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

by

Michael R. L. Bennett

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

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

**The Master of Engineering in Civil and Environmental Engineering
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Department of Civil and Environmental Engineering

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

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Sandwich Plate System:
Experimental Investigation**

**submitted by
Michael R. L. Bennett, B. Eng.**

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

J. L. Humar, Chair, Department of Civil and Environmental Engineering

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Ottawa, Ontario, Canada**

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Abstract

Legislation has been implemented in most jurisdictions around the world, dictating the use of double hull construction for all new product oil tankers in an attempt to reduce the probability of oil outflow from tankers subjected to accidental or extreme load events. Statistical evidence indicates that groundings and collisions will continue to occur even with the advances in navigational technology. During these events, it is recognized, however, that tankers built using the state-of-the-art in double hull technology will have a minimum of one cargo tank breached due to crack propagation from the ruptured outer hull.

It is recognized by industry, researchers and Classification Societies that the energy absorption capacity of the hull configuration must be increased by engaging a larger portion of the vessel during accidental or extreme load events, that the propagation of cracks into the inner hull must be prevented, and that localized strain concentrations must be dissipated to prevent premature rupture of the hull plating. Whatever the development in new double hull tanker configurations, it must also address corrosion issues, fatigue-related problems, production engineering and the provision of sufficient residual strength for post-accidental or post-extreme load events. Implementation of the new technology must have Classification Society approval and sufficient economic advantages to entice shipowners and shipbuilders to invest.

In response, the Unidirectional Double Hull Sandwich Plate System (UDHSPS) was developed. It addresses the issues associated with all-steel double hull construction by introducing a composite structural laminate plate into the hull structure and by providing a design philosophy which gives a structure that is satisfactory for normal

operating loads, provides superior performance for accidental or extreme load events, and has sufficient residual strength. The structural core eliminates the probability of oil outflow due to crack propagation by providing an effective crack arrest barrier for the cargo tanks. A detailed cost analysis of a typical UDHSPS cargo tank from a 40,000 DWT product oil tanker demonstrated an 11.6 percent as-built cost saving over a similar sized tank from a conventional double hull design.

As proof of this concept, an experimental investigation of a typical double hull section (large-scale) was required to verify the design philosophy and to provide data to calibrate numerical simulation procedures. The fabrication of the large-scale test specimen also demonstrated the feasibility of this type of construction.

A large-scale test specimen of the UDHSPS double hull section, measuring 2600 mm by 5000 mm in plan, was modelled after the keel of a baseline conventional double hull structure. It incorporated two 700 mm deep longitudinal girders, a transverse floor, a composite structural laminate outer hull and a single steel plate inner hull. The specimen was fixed to the horizontal reaction frame, which simulated the stiffness of the surrounding ship structure. Under transverse loads, the outer hull plate developed inelastic membrane action between the longitudinal girders followed by the progressive inelastic buckling of the transverse floor and longitudinal girder webs and the subsequent spread outward of the inelastic membrane action. The outer hull ruptured due to a shear-tension failure at the horizontal reaction frame boundary. The experimental evidence confirmed the design philosophy and demonstrated the effectiveness of the composite structural laminate in dissipating localized strain concentrations that otherwise would have caused premature rupture of the hull plating.

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List of Symbols

A	Horizontal clearance term
AASHTO	American Association of State Highway and Transportation Officials
ABL	Above baseline
ABS	American Bureau of Shipping
AC	Human access term
ADH	Advanced Double Hull
ASTM	American Society of Testing and Materials
b	Plate width
B	Molded breadth
B	Vertical clearance term
C_b	Block coefficient
CCG	Canadian Coast Guard
CDH	Conventional Double Hull
CDNSWC	Carderock Division Naval Surface Warfare Center
CF1	Complexity factor to account for access during welding
CF2	Complexity factor to account for access and surface orientation during painting
CF3	Complexity factor related to quality assurance
CO	Collision
d	Molded draft
D	Molded depth

DnV	Det Norske Veritas
DWT	Deadweight tonne
E	Young's modulus
E_{ST}	Strain hardening modulus
EGW	Electro-gas welding
ESW	Electroslag welding
EPOCH	Easy handling/maintenance product oil carrier by Hitachi-Zosen
F_{SW}	Still-water shear force
F_{WN}	Wave-induced negative shear force
F_{WP}	Waved-induced positive shear force
F_y	Yield strength
Fab\$	Fabrication cost
FCAW	Flux core arc welding
F/E	Fire/explosion
GT	Gross tonnage
GRN	Grounding/stranding
h	Double hull bottom spacing requirements
HS	High strength
HSLA	High strength low alloy
I	Hull-girder moment of inertia
IMO	International Maritime Organization
L	Rule ship length = 97% of LBP
LBP	Length between perpendiculars

LOA	Length overall
LST	Large-scale test elastomer
LVDT	Linear variable differential transformer
M_H	Still-water hogging bending moment
M_S	Still-water sagging bending moment
M_{TH}	Total hogging moment
M_{TS}	Total sagging moment
M_{WH}	Wave-induced hogging bending moment
M_{WS}	Wave-induced sagging bending moment
mt	Metric ton
MARAD	Maritime Administration
MISO	Multilinear isotropic hardening
N/K	Not known
NLGEOM	Large deformation effects
NROPT	Newton-Raphson option
OPA	Oil Pollution Act
OS	Ordinary strength
PAWS	Programmable automated welding system
PCF\$	Plate cutting and forming cost
RIM	Reaction injection molding
RMHP\$	Raw material handling and surface preparation cost
SAW	Submerged arc welding
S/H	Structure, hull or machinery failure

SM	Section modulus
SO	Surface orientation term
SOLAS	Safety of Life at Sea
SSTIF	Stress stiffening
t	Thickness of plate
TC	Test cavity elastomer
TSAMC	Tanker structural analysis for minor collisions
UDHSPS	Unidirectional Double Hull Sandwich Plate System
VLCC	Very large crude carrier
w	Side shell double hull spacing requirements
x	Distance between parallel welds within a confined space
x	Horizontal distance between the inner and outer hull stiffening elements in side or horizontal distance between girders in bottom
y	Vertical distance (headroom)
β	Plate slenderness parameter
ϵ_{avg}	Average strain
ϵ_f	Strain at failure
ϵ_{α}	Strain at the onset of strain hardening
ϵ_{ult}	Ultimate strain
ϵ_y	Yield strain
ν	Poisson's ratio
σ_{avg}	Average stress
σ_f	Stress at failure

σ_{ry}	Static yield stress
σ_{ult}	Ultimate stress
σ_y	Yield stress

Chapter 1

Introduction

1.1 General

With the publicity surrounding the grounding of the Exxon Valdez in 1989 in Alaskan waters, the western world was incited into action in an attempt to prevent similar incidents from occurring by invoking legislation. The most influential legislation was that introduced by the American Congress, the *Oil Pollution Act of 1990* (OPA 90), which was modelled after MARPOL 73/78 requirements. This legislation requires that all oil tankers entering U.S. waters commissioned after June 30, 1990, or whose delivery was arranged on or after January 1, 1994, be of double hull tanker construction. Existing single hull tankers are required to be replaced when retired, according to the phase-in schedule. The Canadian Coast Guard (CCG) also recognized the need for improved oil tanker design and construction, and implemented a similar standard, *Standards for the Double Hull Construction of Oil Tankers (TP 11710)*, for tankers operating in Canadian waters. Although there are other design alternatives currently available to minimize the risk of oil outflow, the most widely accepted is the double-side, double-bottom configuration (herein referred to as a double hull), in which the inner hull cargo tanks are completely surrounded by an outer hull, typically spaced at 2.0 meters.

Double hull design technology for oil tankers is in its infancy, with the first commercial vessel being the EPOCH Mark II introduced by Hitachi-Zosen in 1985. Since that time, a number of design configurations have been introduced, each of which has its own merits and deficiencies. The Conventional Double Hull (CDH), the Advanced Double Hull (ADH) and the curved plate variant of the ADH, the MarC Guardian, are examined in the following paragraphs.

A typical CDH design consists of a complex series of longitudinally and transversely framed elements that separate the inner and outer hulls, as illustrated in Figure 1.1. Due to this high level of complexity, the structure is more difficult to analyze, design, inspect and maintain. Also, there are a large number of structural intersections (discontinuities) between framing elements that lead to fatigue problems, crack initiation and hard points that create localized stress and strain concentrations possibly leading to premature rupture of the plate. It has been demonstrated by large-scale grounding tests at the Carderock Division of the Naval Surface Warfare Center (CDNSWC) that the outer hull is ruptured by a rock. As the ship travels forward over the rock, the tear (crack) in the outer hull propagates ahead of the rock towards the stern. This crack, in combination with others caused by the rock passing through transverse elements (floors, stools and bulkheads), propagates into the inside hull, breaching the cargo tanks. Regardless of any variations in the design, it is estimated that a minimum of two cargo tanks will be ruptured during a typical grounding event, (Rodd, 1996).

The ADH, illustrated in Figure 1.2, offers a simplified structural layout that predominantly consists of unidirectional longitudinal framing. Transverse elements (stools) still exist at transverse bulkhead locations. This creates a structural design that is

Figure 1.1: Typical Conventional Double Hull Tanker

Figure 1.2: Advanced Double Hull Tanker (after Okamoto *et al.*, 1985)

easier to analyze since there is a reduction in the number of elements, which lends itself to automated welding and to modular construction, manifesting itself as a reduction in fabrication costs. Eliminating the transverse flooring elements reduces the number of structural intersections which, in turn, lessens the likelihood of potential fatigue cracks at these fatigue prone details, and it also reduces the number of hard points which are a source of hull plate rupture. The simpler construction reduces surface area and allows for better painting and coating application. This same simplicity makes inspection and maintenance less costly.

Cited disadvantages of the ADH design alternative are: (a) for comparative grounding events, the extent of longitudinal damage is greater as there are fewer transverse floor elements to absorb energy; (b) the tight spacing of the horizontal stringers and longitudinal girders yields short, unsupported spans of the hull plate panel, which under certain collision and grounding events can lead to early rupture with a small degree of penetration, thereby reducing the global energy absorption capacity, (Kitamura, 1996); and (c) the ADH designs are not practical for tankers larger than 40,000 to 50,000 DWT as the longitudinal girder spacing becomes too small (related to the buckling capacity of the unstiffened hull plate between the longitudinal girders), (Sikora *et al.*, 1995).

An alternate ADH novel design is the MarC Guardian curved plate concept, which, as the name implies, uses unstiffened curved plating to form the outer and inner hull shell elements, as illustrated in Figure 1.3. The curved hull plates increase the buckling capacity of the hull plate, over equivalent flat plates, for in-plane loads due to hogging and sagging moments, (Melton *et al.*, 1994). This results in a larger spacing

between girders, which reduces the number of subdivisions, hence it increases accessibility, makes inspection and maintenance easier, and allows for better coating application. The possibility of adverse effects on propulsion efficiency and hull strength for dry docking need to be studied. During strandings, however, the energy dissipation was significantly less, (Melton *et al.*, 1994).

Figure 1.3: MarC Guardian Curved Plate Concept Tanker (after Goldbach, 1994)

Even with the advances in navigational technology, groundings and collisions will continue to occur and the development of double hull technology must be advanced to minimize or eliminate oil outflow. It would be simple to modify existing double hull technology to incorporate wider double sides, deeper double bottoms, smaller cargo tanks, increased scantlings or structural members, and improved mechanical properties of materials that would enhance the crashworthiness and minimize oil outflow. However, such design alternatives would not necessarily solve the problem, they are not

economical and, therefore, are not realistic. There exists a need for the development of an innovative, practical, and cost-effective design to minimize or mitigate oil outflow in the case of accidental or extreme load events.

The Unidirectional Double Hull Sandwich Plate System (UDHSPS), where “unidirectional” refers to the orientation of the framing elements, is illustrated in Figures 1.4 and 1.5. The design integrates a composite structural laminate into the hulls and builds on the advantages of the ADH design concept, which addresses the major problems associated with the all-steel double hull systems. The proposed laminate consists of a steel-structural plastic-steel sandwich. The UDHSPS design eliminates the longitudinal stiffeners of the inner and outer hull plates, providing a simple, less complex structural system composed of flat surfaces. The structural composite laminate gives equivalent in-plane load capacity without decreasing the longitudinal girder spacing, as required for the ADH. The UDHSPS design reduces the number of structural intersections characteristic to the CDH and ADH designs, thereby decreasing the number of fatigue-prone details. Much of the steel used for stiffening the outer and inner hull plates is allocated to the hull plating, thereby increasing the energy absorption characteristics of the plating. Hard points at longitudinal girder/transverse floor and hull plate/stiffener intersections are diminished due to the core material that dissipates the localized strains over a larger field. The structural plastic core material, with appropriate construction and fatigue detailing, provides an effective crack arrest barrier for crack propagation into the inner hull. The flat surfaces of this system lend themselves to better coating application and less corrosion-related problems. The elimination of stiffeners and the reduction of structural intersections give increased accessibility for improved

Figure 1.4: Isometric View of the UDHSPS Tanker

Figure 1.5: Midship Section of UDHSPS Tanker

inspection and maintenance. It is anticipated that, with the inherent simplicity of the design, a reduction in steel costs, fabrication costs and maintenance costs are to be expected.

Recognizing the advantages of the proposed structural system, the next step was to select the appropriate structural plastic core material. Linder, (1995) examined a polyurethane foam material as a plausible candidate due to its inherent material characteristics. Such attributes included: lightness; its use in other structural sandwich applications; insulation characteristics; energy absorption characteristics (compressibility); and with additives, the foam can offer excellent fire resistance. However, the selected foam was found not to offer sufficient in-plane capacity to replace the stiffened plate system due to the lack of bond capacity and compressive and tensile resistance.

Another probable alternative was a thermoplastic polyurethane elastomer, since it has substantially increased compressive and tensile strength, does not require mechanical fasteners or chemical adhesives for bond, and offers many of the desirable features of the foam. For this to be a suitable core material, several structural and production issues needed to be addressed:

- (1) Could the elastomer be produced with the desired properties (characteristics)?
- (2) Would the composite structural laminate be equivalent to typical stiffened plate panels during normal operating loads?
- (3) Could it be constructed under shipyard conditions and at the scale anticipated?

- (4) Would it be cost effective?
- (5) Would the fire resistance be acceptable to meet the *Safety of Life at Sea* (SOLAS) requirements?

Given that this core material was a suitable candidate, a design philosophy had to be developed to incorporate the composite structural laminate into the design of a double hull tanker, such that it would provide equivalent strength for normal operating loads and superior performance during accident and extreme load events. Both analytical (Ferro, 1998) and experimental research was required to address the above questions of the core material and for the development of the structural system. It is the experimental research component that forms the basis of this thesis.

1.2 Objectives and Scope

The objectives of this study are:

- (1) to conduct a state-of-the-art literature review to establish the problems that exist with current tanker technology, to examine the direction in which research is heading, and to determine the limitations of current or proposed designs;
- (2) to integrate the composite structural laminate for advanced double hull systems;

- (3) to develop a design philosophy which gives equivalent in-plane capacity for operational loads and maximizes the energy absorption capacity for accidental or extreme load events;
- (4) to produce the UDHSPS design that is equivalent to a CDH design for a typical 40,000 DWT tanker;
- (5) to perform a detailed cost analysis of the UDHSPS design concept versus an all-steel CDH design to establish its economic feasibility;
- (6) to design and construct a large-scale test specimen that is representative of the anticipated construction of a portion of the double hull;
- (7) to conduct a large-scale test:

 - (a) to determine the inelastic behaviour and interaction of the longitudinal girders, the transverse floor, and the composite structural laminate outside hull;
 - (b) to observe the progression of inelastic membrane action of the outside hull panels adjacent to the loaded panel as the longitudinal girders collapse to verify the energy absorption design philosophy;
 - (c) to make observations and measurements in which the finite element analysis can be verified (Ferro, 1998); and
 - (d) to make observations related to the dissipation of strains, and puncture and tear resistance at load points and at or near the transverse floor; and

- (8) to make recommendations for future work that will further characterize the composite structural laminate for fatigue behaviour, in-plane capacity, flexural capacity, bond strength and fire resistance.

The large-scale test specimen measured 5000 mm by 2600 mm in plan and consisted of a composite structural laminate outer hull, a pair of longitudinal girders, a transverse floor, and an inner hull. The outer hull was comprised of two 7.94 mm thick steel plates and a 50 mm polyurethane elastomer structural core. The longitudinal girders and transverse floor were 12.7 mm thick and measured 700 mm deep. The inner hull was 7.94 mm thick and had a series of oval shaped holes, measuring 200 mm by 300 mm, that allowed access for linear variable differential transformer (LVDT) measurements and other observations during the test. The steel section was fully welded to the horizontal reaction frame, which represented the stiffness of the surrounding ship structure and provided the necessary in-plane and transverse restraint to develop membrane action. The elastomer was formulated from two components, a resin and an “iso”, that were mixed and injected under pressure into the steel cavity, then allowed to set. The outer hull, between the longitudinal girders aligned with the longitudinal axis, was transversely loaded with four point loads, each with an area of 200 mm by 300 mm. The load was applied quasi-statically and monotonically to failure. Load, displacement and strain measurements were recorded throughout the duration of the test with a data acquisition system. The test was conducted under laboratory conditions.

A statistical sample of ancillary tests was conducted in accordance with *American Society of Testing and Materials* (ASTM) standards on the steel tension coupons,

polyurethane elastomer compression disks and polyurethane elastomer tension coupons to determine the mechanical properties. Steel-polyurethane elastomer-steel shear-bond tests were also conducted to quantify the bond capacity of the steel-elastomer interfaces for steel plates with and without mill scale.

1.3 Outline

A state-of-the-art literature review and proposed design concept is presented in Chapter 2. The state-of-the-art literature review includes a summary of 50 major oil spills from oil tankers and combined carriers in 1960, and from 1965 to 1989, and any additional oil spills which occurred from 1990 to date; a review of current legislation and standards regarding double hull construction; research and development of current double hull tanker designs; production engineering of double hull oil tankers; effectiveness of double hull tankers during collisions, groundings and strandings; and instability, collapse and fatigue strength.

Chapter 3 describes the design objectives and philosophy, the design of the large-scale test specimen and the experimental program. The experimental program section describes the test objectives, the test set-up, instrumentation, experimental method and the corresponding ancillary tests for the steel, elastomer and steel-elastomer-steel composite structural laminate materials.

The test results are presented and discussed in Chapter 4. The general behaviour, load-displacement curves, strain distributions, stress distributions and failure modes of

the large-scale test specimen are presented along with the material properties of the steel, elastomer and steel-elastomer-steel composite structural laminate.

A summary, conclusions, recommendations and future work are presented in Chapter 5.

Chapter 2

A State-of-the-Art Literature Review and Proposed Design Concept

2.1 The Need and Standards for Double Hull Oil Tankers

The current world cargo-carrying fleet measures a total of 266 million deadweight tonnes (DWT), down from a peak in the early 1980's of 330 million DWT. Of that total, the largest portion is made up of oil tankers, with a fleet of 6761 vessels. Oil tankers have also led the world in total cargo loss, with 38 percent of all ship-related accidents, (Ohtsubo *et al.*, 1997). Table 2.1 lists 50 major oil spills from tankers and combined carriers in 1960 and from 1965 to 1989. Since that time, 188 ships greater than 100 gross tonnage (GT) were declared total losses in 1995, representing 62 percent of the gross tonnage lost during that year, (Lloyd's Register, 1996). Table 2.2 lists 18 tanker and combined carrier oil spills from 1990 to 1997 that were newsworthy. It is apparent that even with improved navigational capabilities, automation and highly trained mariners, shipping lanes will remain congested and accidents or extreme load events will continue to occur. Hence, it is imperative that better structural systems be developed to mitigate or, if possible, eliminate any oil outflow that may result.

The most noted oil spill that caused a turning point in legislation for oil tanker technology was that of the grounding of the Exxon Valdez in March 1989. Even though

Table 2.1: 50 Major Oil Spills from Tankers and Combined Carriers in 1960 and from 1965-1989 (National Research Council, 1991)

Rank	Name	Tanker Size (DWT)	Spills (No.)	Location	Date				
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, Mediter.	1989			x	
12	Nova	239,435	68	Iran, Gulf of	1985	x			
13	Wafra	68,600	62	South Africa	1971		x		
14	Epic Colocotronis	64,000	58	West Indies	1975		x		
15	Sinclair Petrolere	56,000	57	Brazil	1960			x	
16	Yuyo Maru No 10	52,836	42	Japan	1974	x			
17	Assimi	59,032	50	Oman	1983			x	
18	Andros Partia	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		

Table 2.1 (cont'd): 50 Major Oil Spills from Tankers and Combined Carriers in 1960 and from 1965-1989

Rank	Name	Crude Oil Spilled (bbl)	Spillage Category	Location	Date	11	13	12	12	2
27	Napier	38,561	37	Chile	1973		x			
28	Exxon Valdez	214,861	36	USA, Alaska	1989		x			
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**	Bermuda	1970				x	
36	Irenes Challenge	34,884	31	Pacific Ocean	1977				x	
37	Argo Merchant	29,691	28	USA, East Coast	1976		x			
38	Heimvard	55,000	31	Japan	1965	x				
39	Pegasus	---	25**	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			
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, Newfoundland	1988			x		
48	Venail	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 of Accidents

11 13 12 12 2

Total Metric Tons ($\times 10^3$)

792 693 851 653 52

Table 2.2: Major Oil Spills from Tankers and Combined Carriers from 1990-1997 (Facts on File, 1990 to 1997)

Rank*	Name	Tanker Size (DWT)	Spillage (x 10 ³)	Location	Date	CO	GRN	F/E	S/H	N/K
1	Haven	---	150	Italy, Gulf of Genoa	1991				x	
2	Mega Borg	---	145	USA, Gulf of Mexico	1990				x	
3	Braer	---	98	Scotland, Shetland Isld.	1993			x		
4	Aegan Sea	---	81	Spain, La Coruna	1992			x		
5	Agip Abruzzo	---	70	Italy, Livorino Harbour	1991	x				
6	Sea Empress	147,000	59	West Wales, Great Britain	1996			x		
7	Maersk Navigator	---	25	Straits of Malacca	1993	x				
8	Evoikos	130,000	25	Singapore Strait, Singapore	1997	x				
9	Nassia	133,000	18	Bosphorus Strait, Turkey	1994	x				
10	Diamond Grace	147,000	12	Yokohama, Japan	1997		x			
11	Kirki	---	11	Australia, Indian Ocean	1991				x	
12	Nakhodka	20,000	5	Japan	1997		x			
13	Morris J. Berman	---	3	Puerto Rico, San Juan	1994	x				
14	North Cape	---	2.6	Point Judith, Rhode Island	1996			x		
15	Shinoussa	47,000	2	USA, Texas	1990	x				
16	American Trader	---	2	USA, California	1990					x
17	B. T. Nautilus	65,000	2	USA, New York		x				
18	Iron Baron	---	0.8	Tasmania, Australia	1995		x			
Total Number of Accidents					7	7	3	1	0	
Total Metric Tons (x 10 ³)					145	258.4	306	2	0	

* Ranking is based on barrels spilled.

** Spillage data in doubt.

Code:

CO = Collision

GRN = Grounding/Stranding

F/E = Fire/Explosion

S/H = Structure, Hull, or Machinery Failure

N/K = Not Known

the Exxon Valdez was in full compliance with MARPOL 73/78 requirements, which were the standard for vessel design in 1989, the structure was not sufficient to prevent oil outflow due to a low-energy grounding event. The oil spill in environmentally sensitive waters and subsequent public pressure led to the implementation of U.S. legislation in 1990, the U.S. *Oil Pollution Act of 1990* (OPA 90). OPA 90 dictates that for tankers of U.S. registry and those of foreign registry operating in U.S. waters:

Double hull construction is required for all new oil tankers 5,000 gross tons and over. For new tankers less than 5,000 gross tons, double hull construction or a "double containment" system of equivalent effectiveness is required. These requirements apply to oil tank vessels whose construction or major conversion contract was placed on or after 30 June 1990, or whose delivery was arranged on or after 1 January 1994.

