , σ_u , ϵ_p and ϵ_u are defined for the proportional limit and ultimate load capacity only and are given in Table 4.6. These properties may not be representative of those experienced in the sandwich plate test

and should be investigated further.

4.3 Composite Steel-Polyurethane Foam-Steel Sandwich Plate Test Results

4.3.1 General Behaviour

The general behaviour of the composite steel-polyurethane foam-steel sandwich plate is characterized by the total applied load versus midpanel deflection curve shown in Figure 4.6. Points on the curve indicate major events that occurred during the test as the progression of failure took place. These events include the decrease of flexural stiffness due to shear bond failure between the plate and the core, yield of top and bottom plates, development of elastic and inelastic membrane behaviour, collapse of internal cell plates and crushing of foam, and shear tension failure of the loaded bottom (outside) hull plate. Load levels and deflections at which these events occurred are given in Table 4.7.

Based on calculations for two plates including the shear capacity of the internal cell plates and two plates with full shear capacity the load to cause the first plastic hinge to form would be 611 kN and 5615 kN, respectively. Point 1 delineates a change in stiffness of the composite plate system at a load of 423 kN. This load level is consistent with the flexural failure of two plates with a weak core material that has failed in shear. As presented subsequently strain values measured at the top and bottom plates will corroborate this conclusion. Subsequent to this breakdown in flexural capacity, the plates

behave as two independent elastic membranes separated by the core. Point 2 in Figure 4.6 coincides with first yield along the transverse direction at midpanel. Figure 4.7 and 4.8 give a summary of the load levels (kN) at which yielding took place for gauges located at the top and bottom plates respectively. If a value is not indicated then that region of the plate remained elastic throughout the entire test. Yielding in the transverse direction began at midpanel and spread along the longitudinal axis and out towards the long edges transversely to the top and bottom plates as the loads increased. Localized yielding occurred in the longitudinal direction on the top plate over top of the internal cell plates in the central portion around the load point at relatively low load levels (between 1200 and 2150 kN). Yielding in this direction was the result of localized bending of the top plate over the internal cell plates and not due to longitudinal straining. Load levels in which yielding took place for gauges 1 to 8 (located on the bottom plate in Figure 4.8) are difficult to interpret due to larger local deformations and restraint due to the load plates. The decrease in load and deflection of the top plate after point 3, shown in Figure 4.6, is the result of local buckling and collapse of the cell plate underneath load plate number one and extensive densification of the foam core. The bottom plate continues to deflect and load is redistributed between the plates. Figure 4.9 (a) and (b) are photographs of transverse and longitudinal cross-sections of the composite plate. Figure 4.9 (a) clearly shows the collapsed internal cell plate and the crushed foam core which has a thickness of 29.9 mm locally. Figure 4.9 (b) shows the variation in thickness in the transverse direction adjacent to load plate number one. Voids and some diagonal and longitudinal cracks in the foam core are evident. As a result of debonding, the plates have separated

from the core at the ends. The diagonal cracks are similar to the cracks or tears of the compression samples. The longitudinal crack is the result of elastic rebound of the bottom plate tending to separate it from the top plate. Figure 4.10 gives the measurements of core thickness and the final deformed shape of this portion of the transverse section. Subsequent to densification of the foam core and collapse of the internal cell plates over load points no's 1,2 and 3 the top and bottom plates behaved as two inelastic membranes with strains in the transverse direction exceeding strains within the strain hardening range. Point 4 represents the ultimate load which was dictated by a shear tension failure due to concentrated local indentations under the corner of load plate number one. At around 7000 kN these indentations measured 18 mm. Measurements of the plate thickness at this location of 8.25 mm give shear strains in the order of 290 000 μ s. After this load level the test specimen continued to deflect with application but decreasing load.