Existing single-hulled tankers 5,000 gross tons and above are required to be retrofitted with a double hull or be removed from service (subject to a number of exemptions) by 1 January 2010. To alleviate the impact on existing tanker fleet, OPA 90 phases out single-hulled tankers using an incremental schedule based on vessel size and age.

Existing tank vessels of less than 5,000 gross tons; those vessels delivering oil to a lightering operation more than 60 miles offshore; and those vessels unloading at a licensed deepwater port are exempt from the double hull requirements until 1 January 2015. Existing tankers with an effective double bottom or double sides in way of the cargo compartment are permitted a five year extension beyond their mandated retirement date, provided this does not exceed the absolute deadline of 2015, (Ryan, 1994).

Canada also took steps to implement requirements for new and existing tankers operating in Canadian waters by introducing the *Standards for the Double Hull Construction of Oil Tankers* (TP 11710). As of July 6, 1993, it is required that:

For new oil tankers and existing crude oil tankers of 20,000 tonnes deadweight (DWT) or greater and existing product carriers of 30,000 tonnes deadweight (DWT) or greater, the International Maritime Organization's (IMO) International Convention for the Prevention of Pollution from Ships, 1973 and its associated Protocol of 1978 (MARPOL

73/78), specifically Regulations 13F and 13G and related amendments to Annex I of MARPOL 73/78; and

For existing crude oil tankers less than 20,000 tonnes deadweight (DWT) and existing product carriers less than 30,000 tonnes deadweight (DWT), the U.S. Oil Pollution Act of 1990, with specific reference to the United States Coast Guard Interim Final Rule on Double Hull Standards for Vessels Carrying Oil in Bulk, issued 12 August 1992, (Canadian Coast Guard, 1993).

With respect to double hull construction, the U.S. legislation and Canadian regulations only prescribe the minimum hull spacing requirements and the structural requirements are covered by one of the Classification Societies.

The requirements for double hull construction of new tankers are such that the entire length of the cargo tanks are protected by ballast tanks or spaces other than cargo and oil fuel tanks as follows:

- (a) wing tanks or spaces shall extend 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, nowhere less than the distance w which, as shown in Figure 2.1, is measured at any cross-section at right angles to the side shell, specified as follows:
 - (1) for tankers of 5,000 DWT or greater, w is taken as the lesser of 2.0 m or that given by Equation 2.1, but should not have a value less than 1.0 m.

$$w = 0.5 + \frac{\text{DWT}}{20,000} \quad (2.1)$$

Note: h and w are expressed in meters

Figure 2.1: Double Hull Spacing Requirements (after Canadian Coast Guard, 1993)

- (2) for tankers of less than 5,000 DWT,

$$w = 0.4 + \frac{2.4 \text{ DWT}}{20,000} \quad (2.2)$$

with a minimum value of $w = 0.76$ m;

- (b) at any cross-section, the depth of each double bottom tank or space shall be such that the distance h between the bottom of the cargo tanks and the moulded line of the bottom shell plating measured at right angles to the bottom shell plating as shown in Figure 2.1 is not less than specified as follows:

- (1) for tankers of 5,000 DWT or greater, h is taken as the lesser of 2.0 m or that given by Equation 2.3, but should not have a value less than 1.0 m

$$h = \frac{B}{15} \text{ (m)} \quad (2.3)$$

where, B is the breadth of the tanker in meters,

(2) for tankers less than 5,000 DWT

$$h = \frac{B}{15} \text{ (m)} \quad (2.4)$$

but not less than 0.76 m.

It was demonstrated by Monnier, (1995) using the Det Norske Veritas (DnV) probabilistic model for the calculation of oil outflow from tankers during collisions and groundings developed by *Hysing et al.*, (1993), that a 280,000 DWT double hull tanker performed better than a similar sized single hull tanker with a probability of no oil outflow from a grounding accident of 82 percent compared to 10 percent for the single hull tanker. When coupling the oil outflow model with a cost model, it was demonstrated that the maximum oil spill cost for the double hull tanker was \$1.6 billion US, whereas, the single hull tanker would cost nearly four times that figure at \$6.3 billion US. The mean cost values of the oil spills were \$16.68 million US and \$139.95 million US, for the double hull and single hull tankers, respectively. The above figures were based on the information gained from 10,000 accident simulations.

The average spill cost, based on 1992 US dollars, was \$42,452 per ton as determined by Mercer Management Consulting. An oil spill cost database was prepared by Mercer based on information from 21,511 tanker vessel oil spills, accounting for over 356,679 metric tons of oil shed in US waters, to generate a cost figure for oil spills in US waters.

The punitive damages, excluding the clean-up costs that have exceeded \$2.0 billion US, resulting from the Exxon Valdez grounding accident which merely spilled 20 percent of its total cargo, has reached \$5.058 billion US, (National Research Council, 1991). After the Braer accident, salmon fish farmers were awarded compensation totaling over \$45 million US due to the replacement of the salmon population in fish farms.

2.2 Advanced Double Hull Tanker Designs and Design Criteria

This section presents a representative sample of the state-of-the-art designs in oil tanker technology that were published and the design criteria for Advanced Double Hull (ADH) tankers. The designs include: the EPOCH Mark II Unidirectional-Girder-System introduced by Hitachi-Zosen; a baseline 40,000 DWT Conventional Double Hull (CDH) produced by the Maritime Administration (MARAD) with subsequent ADH alternatives produced by the Carderock Division, Naval Surface Warfare Center (CDNSWC); and the MarC Guardian curved plate design produced by Marinex. Section 2.4 presents a discussion on the behaviour of some of these designs to collisions, groundings and strandings, in-plane loading and fatigue strength.

2.2.1 EPOCH Mark II Unidirectional-Girder-System Design Concept, Okamoto *et al.*, (1985)

The EPOCH Mark II (Easy handling/maintenance Product Oil Carrier by Hitachi-Zosen) 40,000 DWT tanker, as illustrated in Figure 2.2 (a), incorporated the unidirectional-girder-system that completely isolated the inner hull with only longitudinal girders. Each cargo hold was entirely surrounded by a double hull structure consisting of a double bottom, double side shells, a double deck and double-skin transverse bulkheads. The double hull spaces were used as segregated ballast tanks. The general Rules and Regulations from the American Bureau of Shipping (ABS), Lloyd's Register and Det Norske Veritas (DnV), for sizing strake scantlings, were found not to be applicable for

the design. The Rules and Regulations, however, allowed for innovative designs to be approved by the Classification Societies by using the “design by analysis” approach which demonstrated that it satisfied all the general requirements.

Figure 2.2: EPOCH Mark II Design Concept (after Okamoto *et al.*, 1985)

Based on the “design by analysis” approach, Okamoto *et al.*, (1985) concluded that, as a result of no transverse elements between bulkheads, the lateral loads were directly transmitted to the transverse bulkheads through the longitudinal girders in the double bottom and the stringers in the side shell, as illustrated in Figure 2.2 (b). To account for the lack of transverse framing, the transverse bulkheads were made stiffer than conventional designs and each transverse bulkhead required a strong stool to ensure compatible transverse stiffness to the elements (longitudinal girders) framing into it. The intersection of the longitudinal girders at the stools was a fatigue-sensitive detail and had to be designed accordingly.

Furthermore, it was found the transverse in-plane strength of the hull plates was small and considered negligible for design. The transverse bulkhead carried the transverse forces from the sea hydrostatic pressures that caused compression in the side shell hull plates.

It was also found that the girder spacing had to be made small (equivalent spacing to stiffener spacing in the CDH design) to decrease the longitudinal flexibility and to limit the transverse deflection to 1/1000 of the span for the severest loading condition. In addition, horizontal and vertical stiffeners were required on the longitudinal girders to prevent buckling.

As part of the “design by analysis” approach, Okamoto *et al.*, (1985) explicitly considered the MARPOL provisions related to damaged conditions for ship stability and the prevention of ocean pollution. The failure modes cited as being included in the design were: stranding, where the double bottom is penetrated or where the outer hull has been breached and the longitudinal girders damaged (buckled); slamming, where a portion of the outer bottom hull is considered ineffective; and transverse buckling of the inner and outer bottom hull due to transverse and longitudinal load effects. The residual strength of the damaged vessel was also considered in the design.

Okamoto *et al.*, (1985) concluded that the unidirectional-girder-system was structurally feasible and realized the following advantages:

- the number and extent of structural intersections (discontinuities) was reduced;
- due to simplicity of the design, it was easier to analyze the strength of each member;

- assembly and construction was simplified;
- the completely flush cargo tank surface facilitated painting, reduced unloading time and permitted quicker and easier cleaning of tanks;
- inspection was made easier;
- since all members, except the transverse bulkheads, provided longitudinal strength to the structure, the unidirectional-girder-system provided a larger section modulus than conventional systems; and
- the torsional strength and stiffness of the hull-module were increased with the use of a multi-cell structure.

2.2.2 MARAD PD-328 Design Concept and ADH Design Alternatives, Melton *et al.*, (1994) and Sikora *et al.*, (1995)

The design of a series of 40,000 DWT double hull product tankers was undertaken by the Maritime Administration (MARAD) as directed by the Secretary of Transportation and by the Carderock Division, Naval Surface Warfare Center (CDNSWC). The principal dimensions and weights of the MARAD PD-328 (Maritime Administration, 1991) CDH baseline design vessel and the ADH design alternatives proposed by the CDNSWC (Melton *et al.*, (1994) and Sikora *et al.*, (1995)) are given in Table 2.3. The MARAD PD-328 represented a commercial product tanker design with a service speed of approximately 14 to 15 knots.

The PD-328 double hull tanker was designed to meet side raking (Beam/20 transverse penetration and 0.5 length between perpendiculars (LBP) longitudinal extent

of damage from the forward perpendicular aft) and bottom raking (Beam/20 transverse penetration and 0.75LBP longitudinal extent of damage from the forward perpendicular aft) damage criteria, (Maritime Administration, 1991).

The layout of the PD-328 had the 126.7 m of available tank length divided into 5.6 m fuel tanks, a 4.2 m slop tank, and five port/starboard pairs of cargo tanks, one of which was 19.6 m long, three 24.5 m long and one 22.4 m long, as illustrated in Figure 2.3. The cargo capacity at 100 percent was approximately 54,400 m³. Water ballast was carried in the double bottom which spans the full breadth, port and starboard wing tanks in the double hull spaces, and in the peak tanks. A midship section is shown in Figure 2.4.

Table 2.3: Design Characteristics of the MARAD PD-328 Baseline Design Vessel and CDNSWC Design Alternatives (after Melton *et al.*, 1994 and Sikora *et al.*, 1995)

Design	MARAD PD-328	CDNSWC Mod.1	CDNSWC Mod.2	CDNSWC ADH-II
Length Overall (LOA)	192.0 m	192.0 m	192.0 m	192.0 m
Length Between Perpendiculars (LBP)				
Beam, Molded (B)	178.5 m	178.5 m	178.5 m	178.5 m
Depth at Side, Molded (D)	32.2 m	32.2 m	32.2 m	32.2 m
	18.9 m	18.9 m	18.9 m	18.9 m
Design Draft (T)	11.48 m	11.48 m	11.48 m	11.48 m
Displacement at Design Draft	53,560 mt	53,560 mt	53,560 mt	53,560 mt
Deadweight, total	43,660 mt	43,135 mt	43,760 mt	43,798 mt
Cargo capacity, 100%, ex slops	54,400 m ³	51,900 m ³	52,200 m ³	52,200 m ³
Ballast capacity, ex peak tanks	16,600 m ³	15,800 m ³	15,800 m ³	15,800 m ³
Weights:				
- Steel	7,430 mt	7,890 mt	7,350 mt	7,292 mt
- Outfit	1,610 mt	1,675 mt	1,590 mt	1,610 mt
- Machinery	860 mt	860 mt	860 mt	860 mt
- Light Ship	9,900 mt	10,425 mt	9,800 mt	9,762 mt
Other:				
- Cargo Tank Arrangement	5 x 2	8 x 2	8 x 1	5 x 2
- Longitudinal Centerline Bulkhead	Yes	Yes	No	Yes
- Deck Camber	0	0.61 m	0.61 m	0.61 m

Note: CDNSWC Modification 1 is also referred to as ADH-I in Sikora *et al.*, (1995)

Figure 2.3: Profile of MARAD PD-328 40,000 DWT Double Hull Tanker (after Melton *et al.*, 1994)

Figure 2.4: MARAD PD-328 40,000 DWT Tanker Midship Section (after Melton *et al.*, 1994)

The longitudinal framing was conventional with nominal transverse web spacing of 2.8 m. Oiltight bulkheads were of conventional stiffened plate construction. Produceability, ease of access for inspection and maintenance of the cargo tanks, and efficiency were considered as part of the design criteria. Using the current ABS Rules

and Regulations, a midship was designed and then redesigned using ABS's OMSEC computer program to reduce the total estimated steel weight by 11 percent. The repetition within the cargo length allowed for the use of modular construction during fabrication.

Using the MARAD PD-328 design as a baseline, modifications were made to create three new design alternatives: the CDNSWC Mod. 1 (also referred to as the ADH-I in reference Sikora *et al.*, 1994), the CDNSWC Mod. 2 and the CDNSWC ADH-II. These modifications were brought about as a result of the comments of several U.S. flag tanker operators regarding cargo tank geometry and structural details. The following design changes were incorporated:

- (1) The wing and double bottom depths were reduced from 2.25 m to 2.13 m, while maintaining sufficient volume for ballast.
- (2) A deck camber of 0.6 m was introduced.
- (3) The unidirectional framing system of the ADH alternatives limited the cargo tank length to approximately 15.0 m and, as such, the cargo tank arrangement was increased to an 8 by 2 array. Two alternatives were considered: (a) with a longitudinal centerline bulkhead (CDNSWC Mod.1) and (b) without a centerline bulkhead (CDNSWC Mod. 2).
- (4) A third alternate design (CDNSWC ADH-II) incorporated the original cargo tank arrangement and used a mid-tank frame (transverse web frame at mid-bay) to provide the necessary transverse support.
- (5) The longitudinal bulkheads, where fitted, were of a vertically fluted configuration with 0.9 m by 0.9 m corrugations and 1.5 m high longitudinal stools to limit vertical spans and to facilitate cleaning.

- (6) The transverse bulkheads were of double plate construction with 0.9 m spacing between plate surfaces and were fitted with 1.5 m high transverse stools.
- (7) The angle section deck longitudinals of the MARAD PD-328 were replaced with flat bars to facilitate cleaning and three equally spaced transverse deck webs were required.

The revised midship sections with the incorporated alternatives, CDNSWC Mod.1, CDNSWC Mod.2 and CDNSWC ADH-II are illustrated in Figures 2.5, 2.6 and 2.7, respectively. (Note: the unidirectional cellular framing as proposed in the ADH concepts is not explicitly described by ABS Rules and Regulations. The scantlings sizes were determined by assuming that the longitudinal girder, inner and outer hull plating form an I-beam, which was subjected to a combination of axial loads from hull-girder bending and secondary forces due to cargo and seaway hydrostatic pressures, and used the appropriate rational analysis.)

Sikora *et al.*, (1995) performed a detailed cost estimate of the baseline MARAD PD-328, the ADH-I and the ADH-II design alternatives which included labour, material properties and material costs and found that the alternative ADH designs were cost competitive. Table 2.4 details the weight and cost estimates for cargo sections of the 40,000 DWT tankers. (Note: the CDNSWC Mod. 2 referenced in Melton *et al.*, (1994) was not included in this cost comparison analysis.)

Figure 2.5: ADH CDNSWC Modification 1 (after Melton *et al.*, 1994)

Figure 2.6: ADH CDNSWC Modification 2 (after Melton *et al.*, 1994)

Figure 2.7: CDNSWC ADH-II (after Melton *et al.*, 1994)

Table 2.4: Weight and Cost Estimates for Cargo Sections of 40,000 DWT Tankers (after Sikora *et al.*, 1995)

Ship	MARAD PD-328	CDNSWC ADH-I	CDNSWC ADH-II
Labour Hours:			
- Cutting and Welding	258,249	239,034	222,978
- Handling, Jigs, Lofting, etc.	46,238	49,998	46,640
- Supervision and Overhead	112,660	106,942	99,782
- Total Hours	417,147	395,974	369,400
Labour Costs (\$US M)	12.514	11.879	11.082
Material Properties:			
- Plates and Shapes (mtons)	6,231	6,896	6,432
- Weld Length (ft)	453,441	454,942	424,333
- Scrap (mtons)	1,059	1,034	965
- Steel Ordered (mtons)	7,290	7,930	7,397
Material Costs (\$US M):			
- Plates and Shapes	4.421	4.854	4.528
- Weld Wire, Jigs, etc.	0.426	0.500	0.467
- Scrap Credit	-0.058	-0.057	-0.053
Total Material Costs	4.789	5.297	4.942
Total Costs (\$US M)	17.303	17.176	16.364

2.2.3 MarC Guardian Curved Plate Design Concept, Goldbach, (1994)

The MarC Guardian curved plate concept, illustrated in Figure 2.8, was developed with close cooperation between Marinex, Metro and Ingalls Shipbuilding, Inc. The objective was to design two tankers, a 348,000 DWT Very Large Crude Carrier (VLCC) and a 125,000 DWT tanker, that would be an efficient and reliable design that American shipbuilders could produce effectively. Marinex claims that structural form and construction of the MarC Guardian will benefit shipowners, shipbuilders and the environment.

Figure 2.8: Midship Section of the MarC Guardian Curved Plate Tanker Concept (after Goldbach, 1994)

Cited beneficial claims include:

- (1) The use of all mild steel construction resulted in lower stresses due to operational loads, increased fatigue and corrosion resistance, and hence, extended vessel life. (Note: the steel weight was approximately 20 percent higher by using all mild steel construction over high-strength steel.)**
- (2) Improved coating application for both the cargo and ballast tanks was expected due to the flush steel surfaces that resulted by avoiding the complexity of stiffened plates.**
- (3) The reduced structural complexity led to increased accessibility, and ease of inspection and maintenance.**
- (4) A reduction of welding and steel surface areas in ballast tanks resulted in lower construction and vessel maintenance costs.**
- (5) By maximizing the parallel midbody size (cargo tank area) and by incorporating modular construction (standardization of curved plate cells), the design resulted in a reduction in fabrication costs and an increased cargo tank capacity of approximately 20 percent over traditional designs.**

Note (1) The added cargo tank capacity was cited as the most advantageous economic benefit for this design, since it more than compensated for the lower speed and/or higher fuel consumption, expressed as a function of the block coefficient, C_b , which for this design was 0.89. Typical ADH tankers have a block coefficient of 0.8 (a higher C_b results in a less efficient design).

Note (2) The curved plate concept can only be incorporated within the parallel midbody portion of the tanker hull between the bow and stern, beyond which conventional stiffened-flat plate construction was used.

- (5) Inner hull retention of oil during a grounding.**
- (6) A high-energy absorption in the curved plate double hull.**

Although the MarC Guardian design did not necessarily produce the lowest initial construction cost, its high quality, environmental features, lowest cost per ton of oil delivered and long life expectancy made it a viable option.

2.3 Production Engineering of Double Hull Oil Tankers

Advanced double hull designs explicitly recognize in their design the advantages of modular construction by creating modular cargo tank sections (cells) that simplify the structure and maximize the number of long, straight welds and minimize the number of complex weld details. By doing the latter, the use of robotics can be introduced to complete the majority of the welding process. The CDNSWC investigated certain aspects of production engineering (modular construction and the application of welding technology) for the building of naval combatants and commercial tank vessels of advanced double hull construction. The results are discussed in the following sections.

2.3.1 Modular Construction, Sikora *et al.*, (1995)

Each ADH cargo tank section was subdivided into seven, fifty feet long modules: a strength deck, a double bottom structure, port and starboard side shell cells, port and starboard bilge keels and a longitudinal bulkhead, as illustrated in Figure 2.9. Transverse bulkheads were also of modular construction. The bow and stern sections of the hull were built and outfitted conventionally and added to the cellular cargo tank portion of the hull in the dry-dock. Separate bilge keel assemblies allowed for the use of flat plate construction for the double bottom. Cells oriented parallel to the ship centerline allowed for changes in the ship's beam away from the midship section. Similarly, a reduction in the number of cells in the side shell modules accommodated changes in the draft. The side and bottom longitudinal web members were made of identical flat plates with straight longitudinal stiffeners fabricated at the shipyard or at a remote site. Strength deck and longitudinal bulkheads used either longitudinally oriented ADH cells or conventional grillages. It was found that most American shipyards preferred to incorporate cellular longitudinal bulkheads in their designs.

By utilizing standardized modules, a large amount of steel handling, cutting, and welding could be performed at remote sites. Assembly of these modules could be done in the shipyard outside the dry-dock. These added features illustrate how the ADH structure lends itself to cost-effective modular construction techniques.

Figure 2.9: Modular Construction of an ADH Tanker (after Sikora *et al.*, 1995)

2.3.2 Applicable Welding Technology, Sikora *et al.*, (1995)

The cellular structure of the ADH design concept is well suited for assembly line techniques in which both the shell plates and internal web plate are joined using the high-deposition-rate (approximately 23.2 kg/hr), 100 percent duty cycle electro-gas welding (EGW) process. The EGW process is a vertical-up casting process which uses a welding arc to provide the molten filler metal which produces its own protective shielding gas and fluxing agent. The process virtually eliminates slag removal, grinding and problems

associated with two-sided welding. The electrode is supplied in a drum which is carried alongside the welding machine, along with the wire feeder, arc controller, cab speed controller, oscillator controller, compressed air and cooling water manifolds, manual torch positioning slides and the welding operator. Other costs associated with plate preparation, fit-up and assembly are reduced as the modular construction utilizes simplified joint details (one square flame cut per plate edge, no edge grinding). When compared to standard shipyard welding processes, such as flux cored arc welding (FCAW) and submerged arc welding (SAW), which require 10 to 16 hours and 5 to 8-1/2 hours for the same weld, respectively, it is evident that the EGW process may be advantageous. However, cited constraints of the EGW process which may limit its applicability include:

- (1) it requires close operator attention;
- (2) it can be used for vertical welding only;
- (3) it is not completely automated; and
- (4) it can only be used on plates of an approximate thickness of 25 mm.

The electroslog welding (ESW) process, although somewhat similar to EGW, uses a granulated flux to protect the weld pool and relies on the resistance heating of a molten flux bath to create a weld that is more tolerant to environmental and operational changes. The ESW process is expected to have the same weld deposition rates as the EGW process and can also be used in the down welding position. However, the same plate thickness constraint for the EGW process also applies to the ESW process.

The Programmable Automated Welding System (PAWS) can be used on a varying degree of plate thicknesses and incorporates off-line programming with real-time process controls. The off-line programming allows the user to model the parts to be welded and the type of weld to be used. The real-time controller utilizes multiple sensors to detect the joint, to measure the topography, to monitor weld pool location and size, to determine weld metal cooling rates, to monitor arc stability and contact tip wear, and to measure hydrogen in the arc shielding gas. In addition to these features, PAWS can be operated both vertically and horizontally and the equipment can fit into cells measuring one meter by one meter. Fatigue tests conducted by Sikora *et al.*, (1995) indicated that original and repaired weld joints made with PAWS exceeded the fatigue requirements (typical of ship structures and similar to those found in highway bridges) dictated by the American Association of State Highway and Transportation Officials (AASHTO).

2.4 Effectiveness of Double Hull Oil Tankers

2.4.1 Behaviour of Double Hull Oil Tankers During Collisions, Groundings and Strandings

The objective of any new double hull design is to reduce or, if possible, prevent oil outflow for accidental or extreme load events and provide sufficient residual hull strength for salvage operations. The designs must meet all the structural requirements under normal operating conditions as specified by Classification Societies, while remaining cost effective by reducing complexity and weight of the vessel and by ensuring

produceability and ease of inspection and maintenance. The following sections present a summary of relevant papers on the behaviour of double hull oil tankers during collisions, groundings and strandings which gives the state-of-the-art in understanding the performance of this technology under accidental or extreme load events.

2.4.1.1 Tanker Structure Behaviour During Collisions and Groundings, Daidola, (1995)

Daidola, (1995) presented a summary of test data and inelastic finite element analysis comparisons for: (1) low-energy impact collisions on full-scale inland waterway 1000 DWT tankers; (2) cone penetration tests for groundings on a model of a 280,000 DWT double bottom tanker (scale not given); and (3) raking damage for single and double hull models of VLCCs (scale not given). Collisions are characterized into one of the following categories: (a) tanker colliding with a rigid object or another vessel, or (b) another vessel colliding with a tanker. The former leads to bow damage and the latter leads to side-shell, deck and bottom damage as the colliding ship penetrates the struck tanker. Groundings are characterized by an impact to the bottom hull structure by an underwater obstruction with raking damage as the tanker passes over that obstruction. Further damage may result if the vessel becomes stranded on the obstruction. The experimental results were compared against a simplified static analysis method for calculating structural damage and the energy absorption in low-energy collisions. The method, called the Tanker Structural Analysis for Minor Collisions (TSAMC), was developed by the U.S. Coast Guard, and was verified and confirmed against MSC/DYA for side shell, web frame and deck distortions due to collisions and bottom structural

damage due to groundings. TSAMC utilized a simplified model for analyzing the behaviour of a struck ship which involved longitudinal plastic bending and plastic membrane tension of the stiffened hull plating and deck; and the yielding, buckling, and/or shearing of web frames. Details of mesh and element types, boundary conditions and solution strategies for MSC/DYA were not given.

Based on observations of the experimental test results and the analysis, Daidola, (1995) summarized the behaviour and energy absorption capacity for various structural elements (stiffeners, web frames, transverse floors, longitudinal girders, brackets, deck edges, outboard bilges and transverse bulkheads) found in conventional single, CDH and ADH designs during accidental or extreme load events. In addition, Daidola, (1995) made recommendations with respect to the designs to increase the energy absorption capacity. The summary and recommendations are outlined in the following paragraphs.

The use of stiffeners on shell plating restricts membrane tension of the shell until the stiffeners buckle or rupture. If the stiffener tip ruptures, it is assumed that the rupture opens up and the crack propagates into the supporting shell plating. If the stiffener buckles, the stiffened plates are assumed to go from bending to membrane tension until rupture is initiated. With rupture, the fracture will propagate throughout the stiffened plate.

The rigid web frames in a single-hulled vessel do not allow for significant energy absorption of the shell plating during a collision. The web frames in a double-hulled vessel must be spaced as far apart as possible and be made as flexible as permissible (web distortions, buckling) to ensure membrane tension in the side shell prior to premature rupture of the shell plating.

The transverse floors behave similar to web frames during groundings. During groundings with raking damage, the floors absorb energy in distortion and rupture as the rock passes through it. By adding more transverse floors, the length of raking damage, relative to systems with fewer floors, is shortened. However, the lateral deflections and energy absorption capacity are lessened for stranding conditions.

The longitudinal girders of the ADH design transmit loads to the transverse bulkhead and support the outer shell plating. It has been theorized that in order to be effective in energy absorption capacity, the inner and outer hulls of the ADH design must distort simultaneously under membrane tension. To achieve this, a balance between the arrangement and scantlings of the shell plating, girders and transverse bulkheads must be achieved to maintain structural integrity, stability, collapse and fatigue requirements without premature rupture of the shell plating.

Brackets are rigid supports that create hard spots in the shell plating which can lead to premature fracture or rupture.

A strong deck capable of ripping free of the side shell plating, web frames and transverse bulkheads during a side collision is recommended. In this form, the deck absorbs energy through failure by transverse rupture from longitudinal membrane tension.

The outboard bilge forms a rigid support that, during a collision, may cut into the bow of the striking vessel, exposing sharp structures which may puncture the struck tanker's side before full membrane tension can be developed. To increase the energy absorption capacity of the side shell, a less-stiff outboard bilge should be used.

The transverse bulkheads are very stiff structural elements that provide transverse rigidity to the tanker. Typically, these do not buckle, yield or rupture and may limit the extent of side shell deflections and the energy absorption capacity. To increase the energy absorption, either flexible bulkheads should be incorporated or the distance between the bulkheads should be increased.

2.4.1.2 Collision Resistance of Side Structures, Kitamura, (1996)

The relevance of presenting the Kitamura, (1996) paper is that it describes an analytical technique for determining the relative energy absorption capacity of different designs for accidental or extreme load events. The energy absorbed can then be compared with the kinetic energy which must be dissipated for these events. Analysis of this type may give guidance to Classification Societies in stipulating the design requirements for these load events.

Large-scale collision experiments were conducted by Kitamura, (1996) to study the crashworthiness of two proposed double hull side structures subjected to a simulated collision event. The first proposed design concept substituted the conventional stiffened plate side shell with a fabricated double plated panel side shell with inside longitudinally inclined frame webs (designated as frame panel). The second proposed design concept was a corrugated plate panel; however, details were not given since this concept was discarded during the investigation as being a non-viable alternative. Dynamic collision experiments were performed on the frame panel and corrugated panel to investigate the

damage and failure modes, and to provide experimental data for which numerical simulations could be verified.

Kitamura, (1996) hypothesized that:

- the frame panel would offer improved energy absorption capacity by increasing the elongation of the hull plates by reducing the stress concentrations (which might cause premature rupture of the hull plate). These elongations may be further increased by the addition of soft nose connecting brackets arranged with shorter spacing and/or outer plates that are not directly connected with transverse webs; and
- the void space of the frame panel (approximately 300 mm) might combat against potential buoyancy loss in case of raking damage.

The cited potential disadvantages of the frame panel are a reduction in global bending rigidity and in the effective shear area against hull-girder shear forces.

Kitamura, (1996) hypothesized that the corrugated panel would have the same improved energy absorption characteristics as the frame panel. In addition to the disadvantages cited above, the corrugated panel would have diminished fatigue strength due to the residual stresses in the corner radii of the corrugated steel plate.

The construction of the test models and the test support system were not well defined in the paper. The cited test specimen dimensions were a plate thickness of 6 mm, a width of 750 mm and a depth of 150 mm for the frame panel and a plate thickness of 6 mm, a pitch of 250 mm and a depth of 100 mm for the corrugated panel. The frame and corrugated test panels were supported by rigid transverse webs (spaced at 2000 mm)

vertically with connecting brackets and by thick extension plates in the longitudinal direction. The panels were repeatedly struck by a free-falling rigid bow (shape not defined), weighing 8.44 tons, which was dropped from a height of 4.9 m at a maximum speed of 10 m/s and with a maximum energy supply of approximately 405,300 J. The bow was guided via guide rails mounted on the side frames. The bow penetration, panel deflection, contact force between the bow and the panel and strains were measured.

At the first stage of bow penetration, the inclined webs of the frame panel buckled locally and the upper plate (contact surface) was indented at the center. At the middle stage, the upper plate was deformed extensively and the webs were collapsed along their length. During the last stage, a crack initiated at the outer edge of the upper plate and propagated transversely into the webs causing a partial rupture. The average plastic strain at this stage was approximately 15 percent. The first, middle and last stages were not defined.

At the first stage of bow penetration, the corrugated panel had large deflections and the side edges turned upwards due to the difference in lateral bending rigidity. The corrugated panel ruptured after “one plus alpha” (not defined) collisions with an average plastic strain of approximately 5 percent.

Numerical simulations were performed for each panel type with measured geometric and material properties (further numerical simulation details were not given). Good correlation with the test results was achieved for the frame panel only. Based on this test alone, Kitamura, (1996) used an explicit finite element core code (Livermore Software: LS-DYNA3D) and newly developed user subroutines, specially customized for ship collision and groundings, to analyze a midship tank structure. The model was a half-

tank length side shell structure (port side only) subjected to a collision for various side shell designs to evaluate their energy absorption capacity, as listed under the design heading in Table 2.5. The energy absorption capacity of a standard double side 300,000 DWT VLCC was used as the baseline, designated as “A” in Figure 2.10, and assessed a crashworthiness index of 100 based on the energy absorbed, 230,400 KJ. Key indicators of the energy absorption capacity for each design are illustrated in Table 2.5 and Figure 2.10 and are defined as the bow penetration of the striking vessel in which rupture of the struck vessel’s side shell occurred with its corresponding energy absorption. The same is given for the rupture of the longitudinal bulkhead of the struck vessel. The crashworthiness index listed in Table 2.5 is defined as the ratio of the absorbed energy of the longitudinal bulkhead of each design with respect to the baseline vessel. A summary of the energy absorption-bow penetration associated with side shell and longitudinal bulkhead rupture for each design are given in Table 2.5 and illustrated in Figure 2.10.

Table 2.5: Numerical Collision Simulation Results and Crashworthiness Indices (after Kitamura, 1996)

Design	Side Shell		Longitudinal Bulkhead		Crashworthiness Index
	Rupture Penetration (m)	Absorbed Energy (kJ)	Rupture Penetration (m)	Absorbed Energy (kJ)	
Standard VLCC	1.750	73,600	5.000	230,400	100
YP350 Class HT	2.125	106,000	5.250	286,800	124
Additional Side Stringers	1.750	76,700	4.875	248,600	108
Top Side Tank	1.875	84,000	5.625	283,100	123
Strut in Cargo Oil Side Tank	1.750	76,700	5.000	228,800	99
ADH	2.000	83,500	5.000	287,300	125
Frame Panel	2.000	131,500	5.375	404,500	176

Figure 2.10: Comparison of Energy Absorption Capacity (after Kitamura, 1996)

The model of the striking ship was taken from the bow a 150,000 DWT Suez Max class having a raked stem of blunt form with a regular flat bulbous bow. The VLCC was constructed with YP315 HT steel. The struck ship was assumed fully loaded while the striking ship was considered in normal ballast condition. The upper part of the double side struck ship was assumed to be damaged by the striking bulbous bow which was in line with the transverse web at the midpoint between the transverse bulkheads.

Six other designs were tested, including a modified side shell VLCC with the frame panel, to compare the energy absorption capacity of the designs to the baseline design. The first design alternative, "B", was the VLCC completely constructed with

YP350 class high tensile steel. The second design alternative, "C", had three additional side stringers arranged in the double side. The third design, "D", incorporated a top side tank (similar to ore/bulk/oil carriers) at the top of the double side. An additional side stringer was provided at the bottom of the top side tank. The web thickness of the double side was reduced slightly due to the shorter unsupported span. The fourth design, "E", had struts that were arranged in the cargo oil side tank that supported the double side directly. The transverse web thickness was reduced accordingly. The fifth design, "F", was an ADH design with side stringers stiffening the shells and longitudinal bulkhead. The sixth design, "G", was the modified VLCC with the frame panel in the side shell as illustrated in the midship section in Figure 2.11. Longitudinal girders were used in the top side and hopper tanks. Struts were used in the cargo oil center tank to avoid direct connection to the frame panel.

Figure 2.11: Midship Section of Frame Panel VLCC (after Kitamura, 1996)

Kitamura, (1996) demonstrated that with the frame panel concept, which eliminated hard spots associated with stiffeners and provided a simplified structural form, that the crashworthiness could be improved by 76 percent. (Note: the production engineering, costing and feasibility of the frame panel design still need to be investigated.)

2.4.1.3 Large-Scale Double Hull Tanker Grounding Experiments, Rodd and McCampbell, (1994) and Rodd, (1996)

The CDNSWC conducted five large-scale grounding experiments, as part of an eleven large-scale grounding experimental series, to evaluate the “crashworthiness” (structural failure mechanisms and energy absorption capacity) of various oil tanker structural configurations. The designs represented tankers in the 30,000 to 40,000 DWT size range. These grounding experiments are being used to validate and improve similar investigations using analytical and numerical modelling techniques.

The Grounding Test Machine was used to test the one-fifth scale models. The test model was carried on a test vehicle (ballast railway cars with a floating shock platform) that traveled down inclined tracks to the impact area. The impact area had an exposed instrumented mechanical “rock” that was fixed to a 1200 tonne reinforced concrete foundation. An inclined model was used to observe the actual initiation of inner hull rupture along the length of the model. The position of the model was such that the rock tip would enter the model at the leading edge just below the inner hull and exit high enough above the inner hull to ensure its rupture, as shown in Figure 2.12. The test model

generally impacted the rock at speeds of 12 knots (6.1 m/s). The rock was shaped like a 90 degree cone, 915 mm (36 inches) high, with a 178 mm (7 inches) radius spherical tip. Fore/aft and vertical base reaction forces at the rock were recorded.

Figure 2.12: Double Hull Grounding Test Model Installation (after Rodd and McCampbell, 1994)

The first experiment (designated as CONV DH) represented a conventionally framed double hull tanker bottom (representative of the Navy T-5 *Paul Buck* class tanker) which provided the baseline information for the single cargo tank length ADH design experiments. The CDH model structure included the double bottom structure beneath one oil tank, the forward and aft transverse bulkheads, and heavy sideplates that represented the vertically stiff structure along the side of the cargo tank, as illustrated in Figure 2.13.

The MARAD PD328 CDH (designated as CONV/PD328) tanker was selected to represent the baseline for the double cargo tank length ADH models, as illustrated in Figure 2.14.

The first ADH model (designated as ADH/PB) was used as a comparison to the conventionally framed *Paul Buck* (PB) class which was characterized by wide spacing of

longitudinal girder webs, conventional plate/stiffener bulkheads and thinner plating for the inner hull as compared to the other designs in the series. The ADH/PB is illustrated in Figure 2.15.

The next ADH model (designated as ADH/PD328) was an ADH alternative to the MARAD PD328 CDH. The model featured closer spacing of the longitudinal girders, double-plate transverse bulkheads with haunched stools and a slightly thicker inner hull plating, as illustrated in Figure 2.16.

The final ADH model (designated as ADH/PD328V) was similar to the ADH/PD328 except that closely spaced vertical stiffening was used on all longitudinal girders and the web plating thickness was increased to that of the inner hull plating, as illustrated in Figure 2.17. This model was used to evaluate the merits of tightly spaced vertical stiffeners on longitudinal girders since buckling of these elements will limit inner hull lift. Inner hull lift will provide the design with more than 2.0 meters of protection.

Figure 2.13: CONV DH Grounding Model (after Rodd and McCampbell, 1994)

Figure 2.14: Structural Details of CONV/PD328 Grounding Model (after Rodd, 1996)

Figure 2.15: Structural Details of ADH/PB Grounding Model (after Rodd, 1996)

When comparing the various models, the first parameter was the amount of vertical intrusion tolerated before the rupture of the inner hull. The second parameter was the amount of energy absorbed once inner hull rupture had occurred. Once the inner hull was ruptured, a higher energy absorption capacity would stop the vessel sooner and leave fewer tanks breached.

Rodd and McCampbell, (1994) made several observations of the first grounding experiment on the single cargo tank length CONV DH model:

- (1) The outer hull was ruptured and plastically deformed to the shape of the rock along the full length of the model.
- (2) The rock passed through the aft transverse bulkhead rupturing it at the point of impact and plastically deformed it and curled the ruptured edges up and away from the centerline.
- (3) As expected, the model decelerated while passing over the rock from 11.9 knots (6.1 m/s) to 7.2 knots (3.7 m/s). Residual velocity indicated that further raking damage would have occurred along the tanker length.
- (4) The kinetic energy dissipated was 2.7×10^6 N-m (2.0×10^6 ft-lbs). Energy dissipation was measured from either the area under the load-deflection curve or by using $1/2mv^2$.
- (5) The initiation of inner hull rupture resulted from the propagation of vertical cracks from the transverse floors that encountered the rock. This design resulted in a Type I crack propagation event which was subsequently characterized by Rodd, (1996) in a detailed description that follows.

- (6) The inner hull crack split and propagated forward toward the oncoming rock tip and aft 305 mm (12 inches) to 610 mm (24 inches) in advance of the rock. The latter crack propagated into the next oncoming transverse floor generating other cracks that propagated into these members cutting the structure in half before the rock tip reached the transverse member. Once the rock tip encountered the propagating cracks of the inner hull, membrane stressing resulted in a catastrophic, high-speed rupture of the plating.

Based on the above test and subsequent grounding tests, Rodd, (1996) characterized rupture and crack propagation events as one of three types, as illustrated in Figure 2.18. All crack propagation events began with longitudinal cracks that extended forward of the initial rupture of the outer hull that was caused by the penetration of the rock tip locally. As this longitudinally oriented crack propagated past the first transverse weld, a Type I crack propagation event occurred. This event was characterized by a series of propagating cracks that first propagated vertically through the transverse floor. As the rock passed through that member, the vertical crack then opened and continued to propagate at high-speed into and along the inner hull where the crack then split and propagated in the forward- and aft-moving directions. Because of the sloped rock face, this occurred before the rock tip arrived at the transverse floor intersection. Catastrophic longitudinal inner hull rupture occurred as the forward-moving crack met the oncoming rock tip since the hull plating was already transversely stressed by the rock tip.

Figure 2.18: Crack Propagation Event Types (after Rodd, 1996)

A Type II crack propagation event occurred in double hull structures with tightly spaced longitudinal girders or for those grounding events where the rock penetrated the outer hull close to the longitudinal girder. The crack in the outer hull propagated alongside the longitudinal web until it reached the transverse element. It then propagated vertically at or near the vertical intersection of the transverse and longitudinal elements. The transverse floor separated from the longitudinal girder at the weld as the longitudinal member was displaced outward by the rock. The transverse floor was pushed upward and aft by the penetrating rock and the vertical crack continued to propagate into the inner hull.

A Type III crack propagation event involved the splitting of the longitudinal crack into two transverse cracks as the rock impacted the transverse floor or bulkhead and changed the stress field in the outer hull plate. These cracks then propagated into the longitudinal girders and turned upward into the web towards the inner hull. As the base of the transverse floor or bulkhead was pushed back by the penetrating rock, these vertical cracks opened and continued to propagate into the inner hull.

Table 2.6 lists the double hull model type with the general design characteristics, the crack propagation event type and the corresponding kinetic energy dissipated by the model. All four grounding models experienced breach of the inner hull at the mid-length transverse bulkhead, but with different failure mechanisms.

Rodd, (1996) concluded that inner hull rupture occurred when the penetrating rock, which was in contact with the highly membrane stressed inner hull, passed through the next full depth transverse element and propagated a vertical crack into the hull plating which propagated back to meet the rock tip in the stressed plating. In virtually all grounding experiments to date, inner hull rupture initiation can be traced back directly to crack propagation in the intermediate members from the original large rupture of the outer hull near a transverse element. For an ADH tanker design to prevent oil outflow, it must be capable of sustaining the initial rupture of the outer hull and prevent the propagation of cracks into the inner hull as the tanker moves forward and the rock passes through the transverse element.

Table 2.6: Summary of Double Hull Grounding Experiments

Model Designation	Description of Grounding Event
<p>CONV/PD328</p> <p>Baseline design for double cargo tank length with bottom structure identical to MARAD PD328</p>	<ul style="list-style-type: none"> - Initial impact velocity was 14 knots. - The rock passed completely through the model with only 76 percent kinetic energy dissipated (3.25×10^6 J). - Unlike the preliminary CDH grounding experiment (Rodd and McCampbell, 1994) which demonstrated a catastrophic Type I inner shell rupture, this design had every frame torn away cleanly from the inner hull without causing rupture until the transverse bulkhead was reached. - Type III tearout of the centerline longitudinal girder at the transverse bulkhead. - Over the same stopping distance, the ADH/PD328 dissipated 73 percent more energy.
<p>ADH/PB</p> <p>Wide longitudinal girder spacing with conventional plate/stiffener transverse bulkheads and thinner inner hull plating</p>	<ul style="list-style-type: none"> - Initial impact velocity was 12 knots. - Type III longitudinal web tearout occurred on the longitudinal girder web nearest to the centerline of the rock on the starboard side, with both forward and aft moving cracks rupturing the inner hull. - Rock passed completely through the model with 94 percent of the kinetic energy dissipated (2.66×10^6 J).
<p>ADH/PD328</p> <p>Closer longitudinal girder spacing, double-plate transverse bulkheads with haunched stools and slightly thicker inner hull plating</p>	<ul style="list-style-type: none"> - Initial impact velocity was 14 knots. - Type II failure occurred with the bulkhead plating being torn away from the longitudinal girder web which caused a short breach of the forward tank to propagate and rupture the aft tank plating. - A similar Type II failure occurred at the aft haunch. - Due to the high density of the longitudinal structure that was impacted, crushed, and displaced by the rock and the high level of friction involved, the structure came to rest on the rock with a kinetic energy dissipation (5.34×10^6 J). - Crashworthy design.
<p>ADH/PD328V</p> <p>Same as ADH/PD328 except with closely spaced vertical stiffening on longitudinal girders and web plating was increased to that of the inner hull</p>	<ul style="list-style-type: none"> - All kinetic energy was dissipated as the model came to rest on the rock just forward of the aft bulkhead. - The stopping distance and energy dissipation (6.03×10^6 J) was similar to that of the ADH/PD328. - Vertical stiffening of girder webs resulted in a more severe damage pattern. Rather than buckling and crushing the webs, the rock caused those in contact to be rotated outward as if hinged at the inner hull weld. - Inner shell rupture was a Type III longitudinal girder web tearout that occurred at the mid-bulkhead forward haunch and resulted in the removal of an inner hull and longitudinal girder plating section. Three separate Type II crack propagations occurred at the aft haunch, one of which opened the aft tank plating. - Numerous small inner hull ruptures occurred form the longitudinal girder web vertical stiffeners as the webs were pushed flat against the hull plating.

2.4.1.4 Stranding Experiments on Double Hull Tanker Structures, Rodd *et al.*, (1994)

The CDNSWC conducted three one quarter-scale stranding experiments to evaluate the crashworthiness of tankers under stranding conditions in which the structure was subjected to the vertical indentation of a rock tip. When a ship becomes stranded, the subsequent wave and tide action progressively ruptures the outer hull, crushes the bottom structure and eventually ruptures the inner hull. The designs were representative of tankers in the 30,000 to 40,000 DWT size range since U.S. coastal operations are primarily medium and small tankers and these geometries (rather than supertankers) are more likely to experience grounding and/or stranding accidents. The selected tankers were: a 30,000 DWT CDH Navy T-5 tanker used as a baseline; an ADH equivalent (Sikora *et al.*, 1989) of the CDH Navy T-5 tanker with collapse-resistant longitudinal girder webs; and a 40,000 DWT curved plate variant of the ADH concept, similar to the MarC Guardian. The models represented the double bottom structure and extended one tank length between transverse bulkheads, with a width nearly half the beam of the ship. The CDH and ADH variant are illustrated in Figures 2.19 and 2.20, respectively. The rock tip was modelled as a single point load applied centrally between the longitudinal girders and the transverse floors, where applicable, through a 90 degree steel cone with a rounded tip. The experimental test results were used to verify numerical modelling tools and analysis methods that will subsequently be used to optimize bottom and inner shell geometries for the dissipation of the total kinetic energy during a grounding and/or a stranding.

Figure 2.19: CDH Stranding Model (after Melton *et al.*, 1994)

Figure 2.20: ADH Stranding Model (after Melton *et al.*, 1994)

The characteristic load-cone penetration curves for each hull section is illustrated in Figure 2.21. (Note: the hull model was a fixed-fixed beam and, under load, experienced global deformations. For example, for the CDH design, the rock tip did not make contact with the inner hull until the cone penetration was approximately 21 inches and the actual depth of the model is 18.5 inches. This implies global bending deflections of approximately 2.5 inches at midspan.) The corresponding descriptions of the behaviour (which includes observations concerning rupture, crack propagation and plastic distortion) are given in Table 2.7, Table 2.8 and Table 2.9 for the CDH, ADH and ADH curved plate variant, respectively. (Note: in both the Figure and the Tables, the key loading events associated with changes in behaviour are marked as points (a) – (m).) The total energy dissipated was expressed as a function of the area under the load-cone penetration curve and is given in the Tables for each model.