4.3.2 Deflection Characteristics

As shown in Figure 3.8 and in the photograph of Figure 4.12, a series of 30 LVDTs were used to measure the transverse deflections of the top plate. Five LVDTs were used to monitor the displacement of the actuator head. One of these is shown in Figure 4.13 which is an end view of the actuators and shows the underside of the deformed composite plate at a load level of 7500 kN. Due to symmetry the quadrant shown in Figure 4.11 was heavily instrumented to give the most complete picture of the

deformed shape that was possible. Characteristic displacements are the best indicator of behaviour and will be used in a subsequent thesis for comparison with the appropriate finite element analyses. The behaviour as related to these measured deflections will briefly be discussed.

Figures 4.14, 4.15, and 4.16 give the total load deflection curves for the top plate directly located over each load plate and the displacement of the corresponding actuator. The difference of displacement between these two curves at any load level gives a measure of the change in thickness of the composite plate, very small deformations of knife-edge-roller plate assembly and the vertical displacement of the reaction frame to the strong floor. The latter of these is in the order of 10 mm. The change in thickness can be estimated assuming that these lateral displacements are linear with respect to load. The five sets of curves clearly show the crushing of the foam and collapse of the cell plates at a load of 7690 kN. After this point the bottom plate deflects more rapidly than the top plate indicating that the load percentage of this additional applied load is being carried by the bottom plate in inelastic membrane behaviour.

Figure 4.17, 4.18, 4.19 and 4.20 give the total load displacement curves for LVDTs aligned in the transverse direction along the lines at $1/2$, $3/8$, $1/4$, and $1/8$, of the length, respectively. LVDT 11 gave unreliable data. Taking advantage of symmetry, the displacements for LVDT 11 were replaced with the corresponding displacement measurements of LVDT 24. Figure 4.21 gives the load deflection curves for the LVDTs aligned with the longitudinal axis. The curves indicate that the changes in thickness of the composite plate are local to the load points and diminish rapidly to essentially zero within

a distance equal to the largest plan dimension of the load plate. Figure 4.22 gives the displacement values for the specified quadrant for total loads that correspond to the events described in Table 4.7. Figure 4.23 shows the deflected shape of the top plate in the longitudinal and transverse direction at ultimate load. The corresponding LVDT numbers for displacements are shown. The deformed shape of the top plate is consistent with the shape of transversely loaded plates that behave as membranes with a maximum deformation of $l/13$ of the transverse span.

4.3.3 Measured Strains

Figures 3.6 and 3.7 show the locations and identification numbers of the 94 strains measured orthogonally (aligned with the principle directions of the composite plate) on the top and bottom plates, respectively. Due to symmetry the strains measured along the transverse and longitudinal axis are equal to the principal strains. The strain values and distributions presented subsequently will be used to verify the finite element analyses. The total load principal strain curves for the strains measured on the top plate along the longitudinal axis are given in Figure 4.24. Odd numbers indicate transverse strain and even numbers longitudinal strains. The large values associated with the longitudinal strains are attributed to large local bending strains over the internal cell plates and are not due to membrane strains in that direction. Strain gauge no. 73 did not work properly during the testing of the instrumentation functionality and was therefore eliminated for the data set. In Figure 4.24 strain gauge no.'s 73 and 74 were replaced by their

symmetrical duplicates no.'s 57 and 58. The total load principal strain curves for strains measured on the top and bottom plates at $b/4$ and $b/8$, along the transverse axis are shown in Figure 4.25. Strain values for the top and bottom plate, for the specified quadrant, are given at the load events described in Table 4.7 in Figures 4.26 and 4.27, respectively. The maximum strains in tension and compression at a load level of 423 kN are $364 \mu\text{s}$ and negative $313 \mu\text{s}$ respectively. These are significantly less than the yield strain and are consistent with the conclusion drawn previously of flexural failure due to shear bond failure between the steel plate and the foam core. The strain values are consistent with composite flexural action of the plate. Inelastic membrane action started with yielding of the top plate in transverse direction along the longitudinal axis. The corresponding transverse strains on the bottom plate were elastic at this load level. The maximum transverse strains on the top plate at the point of collapse of the internal cell plates and crushing of the foam were approaching the strain hardening strain along the longitudinal axis. These dissipated to elastic strain values at the edge of the plate near the reaction frame. The strain values in the transverse direction of the bottom plate were essentially elastic at this load level. Subsequent to collapse of the cell plate and crushing of the foam the additional applied load was carried by the bottom plate. This is supported by the strain values shown in Figure 4.6 (d) and Figure 4.7 (d). Transverse strains for the top plate remained relatively constant while transverse strains for the bottom plate yielded throughout and approached strain hardening strain values.