In summary, the first noted damage in all hull configurations was rupture of the outer hull in which the cracks then propagated longitudinally to the transverse floor, where applicable, and subsequently other cracks propagated transversely to the longitudinal girders, where applicable. The CDH design proved very effective under stranding conditions, as the longitudinal girder/transverse floor intersections transmitted significant force to the inner hull and provided more than 2.0 m spacing by lifting the inner hull up and away from the rock tip. The ADH provided equivalent performance through the use of the collapse-resistant longitudinal girders. The ADH curved plate variant had considerably less energy absorption capability. Although the curved plate model had a much lower load capacity than the CDH and ADH models, the curved plate model had a scale factor of 4.8 compared to 4.25 for the CDH and ADH models and, as a

Figure 2.21: Load-Cone Penetration Curves for the CDH, ADH and ADH Curved Plate Variant (after Rodd *et al.*, 1994)

Table 2.7: Experimental Stranding Loading Behaviour and Total Energy Dissipated for the CDH Design

Load (lbs)	Description of Behaviour
185,000	Outer shell ruptured around the perimeter of the rock tip with 6.2" of penetration. Point (a).
200,000	Outer shell rupture widened with crack propagation in the longitudinal direction extended beyond the transverse floor/ outer shell weld joint. Approximately 15" of penetration.
140,000	As the rupture extended beyond this weld, the outer shell crack opened the weld site considerably and initiated a secondary rupture in the transverse floor that propagated vertically to the inner hull. Point (b).
335,000	New transverse crack was noted at one side of the longitudinal opening, near the center of the model. Rock tip penetration was 19.3". Point (c).
400,000	Rock tip made contact with the inner hull with 21.1" of penetration. Point (d).
450,000	Rock contacted the longitudinal girder/transverse floor intersections. Point (e).
600,000	Loss of stiffness in the model as the intersections were crushed. Point (f).
750,000	Inner hull ruptured. Rock tip penetration was 29.8". The energy dissipated was 776,000 ft-lbs.

Table 2.8: Experimental Stranding Loading Behaviour and Total Energy Dissipated for the ADH Design

Load (lbs)	Description of Behaviour
170,000	Outer shell rupture oriented longitudinally near the rock tip with 6.2" of penetration. Point (g).
100,000	Outer shell rupture extended longitudinally.
170,000	Transverse rupture in the outer shell extended beyond the nearest longitudinal girder. This rupture was initiated from the center of the original rupture opening. A secondary vertical crack propagated into the longitudinal girder. Rock tip penetration was 11.6". Point (h).
190,000	Rock made contact with the collapse-resistant longitudinal girders at a penetration of 13.0". Point (i).
720,000	Vertical crack in the longitudinal girder propagated into the inner hull and initiated an inner shell crack at the weld, which propagated longitudinally. Rock tip penetration was 30.4". The energy dissipated was 807,000 ft-lbs.

Table 2.9: Experimental Stranding Loading Behaviour and Total Energy Dissipated for the ADH Curved Plate Variant Design

Load (lbs)	Description of Behaviour
50,000	Convex surface of the outer shell flattened over a considerable length and became noticeably concave with approximately 3.45" of penetration. Point (j).
93,000	Outer shell rupture with 7.1" of penetration. Point (k).
65,000	Outer shell ruptured near the edge of the contact area and propagated 9" longitudinally. Point (l).
99,000	Outer shell rupture propagated 53" longitudinally. Longitudinal web crippling extended vertically to the nearest longitudinal stiffener. Rock tip penetration was 14.0".
125,000	Rock tip made contact with the inner hull with 18.5" of penetration. Point (m).
130,000	Curved inner hull shell plating at the tip of the rock flattened. The outer shell rupture propagated transversely to the nearest longitudinal girder. A new crack in the outer shell, initiated from the first rupture, propagated transversely to the nearest longitudinal girder on the opposite side. Rock tip penetration was 19.0"
180,000	Inner shell deformation was limited to the centerline cell.
210,000	A crack in the inner shell along the longitudinal girder weld nearest the rock tip. The energy dissipated was 181,000 ft-lbs.

result, the curved plate model penetration and load values must be increased by 1.13 and 1.28, respectively. It is therefore cautioned that preliminary conclusions should not be drawn based on load levels alone as it may be decided that the primary criteria for evaluating crashworthiness is by vertical penetration alone, in which case, all three models were approximately equivalent. The three models ultimately failed with inner hull rupture, with the CDH design having rupture around the rock tip, the ADH due to crack propagation and the ADH curved plate variant with cracks initiating in the hull plate/longitudinal girder weld (the deformation of the curved plate inner hull opens the longitudinal fillet weld).

The CDH design was very effective during stranding events due the transverse floor/longitudinal girder intersections that transmitted load to the inner hull and lifted it away from the rock tip. However, related studies have demonstrated that the transverse webs created adverse localized effects from groundings. The collapse-resistant longitudinal girders of the ADH provide comparable performance to the CDH.

2.4.2 Instability and Collapse Strength of Advanced Double Hull Structures, Devine *et al.*, (1994)

Devine *et al.*, (1994) reported on the research that has been conducted at Lehigh University and the CDNSWC to identify the instability modes and evaluate the strength of ADH designs. During operational loading and during accidental or extreme load events, compressive stresses develop which can strongly affect the overall stiffness and strength of the hull.

Three principal modes of compressive instability that must be considered for ADH designs include local plate instability, beam-column instability and cellular grillage instability, as shown in Figure 2.22 (a), (b) and (c), respectively. Local instability occurs if the plate slenderness ratios (width-to-thickness, b/t) are high. Beam-column instability involves overall buckling of the cells between transverse bulkheads and is usually coupled with local plate instability. Cellular grillage is the instability behaviour of a panel of box cells supported on four edges.

Local plate instability involves buckling of the hull plating between the longitudinal girders. In an ADH structure, plate panels are loaded along the shorter edges. The local buckling of the hull plating does not necessarily result in failure. After local buckling, the in-plane stress is redistributed and a higher stress develops along the longitudinal girder edges. If the hull plating is slender (b/t greater than 80 for mild steel) and with the unloaded edges constrained, then appreciable post-buckling strength will exist. For sturdier hull plates (low b/t), local buckling is essentially inelastic and is generally coincident with plate failure. In summary, the following factors influence plate local buckling behaviour: (1) plate slenderness; (2) material properties; (3) load pattern; (4) edge fixity and constraint; (5) residual stresses; and (6) geometric imperfections.

Beam-column and cellular grillage instability involve the entire hull, limiting it from carrying any additional load and resulting in failure. Beam-column and cellular grillage instability are considered a primary mode failure, where post-buckling strength does not exist.

Figure 2.22: Principal Modes of Compression Instability for ADH Structures (after Devine *et al.*, 1994)

The Lehigh University study was directed towards naval surface combatant ships fabricated with high-strength steel (HSLA-80) and included seven full-scale and two half-scale specimens tested to ultimate compressive collapse. The specimens included multicellular stub column tests, single-cell long column tests and multicellular beam-column tests to examine the effects of cell numbers, plate slenderness (b/t) ratios ranging from 48 to 96, overall length and lateral pressure loading on instability failure modes. A cell is defined as a rectangular shaped box formed by the longitudinal girders and the hull plates. A summary of the geometry, slenderness ratio, width factor for the plates β (where

$\beta = (b/t)\sqrt{F_y/E}$ and the ratio of ultimate strength to yield strength (σ_u/σ_y) is given in Table 2.10 for the stub-column, beam-column and pinned-end column tests. The yield strengths for the full- and half-scale specimens were 601.9 MPa (87.3 ksi) and 557.8 MPa (80.9 ksi), respectively.

Table 2.10: Summary of the Lehigh University Column Test Results (after Devine *et al.*, 1994)

Lehigh Stub-Column Tests

Specimen	Number of Cells	Material	Nominal Cell Size (in. x in.)	Length (in.)	Plate Thickness (in.)		Average b/t Ratio	Average β	Ratio
					Flange	Web			
1-3636-10	1	HSLA-80	36 x 36	120	0.403	0.386	96	5.04	0.369
2-3036-6	2	HSLA-80	30 x 36	72	0.390	0.392	96	5.05	0.420
3-1818-6	3	HSLA-80	18 x 18	72	0.396	0.396	48	2.49	0.752
3-2430-6	3	HSLA-80	24 x 30	72	0.388	0.391	80	4.15	0.479
3-3036-6	3	HSLA-80	30 x 36	72	0.384	0.391	96	5.16	0.403

Note: Specimen designation is given as the number of cells – cell dimension in inches – cell length in feet.

Lehigh Beam-Column Tests

Specimen	Number of Cells	Material	Nominal Cell Size (in. x in.)	Length (in.)	Plate Thickness (in.)		Average b/t Ratio	Average β	Ratio
					Flange	Web			
3-1818-18	3	HSLA-80	18 x 18	216	0.404	0.420	48	2.39	0.696
3-3636-36	3	HSLA-80	36 x 36	432	0.394	0.398	96	4.88	0.376

Lehigh Pinned-end Column Tests

Specimen	Number of Cells	Material	Nominal Cell Size (in. x in.)	Length (in.)	Plate Thickness (in.)		Average b/t Ratio	Average β	Ratio
					Flange	Web			
3-3636-18	1	HSLA-80	36 x 36	216	0.394	0.384	96	5.07	0.358
3-3636-35	1	HSLA-80	36 x 36	420	0.390	0.390	96	5.04	0.353

The CDNSWC study was directed towards both naval and commercial ships constructed from ordinary and high strength steels and included eleven small-scale and five large-scale longitudinal column collapse specimens. The small-scale specimens consisted of eight 6 inch square single-cell columns with b/t ratios ranging from 59 to 103 and three 6 inch square triple-cell columns with similar b/t ratios. Five of the single-

cell columns were 6 inches long and the remaining were 20 inches long. The triple-cell columns were 54 7/8 inches long. Three different plate thicknesses of ordinary strength steel were used to model the range of plate slenderness ratios representative of typical ships. The yield strengths for the single-cell columns were 211.0 MPa (30.6 ksi), 197.2 MPa (28.6 ksi) and 193.1 MPa (28.0 ksi) for the plating thicknesses of 2.6 mm (0.101 in.), 1.8 mm (0.071 in.) and 1.5 mm (0.058 in.), respectively. The yield strengths for the triple-cell columns were 231.0 MPa (33.5 ksi), 219.3 MPa (31.8 ksi) and 77.2 MPa (11.2 ksi) for specimens 3-55-105, 3-55-077 and 3-55-058, respectively. A summary of the geometry, slenderness ratio, width factor for the plates β , and the ratio of ultimate strength to yield strength (σ_u/σ_y) is given in Table 2.11 for each of the small-scale specimens.

Table 2.11: Summary of the Small-Scale CDNSWC Column Test Results (after Devine *et al.*, 1994)

CDNSWC Small-Scale Single-Cell Column Tests

Specimen	Number of Cells	Material	Nominal Cell Size (in. x in.)	Length (in.)	Plate Thickness (in.)		Average λ Ratio	Average β	σ_u/σ_y
					Flange	Web			
1-6-101-a	1	OS	6 x 6	6	0.101	0.101	59.4	2.037	0.8219
1-6-101-b	1	OS	6 x 6	6	0.101	0.101	59.4	2.037	0.8271
1-6-101-c	1	OS	6 x 6	6	0.101	0.101	59.4	2.037	0.8368
1-6-071	1	OS	6 x 6	6	0.071	0.071	84.6	2.763	0.6877
1-6-058	1	OS	6 x 6	6	0.058	0.058	103.1	3.300	0.5957
1-20-101	1	OS	6 x 6	20	0.101	0.101	59.4	2.037	0.8356
1-20-071	1	OS	6 x 6	20	0.071	0.071	84.6	2.763	0.6340
1-20-058	1	OS	6 x 6	20	0.058	0.058	103.1	3.300	0.5456

Note: Specimen designation is given as the number of cells – cell length in inches – plate thickness.

CDNSWC Small-Scale Triple-Cell Column Tests

Specimen	Number of Cells	Material	Nominal Cell Size (in. x in.)	Length (in.)	Plate Thickness (in.)		Average λ Ratio	Average β	σ_u/σ_y
					Flange	Web			
3-55-105	3	OS	6 x 6	54.88	0.105	0.105	57.0	1.903	0.6613
3-55-077	3	OS	6 x 6	54.88	0.077	0.077	78.0	2.968	0.4880
3-55-058	3	OS	6 x 6	54.88	0.058	0.058	101.0	3.300	0.6767

The five large-scale CDNSWC specimens consisted of cellular structures (either 3 or 5 cells) made of flat plates or curved shell plates (CP) with flat longitudinal webs with localized transverse stiffening. The test specimens were constructed of either ASTM A-36 ordinary strength steel (OS) or high strength steel (HS). The yield strengths for the OS3, OS5, HS3 shell plating, HS3 web plating, CP3 and CP5 tests specimens were 293.0 MPa (42.5 ksi), 279.2 MPa (40.5 ksi), 412.3 MPa (59.8 ksi), 508.1 MPa (73.7 ksi), 344.0 MPa (49.9 ksi) and 274.4 MPa (39.8 ksi), respectively. A summary of the geometry, slenderness ratio, width factor for the plates β , and the ratio of ultimate strength to yield strength (σ_u/σ_y) is given in Table 2.12 for each of the large-scale flat and curved plate specimens. Details of the test specimens OS3 and CP3 are illustrated in Figures 2.23 and 2.24, respectively.

Table 2.12: Summary of the Large-Scale CDNSWC Column Test Results (after Devine *et al.*, 1994)

CDNSWC Large-Scale Flat Plate Column Tests

Specimen	Number of Cells	Material	Nominal Cell Size (in. x in.)	Length (in.)	Plate Thickness (in.)		Average λ/λ_c Ratio	Average β	σ_u/σ_y
					Shell	Web			
OS3	3	OS	24 x 21	150	0.300	0.300	76.3	2.853	0.5270
OS5	5	OS	24 x 21	150	0.319	0.319	72.0	2.661	0.5437
HS3	3	HSS	24 x 21	150	0.328	0.249	76.6	3.526	0.4654

Note: Specimen designation is given as steel type and the number of cells.

CDNSWC Large-Scale Curved Plate Column Tests

Specimen	Number of Cells	Material	Nominal Cell Size (in. x in.)	Length (in.)	Plate Thickness (in.)		Average λ/λ_c Ratio	Average β	σ_u/σ_y
					Shell	Web			
CP3	3	OS	24 x 21	150	0.251	0.259	71.1	3.028	0.6010
CP5	5	OS	24 x 21	150	0.251	0.259	72.5	2.686	0.7479

Figure 2.23: Column Collapse Model OS3 (after Devine *et al.*, 1994)

Figure 2.24: Column Collapse Model CP3 (after Devine *et al.*, 1994)

The ultimate load for each of the Lehigh University specimens was governed by severe local plate buckling. The initial local buckling appeared as a sinusoidal pattern and as the load was increased, the local buckling became more prominent, with concentrations of inelastic local buckling occurring near specimen midheight. Specimens with larger b/t ratios had appreciable post-buckling strength. For a constant nominal b/t ratio, effects of varying the specimen length and number cells had little effect on the ultimate load capacity of the specimen. The length of the specimen did however appear to have a significant effect on load redistribution during collapse following the ultimate load.

The ultimate strength behaviour for each of the CDNSWC small-scale specimens was governed by local plate buckling, where buckling had a direct correlation with plate slenderness. For increasing column length, the single-cell specimens demonstrated a strength reduction for those comprised of more slender plating. However, with less slender plating, for b/t equal to 59, the longer column shows a slightly higher strength and length does not clearly shown an impact on strength. The triple-cell specimens demonstrated similar behaviour to the single-cell specimens, however, these specimens suffered the effects of weld residual stresses and fabricating imperfections, unlike the single-cell specimens. This was attributed to the fact that the single-cell specimens were constructed from a single steel plate with four corner folds and a single longitudinal butt weld, thereby being mostly free of residual weld stresses. In contrast, the triple-cell specimens were constructed using a combination of folded angle sections and flat plates causing weld residual stresses along six of the eight corner boundaries.

The typical behaviour of the CDNSWC large-scale column tests is illustrated in the axial load-deformation curve and the non-dimensionalized form of that, that is, the load-shortening curves (σ_m/σ_y versus $\varepsilon_m/\varepsilon_y$) in Figures 2.25 and 2.26, respectively. The ultimate strength behaviour for each of the CDNSWC large-scale specimens was governed by local plate buckling. No weld failures or snap-through buckling events were observed. The flat plate specimens were not particularly sensitive to the number of cells for ultimate load capacity. The number of cells appeared to affect the post-ultimate collapse behaviour by having a higher strength for a higher number of cells. For the flat plate specimens, the square sinusoidal wave buckled shape was amplified until the ultimate capacity of the section was reached which occurred when the normal stress at the hull-plate/longitudinal junction yielded. In contrast, the curved plate specimens retained their mid-plate effectiveness until an ultimate load was reached at a significantly higher level than that provided by the effective-width hard-corner structure of the flat plate specimens. The load-shortening curves for the curved plate specimens remained relatively linear with no indication of buckling until just prior to ultimate load. At this point, plastic buckling suddenly occurred and collapse proceeded quickly. Buckling of the longitudinal webs, which had very low slenderness (b/t less than 32), was evident prior to buckling of the curved plates. All configurations exhibited a significant post-buckling capacity that diminished as the axial deformations increased.

Figure 2.25: Load-Deflection Curves for CDNSWC Large-Scale Column Tests (after Devine *et al.*, 1994)

Figure 2.26: Load-Shortening Curves for CDNSWC Large-Scale Column Tests (after Devine *et al.*, 1994)

Based on the experimental test results, Devine *et al.*, (1994) concluded that:

- (1) local plate buckling governed the collapse for all cases;
- (2) as anticipated, buckling characteristics of the plates were typical: post-buckling for slender plates where the effective-width concept could be applied; square sinusoidal waves for flat plates; and the buckling load was relatively independent of length, boundary conditions at the ends, lateral load and the number of cells; and
- (3) the curved plate specimens, CP3 and CP5, had an increased strength of 1.14 and 1.38 over their counterpart flat plate specimens, OS3 and OS5, respectively, due to an increased energy mode for buckling the curved plating. It should be noted, however, that part of this increase in capacity could be attributed to the stiffening elements used on the curved plate specimen longitudinal girders.

Perhaps the most significant observation that can be made from the experimental testing, other than that of the failure being controlled by local buckling, is the ratio of the critical buckling load to the yield load that varied from 0.47 to 0.54 for the flat plate tests. This indicates that the hull-plate in an all-steel ADH structure does not effectively use the steel plate for in-plane loads.

2.4.3 Fatigue Strength of Advanced Double Hull and Curved Plate Concept, Melton *et al.*, (1994)

The relevance of presenting the Melton *et al.*, (1994) paper is that the design against fatigue is an important aspect of the overall design and detailing of a ship, as outlined in Section 5/2A5.2.3 in ABS. A significant amount of the maintenance for a ship is related to corrosion, which is accelerated in fatigue prone areas, and crack propagations related to fatigue of standard construction details. Fatigue-related problems are a function of the geometry of the construction details and can only be detected through extensive testing and/or experience. New design configurations, like the ADH and MarC Guardian, required the corresponding fatigue detail investigations.

A series of fatigue tests were conducted by the CDNSWC to evaluate the susceptibility of an ADH tanker to fatigue failure. The tests were conducted for both the ADH and the MarC Guardian curved plate concept to examine the fatigue behaviour of the designs under a range of constant amplitude, fully reversed loadings, typical of those for hull loading due to longitudinal bending. The effects of using different grades of steel and different levels of fabrication qualities were also considered in the fatigue testing.

To examine the local behaviour of ADH flat plate specimens, a cruciform-shaped test specimen of full- or large-scale plate thickness was chosen, since it is representative of the most common construction detail, which is, intersecting plates that are fillet welded. The intersecting elements were divided into either continuous or discontinuous, and are illustrated in Figures 2.27 (a) and (b), respectively. The discontinuous elements were further divided into “aligned” and “misaligned” specimens, illustrated in Figure 2.27 (c), to replicate the difficulty in aligning the pieces during construction. Three

grades of steel were tested: ordinary steel (OS) with a nominal yield strength of 241 to 310 MPa (35 to 45 ksi); high-strength steel (HS), 379 to 448 MPa (55 to 65 ksi); and high-strength low-alloy steel (HSLA-80), 552 to 621 MPa (80 to 90 ksi).

Figure 2.27: Cruciform Test Specimen Configurations (after Melton *et al.*, 1994)

For the MarC Guardian, small-scale curved plate tee-stubs and three large-scale hull plate longitudinal I-sections were tested to examine the fatigue strength of large-electro-gas weld joints and other standard weld joints. The tee-stubs were loaded through a flat web member placed vertically at midspan of the curved hull plate that bends it about its weak axis. The ends of the plate were simply supported. The large-scale tests were loaded as simply supported beams with two point loads providing a constant moment region with the curved hull plates forming the flanges of the I-section and bending occurring about the strong axis.

The data from approximately 200 fatigue tests on ADH details was statistically analyzed to produce geometric mean lives and standard deviations to compare the effects of yield strength and fabrication quality on the fatigue strength. The test results indicated that:

- yield strength had a negligible effect on the fatigue strength for low to intermediate stress ranges;
- for large stress ranges approaching yield, a decrease in fatigue strength was observed for specimens with a decreasing yield strength;
- continuous specimens had a slightly longer mean fatigue life compared to the discontinuous specimens;
- as could be expected, the misaligned specimens showed a marked decrease in fatigue strength when compared to the corresponding specimens;
- the continuous and discontinuous specimens, for a given stress range, had fatigue lives 2 to 10 times greater than the conventionally stiffened plate; and
- the misaligned specimens had a smaller fatigue life at higher stress ranges and a better fatigue life at low stress ranges than conventionally stiffened plates.

The MarC Guardian curved plate fatigue tests demonstrated that the electro-gas weld had adequate fatigue strength when compared to other standard weld types.

2.5 Proposed Design Concept of Unidirectional Double Hull Sandwich Plate System

It is generally recognized that groundings and collisions will continue to occur, despite navigational advances and the use of highly trained mariners to operate oil tankers. Existing double hull tanker technology has not yet eliminated the possibility of oil outflow in the case of accidental or extreme load events. Clearly, there remains a need to develop of an improved structural system, which will prevent oil outflow in the case of these load events, and which will be constructable and cost-effective.

With this objective in mind, a composite structural laminate plate composed of steel-polyurethane elastomer-steel was developed and incorporated into the design of an advanced double hull 40,000 DWT product oil tanker. It is referred to as the Unidirectional Double Hull Sandwich Plate System (UDHSPS), where “unidirectional” refers to the orientation of the framing elements. The UDHSPS design, illustrated in Figure 2.28, provides an equivalent structure for normal operating loads, as defined by the Classification Societies, and provides superior energy absorption capacity during accidental or extreme load events. It also eliminates the possibility of oil outflow due to crack propagation and provides sufficient residual strength for post-accident events.

The original objective in the development of the UDHSPS concept was to increase the amount of energy absorption capacity. The research community generally recognizes that to maximize the energy absorption capacity of the hull plating, the system must be capable of developing inelastic membrane action. With the CDH designs, the presence of longitudinal stiffeners and transverse floors limit the development of membrane action to a single plate, as rupture will occur at or near the supporting

Figure 2.28: Proposed Unidirectional Double Hull Sandwich Plate System

elements, (Daidola, 1995). The energy absorption capacity can be increased if the transverse floors are removed, the spacing of the supporting elements is increased, and if these elements are designed to collapse prior to rupture of that plate, thereby engaging the adjacent plates into membrane action. The removal of the transverse floors transforms the CDH configuration into an ADH design. However, by removing the longitudinal stiffeners, the required in-plane buckling capacity of the thin steel plates cannot be achieved. Okamoto *et al.*, (1985) concluded that for ADH designs, the longitudinal girder spacing needed to be decreased to approximately the same spacing as the longitudinal stiffeners in the CDH design to obtain the equivalent in-plane buckling capacity. To

increase the longitudinal girder spacing to the maximum practical distance, the buckling capacity of the hull plate must be increased. This led to the development of a composite structural laminate that would be substituted for the steel hull plating. Other structural properties that enhance the energy absorption capacity that naturally accrue from the composite structural laminate design concept are:

- a portion of the stiffener material that is eliminated can be reallocated to the hull plating which maximizes the steel acting membrane action;
- the ductile elastomer core dissipates localized strain concentrations due to bending at the longitudinal girder/hull plate intersection, thereby reducing the possibility of premature rupture of the hull plating at this hard-spot; and
- the elastomer absorbs energy as it is compressed and transfers load from the outer steel plate to the inner steel plate upon contact with the external force or object.

Incorporating a design philosophy that maximizes plastic deformation and the progressive collapse of the longitudinal girders can further increase the energy absorption capacity. For ADH designs, like the EPOCH Mark II, Okamoto *et al.*, (1985) concluded that the transverse in-plane strength was small and should be considered negligible for design. As a result, the transverse bulkhead must be stiff and have sufficient strength to carry the transverse hydrostatic forces (normally carried by the transverse floors in the CDH design) for a full cargo tank length. Another natural by-product of the composite structural laminate is that it has sufficient transverse in-plane capacity to transmit these hydrostatic forces.

In addition to the increased energy absorption capacity developed by removing the longitudinal stiffeners, the concept reduces the complexity as related to the CDH design. Reduced complexity translates into the following advantages:

- fewer structural intersections, which reduces the number of fatigue-prone details and the corresponding number of hard-spots which can lead to premature rupture of the shell;
- improved coatings application due to the flush cargo/ballast tanks means less corrosion and longer life expectancy;
- increased accessibility for inspection and maintenance; and
- substantial cost savings in welding, detailing, labour and materials related to the elimination of the longitudinal stiffeners.

In addition to the advantages already cited, the proposed design reduces the weight capacity by approximately 13 percent over an equivalent all-steel CDH design, resulting in lower operating costs.

The thermoplastic polyurethane elastomer was selected as a candidate core material because of its range of physical properties and characteristics that make it suitable for this application, illustrated in Table 2.13. Other characteristics include: compressive strength, dynamic resilience, fuel and oil resistance, hydrophobic and insulation properties.

Table 2.13: Thermoplastic Polyurethane Elastomer Characteristic Properties (after ASM International, 1988)

Material Characteristic	Range
Elongation, %	100 - 800
Tensile strength, MPa	20 - 55
Tensile stress at 100% elongation, MPa	2 - 35
Flexural Modulus, Gpa	< 0.02 - 1.035
Tear strength, N/mm	25 - 260
Hardness, Shore	70A - 80D
Glass transition temperature, °C	-70 - 15
Bayshore rebound, %	20 - 60

In addition, sufficient bond capacity, fire resistance, toxicity of off-gases and the environmental issues concerning disposal of used elastomer material were investigated and confirmed that this material is suitable. The elastomer provides sufficient bond capacity without the aid of mechanical or chemical fasteners. When exposed to an open flame, the elastomer chars and produces comparable off-gases similar to wood. Fire retardants, such as antimony oxide and an organic chlorine- or bromine-containing compound, and smoke suppressants, such as aluminum trihydrate, magnesium carbonate, and magnesium hydroxide, can be added to the elastomer material to increase fire resistance and reduce toxic off-gases, respectively (Sevart *et al.*, 1990). The elastomer is a thermo-recyclable material.

The remaining outstanding feasibility issues that were addressed included:

- Could the elastomer be integrated into the design of an oil tanker?
- Could the design be constructed?
- Could the design be repaired in the case of an accidental or extreme load event? and
- Would the design be cost-effective?

Details were developed to integrate the composite structural laminate with longitudinal girders, stools and transverse bulkheads that specifically recognize known fabrication and welding techniques. The result incorporates modular construction in which the elastomer is injected under pressure through ports into the voids of the laminate component in the hull-girder, side structure, bilge, sheerstrake, deck and transverse bulkhead modules as they are fabricated. As these modules are joined to form cargo tank sections, the corresponding interconnecting voids are then filled with elastomer.

Details of production engineering are shipyard dependent and can only be resolved once the design is finalized and moved into production.

As part of the design to ensure constructability, details were developed to use an elastomer layer to isolate the inner cargo steel plate lining from all other steel parts to prevent the problem of cracks propagating into and breaching the cargo liner. This will solve the crack propagation problems that are initiated during a grounding situation as described by Rodd, (1996). It is anticipated that this composite structural laminate system will sustain substantial penetration into the inner hull prior to being breached.

It is also anticipated that damaged sections will be cold-cut from the adjoining structure, a local strip of elastomer shot-blasted clean, and a new sub-section welded and cast in a similar method to its construction.

The build-cost was evaluated for an all-steel 40,000 DWT CDH product oil tanker and an equivalent UDHSPS design. The detailed cost analysis is contained in three sections describing the cost evaluation details, the cost spreadsheet for the CDH, and the cost spreadsheet for the UDHSPS in Appendices A, B and C, respectively. Appendix A

assesses the proportion of costs associated with the hull structure, propulsion system, machinery, paint and tank coatings, hull outfit and electrical components in an all-steel 40,000 DWT CDH 7-tank tanker with an approximate total cost of \$40 million US. The hull structure and the cost of painting and tank coating were separated from the total, since these costs would be shared by either design. This cost represents 43 percent of the total cost at an approximate total cost of \$17.1 million US, or \$2.44 million US per cargo tank. The analysis gives estimates of the cost of raw material handling and preparation; plate cutting and forming; fabrication; and erection unit joining and painting and coating application. A quantity take-off was completed for a typical cargo tank of the all-steel CDH for steel weight and surface area, number of pieces, weld lengths and volume, and the number of intersections, as given in Appendix B. This cost per cargo tank, \$2.49 million US, as summarized in Table 2.14, was compared to the industry standard of \$2.44 million US. Once the algorithm was calibrated, it was then used to determine the cost per cargo tank of the UDHSPS design with new cost estimates for the elastomer raw material and for placement (based on industry estimates). The summary is given in Table 2.15. The UDHSPS cargo tanks represent an 11.6 percent reduction in hull costs and a 5.2 percent reduction in the total build cost.

It is anticipated that the UDHSPS design will accrue other cost benefits, not yet quantified, that are inherent with the design technology. These additional cost benefits are expected in the areas of operating costs, inspection and maintenance costs, and accidental or extreme load event costs. In terms of operating costs, benefits attributed to the composite structural laminate include a reduction in in-transit heating costs, due to the

Table 2.14: Cost Parameters for an All-Steel 40,000 DWT Conventional Double Hull Tanker

Cost Parameter per Cargo Tank	Value
Total Steel Weight	859.3 t
Surface Area	15,105 m ²
Number of Pieces	1,384
Butt Weld Length	2,674 m
Fillet Weld Length	15,159 m
Number of Intersections	5,020
Butt Weld Volume	538,600 cm ³
Fillet Weld Volume	1,232,000 cm ³
Raw Material Handling and Preparation Cost (\$US)	\$ 0.80 million
Plate Cutting and Forming Cost (\$US)	\$ 0.06 million
Fabrication Cost (\$US)	\$ 1.63 million
Total Cost (\$US)	\$ 2.49 million
Target Cost for Calibration (\$US)	\$ 2.44 million

Table 2.15: Cost Parameters for a 40,000 DWT Unidirectional Double Hull Sandwich Plate System Tanker

Cost Parameter per Cargo Tank	Value
Total Steel Weight	515.8 t
Total Elastomer Weight	234.1 t
Total Weight	749.9 t
Surface Area	19,306 m ²
Number of Pieces	738
Butt Weld Length	4,511 m
Fillet Weld Length	4,833 m
Number of Intersections	292
Butt Weld Volume	172,700 cm ³
Fillet Weld Volume	47,500 cm ³
Raw Material Handling and Preparation Cost (\$US)	\$ 0.60 million
Plate Cutting and Forming Cost (\$US)	\$ 0.04 million
Fabrication Cost (\$US)	\$ 0.46 million
Elastomer Cost (Including Placement) (\$US)	\$1.10 million
Total Cost (\$US)	\$ 2.20 million
Total Cost for All-Steel CDH (\$US)	\$ 2.49 million

insulation of the cargo tanks provided by the elastomer, and reduced fuel consumption due to a lighter weight tanker. With a less complex structural configuration, inspection and maintenance are not only easier to perform but are be done with better quality in less time. The simplicity of the design leads to fewer fatigue prone details and to increased corrosion resistance. This equates to reduced maintenance costs, less time spent in the dry-dock, and a greater life expectancy (the shipowner can transport more oil during the vessel life). The UDHSPS offers improved energy absorption capacity, good residual strength and a decreased probability of oil outflow for collisions and a high probability of zero oil outflow for groundings with penetrations less than 2.0 m. Any reduction in oil outflow leads to reduced clean-up costs and subsequent punitive damages, (Monnier, 1995).

Chapter 3

Design Objectives and Experimental Program

3.1 Design Objectives

3.1.1 Development of an Equivalent Conventional Double Hull Design

A typical all-steel 40,000 DWT conventional double hull tanker was designed according to ABS Rules and Regulations and is illustrated in Figure 3.1.

Figure 3.1: Typical All-Steel 40,000 DWT Conventional Double Hull Tanker

The design was undertaken for several reasons:

- (1) to understand the state-of-the-art of oil tanker design and, if possible, to understand the rationale for the current set of design rules;**
- (2) to examine the limitations and assumptions imposed on current design practices;**
- (3) to determine a configuration for a baseline design for a 40,000 DWT conventional double hull oil tanker that can be used to develop an equivalent UDHSPS design;**
- (4) to examine current construction practices and limitations; and**
- (5) to compare different Classification Societies Rules and Regulations to determine whether or not they yield similar scantling (dimension) results.**

The principal design parameters for a 40,000 DWT CDH design are tabulated in Table 3.1. Table 3.2 gives the hull-girder design loads for still-water and wave-induced conditions, and minimum section modulus (similar to a serviceability limit state) and moment of inertia, along with the corresponding applicable sections from ABS Rules and Regulations from which these values can be calculated. Table 3.3 lists the scantlings as determined by Lloyd's Register (Ferro, 1998) and ABS Rules and Regulations. Selection of nominal rolled plate thicknesses that satisfy these minimum scantlings lead to similar thicknesses, in most cases; hence, the rules generated comparable designs.

Table 3.1: Principal Design Parameters for a 40,000 DWT CDH Tanker

Principal Design Parameters	ABS Class	Value
Deadweight, DWT	---	40,000 t
Length between Perpendiculars, <i>LBP</i>	---	180.0 m
Rule Length (97% of LBP), <i>L</i>	3/1.1	174.6 m
Molded Breadth, <i>B</i>	3/1.3	32.0 m
Molded Depth, <i>D</i>	3/1.5	20.0 m
Molded Draft, <i>d</i>	3/1.7	12.0 m
Block Coefficient, <i>C_b</i>	---	0.85

Table 3.2: Hull-Girder Design Loads and Minimum Section Modulus and Moment of Inertia for a 40,000 DWT CDH Tanker

Design Parameters		Design Value
Still-Water Bending Moments, 3/6.3.2	Hogging, <i>M_H</i>	1000.64 x 10 ³ kNm
	Sagging, <i>M_S</i>	-918.54 x 10 ³ kNm
Wave-Induced Bending Moments, 3/6.3.3(a)	Hogging, <i>M_{WH}</i>	1472.40 x 10 ³ kNm
	Sagging, <i>M_{WS}</i>	-1554.45 x 10 ³ kNm
Total Moment, 3/6.3.4(a)(1)	Hogging, <i>M_{TH}</i>	2472.99 x 10 ³ kNm
	Sagging, <i>M_{TS}</i>	-2472.99 x 10 ³ kNm
Still-Water Shear Force, 3/6.3.2	Positive, <i>F_{SW}</i>	278.15 x 10 ³ ton
Waved-Induced Shear Force, 3/6.3.3(c)	Positive, <i>F_{WP}</i>	17.00 x 10 ³ kN
	Negative, <i>F_{WN}</i>	17.00 x 10 ³ kN
Required Section Modulus, 3/6.3.4(a)	<i>SM</i>	141.31 x 10 ³ m cm ²
Minimum Section Modulus, 3/6.3.4(a)(2)	<i>SM</i>	141.31 x 10 ³ m cm ²
Hull-Girder Moment of Inertia, 3/6.3.4(b)	<i>I</i>	740.94 x 10 ³ m ² cm ²

Table 3.3: Comparison of Element Scantlings Determined by Lloyd's Register and ABS Rules and Regulations

Element	Lloyd's Thickness (mm)	ABS Thickness (mm)	Applicable ABS Clause
Strength Deck Plating	12.00	12.65	3/16.5.1(a)(1)(b)
Sheerstrake Plating	12.26	12.65	3/15.3.6
Side Shell Plating	12.26	12.53	3/15.3.5
Bilge Plating	12.99	13.11	3/15.3.8(b)
Bottom Shell Plating	12.99	13.11	3/15.3.8(b)
Keel Plating	15.08	14.61	3/ 4.1
Inner Bottom Plating	13.91	13.36	3/7.5.1
Center Girder Plating	13.85	15.28	3/7.3.2(b)(1)
Side Girder Plating	12.00	12.49	3/7.3.4(a)
Longitudinal Bulkhead Plating	12.63	12.50	3/12.7.1
Corrugated Transverse Bulkhead	14.83	15.00	3/12.7.3
Floor Plating	12.00	12.49	3/7.3.4(a)
Deck Transverse Plating	13.00	14.00	3/11.7.3

Note: (1) The stiffener sizes are not listed.

- (2) For a typical longitudinal girder spacing of 3000 mm, 4 equally spaced 25 by 400 mm flat-bar stiffeners are used, yielding an average hull plate thickness of 26.3 mm.

In order to design an impact resistant vessel, it is necessary to design a new structural arrangement and construction that may not be explicitly described by current design Rules and Regulations. ABS allows for alternative designs, as long as equivalency can be established. Section 5/2B.1.2(c) states:

“Where it can be shown that the calculated stresses using the loading conditions specified in 5/2B.27.3 are less than those stated to be permissible, consideration will be given to scantlings alternative to those recommended by this section.”
(American Bureau of Shipping, 1996).

3.1.2 Design Objectives of an Equivalent Unidirectional Double Hull Sandwich Plate System

The Unidirectional Double Hull Sandwich Plate System (UDHSPS) was designed for all limit states associated with normal operating loads and those during accidental or extreme load events. Under normal operating conditions, the outer and inner hulls must resist in-plane loads due to hogging, sagging, and shear, and provide a minimum section modulus and moment of inertia. The stools, transverse bulkheads, and deck must be designed for other forces that include static loads (dry-docking and thermal loads), slowly varying loads (sloshing of liquid cargoes, launching and berthing loads), and rapidly varying loads (slamming, and other dynamic loads), (Hughes, 1983). Detailed designs of these elements are outside the scope of this thesis, which pertains to the bottom hull and side structure only. For accidental or extreme loading events (collisions or groundings), the hull structure should be designed:

- (1) to maximize plastic distortion/deformation without rupture of the inner hull;
- (2) to absorb energy through plastic deformation of the outer hull, girders and transverse floor, and elastic deformation of the inner hull; and
- (3) to maintain inner hull structural integrity in order to retain sufficient residual strength for successful removal of the damaged vessel during stranding, rescue, and repair operations.

3.1.3 Design Philosophy for Accidental or Extreme Load Events and Design of the Large-Scale Test Specimen

The sizing of the hull plate and core for in-plane capacity and the determination of core properties to establish an equivalent hull element for operational loads were completed by Ferro, (1998). Development of the design philosophy for accidental or extreme load events and the sizing of the outside hull, longitudinal girders, and transverse floor elements formed the basis for the design of the large-scale test specimen, which was only being loaded transversely. The full-scale plate sizes and girder spacing had to be comparable to those determined for the all-steel 40,000 DWT CDH designed according to ABS Rules and Regulations and Lloyd's Register, as illustrated in Table 3.3. Additional constraints for the design of the test specimen were: (1) the test specimen size had to be large enough to provide sufficient access for instrumentation purposes, to avoid scaling effects which could alter stability and behaviour characteristics, and to eliminate any difficulties associated with scaling welds and plates below practical sizes; (2) the maximum transverse load that could be applied to the specimen was 17,800 kN; and (3) the maximum plan dimensions were 5000 mm by 2600 mm, and the maximum depth between hulls was limited to 700 mm. The test specimen was a scaled equivalent version of the keel of a 40,000 DWT oil tanker with two longitudinal girders and a transverse floor, as illustrated in Figure 3.2, shown in relation to the tanker from which it was taken. (Note: not all elements were scaled by the same amount, due to the implementation of a different design philosophy and for the selection of realistic plate sizes).

An all-steel test specimen was used for the initial investigation into the behaviour of longitudinally-framed double hull tankers to maximize the energy absorption capacity. An equivalent sub-section representing the cross-section of the test specimen was taken from the keel of a 40,000 DWT tanker, illustrated in Figure 3.3. Identified in the figure are the inner hull, longitudinal girders, transverse floor, and the outer hull where panel #2 was transversely loaded and panels #1 and #3 are defined as those adjacent to the loaded panel.

Figure 3.2: Location of Large-Scale Test Specimen Relative to the 40,000 DWT Tanker

Figure 3.3: Definition of Large-Scale Test Specimen Elements

Based on a recent presentation on the state-of-the-art for inelastic membrane analysis of continuous steel plates (Bond, 1995), it was anticipated that the following failure modes, as illustrated in Figures 3.4, 3.5 and 3.6, may exist for this longitudinally framed structure:

- (1) If the longitudinal girders and inner hull are stiff and remain elastic, elastic-inelastic membrane action of panel #2 of the outer hull would develop, followed by rupture at or near the longitudinal girder junction where the maximum bending and membrane strains occur.
- (2) If the longitudinal girders are slender enough, panel #2 would develop inelastic membrane action, the longitudinal girders would buckle elastically or inelastically, and panels #1 and #3 would develop membrane action. The continuous collapse of the longitudinal girders and the spread of membrane action would continue away from the loaded panel and could continue until the outer hull made contact with the inner hull. Obviously, with the

Comments:

Outside hull panel #2 will rupture. The remaining structure will remain elastic and will have sufficient residual strength for post-accident or post-extreme load event.

Figure 3.4: Failure Mode #1

Comments:

Elastic-inelastic membrane action of outside hull panel #2.

Figure 3.5 (a): Failure Mode #2

Comments:

Buckling or collapse of longitudinal girders followed by elastic-inelastic membrane action of the outer panels #1 and #3.

Figure 3.5 (b): Failure Mode #2 (Continued) – Buckling or Collapse of Longitudinal Girders

Comments:

Contact of outer and inner hulls. This may require an increase in contact area of the colliding object.

Figure 3.5 (c): Failure Mode #2 (Continued) – Contact of Outer Hull and Inner Hulls

Comments:

Collapse of adjacent longitudinal girders and development of membrane action in outer panels in combination with elastic-inelastic membrane action of the inner hull. Residual strength is in the portions of the structure not damaged.

Figure 3.5 (d): Failure Mode #2 (Continued) – Collapse of Adjacent Longitudinal Girders

Comments:

Elastic and inelastic membrane action of outer and inner hulls.

Figure 3.6: Failure Mode #3

collapse of the longitudinal girders, little load would be transmitted into the inner hull which would, therefore, remain elastic. Once the outer hull is in contact with the inner hull, there may be one of the following failure modes:

- (a) with sufficient local shearing, a rupture of the outer hull may occur;
 - (b) large bending strains resulting from the collapse of the longitudinal girders may cause a rupture of the outer hull; or
 - (c) the inner and outer hulls may rupture simultaneously, where the strain capacity is exceeded at or near a weld or heat affected zone (HAZ) in the structure.
- (3) If the longitudinal girders are stiff enough to transmit load into the inner hull, then inelastic membrane action could be developed in panel #2, then in adjacent panels #1 and #3, and then in all panels of the inner hull before a failure occurs where the strain capacity is exceeded at or near a weld or HAZ in the structure.

The task was to determine what was practical with respect to plate sizes and cost, balanced with maximizing the energy absorption of the system while providing a structure that had sufficient residual strength after the accidental or extreme load event. Given the plan dimensions of the test specimen, the hull spacing of 700 mm, and typical stress-strain characteristics for steel, a series of inelastic, non-linear geometric analyses were performed using ANSYS 5.3 (a commercially available, advanced finite element analysis package developed and supported by Swanson Analysis Systems Inc. of Houston, PA.) on a three-

panel, two-longitudinally girder framed section with a transverse floor, where the center panel was transversely loaded. The elements were modelled with an assemblage of plastic shell elements (SHELL43). These elements, well suited to model nonlinear, flat or warped, thin to moderately thick shell structures, have six degrees of freedom at each node, those being translations in the x-, y- and z-directions and rotations about the x-, y- and z-axes. The deformation shape is linear in both in-plane directions. The out-of-plane motion is predicted using a mixed interpolation of tensorial components. The SHELL43 element is defined by four corner nodes, four thicknesses and orthotropic material properties (ANSYS, 1995).

To describe the plastic behaviour, the Multilinear Isotropic Hardening (MISO) option in ANSYS was used. The MISO option uses the von Mises criteria coupled with an isotropic work hardening assumption. This option was used since it gives the best results for large strain analyses. The Newton-Raphson option (NROPT) was left for the program to choose which options (solution strategy) to use. Large deformation (NLGEOM) effects (large deflections, large rotations, and/or large strains) and stress stiffening (SSTIF) effects were included in the analysis.

A static nonlinear analysis, which corresponds to the static loading condition of the experimental program, was used. To trace the post yield behaviour, incremental loading was used. The edges of the model were fixed against all translations and rotations. The loading was applied as pressure loads with dimensions of 200 mm by 300 mm which corresponded to the contact surface of the load plates. A refined mesh was used directly under the load. The finite element meshes used for the inner hull, the longitudinal girders

and transverse floor, and the outer hull of the model are illustrated in Figures 3.7 (a), (b) and (c), respectively.

The plate sizes for each were varied to extremes beyond practical limits and the analysis was conducted to determine the failure modes that were achievable. A review of load-deflection curves, stress distributions and behaviour of each analysis revealed that the failure mode #1 was plausible but did not offer sufficient energy absorption capacity to meet the design criteria, and failure mode #3 could not be achieved. The remaining task was to size the elements to maximize the energy absorption capacity of failure mode #2.

The subset of design alternatives used to achieve the optimum configuration are summarized in Table 3.4. The first set of plate sizes (design alternative #1) was selected to give equivalent thicknesses of an equivalent composite structural laminate of both the outer and inner hulls, and a realistic slenderness value for the longitudinal girders to achieve some form of inelastic behaviour.

Table 3.4: Summary of ANSYS Test Plate Design Alternatives

Design Alternative	Outer Hull Plate Thickness (mm)	Inner Hull Plate Thickness (mm)	Longitudinal Girder and Transverse Floor Plate Thickness (mm)
1	15.0	20.0	10.0
2	15.0	12.0	10.0
3	16.0	12.0	14.3
4	16.0	7.94	12.7

Figure 3.7 (a): Finite Element Mesh of Inner Hull

Figure 3.7 (b): Finite Element Mesh of Longitudinal Girders and Transverse Floor

Figure 3.7 (c): Finite Element Mesh of Outer Hull

In general, the behaviour of design alternative #1 followed that described by failure mode #2, with buckling of the longitudinal girders at an approximate total applied load of 6600 kN. The analysis was concluded at the buckling of the longitudinal girders. It is anticipated that continued loading would further develop the inelastic membrane action of the outer panels #1 and #3 of the outer hull. The examination of von Mises stresses indicated the inside hull plate reached a maximum stress of 166 MPa.

Given the maximum stress value of the inside hull for design alternative #1, the thickness of the inside hull plate was reduced from 20.0 mm to 12.0 mm, which should have increased the von Mises stresses of the inside hull plate to approximately 276 MPa. The same general behaviour was exhibited in design alternative #2 with buckling of the longitudinal girders at an approximate total applied load of 5800 kN. The lower buckling load resulted from a reduced fixity at the girder-inner hull junction. The maximum von Mises stress of the inner hull plating was approximately 206 MPa.