4.3.4 Comparison with IMASP Model

As demonstrated by Ratzlaff (1985, 1986), Kennedy (1987), and Bond (1995) the membrane behaviour of this composite plate may be modelled as transversely loaded wide beam with a length equal to the transverse span of 2392 mm, a width equal to the length of applied load of 2750 mm, and a depth of 123.12. Figure 4.28 shows the total load centre span deflection curves for the test and for the IMASP predictions with infinite in-plane boundary stiffness and an estimated in-plane boundary stiffness of 8000 kN/mm of the horizontal reaction frame. As shown by a comparison of the experimental and predicted curves, there is relatively close correspondence in the inelastic membrane behaviour region. The increased load capacity for a given deflection in the flexural elastic membrane region is the result of the core and cell plates. The sharp increase in stiffness of the composite plate after 7690 kN is the result of the changing through thickness of the composite plate and redistribution of membrane forces between the plates. IMASP predicts a fracture at the edge of the plate panel at a load of 12021 kN and a deflection of 234 mm. This is viewed as the maximum probable load that a system could carry if premature rupture due to a shear tension failure under the load plate did not occur.

Table 4.1 Chemical Composition and Mechanical Properties of the Structural Steel Plate from the Mill Certificates

Plate (mm)	Grade	Manufacturer	Heat No.	Yield Point (MPa)	Ultimate Tensile Strength (MPa)	Elongation (%)	Chemical Composition (%)			
							C	S	P	Si
12.70 x 2981 x 10942	ASTM A 36	Ilych Iron Steel Works Mariupol, Ukraine	242156	312.5	447.5	30.0	0.18	0.19	0.12	0.25

Table 4.2 Characteristic Parameters for Steel Plate determined from Uniaxial stress strain curves for the 12.7 mm Plates in the Longitudinal Direction

Coupon No.	E (MPa)	ν	σ_y (MPa)	σ_y (MPa)	σ_u (MPa)	σ_t (MPa)	$\epsilon_y \times 10^{-6}$	$\epsilon_u \times 10^{-6}$	$\epsilon_t \times 10^{-6}$	E_u (MPa)
1	207300	-	357.1	351.9	499.0	488.4	1648	25628	181906	5970
2*	203400	0.274	364.5	359.8	501.9	493.6	1706	20838	170000	4121
3	208900	-	367.7	333.4	506.0	502.1	1666	29648	170854	5866
4	207000	-	372.3	356.0	509.9	499.5	1709	-	181910	-
5	204000	0.280	373.9	361.4	504.8	482.0	1766	18729	190000	8157
6	206400	-	376.2	369.2	517.9	491.0	1730	20500	180000	3214
7	204600	0.281	378.7	373.4	514.7	498.6	1763	19369	156000	5637
8	209700	-	374.3	371.9	514.7	498.7	1747	-	231217	-
9	209400	-	376.2	370.9	513.1	491.6	1689	24000	169500	1908
10**	208900	-	373.8	370.7	508.8	-	1746	20000	183500	2589
Avg.	207000	0.278	373.5	361.9	509.1	493.9	1717	22389	181489	4683

Notes: * ν is determined from one transverse reading only

**results are determined from one longitudinal reading only

Table 4.3 Characteristic Parameters for Steel Plate determined from Uniaxial Stress-Strain Curves for the 12.7 mm plates in the Transverse Direction

Coupon No.	E (MPa)	ν	σ_y (MPa)	σ_y (MPa)	σ_u (MPa)	σ_f (MPa)	$\epsilon_y \times 10^{-6}$	$\epsilon_u \times 10^{-6}$	$\epsilon_f \times 10^{-6}$	E_u (MPa)
11	224900	-	368.8	343.3	508.5	452.0	1580	25126	149452	4139
12	218500	-	375.2	339.9	517.0	514.8	1740	17588	145729	1672
13	207700	0.286	377.1	361.2	514.7	506.0	1748	20896	154229	2665
14	206400	0.284	379.4	356.1	512.8	508.8	1744	27638	160804	3993
15	207300	-	372.4	370.0	517.2	509.6	1689	25125	173367	2871
16	206500	0.284	380.0	353.4	509.4	502.6	1744	-	145000	-
17	208100	-	373.3	373.2	510.2	495.5	1691	29146	158292	4030
18*	-	-	-	-	-	-	-	-	-	-
19	213000	-	376.7	378.2	512.2	440.9	1670	21106	178392	2601
20	207200	-	376.3	358.6	515.0	444.2	1816	25628	144724	5423
Avg.	211000	0.285	375.5	359.3	513.0	486.0	1714	24032	156665	3424

Note: *data file was corrupt

Table 4.4 Characteristic Parameters for Polyurethane Foam determined from Uniaxial Compression Test Data

Sample No.	Length (mm)	Width (mm)	Diameter (mm)	Thickness (mm)	E (kPa)	σ_p (kPa)	σ_n (kPa)	$\epsilon_p \times 10^{-6}$	$\epsilon_n \times 10^{-6}$
1	127.00	101.60	88.90	50.80	1778	183.0	302.3	55425	176613
2	127.00	101.60	88.90	50.80	1750	181.9	255.6	48800	125375
3	127.00	101.60	88.90	50.80	1786	163.4	230.9	44194	130493
4*	127.00	101.60	88.90	50.80	-	-	563.3	-	728701
Avg.	127.00	101.60	88.90	50.80	1771	176.1	262.9	49473	144160

Note: * values not included in mean value calculation

Table 4.5 Characteristic Parameters for Polyurethane Foam determined from Uniaxial Tension Test Data

Sample No.	Width (mm)	Thickness (mm)	E (kPa)	σ_u (kPa)	$\epsilon_u \times 10^{-6}$
1	12.70	11.39	1705	327.6	208425
2	12.70	11.39	2449	146.7	89803
Avg.	12.70	11.39	2077	237.1	149114

Table 4.6 Characteristic Parameters for Polyurethane Foam determined from Uniaxial Shear Test Data

Sample No.	Diameter (mm)	Thickness (mm)	σ_p (kPa)	σ_a (kPa)	$\epsilon_p \times 10^{-6}$	$\epsilon_a \times 10^{-6}$
1	25.15	4.85	78.7	433.6	71113	824504
2	25.15	3.89	54.2	482.8	40399	947970
3	25.15	4.22	95.9	589.1	77080	937767
Avg.	25.15	4.32	76.3	501.8	62864	903414

Table 4.7 Progression of Failure

Event	Load (kN)	Deflection (mm)	Description
1	423	14.0	<ul style="list-style-type: none"> • decrease in flexural stiffness which is probably a result of a bond failure between the plate and the core. • onset of elastic membrane behaviour.
2	2031	65.4	<ul style="list-style-type: none"> • first yield in transverse direction at strain gauge 63. • onset of inelastic membrane behaviour.
3	7690	162.8	<ul style="list-style-type: none"> • collapse of internal cell plates locally over load points no's 1, 2, and 3. • crushing of foam core locally over load points no's 1, 2, and 3.
4	8992	169.1	<ul style="list-style-type: none"> • Shear tension failure of the bottom (outside hull) plate underneath the corner of load plate no. 1. • Termination of test.