For design alternative #3, the outer hull plating was increased from 15.0 mm to 16.0 mm to reflect realistic plate sizes. The longitudinal girders and transverse floor thicknesses were increased from 10.0 mm to 14.3 mm to increase the load transfer to the inner hull plating in an attempt to promote inelastic membrane action of the hull. The overall behaviour was as described by failure mode #2 with buckling of the longitudinal girders at an approximate total applied load of 7600 kN. The transverse deflection of panel #2 was approximately 155 mm, giving a span-to-deflection ratio of 5.8 which was very close to ratios found in other tests where the loaded plate ruptured. The maximum von Mises stress in the inner hull plate was approximately 286 MPa, still well below the plate yield stress of 322 MPa.

The thicknesses of the longitudinal girders and transverse floor were reduced for design alternative #4 from 14.3 mm to 12.7 mm. The inner hull plating was also reduced from 12.0 mm to 7.94 mm and holes were also placed on the plating for access to LVDT's and other observations during the large-scale test. The combination of reduced plate thicknesses was done in an attempt to promote yielding of the inner hull plating without rupture of panel #2 prior to buckling of the longitudinal girders. The corresponding total applied load-deflection curves for the center load point, longitudinal girders, transverse floor, inner hull plate at the longitudinal girder junction, and the outer hull plate at the longitudinal girder junction are illustrated in Figure 3.8. The corresponding von Mises stress distributions for an isometric view of the design model, the inner hull, the outer hull, the transverse floor, and the longitudinal girders are illustrated in Figures 3.9 (a), (b), (c), (d) and (e), respectively. The longitudinal girders buckled at an approximate total applied load of 7560 kN, and a maximum von Mises stress of the inner hull was in the range of 272 to 324 MPa with signs of local yielding around the access holes.

Figure 3.8: Total Applied Load-Deflection Relationships for Design Alternative #4

Figure 3.9 (a): Isometric View of von Mises Stress Distribution for Design Alternative #4

Figure 3.9 (b): Stress Distribution of the Inner Hull Plate for Design Alternative #4

Figure 3.9 (c): Stress Distribution of the Outer Hull Plate for Design Alternative #4

Figure 3.9 (d): Stress Distribution of the Transverse Floor Plate for Design Alternative #4

Figure 3.9 (e): Stress Distribution of the Longitudinal Girder Plate for Design Alternative #4

With realistic plate sizes consistent with those produced by current design Rules and Regulations, and based on the parametric FEA study conducted, it was determined that it may not be possible to promote yielding of the inner hull plating prior to the outer hull making contact and with the inner hull and the two hull plates being loaded simultaneously. In the UDHSPS design, the inner hull plate consisted of a minimum (for corrosion, fabrication and handling) of two 8 mm plates and a 50 mm core; hence, yielding of this plate prior to contact was even more remote. It should be noted that although an all-steel section was investigated, it was anticipated that the composite structural laminate proposed for the UDHSPS would provide superior resistance since it would be able to dissipate strains over a larger area, thus delaying the initiation of rupture. Design alternative #4 presents the final plate thicknesses for the large-scale test specimen.

3.2 Experimental Program

3.2.1 Large-Scale Test Specimen Geometry

An experimental program was undertaken to determine the behaviour of a typical composite structural laminate plate used in the construction of both the inner and outer hulls of the UDHSPS tanker, when subjected to transverse loads. A typical 40,000 DWT midship cross-section of the UDHSPS is illustrated in Figure 3.10.

Figure 3.10: Midship Section of Unidirectional Double Hull Sandwich Plate System

The test specimen represented a large-scale (between one-third and one-half size) model of the keel of a double hull 40,000 DWT oil tanker. The double hull test specimen consisted of a composite structural laminate outside hull composed of two steel plates, 7.94 mm (5/16 inch) thick, measuring 5000 mm by 2600 mm in plan separated by 50 mm of polyurethane elastomer structural core material; two longitudinal girders, and a transverse floor, each 700 mm deep and 12.7 mm (1/2 inch) thick; and a 7.94 mm (5/16 inch) thick steel plate that represented the composite structural laminate inner hull. The construction of the outer hull, longitudinal girders and transverse floors were typical of those proposed for this system. The latter two were connected to the inside plate of the outside hull. Spacer plates ran the length of the plate at mid-distance between longitudinal girders and ran transversely at 1400 mm from the east side of the horizontal reaction frame to separate the inner and outer plates of the outside hull by the prescribed core thickness and provided cavities of a given volume to ensure that these could be filled within the projected set time and pump volume limitations. An assembled and an exploded view of the test specimen are shown in Figures 3.11 and 3.12, respectively. The transverse floor, longitudinal girders, inner and outer hull plates of the outside hulls and the inner hull with access holes are identified in the corresponding Figures. The detailed shop drawings of the test specimen, shown in Figures 3.13 (a) and (b), identify the outer hull plates, longitudinal girders, transverse floor, inner hull, cover plates and cavity spacers. The longitudinal spacer and weld details joining adjacent panels represented typical modular construction details used in the full-scale oil tanker.

The exploded view of the test specimen shown in Figure 3.12 clearly illustrates the stiffened border which surrounded the composite outer hull. The stiffened border

transmitted the forces from the loaded specimen to the horizontal reaction frame and allowed for welding and cutting of the test specimen to and from the frame without damaging the elastomer core. The supporting horizontal reaction frame represented the stiffness of the surrounding ship structure and provided the in-plane and transverse restraint to develop membrane action.

Figure 3.11: Assembled View of Large-Scale Test Specimen

Figure 3.12: Exploded View of Large-Scale Test Specimen

Figure 3.13 (a): Detailed Shop Drawings of the Outer Hull, Longitudinal Girders and Transverse Floor of the Large-Scale Test Specimen

DETAIL "C"

3.2.1.1 Large-Scale Test Objectives

The objectives were:

- (1) to determine the inelastic behaviour and the interaction of the longitudinal girders, the transverse floor, and the composite outside hull when subjected to transverse loads;
- (2) to observe the progression of inelastic membrane action of the outside hull panels adjacent to the loaded panel as the transverse floor and the longitudinal girders collapsed;
- (3) to make observations and measurements in which the finite element analysis could be calibrated; and
- (4) to make observations related to puncture and tear resistance at load points and at or near the transverse floor with respect to crack propagation.

3.2.1.2 Large-Scale Test Specimen Elastomer Injection

A two-component polyurethane elastomer, trade name “Elastocast”, was selected from BASF as the core material. The two components were an “iso” and a resin. The “iso” was made from WUC 3092T isocyanate (the common chemical name is diphenylmethane diisocyanate prepolymer). The resin was made from the Elastocast 70559R Resin (the common chemical name is urethane resin component).

Discussions with BASF and a local contractor specializing in non-structural sandwich panel construction led to the decision to sub-divide the cavity transversely, as

illustrated in Figure 3.12, to ensure that the cavity could be completely filled with the injected elastomer material prior to setting. The elastomer was injected after the test specimen was welded into place to prevent any damage to the elastomer from the heat of the welding process. The Gusmer pump used for the elastomer injection is illustrated in Figure 3.14 (a). A stiffened edge with cover plates for the inner plating of the outside hull and continuous outer plating of the outside hull was provided to facilitate the welding and subsequent removal of the test specimen to and from the reaction frame. Each cavity was provided with four air vents at each of the corners. A central injection port was drilled and threaded into each of the cavities. The injection gun was then threaded into the injection ports to fix it to the plate during the injection process, as illustrated in Figure 3.14 (b). To ensure that the test specimen cells would be filled with a quality mix (no discontinuities, air bubbles, voids, etc.), two test cavities with similar cavity volume and measuring 815 mm by 4647 mm were built, as partially illustrated in Figure 3.14 (c). The steel plates were separated by 50 mm. A $\frac{1}{4}$ inch thick plexiglass was placed around the perimeter to allow the injection process to be observed. Once the cavities were cast and the quality of the mix assured, the injection of the large-scale test specimen followed.

3.2.1.3 Test Set-Up

The test set-up consisted of a horizontal frame, support columns, high-strength tie-down rods, a stiffened plate girder, hydraulic actuators, electronic instrumentation, a data acquisition system and the large-scale test specimen. Figures 3.15 and 3.16 give an

(a) Gusmer Pump

(b) Static Mixer

(c) Test Cavity Composite Structural Laminate

Figure 3.14: Polyurethane Elastomer Casting Process

Figure 3.15: Cross-Section of Test Specimen within Test Apparatus

Figure 3.16: Plan View of Test Specimen within Test Apparatus

elevation and plan view of the test specimen in the test apparatus, respectively. For clarity, the instrumentation has not been shown.

3.2.1.4 Loading Apparatus

The four transverse loads were applied to the test specimen along the longitudinal centerline axis, as shown in Figure 3.17. The loads were transmitted from an actuator through a knife-edge roller system to the outer plate of the outside hull over an area measuring 200 mm by 300 mm. This knife-edge roller system allowed for a relative rotation between the actuator and plate up to 30 degrees and a lateral displacement up to 70 mm and ensured that the applied loads were vertical. A series of copper plates placed between the knife-edge roller plate and the test specimen rounded the sharp edges of the load plates to help reduce the local shearing and tearing at these locations. The reactions from the hydraulic actuators were transmitted to the laboratory strong floor by way of a stiffened plate girder. The hydraulic system was comprised of an electric Enerpac pump, a hydraulic reservoir, solenoid valves and manifolds.

3.2.1.5 Horizontal Reaction Frame

The test specimen was welded all around to a horizontal reaction frame, which simulated the stiffness of the surrounding ship structure. The frame also provided the in-

Figure 3.17: Load Application Points for the Outer Plate Outside Hull of Test Specimen

plane and transverse restraint required to develop membrane action. The frame was designed to behave elastically under ultimate test conditions, thus permitting future use of the frame. It was restrained from vertical displacements with eight columns and twelve high-strength tie-down rods. The columns supported the dead load of the frame and test specimen and, with the rods, worked together to produce a force couple that removed the torque applied to the frame through the loaded specimen.

3.2.1.6 Instrumentation and Measurement

The applied loads, strains and displacements were continuously measured during the test using four pressure transducers, 100 electrical resistance gages and 35 linear

variable differential transformers (LVDTs), respectively. Figures 3.18 (a), (b), (c) and (d) indicate the LVDT locations on the outer plate of the outside hull, the inner plate of the outside hull, the longitudinal girders and the transverse floor elements, respectively. (Note: out-of-plane displacements for the longitudinal girders and transverse floor were measured on the corresponding side where the access holes were located). Figures 3.19 (a), (b), (c), (d), and (e) indicate the strain gage locations of the outer plate of the outside hull, the inner plate of the outside hull, the inner hull plating, the longitudinal girders, and the transverse floor, respectively. (Note: strain gages on the inner hull with circles represent those located on the outside surface of the inner hull. Strain gages on the longitudinal girder and transverse floor with circles represent those located on the plate surface away from the central loaded panel). Figures 3.19 (a), (b), (c) and (e) illustrate the designated north-south orientation of the large-scale test specimen which will be used to reference the plate surfaces of the longitudinal girders and transverse floor.

3.2.1.7 Installation, Alignment and Instrumentation of Test Specimen

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

- (1) the outside hull, longitudinal girders and transverse floor section was fitted to the horizontal reaction frame;
- (2) strain gages were installed on the outer plate of the outside hull;
- (3) the knife-edge-roller assemblies were lifted into place and positioned;

Figure 3.18 (a): LVDT Locations of Outer Plate Outside Hull

Figure 3.18 (b): LVDT Locations of Inner Hull

Figure 3.18 (c): LVDT Locations of Longitudinal Girders

Figure 3.18 (d): LVDT Locations of Transverse Floor

Figure 3.19 (a): Strain Gage Locations of Outer Plate Outside Hull

Figure 3.19 (b): Strain Gage Locations of Inner Plate Outside Hull

Figure 3.19 (c): Strain Gage Locations of Inner Hull

Figure 3.19 (d): Strain Gage Locations of Longitudinal Girders

Note: Strain gages with circles represent those located on the plate surface away from the central loaded panel.

Figure 3.19 (e): Strain Gage Locations of Transverse Floor

- (4) the panel was lifted into position and welded into place;
- (5) elastomer was injected into the four cavities;
- (6) hydraulic actuators were connected to the hydraulic pump system;
- (7) the cover plates of the outside hull were welded into place;
- (8) strain gages were installed on the inner plate of the outside hull, longitudinal girders and transverse floor;
- (9) strain gages were installed on the outside of the inner hull plating;
- (10) the inside hull plate was welded into place;
- (11) strain gages were installed on the inside of the inner hull plating;

- (12) LVDT support frames were installed above and below the test specimen and were attached to the horizontal reaction frame (all deflections are relative to the reaction frame);
- (13) LVDTs were installed and connected;
- (14) all strain gages and pressure transducers were connected to the data acquisition system;
- (15) the test set-up and data files were generated using LABCON;
- (16) the power supply was adjusted and the hydraulic actuators extended to bring the knife-edge-rollers into contact with the test panel;
- (17) the specimen was subjected to a small pre-load, the instrumentation checked, and malfunctioning instrumentation fixed;
- (18) the test specimen was unloaded and the data acquisition system was zeroed; and
- (19) initial observations and measurements were made.

3.2.1.8 Test Procedure

Load was applied monotonically in load increments, as determined by the load-deformation curve, until rupture of the composite outer hull. After failure, the test specimen was unloaded and residual deformations were recorded. At the end of each load step increment, strain gage, LVDT and pressure transducer readings were taken electronically and recorded through the data acquisition system. Observations were made after every load step and any phenomenon, noises, deflected shape, and progressions of failures were recorded. Photographs were taken periodically to complement the recordings.

3.2.2 Ancillary Tests

3.2.2.1 Steel Strength Test

Tension coupons from both the longitudinal and transverse directions were tested in uni-axial tension to determine the stress-strain relationships for the outer plate of the outside hull, the inner plate of the outside hull, the longitudinal girders and transverse floor, and the inner hull plate. The coupons were taken from pieces of the plates that were flame cut from the parent plates. The coupons were cut and milled according to ASTM E8M-88 using a gage length of 200 mm and width of 40 mm, and are illustrated in Figure 3.20.

Figure 3.20: Steel Tension Coupon Definition

Measurements of the width and thickness of the gage length were taken to determine the reduced cross-sectional area of the coupon. Load, longitudinal and transverse strains were measured. All tension coupon tests were performed in a 1780 kN Tinius Olsen testing machine and in accordance with ASTM E8M-88.

3.2.2.2 Polyurethane Elastomer Strength Tests

Compression, tension and shear-bond tests were performed on the polyurethane elastomer to determine material characteristics.

3.2.2.2.1 Compression Test

Ten elastomer compression disks, with a diameter of 28.6 mm and thickness of 12.5 mm, were cut from sheets cast during the casting of the test cavities (indicated as disks TC-x) and the large-scale test specimen (indicated as disks LST-x) (five from each). The tests were performed according to ASTM D575-83 Method A. The first set of compression disks (TC-x) was tested with variable loading rates to examine the effects of loading rate on the elastomer. Compression disk TC-1 was loaded at a rate of 25.4 mm/min (1.0 in/min), while disks TC-2, TC-3 and TC-4 were loaded at a rate of 12.7 mm/min (0.5 in/min). The final compression disk sample, disk TC-5, was loaded at a rate of 2.5 mm/min (0.1 in/min). The second set of elastomer compression disks (LST-x) was also loaded with variable loading rates. Compression disk LST-1 was loaded at a rate of

25.4 mm/min (1.0 in/min), disk LST-2 at 50.8 mm/min (2.0 in/min), and disks LST-3 and LST-4 at a 2.5 mm/min (0.1 in/min). The final compression disk sample, disk LST-5, was loaded at a rate of 25.4 mm/min (1.0 in/min) and had both loading surfaces greased. Load and deflection readings were taken throughout the test. Stress-strain and load-deflection graphs were then plotted for each specimen.

3.2.2.2.2 Tension Test

Two sets of elastomer tension coupon specimens, with a reduced area length of 59 mm, gage width of 6.0 mm, gage length of 25 mm, and thickness of 3.0 mm, were cut from sheets cast during the casting of the test cavities (indicated as coupon TC-x) and large-scale test specimen (indicated as coupon LST-x). The coupon dimensions are illustrated in Figure 3.21. The tests were performed according to ASTM D412-83 Method A. Three measurements were made for the thickness and width, one at the center and one at each end of the reduced section of the specimen, to determine the reduced cross-sectional area. Load and deflection readings were taken approximately every 5.0 mm (0.2 in) to plot stress-strain and load-deflection curves. The rate of loading was 50.8 mm/min (2.0 in/min). A special knife-edge punch cutting tool was designed and machined to cut these specimens from the 3.0 mm thick sheets.

Figure 3.21: Elastomer Tension Coupon Definition

3.2.2.3 Steel-Elastomer-Steel Shear-Bond Test

A series of tests was conducted on the steel-elastomer-steel section to examine the shear-bond behaviour of the composite and to determine the bond capacity of the elastomer-steel interface. The test specimens consisted of two 6 mm steel plates with 50 mm elastomer in-fill cast by BASF. The prepared specimen was placed in a shear-bond apparatus built specifically to shear one of the steel faceplates from the elastomer core. (Note: the apparatus was designed to minimize the transverse distance between the force pulling on the faceplate and the restraint provided to the core and other face plate in order to minimize the moment required to keep the system in equilibrium. This small value was taken out by restraining blocks at the base on one side and adjustment bolts on the other). The load and relative displacement of the sheared faceplate and core were measured. The maximum load was recorded to determine the shear-bond capacity based on the measured shear area. The test specimen and apparatus are illustrated in Figure 3.22.

Figure 3.22: Shear Specimen in Shear-Bond Test Apparatus

Chapter 4

Test Results and Discussion

4.1 Initial Conditions and Steel Plate Material Properties

The large-scale test specimen was constructed from several steel plates of various thicknesses, as illustrated in Figure 3.12. The outer plate of the outside hull was constructed from two steel plates, 7.94 mm (5/16 inch) thick, connected with a full-length square butt weld at a distance of 873.5 mm from the longitudinal centerline. The inner plate of the outside hull was also constructed from two steel plates, 7.94 mm (5/16 inch) thick, connected with a full-length butt weld along the longitudinal centerline. The inner and outer plates of the outside hull were separated by 50 mm high spacers that ran longitudinally and transversely inside the bottom to subdivide the cavity into four volumes consistent with the pump capacity and gel time of the polyurethane elastomer. The longitudinal spacer was 19.05 mm (3/4 inch) thick and ran along the plate centerline while the transverse spacer was 6.35 mm (1/4 inch) thick and was placed at an approximate distance of 2200 mm from the transverse floor. To provide sufficient bending restraint for the void around the outer hull perimeter of the specimen (all-steel perimeter facilitated welding and removal of the test specimen to and from the horizontal reaction frame), a series of 19.05 mm (3/4 inch) and 38.1 mm (1½ inch) stiffeners were placed in both the longitudinal and transverse directions, as illustrated in Figure 3.12. The

700 mm deep longitudinal girders and transverse floor were constructed from 12.7 mm (1/2 inch) thick steel plates that were welded to the inner plate of the outside hull. Because of the construction sequence for welding the test specimen into the horizontal reaction frame, the casting of the elastomer and the installation of strain gages, the inner hull had to be constructed in 6 pieces, each measuring 7.94 mm (5/16 inch) thick. These pieces were subsequently welded to the longitudinal girders and transverse floor elements with butt welds and to the horizontal reaction frame with full-penetration groove welds. A series of holes measuring 200 mm by 300 mm with rounded corners (75 mm in radii) were made in the inner hull plate assembly to allow for LVDT measurements and other observations during the test. The chemical and mechanical properties reported from the mill certificates for the 12.7 mm (0.5 inch) plates used for the longitudinal girders and transverse floor, and for the 7.94 mm (5/16 inch) plates used for the outer and inner plates of the outside hull and the inside hull plate, are given in Table 4.1. Five coupons from both the longitudinal (coincident with the rolling direction) and transverse directions were cut from a section of parent plate and tested to determine the material properties. The material properties E , ν , σ_y , ϵ_y , σ_{fy} , ϵ_{fy} , E_{α} , σ_{uk} , ϵ_{uk} , σ_f , and ϵ_f along with the mean, standard deviation, and coefficient of variation for the longitudinal and transverse directions are given in Tables 4.2 to Table 4.5 for each of the steel plate tension coupons. Typical stress-strain curves for the longitudinal and transverse directions corresponding to each of the plates are illustrated in Figures 4.1 to Figure 4.4.

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

Plate (mm)	Grade	Manufacturer	Heat No.	Yield Point (MPa)	Ultimate Tensile Strength (MPa)	Elongation (%)	Chemical Composition (%)				
							C	Mn	P	S	
12.70 x 2438 x 6096	CAN/CSA- G40.21-92 50W	Stelco Steel, Hamilton Ontario	433565 534531 633070 633265	393.0	538.5	25	0.18	0.005	0.012	0.27	
				383.3	525.4	23	0.18	0.008	0.007	0.24	
				383.3	524.7	23	0.19	0.008	0.010	0.26	
				399.2	533.7	24	0.18	0.009	0.008	0.26	
7.94 x 2438 x 6096	CAN/CSA- G40.21-92 50W	Stelco Steel, Hamilton Ontario	534728 534729	413.0	541.2	24	0.17	0.008	0.007	0.25	
				399.9	518.5	26	0.17	0.006	0.008	0.27	

Table 4.2 (a): Characteristic Material Properties Determined from Uniaxial Stress-Strain Curves for the 12.7 mm (1/2 inch) Longitudinal Girder and Transverse Floor Members in the Direction Longitudinal to the Rolling

Coupon No.	E (MPa)	ν	σ_y (MPa)	$\epsilon_y \times 10^3$	σ_T (MPa)	$\epsilon_T \times 10^3$	E_T (MPa)	σ_{TS} (MPa)	$\epsilon_{TS} \times 10^3$	σ_u (MPa)	$\epsilon_u \times 10^3$
1L4	214,510	---	377.7	1,761	*	22,620	4115	522.5	160,000	431.3	250,000
1L5	211,780	0.284	376.0	1,775	*	20,750	5323	520.8	193,000	386.7	258,100
1L6	208,760	0.280	371.2	1,778	313.8	19,780	5479	522.0	162,100	418.4	251,900
1L7	210,090	---	373.4	1,777	*	21,590	4249	521.8	162,100	422.0	261,800
1L8	209,390	---	372.6	1,780	*	21,220	6441	523.8	157,100	406.9	251,900
Average	210,906	0.282	374.2	1,774	---	21,192	5121	522.2	166,860	413.1	254,740
Std. Dev.	2,309	---	2.63	7.63	---	1,047.32	959.72	1.10	14,755.78	17.13	4,993.30
ν	0.011	---	0.007	0.004	---	0.049	0.187	0.002	0.088	0.041	0.020

* No static yield values were present for coupons 1L4, 1L5, 1L7 and 1L8

Table 4.2 (b): Characteristic Material Properties Determined from Uniaxial Stress-Strain Curves for the 12.7 mm (1/2 inch) Longitudinal Girder and Transverse Floor Members in the Direction Transverse to the Rolling

Coupon No.	E (MPa)	ν	σ_y (MPa)	$\epsilon_y \times 10^3$	σ_T (MPa)	$\epsilon_T \times 10^3$	E_T (MPa)	σ_{TS} (MPa)	$\epsilon_{TS} \times 10^3$	σ_u (MPa)	$\epsilon_u \times 10^3$
1T4	209,520	---	376.8	1,798	323.0	15,690	4860	525.6	165,000	448.0	245,000
1T5	225,390	---	374.1	1,660	310.1	18,600	4893	518.6	150,000	441.2	245,000
1T6	208,390	0.286	375.1	1,801	*	20,990	5341	524.5	165,000	459.0	237,500
1T7	202,530	0.284	378.3	1,868	334.3	21,610	5867	521.7	165,000	453.3	247,500
1T8	214,390	---	375.2	1,750	324.0	20,490	6356	522.9	160,000	447.6	255,000
Average	212,044	0.285	375.9	1,775	322.9	19,476	5463	522.7	161,000	449.8	246,000
Std. Dev.	8,571	---	1.65	76.93	9.9	2,396.64	645.16	2.72	6,519.20	6.69	6,274.95
ν	0.040	---	0.004	0.043	0.031	0.123	0.118	0.005	0.040	0.015	0.026

* No static yield value was present for coupon 1T6

Table 4.3 (a): Characteristic Material Properties Determined from Uniaxial Stress-Strain Curves for the 7.94 mm (5/16 inch) Inner Plate of the Outside Hull in the Direction Longitudinal to the Rolling

Coupon No.	E (MPa)	ν	σ_y (MPa)	$\epsilon_y \times 10^{-3}$	σ_{UT} (MPa)	$\epsilon_{UT} \times 10^{-3}$	E_{UT} (MPa)	$\sigma_{0.2}$ (MPa)	$\epsilon_{0.2} \times 10^{-3}$	σ_u (MPa)	$\epsilon_u \times 10^{-3}$	R_m (MPa)
2L4	209,710	---	397.3	1,895	*	21,380	5547	525.1	142,100	465.0	219,500	
2L5	208,240	0.280	389.2	1,869	*	21,880	5829	523.3	165,000	457.9	227,500	
2L6	211,910	0.279	387.3	1,828	*	20,920	6363	524.6	154,200	450.6	218,900	
2L7	211,360	---	393.9	1,864	*	19,380	3128	522.0	142,100	462.3	219,500	
2L8	209,620	---	395.5	1,887	*	20,850	4678	530.9	160,000	493.7	225,000	
Average	210,168	0.280	392.6	1,868	---	20,882	5109	525.2	152,680	465.9	222,080	
Std. Dev.	1,473	---	4.24	25.97	---	935.75	1264.41	3.42	10,386.87	16.47	3,915.61	
ν	0.007	---	0.011	0.014	---	0.045	0.247	0.007	0.068	0.035	0.018	

* No static yield values were present for coupons 2L4, 2L5, 2L6, 2L7 and 2L8

Table 4.3 (b): Characteristic Material Properties Determined from Uniaxial Stress-Strain Curves for the 7.94 mm (5/16 inch) Inner Plate of the Outside Hull in the Direction Transverse to the Rolling

Coupon No.	E (MPa)	ν	σ_y (MPa)	$\epsilon_y \times 10^{-3}$	σ_{UT} (MPa)	$\epsilon_{UT} \times 10^{-3}$	E_{UT} (MPa)	$\sigma_{0.2}$ (MPa)	$\epsilon_{0.2} \times 10^{-3}$	σ_u (MPa)	$\epsilon_u \times 10^{-3}$	R_m (MPa)
2T4	206,860	---	392.7	1,898	344.7	20,400	4973	528.0	170,000	469.6	230,000	
2T5	228,080	0.289	393.9	1,727	*	20,490	4616	526.7	167,100	462.1	231,900	
2T6	208,750	0.293	394.9	1,892	*	22,770	5326	529.1	155,000	446.2	225,000	
2T7	218,480	---	392.3	1,796	*	21,220	5184	529.1	157,900	474.7	228,100	
2T8	220,630	---	394.0	1,786	353.7	19,980	5053	529.4	157,900	474.7	238,100	
Average	216,560	0.291	393.6	1,820	349.2	20,972	5030	528.5	161,580	465.5	230,620	
Std. Dev.	8,776	---	1.05	73.65	---	1,099.76	267.71	1.12	6,552.63	11.94	4,896.63	
ν	0.041	---	0.003	0.040	---	0.052	0.053	0.002	0.041	0.026	0.021	

* No static yield values were present for coupons 2T5, 2T6 and 2T7

Table 4.4 (a): Characteristic Material Properties Determined from Uniaxial Stress-Strain Curves for the 7.94 mm (5/16 inch) Inner Hull Plate in the Direction Longitudinal to the Rolling

Coupon No.	E (MPa)	ν	σ_y (MPa)	$\epsilon_y \times 10^{-3}$	σ_T (MPa)	$\epsilon_T \times 10^{-3}$	E_{sh} (MPa)	σ_{sh} (MPa)	$\epsilon_{sh} \times 10^{-3}$	σ_c (MPa)	$\epsilon_c \times 10^{-3}$
3L4	205,220	---	390.0	1,901	*	20,010	5964	528.1	162,100	478.7	246,900
3L5	215,610	0.281	385.2	1,786	*	18,980	5514	526.6	175,000	437.2	245,000
3L6	215,760	0.286	387.3	1,795	336.4	19,360	4750	527.1	165,000	471.8	247,500
3L7	214,930	---	391.7	1,822	*	19,790	6222	531.8	170,000	417.7	247,500
3L8	217,390	---	389.1	1,790	365.5	19,220	4656	530.3	155,000	437.7	245,000
Average	213,782	0.284	388.7	1,819	351.0	19,472	5421	528.8	165,420	448.6	246,380
Std. Dev.	4,871	---	2.50	47.80	---	420.92	703.582	2.21	7,624.43	25.73	1,283.35
ν	0.023	---	0.006	0.026	---	0.022	0.130	0.004	0.046	0.057	0.005

* No static yield values were present for coupons 3L4, 3L5 and 3L7

Table 4.4 (b): Characteristic Material Properties Determined from Uniaxial Stress-Strain Curves for the 7.94 mm (5/16 inch) Inner Hull Plate in the Direction Transverse to the Rolling

Coupon No.	E (MPa)	ν	σ_y (MPa)	$\epsilon_y \times 10^{-3}$	σ_T (MPa)	$\epsilon_T \times 10^{-3}$	E_{sh} (MPa)	σ_{sh} (MPa)	$\epsilon_{sh} \times 10^{-3}$	E_{ch} (MPa)	σ_{ch} (MPa)	$\epsilon_{ch} \times 10^{-3}$
3T4	206,000	---	401.0	2,136	371.6	18,500	4753	536.6	170,000	479.9	240,000	
3T5	217,140	0.292	396.6	1,827	374.8	20,770	5106	530.7	175,000	473.7	245,000	
3T6	203,000	0.282	394.8	1,945	*	22,680	4390	531.0	170,000	476.5	245,000	
3T7	219,430	---	394.0	1,796	358.1	19,940	4548	532.2	170,000	448.7	250,000	
3T8	208,540	---	392.9	1,884	342.0	20,670	6172	529.8	170,000	470.0	230,000	
Average	210,822	0.287	395.9	1,917	361.6	20,512	4994	532.1	171,000	469.8	242,000	
Std. Dev.	7,135	---	3.17	134.68	15.0	1,513.86	710.966	2.68	2,236.07	12.32	7,582.88	
ν	0.034	---	0.008	0.070	0.041	0.074	0.142	0.005	0.013	0.026	0.031	

* No static yield value was present for coupon 3T6

Table 4.5 (a): Characteristic Material Properties Determined from Uniaxial Stress-Strain Curves for the 7.94 mm (5/16 inch) Outer Plate of the Outside Hull in the Direction Longitudinal to the Rolling

Coupon No.	E (MPa)	ν	σ_y (MPa)	$\epsilon_y \times 10^{-3}$	σ_u (MPa)	$\epsilon_u \times 10^{-3}$	E_u (MPa)	$\sigma_{0.2}$ (MPa)	$\epsilon_{0.2} \times 10^{-3}$	σ_r (MPa)	$\epsilon_r \times 10^{-3}$
6L4	207,090	---	390.0	1,883	*	21,580	4668	526.7	157,100	420.2	244,400
6L5	207,000	0.286	386.5	1,867	*	20,740	6629	527.6	150,000	453.6	235,000
6L6	193,050	0.282	398.0	2,062	*	20,910	4109	531.3	160,000	476.2	235,000
6L7	219,520	---	381.4	1,738	353.6	20,500	4067	523.6	175,000	429.6	247,500
6L8	229,910	---	389.2	1,693	*	20,420	5222	524.7	162,100	420.4	241,900
Average	211,314	0.284	389.0	1,849	---	20,830	4939	526.8	160,840	440.0	240,760
Std. Dev.	13,993	---	6.04	144.61	---	462.06	1055.74	2.98	9,141.83	24.39	5,619.88
ν	0.066	---	0.016	0.078	---	0.022	0.214	0.006	0.057	0.055	0.023

* No static yield values were present for coupons 6L4, 6L5, 6L6 and 6L8

Table 4.5 (b): Characteristic Material Properties Determined from Uniaxial Stress-Strain Curves for the 7.94 mm (5/16 inch) Outer Plate of the Outside Hull in the Direction Transverse to the Rolling

Coupon No.	E (MPa)	ν	σ_y (MPa)	$\epsilon_y \times 10^{-3}$	σ_u (MPa)	$\epsilon_u \times 10^{-3}$	E_u (MPa)	$\sigma_{0.2}$ (MPa)	$\epsilon_{0.2} \times 10^{-3}$	σ_r (MPa)	$\epsilon_r \times 10^{-3}$
6T4	217,990	---	398.6	1,829	*	20,530	4536	531.7	162,100	463.3	219,500
6T5	209,610	0.287	392.6	1,873	*	21,410	6237	528.0	167,100	466.6	221,900
6T6	230,570	0.291	398.9	1,730	*	21,970	5699	528.5	147,900	479.6	223,000
6T7	208,160	---	396.0	1,903	*	22,350	6547	528.1	165,000	451.1	225,000
6T8	194,280	---	401.1	2,065	*	20,200	3629	533.0	165,000	467.6	227,500
Average	212,122	0.289	397.4	1,880	---	21,292	5330	529.9	161,420	465.6	223,380
Std. Dev.	13,371	---	3.25	122.19	---	917.34	1221.1	2.33	7,764.47	10.20	3,040.89
ν	0.063	---	0.008	0.065	---	0.043	0.229	0.004	0.048	0.022	0.014

* No static yield values were present for coupons 6T4, 6T5, 6T6, 6T7 and 6T8

Figure 4.1: Typical Stress-Strain Curves for the 12.7 mm Longitudinal Girders and Transverse Floor Members

Figure 4.2: Typical Stress-Strain Curves for the 7.94 mm Inner Plate of the Outside Hull

Figure 4.3: Typical Stress-Strain Curves for the 7.94 mm Inner Hull Plate

Figure 4.4: Typical Stress-Strain Curves for the 7.94 mm Outer Plate of the Outside Hull

4.2 Polyurethane Elastomer Material Properties

Material test specimens for compression and tension tests were taken from the 12.7 mm (1/2 inch) thick and the 3.2 mm (1/8 inch) thick sheets, respectively, that were cast simultaneously with the casting of the test cavity specimens and the large-scale test specimen to give a representative sample of the material.

4.2.1 Compression Tests

Two sets of five elastomer compression disk specimens were cut from the test cavity specimen elastomer (indicated as disk TC-x) and from the large-scale test specimen elastomer (indicated as disk LST-x). The disks were tested in the 90 kN Instron Testing Machine at Carleton University according to ASTM D575-83 Method A testing procedures.

The first set of compression disks was tested with variable loading rates to examine the effects of loading rate on the elastomer. Compression disk TC-1 was loaded at a rate of 25.4 mm/min (1.0 in/min), disks TC-2, TC-3 and TC-4 at a rate of 12.7 mm/min (0.5 in/min) and disk TC-5 at a rate of 2.5 mm/min (0.1 in/min).

The second set of elastomer compression disks was also loaded with variable loading rates. Compression disk LST-1 was loaded at a rate of 25.4 mm/min (1.0 in/min), disk LST-2 at 50.8 mm/min (2.0 in/min), disks LST-3 and LST-4 at a 2.5 mm/min (0.1 in/min) and disk LST-5 at a rate of 25.4 mm/min (1.0 in/min) with both loading surfaces greased. The latter sample was to examine whether or not the lateral restraint provided by

the loading platens via friction would affect the characteristic behaviour of the elastomer. Tables 4.6 and 4.7 summarize the results of the elastomer compression tests for both samples listing σ_p , ϵ_p , σ_u , and ϵ_u . The corresponding stress-strain curves for each set of elastomer compression disks are illustrated in Figures 4.5 and 4.6.

The ultimate stress and strain values measured were not that of the material but that of the maximum load capacity of the testing machine.

The stress-strain curves exhibited similar characteristics with a 16.1 percent coefficient of variation in the proportional limit that was unrelated to the loading rates or to whether the loaded surfaces was greased or not. Consequently, no conclusive statements can be made with respect to the compressive stress-strain characteristics for varying loading rates as applied to the compression disk specimen, the large-scale test specimen or in application for double hull product oil tankers.

4.2.2 Tension Tests

A series of material tension coupons were cut using a die from the sheets of material cast for the test cavity specimens and the large-scale test specimen and are indicated as coupons TC-x and LST-x, respectively. The coupons were tested in the 90 kN Instron Testing Machine at Carleton University according to ASTM D412-83 Method A testing procedures. Tables 4.8 and 4.9 give the tension coupon results of E , σ_p , ϵ_p , σ_u , and ϵ_u for the test cavity and large-scale test specimen coupons, respectively. A typical stress-strain curve is shown in Figure 4.7 for both elastomer specimens. Figure 4.8 (a) illustrates a photograph of the elastomer tension coupon in the Instron Testing Machine.

Table 4.6: Summary of Test Cavity Elastomer Compression Disks

Disk No.	Loading Rate (in/min)	Area (mm ²)	Thickness (mm)	Proportional Limit		Ultimate	
				Stress, σ_p (MPa)	Strain, ϵ_p (mm/mm)	Stress, σ_u (MPa)	Strain, ϵ_u (mm/mm)
TC-1	1.0	653.6	13.0	11.8	0.063	86.2	0.728
TC-2	2.0	652.0	13.1	10.7	0.075	77.7	0.731
TC-3	0.1	647.1	13.0	10.3	0.103	93.3	0.776
TC-4	0.1	649.0	13.0	9.5	0.107	93.1	0.761
TC-5	1.0	651.1	13.2	7.5	0.111	90.2	0.771
Average		650.6	13.1	10.0	0.092	88.1	0.753
Std. Dev.		2.55	0.09	1.61	0.021	6.48	0.02
v		0.004	0.007	0.161	0.233	0.074	0.030

Table 4.7: Summary of Large-Scale Test Elastomer Compression Disks

Disk No.	Loading Rate (in/min)	Area (mm ²)	Thickness (mm)	Proportional Limit		Ultimate	
				Stress, σ_p (MPa)	Strain, ϵ_p (mm/mm)	Stress, σ_u (MPa)	Strain, ϵ_u (mm/mm)
LST-1	1.0	649.5	12.9	7.9	0.900	88.2	0.688
LST-2	0.5	651.3	12.9	8.2	0.100	84.9	0.702
LST-3	0.5	653.4	12.9	3.9	0.127	88.0	0.731
LST-4	0.5	649.8	12.9	2.8	0.095	87.9	0.737
LST-5*	0.1	655.2	12.9	6.7	0.097	81.1	0.732
Average		651.8	12.9	5.9	0.264	86.0	0.718
Std. Dev.		2.43	0.00	2.43	0.356	3.07	0.02
v		0.004	0.000	0.411	1.349	0.036	0.030

* Greased top and bottom loading surfaces

Figure 4.5: Stress-Strain Curves for Test Cavity Elastomer Compression Disks TC-1 to TC-5

Figure 4.6: Stress-Strain Curves for Large-Scale Test Elastomer Compression Disks LST-1 to LST-5

Table 4.8: Summary of Test Cavity Elastomer Tension Coupons

Coupon No.	E (MPa)	Proportional Limit		Ultimate	
		σ_p (MPa)	ϵ_p (mm/mm)	σ_u (MPa)	ϵ_u (mm/mm)
TC-01	410	17.1	0.050	19.0	0.571
TC-02	320	18.7	0.072	21.4	0.831
TC-03	1290 [†]	20.2	0.083	24.8	1.063
TC-04	280	21.2	0.088	24.4	0.938
TC-05	350	20.9	0.067	21.7	0.615
TC-06	350	20.3	0.073	23.5	0.953
TC-07	350	20.2	0.073	19.8	0.492
TC-08	350	20.4	0.070	24.5	0.984
TC-09	450	20.2	0.054	20.8	0.714
TC-10	300	18.1	0.088	22.1	1.032
Average	351	19.7	0.072	22.2	0.819
Std. Dev.	52.31	1.32	0.013	2.05	0.207
v	0.149	0.067	0.178	0.092	0.253

[†] This data was not included in the average, standard deviation and coefficient of variation as it was an outlier.

Table 4.9: Summary of Large-Scale Test Specimen Elastomer Tension Coupons

Coupon No.	E (MPa)	Proportional Limit		Ultimate	
		σ_p (MPa)	ϵ_p (mm/mm)	σ_u (MPa)	ϵ_u (mm/mm)
LST-01	290	21.5	0.095	21.4	0.906
LST-02	360	20.0	0.075	19.5	0.492
LST-03	360	20.8	0.073	24.8	0.969
LST-04	310	19.4	0.067	22.3	0.859
LST-06	360	19.0	0.078	21.4	0.750
LST-07	560 [†]	21.7	0.051	21.7	0.406
LST-08	370	20.0	0.078	26.9	1.047
LST-10	370	21.9	0.060	24.2	0.938
LST-11	370	21.5	0.078	29.9	1.206
Average	349	20.6	0.073	23.6	0.841
Std. Dev.	30.91	1.08	0.013	3.24	0.256
v	0.089	0.052	0.172	0.137	0.305

[†] This data was not included in the average, standard deviation and coefficient of variation as it was an outlier.

* Large-scale test specimen elastomer tension coupons LST-05 and LST-09 had insufficient data

Figure 4.7: Typical Stress-Strain Curves for Elastomer Tension Coupons

(b) Shear-Bond Test Specimen

(a) Tension Coupon Specimen

Figure 4.8: Polyurethane Elastomer Ancillary Tests

4.3 Steel-Polyurethane Elastomer-Steel Shear-Bond Tests

Four shear-bond tests were conducted on two sets of steel-elastomer-steel composite plate sections to quantify the bond capacity of the steel-elastomer interfaces for steel plates with and without mill scale (sandblasted). (Note: mechanical fasteners and chemical adhesives were not used here or for the large-scale test specimen.) The shear-bond test specimens were constructed of two 6 mm steel plates separated by a 50 mm elastomer core. The test specimens were cast by BASF, Inc. and tested at Carleton University in the 1780 kN Tinius Olsen Testing Machine according to the procedure outlined in Section 3.2.2.3. The shear-bond test results are summarized in Table 4.10 and a typical shear load-displacement curve is shown in Figure 4.9 along with a summary of the shear-bond test results. Figure 4.8 (b) illustrates a photograph of the shear-bond test specimen within the apparatus.

There was no apparent difference in the shear-bond capacity for the specimens with and without mill scale.

Table 4.10: Shear-Bond Test Results

Test Specimen	Maximum Load (N)	Contact Shear Area (mm ²)	Average Shear Stress (MPa)
SB-1	17,380	4,430	3.9
SB-2	19,110	4,490	4.3
SB-3	19,010	4,400	4.3
SB-4	19,010	4,590	4.1
Average	18,628	4,478	4.2

Figure 4.9: Typical Shear Load-Displacement Curve for the Shear-Bond Test Specimens

4.4 Large-Scale Test

The large-scale test data given by discrete measurements of displacements and strains throughout the structure was used to characterize the behaviour and subsequently to verify the corresponding numerical simulations, (Ferro, 1998). A complete set of total applied load-displacement curves is presented in Section 4.4.1. A representative sample of strains and the corresponding stresses (where applicable) at key load levels in which different behavioural events occurred are presented in Section 4.4.2. A general description and discussion of the overall behaviour as related to the information provided by these measurements is given in Section 4.4.3, along with the corresponding photographs and finite element analysis von Mises stress distribution plotted on the deformed configuration, (Ferro, 1998).

4.4.1 Deflection Characteristics

The location and orientation for the 35 LVDTs used to measure the displacements of the test specimen are illustrated in Figures 3.18 (a) – (d). Four LVDTs (#1 through #4) were used to monitor the displacements of the actuator heads, as illustrated in Figure 4.10 (a). The actuator numbers are identified in the Figure. Thirteen LVDTs were used to monitor the displacements of the outer plate of the outside hull, of which five LVDTs (#5 through #9) were used to measure the displacements along the longitudinal axis between the actuator heads and two equidistant points outside the outermost load points and two sets of four LVDTs (#10 through #13 and #14 through #17) were used to observe the

(a) Side View of the Undeformed Shape of the Outer Hull Prior to Loading

(b) End View

Figure 4.10: LVDT Identification Numbers and Locations on Outer Plate of Outside Hull

vertical displacements of longitudinal girders #2 and #1, respectively, as partially illustrated in Figure 4.10 (b). Nine LVDTs (#18 through #26) were used to measure the vertical displacements of the inner hull due to the transmission of load through the longitudinal girders and the transverse floor, as illustrated in Figures 4.11 (a) and (b). Two sets of three LVDTs (#27 through #29 and #30 through #32) were used to monitor lateral displacements of longitudinal girders #1 and #2, respectively. The remaining three LVDTs (#33 through #35) were used to measure the lateral displacements of the transverse floor panels.

The total applied load-transverse displacement curves for each of the actuators was given by the measurements of LVDTs #1 through #4 and are illustrated in Figures 4.12 through 4.15, respectively. The locations of the LVDTs on the outer plate of the outside hull are indicated with each of the curves. The LVDTs measured the relative displacement of the centerline position of the knife-edge plate with respect to the horizontal reaction frame. It is implied that this is equivalent to the displacement of the outer hull plate at this location as the load plates were very stiff. Even though the zero-reading was taken at the point of contact between the knife-edge plates and the test specimen, there may have been some small displacements to completely close the gap between the two. This would manifest itself as a “soft” response at the beginning of the load-displacements curves, especially for actuators #2, #3 and #4. The control of the loads at low levels (in the initial stages of loading) was limited by the response time of the pump and the data acquisition system controller. The accuracy of the measurement of loads for the 500 T actuators, including the response time, was estimated to be ± 5 kN for each actuator. The somewhat erratic response of LVDT #4 was related to an electrical

(a) Inner Hull LVDT Identification Numbers

(b) Longitudinal Girder and Transverse Floor LVDT Identification Numbers

Figure 4.11: LVDT Identification Numbers and Locations on Inner Hull Plate, Longitudinal Girders and Transverse Floor

Figure 4.12: Total Applied Load-Transverse Displacement for LVDT #1

Figure 4.13: Total Applied Load-Transverse Displacement for LVDT #2

Figure 4.14: Total Applied Load-Transverse Displacement for LVDT #3

Figure 4.15: Total Applied Load-Transverse Displacement for LVDT #4

contact problem that could not be resolved and was not due to any physical phenomenon within the large-scale test specimen.

LVDT #1 measured the transverse displacements of the actuator that was centered under the transverse floor and its characteristics were dominated by the behaviour of that floor. Points (b), (c), (d), (e), (f), (g), (h), (i) and (l) in Figure 4.12 represent the change in flexural strain distribution in the longitudinal girders and localized yielding of the inner plate of the outside hull/transverse floor web junction; the spread of yield in the transverse floor; the inelastic buckling of the transverse floor; the stable transverse floor web/outer hull plate post-buckled configuration; the outer plate of the outside hull butt weld failure; the corresponding load-displacement point after the repair of the butt weld; the buckling of the longitudinal girders; the development of inelastic membrane action across the full width of the outer hull; and the terminal load point, respectively. Due to the failure of the butt weld, the specimen was unloaded after point (f), a repair was effected, and then reloaded. This is illustrated in all load-displacement curves and has the typical characteristics shown by the unloading/reloading curve after point (f) in Figure 4.12.

The total applied load-transverse displacement curves for actuator LVDTs #2, #3 and #4 illustrate the characteristic behaviour of the overall double hull section due to the transverse loads. Points (a), (j) and (k) in Figures 4.13 through 4.15 indicate load levels associated with the first yield of the outer hull in flexure between the longitudinal girders; rupture of the inner cover plate of the outside hull; and rupture of the outer plate of the outside hull along the north side of the reaction frame, respectively. Points (b), (c), (d), (f), (g), (h) and (i) represent the same characteristic behaviour previously described for

LVDT #1. The actuator load-displacement curves were used as the primary data for verification of the analytical model, (Ferro, 1998).