Figure 4.1 Typical Stress-Strain Curves for Tension Tests of Steel Plates

Figure 4.2 Stress-Strain Curves for Compression Test of Polyurethane Foam

Figure 4.3 Stress-Strain Curve for Extended Loading of Compression Test for Polyurethane Foam

Figure 4.4 Stress-Strain Curves for Tension Test of Polyurethane Foam

Figure 4.5 Stress-Strain Curves for Shear Test of Polyurethane Foam

Figure 4.6 Total Load Midpanel Deflection Curve illustrating Progression of Failure

Figure 4.7 Yielding Summary of Top Plate

Figure 4.8 Yielding Summary of Bottom Plate

(a) Cross-Sectional View of Deflected Shape underneath the Corner of Load Plate no.1

(b) Deflected Shape of Top and Bottom Plate in Transverse Direction

Figure 4.9 Photographs of Deflected Shape

Figure 4.10 Measurements of Core Thickness and Final Deformed Shape of Transverse Section

Figure 4.11 Representative Quadrant for Display of Test Results

Figure 4.12 LVDTs for Vertical Displacement Measurements on Top Plate

Figure 4.13 End View of Actuators and Deformed Plate

Figure 4.14 Total Load Deflection Curves for LVDT 18 and LVDT 31 located at Midpanel

(a) LVDT 19, LVDT 32

(b) LVDT 17, LVDT 33

Figure 4.15 Total Load Deflection Curves for LVDTs 19, 32, 17 and 33

(a) LVDT 14, LVDT 34

(b) LVDT 16, LVDT 35

Figure 4.16 Total Load Deflection Curves for LVDTs 14, 34, 16 and 35

Figure 4.17 Total Load Deflection Curves along Transverse Axis for LVDTs 18, 24, 5 and 1

Figure 4.18 Total Load Deflection Curves along 3/8 (Transverse Direction) for LVDTs 19, 12, 6 and 2

Figure 4.19 Total Load Deflection. Curves along 1/4 (Transverse Direction) for LVDTs 14, 13, 7 and 3

Figure 4.20 Total Load Deflection Curves along 1/8 (Transverse Direction) for LVDs 21, 8 and 4

Figure 4.21 Total Load Deflection Curves along Centreline of Longitudinal Direction for LVDTs 18, 19, 14 and 21

(a) at 423 kN

(b) at 2031 kN

Figure 4.22 Displacement Values in mm for the Specified Quadrant

(c) at 7690 kN

(d) at 8992 kN

Figure 4.22 Continued

(a) Longitudinal Direction

(b) Transverse Direction

Figure 4.23 Deflected Shape of Top Plate at Ultimate Load

(a) at actuator #1

(b) at actuator #2

Figure 4.24 Total Load Principal Strain Curves along the Centreline

(c) at actuator #4

(d) located at l/8

Figure 4.24 Continued

(a) located at b/4

Figure 4.25

Total Load Principal Strain Curves for Top and Bottom Plate along the Transverse Axis

Figure 4.25 Continued

Figure 4.26 Strain Values for Top Plate for the Specified Quadrant

Figure 4.26 Continued

Figure 4.26 Continued

Figure 4.26 Continued

Fig. 4.27 Continued

Figure 4.27 Continued

Figure 4.27 Continued

Figure 4.28 Total Load Centre Span Deflection Curves for IMASP Model and Test Result

Chapter 5

Summary, Conclusions, Recommendations, and Future Work

5.1 Summary and Conclusions

As a result of recent well-publicized oil-tanker failures and spills, the corresponding environmental damage, growing public awareness, and demand for risk-free technology, new national and international standards have been legislated for the operation and design of oil tankers. Consequently a significant amount of research is now being conducted on double hull construction of oil tankers, including survivability and collision resistance and oil outflow management processes.

Based on previous work by Ratzlaff (1985,1986), Kennedy (1987, 1994) and in light of the new legislation, an advanced unidirectional sandwich plate system has been proposed. A major portion of this work was:

1. the determination an appropriate structural plastic material for the use in the composite sandwich system;
2. to design and construct a large scale composite plate;
3. to conduct a large scale test on a steel-polyurethane foam-steel plate to examine its behaviour and to provide data for future analytical work.

The large scale test specimen measured 5000 x 2600 mm in plan and was composed of two steel plates (inner and outer hulls) with a measured thickness of 11.56 mm and a polyurethane foam core (XUS Experimental Polyol/Papi 27 Polymeric MDI, DOW CHEMICAL) with a depth of 100 mm. The foam material was injected into eight cells in the cavity of the steel plates after these were fully welded into the horizontal reaction frame. The test specimen was loaded monotonically, under five transverse point loads with a footprint measuring 300 x 2400 mm, to failure at a load of 8992 kN. The corresponding deflection was 169 mm. The specimen had little flexural capacity and mainly behaved as a membrane. The ultimate capacity was dictated by a shear tension failure of the outside hull plate under the corner of the central load plate.

Based on the fabrication and construction of the test specimen, material investigations for the core, observations and experimental evidence from material and large scale composite plate tests, it can be concluded that:

- (1) Horizontally oriented composite panels require cells to be constructed for the proper coverage and casting of the foam core material;

- (2) The cell plates should be stitch-welded and relatively thin as not to create any hard spots and to give the ability for them to collapse under transverse loads;
- (3) The bond between the steel and the foam is relatively weak and should not be relied on to transfer shear across the interface for flexure;
- (4) Based on preliminary calculations a denser foam with a compressive resistance of approximately 5 MPa is required to prevent large local deformations of the loaded plate and subsequent localized punching or shearing of that plate. This would enable the composite plate to achieve the maximum ductility possible which is governed by a shear tension failure of the heat affected zones for welds aligned in the longitudinal direction of the plate.

5.2 Recommendations and Future Work

This thesis forms a preliminary investigation into the feasibility and behaviour of steel-polyurethane foam-steel sandwich plates for use as the outside hull for oil tankers using the UDHSPS concept. The following are recommendations for future work:

1. To continue the investigation of an appropriate core material with adequate bond capacity and improved compressive and tensile strength. One such material is the polyurethane elastomer material (WUC 50792R / WUC 3253T) produced by the BASF Corporation. Problems with injection and construction must be resolved prior to its use. As part of this investigation, a shear bond test could be conducted

to determine the bond strength between the core material and the steel plates. Additional, ancillary tests should be conducted to determine the in-plane and compressive resistance of the composite section.

2. To increase the flexural capacity of the sandwich plate by incorporating collapsible bulkheads and stiffeners, that span in the longitudinal direction. The corresponding transverse loads would represent the maximum probable load under service conditions.
3. To avoid the accumulation of erroneous displacements related to displacements of the reaction frame and load devices, a support frame for LVDTs measuring displacements of the bottom plate (outside hull plate) should be attached to the underside of the horizontal reaction frame.
4. A finite element analysis should be conducted to compare and to verify the test results. Further finite element analyses should be made to predict the behaviour and load carrying capacity of different structural arrangements for this system and for load combinations that would be difficult to model experimentally.
5. A 30,000 - 40,000 DWT baseline oil tanker should be selected, analyzed and compared to the behaviour of the sandwich plate under service and extreme loads associated with collision and grounding damage. The selection of a tanker of this size would have the following advantages: (a) most new oil tankers which are built according to the advanced double hull technology (like the EPOCH Mark II series of Hitachi-Zosen, Okamoto et al., 1985) are in this deadweight class. Therefore a direct comparison with respect to costs, constructability, and performance can be

made; (b) according to Rodd et al. (1994) reliable scaling rates for experiments should not be smaller than approximately 1:4, because for smaller scaled specimens damage does not scale by a relationship compatible with real distortion damage. To model a ship section which represents the double bottom section extending one longitudinal spacing between transverse bulkheads and about half the beam of a ship, which is the most vulnerable hull section with respect to groundings, Rodd et al. (1994) applied a scaling factor of 1:4.25 for a 30,000 DWT oil tanker, which resulted in a test specimen size of 4267 mm by 2540 mm. This correlates well with the size of test specimen (5000 mm x 2600 mm) used for this study.

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