The total applied load-transverse displacement curves for the intermediate points between the load points and the two equidistant points outside the outermost load points are given by LVDTs #5 through #9, as illustrated in Figure 4.16. The overall characteristic behaviour illustrated by LVDTs #5 through #9 was similar to that of LVDTs #1 through #4 and the data was used in combination with those from the actuators to give the longitudinal displacement profile along the centerline axis.

The remaining transverse displacement measurements on the outer plate of the outside hull were made at the longitudinal girder/transverse floor intersections and along the longitudinal girders at locations aligned with the load points with LVDTs #10 through #17. Due to symmetry, the total applied load-transverse displacement curves for the LVDTs located about the centerline should be characteristically similar and are therefore illustrated in pairs for LVDTs #10 and #14, #11 and #15, #12 and #16, and #13 and #17, as illustrated in Figures 4.17, 4.18, 4.19 and 4.20, respectively. The magnitude of the displacements for any given load point for each pair was different even though the characteristic curves were similar. The larger displacements along longitudinal girder #2 indicate that it was not as stiff or rigid as longitudinal girder #1. This may have been the result of a slight horizontal offset of the entire specimen within the horizontal reaction frame (not more than 1-2 mm), and of different imperfections in the longitudinal girders and the alignment of the top of the girders. There was a change in stiffness for LVDTs #10 and #14 at a total applied load of 3783 kN, as indicated by point (e) in Figure 4.17.

Figure 4.16: Total Applied Load-Transverse Displacement of LVDTs #5, #6, #7, #8 and #9

Figure 4.17: Total Applied Load-Transverse Displacement for LVDTs #10 and #14

Figure 4.18: Total Applied Load-Transverse Displacement for LVDTs #11 and #15

Figure 4.19: Total Applied Load-Transverse Displacement for LVDTs #12 and #16

Figure 4.20: Total Applied Load-Transverse Displacement for LVDTs #13 and #17

This change in stiffness is a result of the buckled transverse floor/outer hull plate displacing and re-establishing a stable post-buckled configuration.

The corresponding transverse displacements for the inside hull plate are illustrated for LVDTs #18 and #22, #19 and #23, #24 and #21 in Figures 4.21, 4.22, 4.23 and 4.24, respectively. The total applied load-transverse displacement curves for LVDTs #20 and #25 are not illustrated, as the electronic response of these LVDTs was unstable and the curves unintelligible. As illustrated by the curves for LVDTs #18 and #22, and #19 and #23, which were essentially identical, it was reasonable to expect that LVDTs #20 and #25 would give similar results for the corresponding LVDTs #21 and #24. This group of curves exhibited the same total applied load-transverse displacement characteristics with linear behaviour up to the load level in which the transverse floor buckled, point (d), followed by a slight decrease in stiffness, and then a decreasing stiffness between points (f) and (h) where the longitudinal girders buckled. The total applied load-transverse displacement responses for LVDTs #19, #23, #24 and #21 exhibited a “zigzag” pattern as the inner hull plate retracted elastically during extended periods (measured in hours) in which load was not being applied. The elastic retraction was simply due to the lateral displacement (creep) of the buckled longitudinal girders. With additional load, the inner hull plate displacements increased and subsequently retracted elastically during the periods of no load application. This effect was not noticeable for LVDTs #18 and #24, which were located at the longitudinal girder/transverse floor intersections, and was most noticeable for the centerline LVDT #24.

Figure 4.25 illustrates the centerline transverse displacements of the inside hull plate as measured by LVDTs #24 and #26. The decrease in displacement for LVDT #26

Figure 4.21: Total Applied Load-Transverse Displacement for LVDTs #18 and #22

Figure 4.22: Total Applied Load-Transverse Displacement for LVDTs #19 and #23

Figure 4.23: Total Applied Load-Transverse Displacement for LVDT #24

Figure 4.24: Total Applied Load-Transverse Displacement for LVDT #21

Figure 4.25: Total Applied Load-Transverse Displacement for LVDTs #24 and #26

reflects the dominance of the imposed rotational displacements at midspan as the longitudinal girders buckled. The displacements at this point decreased from a maximum of 4.2 mm to a minimum of -4.5 mm, as indicated by point (j). The displacements began to increase subsequent to the rupture of the outer plate of the outside hull plate, points (j) through (l).

The total applied load-lateral displacement curves for longitudinal girder #1 and #2 are illustrated by measurements from LVDTs #27, #28 and #29 and from LVDTs #30, #31 and #32, in Figures 4.26 and 4.27, respectively. These curves are characteristic of inelastic web girder buckling problems. The inadequacies of the measurement support system to effectively monitor large displacements without manual adjustments led to discontinuities in the displacements, as represented by the vertical and horizontal “jumps” in the total applied load-lateral displacement curves.

To compare the characteristic behaviour of longitudinal girder #1 and #2, the total applied load-lateral displacement curves for LVDTs #28 and #31, which are located at the same position along the length of the girder, are shown together in Figure 4.28. The total applied load-lateral displacement curves given by LVDTs #28 and #31 clearly show that both girders exhibited essentially identical buckling behaviour, as indicated by point (h), with buckling loads of 7388 kN and 7408 kN, respectively.

The total applied load-lateral displacement curves for the transverse floor panels are illustrated by measurements from LVDTs #33, #34 and #35 in Figure 4.29. The total applied load-lateral displacement at the midpanel locations of the transverse floor outer panels, given by LVDTs #33 and #35, showed little out-of-plane deformation, with a

Figure 4.26: Total Applied Load-Lateral Displacement for LVDTs #27, #28 and #29

Figure 4.27: Total Applied Load-Lateral Displacement for LVDTs #30, #31 and #32

Figure 4.28: Total Applied Load-Lateral Displacement for LVDTs #28 and #31

Figure 4.29: Total Applied Load-Lateral Displacement for LVDTs #33, #34 and #35

maximum displacement of 1.8 mm and 2.6 mm, respectively. The linear response indicated an elastically stable transverse floor web.

The total applied load-lateral displacement response of LVDT #34 located at the quarter height at the centerline of the center (loaded) panel clearly reflected the inelastic buckling load of the transverse floor at point (d) in Figure 4.29; the re-establishment of a post-buckled configuration at point (e); the buckling of the longitudinal girders at point (h); the development of inelastic membrane action across the width of the outer hull at point (i); and the rupture of the outer plate of the outside hull at point (k).

4.4.2 Strain Characteristics

Elastic and inelastic strain measurements for the inner and outer plates of the outside hull and for the shear (outer) panel of the transverse floor were converted to principal stresses and shear stresses (where applicable) with the corresponding orientation using a strain data interpretation program developed by Benichou, (1994). (Note: this transformation was only applicable for strain rosettes that gave sufficient data to determine the principal stresses.) The uniaxial stress-strain curve (modelled as either bilinear or trilinear), the strain measurements for each gage for the duration of the test, the identification of the gages associated with each rosette, and their corresponding orientation were required as input to the program. The program used plasticity algorithms developed for finite element analysis to determine the corresponding elastic and inelastic principal stresses and their orientation. The program was verified by comparing experimental data from tests on biaxially loaded cylinders conducted by Kennedy and

Cheng, (1987) and was used successfully in the interpretation of strain data on the webs of rolled steel shapes subjected to in-plane loads.

The stress-strain data used to model the trilinear curves for the outer and inner plates of the outside hull and for the transverse floor of the large-scale double hull test specimen are given in Table 4.11. The stress-strain curve and corresponding trilinear stress-strain curve for the outer plate of the outside hull are illustrated in Figure 4.30. Obviously, the third segment of the curve is only applicable over a range of strain from ϵ_x to approximately 28000 $\mu\epsilon$.

Table 4.11: Stress-Strain Data Used to Model the Trilinear Curve

Plate Data	Outer Plate Outside Hull	Inner Plate Outside Hull	Transverse Floor
Static Yield, σ_y (MPa)	389.0	392.6	374.2
Young's Modulus, E (MPa)	211,314	210,168	210,906
Poisson's ratio, ν	0.284	0.280	0.282
Slope in Elasto-Plastic Region	0.1	0.1	0.1
Strain Hardening Modulus, E_x (MPa)	4939	5109	5121
Yield Strain, ϵ_y (mm/mm)	0.001849	0.001868	0.001774
Strain Hardening, ϵ_x (mm/mm)	0.020830	0.020882	0.021192
Ultimate Strain, ϵ_u (mm/mm)	0.160840	0.152680	0.166860

Note: Longitudinal material property values were used to model the curve since the transverse values did not differ greatly.

The strains for the outer plate of the outside hull measured at loads of 718 kN, 1020 kN, 1346 kN, 2957 kN, 5718 kN, 5881 kN, 7388 kN and 8201 kN, at the locations in which they were measured (refer to Figure 3.19 (a)), are illustrated in Figures 4.31 (a) through (h), respectively. The plate yield strain is given in the Figures. The load levels cited represent the key behavioural aspects of the outer hull with respect to the overall behaviour of the double hull system and correspond to the points (a), (b), (c), (d), (f), (g),

Figure 4.30: Trilinear Curve Model and Tension Coupon Curve for the Outer Plate of the Outside Hull

(h) and (k) in Figures 4.12 and 4.13. The corresponding principal stresses and orientations (measured counter-clockwise with respect to the longitudinal centerline axis) are illustrated in Figures 4.32 (a) through (h), respectively. The plate yield stress is given in the Figures.

The strains for the inner plate of the outside hull measured at similar load levels, at the locations in which they were measured (refer to Figure 3.19 (b)), are illustrated in Figures 4.33 (a) through (h), respectively. Strain gage #38 was inactive for the duration of the test and is indicated with an '*' in the strain Figures. The plate yield strain is given in the Figures. The corresponding principal stresses and orientations (measured counter-clockwise with respect to the longitudinal centerline axis) are illustrated in Figures 4.34 (a) through (h), respectively. The plate yield stress is given in the Figures. The principal

Figure 4.31 (a): Strains of Outer Plate Outside Hull at Load = 718 kN

Figure 4.31 (b): Strains of Outer Plate Outside Hull at Load = 1020 kN

Figure 4.31 (c): Strains of Outer Plate Outside Hull at Load = 1346 kN

Figure 4.31 (d): Strains of Outer Plate Outside Hull at Load = 2957 kN

Figure 4.31 (e): Strains of Outer Plate Outside Hull at Load = 5718 kN

Figure 4.31 (f): Strains of Outer Plate Outside Hull at Load = 5881 kN

Figure 4.31 (g): Strains of Outer Plate Outside Hull at Load = 7388 kN

Figure 4.31 (h): Strains of Outer Plate Outside Hull at Load = 8201 kN

Figure 4.32 (a): Principal Stresses of Outer Plate Outside Hull at Load = 718 kN

Figure 4.32 (b): Principal Stresses of Outer Plate Outside Hull at Load = 1020 kN

Figure 4.32 (c): Principal Stresses of Outer Plate Outside Hull at Load = 1346 kN

Figure 4.32 (d): Principal Stresses of Outer Plate Outside Hull at Load = 2957 kN

Figure 4.32 (e): Principal Stresses of Outer Plate Outside Hull at Load = 5718 kN

Figure 4.32 (f): Principal Stresses of Outer Plate Outside Hull at Load = 5881 kN

Figure 4.32 (g): Principal Stresses of Outer Plate Outside Hull at Load = 7388 kN

Figure 4.32 (h): Principal Stresses of Outer Plate Outside Hull at Load = 8201 kN

Figure 4.33 (a): Strains of Inner Plate Outside Hull at Load = 718 kN

Figure 4.33 (b): Strains of Inner Plate Outside Hull at Load = 1020 kN

Figure 4.33 (c): Strains of Inner Plate Outside Hull at Load = 1346 kN

Figure 4.33 (d): Strains of Inner Plate Outside Hull at Load = 2957 kN

Figure 4.33 (e): Strains of Inner Plate Outside Hull at Load = 5718 kN

Figure 4.33 (f): Strains of Inner Plate Outside Hull at Load = 5881 kN

Figure 4.33 (g): Strains of Inner Plate Outside Hull at Load = 7388 kN

Figure 4.33 (h): Strains of Inner Plate Outside Hull at Load = 8201 kN

Figure 4.34 (a): Principal Stresses of Inner Plate Outside Hull at Load = 718 kN

Figure 4.34 (b): Principal Stresses of Inner Plate Outside Hull at Load = 1020 kN

Figure 4.34 (c): Principal Stresses of Inner Plate Outside Hull at Load = 1346 kN

Figure 4.34 (d): Principal Stresses of Inner Plate Outside Hull at Load = 2957 kN

Figure 4.34 (g): Principal Stresses of Inner Plate Outside Hull at Load = 7388 kN

Figure 4.34 (h): Principal Stresses of Inner Plate Outside Hull at Load = 8201 kN

stresses at the strain gage rosette for gages #36, #37 and #38 could not be determined since strain gage #38 was inactive.

The strains for the inside and outside surface of the inner hull plate at loads of 718 kN, 2957 kN, 7388 kN, and 8201 kN, at the locations in which they were measured (refer to Figure 3.19 (c)), are illustrated in Figures 4.35 (a) through (d), respectively. The values within the brackets indicate the strains measured for the outside surface of the inner hull plate. The plate yield strain is given in the Figures. As before, the load levels cited relate to the key behavioural aspects of the inner hull with respect to the overall behaviour of the composite double hull system and correspond to points (a), (d), (h) and (k).

The flexural strain distributions for the north and south side surfaces of longitudinal girders #1 and #2 at loads of 718 kN, 1020 kN, 1346 kN, 2957 kN, 5718 kN, 5881 kN, 7388 kN and 8201 kN, at the locations in which they were measured (refer to Figure 3.19 (d)), are illustrated in Figures 4.36 (a) and (b), and Figures 4.37 (a) and (b), respectively. The plate yield strain is given in the Figures. The appropriate scale is indicated on each Figure and the strain distributions between the measured points were assumed to be linear since the span-to-depth ratio was 7.1 and the girders behaved as flexural members. As before, the load levels cited relate to the key behavioural aspects of the longitudinal girders with respect to the overall behaviour of the composite double hull system and correspond to points (a), (b), (c), (d), (f), (g), (h) and (k).

The longitudinal (flexural) strain distributions for the east and west side surfaces of the transverse floor at similar load levels, at the locations in which they were measured (refer to Figure 3.19 (e)), are illustrated in Figures 4.38 (a) and (b), respectively. The corresponding transverse strain distributions for the east and west side surfaces of the

Figure 4.35 (a): Strains of the Inner Hull at Load = 718 kN

Figure 4.35 (b): Strains of the Inner Hull at Load = 2957 kN

Figure 4.35 (c): Strains of the Inner Hull at Load = 7388 kN

Figure 4.35 (d): Strains of the Inner Hull at Load = 8201 kN

Figure 4.36 (a): Flexural Strain Distribution of Longitudinal Girder #1 - North Side

Figure 4.36 (b): Flexural Strain Distribution of Longitudinal Girder #1 - South Side

Figure 4.37 (a): Flexural Strain Distribution of Longitudinal Girder #2 - North Side

Figure 4.37 (b): Flexural Strain Distribution of Longitudinal Girder #2 - South Side

Figure 4.38 (a): Longitudinal (Flexural) Strain Distribution in Transverse Floor – East Side

Figure 4.38 (b): Longitudinal (Flexural) Strain Distribution in Transverse Floor – West Side

transverse floor, at the locations in which they were measured (refer to Figure 3.19 (e)), are illustrated in Figures 4.39 (a) and (b), respectively. The plate yield strain is given in the Figures. The appropriate scale is indicated on each Figure and the distribution between the measured strains was assumed to be linear.

The average shear stress, as determined for a strain gage rosette located on both sides of the transverse floor at the center of the outer web panel, is illustrated in Figure 4.40 as a function of the total applied load. An insert within the Figure shows a typical elastic shear distribution along section a-a of the transverse floor and is based on an assumed effective hull width of 500 mm. For these calculations, the elastomer was transformed to an equivalent steel section and assumed to be fully bonded to the hull plates. Since the transverse floor and inner hull behaved elastically throughout the entire loading history, the shear stress distribution illustrated is a reasonable approximation.

The changes in slope of the shear stress curve clearly illustrate a number of the key behaviour characteristics of the double hull system and are labeled appropriately. The curve gives a measure of the amount of load being carried by the transverse floor webs when plotted as a function of the total applied load of a statically indeterminate system. A smaller positive slope (point 0 to point (a)) indicates that the element (transverse floor) is attracting a larger fraction of the total applied load. Steeper curves (point (a) to point (d)) are the converse and the negative slope between points (h) and (i) indicates a loss of load to the outer hull through inelastic membrane action.

Figure 4.39 (a): Transverse Strain Distribution in Transverse Floor -- East Side

Figure 4.39 (b): Transverse Strain Distribution in Transverse Floor – West Side

Figure 4.40: Total Applied Load-Average Shear Stress of Transverse Floor Outer Panel

4.4.3 General Behaviour

The general behaviour of the large-scale test specimen is characterized by the total applied load-transverse centerline displacement curve for actuator #3, as shown in Figure 4.41. Key behavioural events are indicated on the total applied load-transverse displacement curve and are briefly described in Table 4.12, with further details to follow. For clarity of the description of the general behaviour, a plan view of the large-scale test specimen prior to and subsequent to longitudinal girder buckling is illustrated in Figures 4.42 (a) and (b), respectively. A cross-sectional view and side view of the test specimen are illustrated in Figure 4.42 (c). The Figures include terms, load transfer directions, locations of failures, the transverse floor panels, longitudinal girders, the locations of the load points numbered according to the actuator numbers, and the horizontal reaction frame. The arrows on the transverse floor and longitudinal girders indicate the primary transfer of load in flexure from these elements to the horizontal reaction frame. Also illustrated is the transfer of load from the load points in membrane action of the outer hull plates to the longitudinal girders and, subsequent to their buckling, to the horizontal reaction frame. When membrane action was fully developed, the primary transfer of load from the test specimen to the horizontal reaction frame was across the loaded width as shown. Points (d) and (h) in Figure 4.42 (c) illustrate the point of maximum out-of-plane displacement for the buckled transverse floor and longitudinal girders, respectively. Also shown is the location and length of the butt weld failure of the outer plate of the outside hull, point (f) in Figure 4.42 (a), the subsequent lap-plate repair, the failure of the inner

Figure 4.41: Total Applied Load-Transverse Displacement for Centerline LVDT

Table 4.12: Key Behavioural Events of the UDHSPS During Transverse Loading

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

Figure 4.42 (a): General Behaviour Terms, Load Transfer Directions and Failure Locations of Large-Scale Test Specimen Prior to Buckling of Longitudinal Girders

Figure 4.42 (b): Subsequent to Buckling of the Longitudinal Girders

Figure 4.42 (c): General Behaviour of Transverse Floor and Longitudinal Girders

cover plate and the shear-tension failure of the outer plate of the outside hull, point (k) in Figure 4.42 (b).

The initial portion of the total applied load-transverse displacement curve in Figure 4.41 contains a “soft” portion, a result of closing the gap between the load plate and the test specimen. The erratic shape was due to control of the applied load at low load levels relative to the actuator capacities. This may have been further accentuated since very few readings were taken during the initial stages of loading.

The first behavioural event occurred at or near a total applied load of 718 kN, in which a change in stiffness in the curve indicated first yield of the inner plate of the

outside hull directly above the centerline actuator #3 due to flexure, as illustrated by Figures 4.33 (a) and 4.42 (a).

The second behavioural event occurred at total applied load of 1020 kN, where strain gages #69 and #70, located at the base along the centerline of longitudinal girder #1, and strain gages #81 and #82, at the same location on longitudinal girder #2, had reached maximum average tensile strains of approximately 93 μs and 125 μs , respectively. With increasing total applied load, these values diminished.

The corresponding average flexural strain distribution through the depth of longitudinal girder #2 at this total applied load level (1020 kN) and other key behavioural load levels (1346 kN, 2957 kN, 5718 kN and 7388 kN) is illustrated in Figure 4.43. Based on the measured strains of the inner hull plate and the web of the longitudinal girder, it was reasonable to apply the assumption of plane sections remaining plane for this portion of the welded structure in flexure. The discontinuous curve shown in the Figure resulted from the low shear modulus of the elastomer core. Based on preliminary computations, the shear flow required for flexural behaviour at this load level was less than the maximum value obtained from the shear-bond test.

Figure 4.43: Average Flexural Strain Distributions for Longitudinal Girder #2

The increase in the stiffness of the total applied load-transverse displacement curve after point (b) in Figure 4.41 resulted from the development of elastic and inelastic membrane action of the outer and inner hull plates of the outside hull, as illustrated by the strains in Figures 4.31 (b) and (c) and Figures 4.33 (b) and (c), respectively, and in Figure 4.42 (a).

The slight decrease in the total applied load-transverse displacement curve for LVDT #1, as illustrated in Figure 4.12, indicated a change in stiffness in the local structure above actuator #1. This effect manifested itself as a change in slope in the total applied load-strain curves for strain gages #83 to #86 and #94, all of which were located on the transverse floor above this location. It was hypothesized, based on the strain magnitudes for strain gages #91, #92, #93 and #94 at this load level, that localized yielding was occurring in the inner plate of the outside hull and in the transverse floor web at the north and south edges of the load plate, as illustrated in Figures 4.42 (a) and (c).

Yielding in the transverse floor web spread from this region and, at a total applied load of 1346 kN, yielded the central region around strain gage #93 and #94 for the third behavioural event, point (c), as illustrated in Figure 4.42 (c). The total applied load-strain curves for these gages indicated a rapid increase in strain from values of $-1320 \mu\text{s}$ to $-36650 \mu\text{s}$ (at a total applied load of 3097 kN) and from values of $1342 \mu\text{s}$ to $18645 \mu\text{s}$ (at a total applied load of 3097 kN) to $22954 \mu\text{s}$ (at a total applied load of 3322 kN), respectively, at buckling of the transverse floor at a total applied load of 2957 kN, as indicated by point (d) in Figure 4.41.

There was a further decrease in stiffness for LVDT #1 beyond point (c) in Figure 4.12. The inelastic membrane action of the outer and inner plates of the outside hull continued to spread outward from the load points to the longitudinal girders, as illustrated by the strains in Figures 4.31 (c) and (d) and Figures 4.33 (c) and (d), respectively, and in Figure 4.42 (a). The longitudinal girders effectively contained the inelastic membrane action to the loaded hull plates, and outside this region, the plates were still elastic. There was an apparent increase in the stiffness of the outer hull plates as illustrated by the load-displacement curves for LVDTs #2, #3 and #4 in Figures 4.13, 4.14 and 4.15, respectively. This may have been a result of strain hardening somewhere within the outer hull plates (most likely over the centerline actuator).

Up to the a total applied load level of 1346 kN, the flexural strain distributions for longitudinal girders #1 and #2 had similar shapes for both sides for all load levels, as illustrated by Figures 4.36 (a) and (b) and Figures 4.37 (a) and (b), respectively, and Figure 4.43. Because of the yielding of the transverse floor at this load level, the longitudinal girders attracted more of the load and transmitted it over their full length to the horizontal reaction frame. This effect was manifested as a significant change in the slope of the flexural strain distributions through the depth of the longitudinal girders between the total applied load of 1346 kN and 2957 kN, as illustrated in the previously cited Figures.

Inelastic buckling of the transverse floor occurred at a total applied load of approximately 2957 kN, point (d) in Figures 4.41 and 4.42 (c), and is clearly illustrated by the change in slope of the of the total applied load-lateral displacement curve for LVDT #34 at point (d), illustrated in Figure 4.29. The exact load level was difficult to

determine since the system was statically indeterminate and there were only a limited number of discrete measurements to illustrate the overall behaviour. However, from the total applied load-strain curves for strain gages #91, #93 and #94, it was clear that the transverse floor web had yielded and strain hardened over a substantial area prior to becoming unstable. The latter was manifested as an abrupt change in the strain rates from positive to negative and in the rate of change of the average shear stress of the transverse floor outer panel, as shown in Figure 4.40. The original undeformed and the buckled shape of the transverse floor are illustrated in Figures 4.44 (a) and (b), respectively.

From a total applied load of 2957 kN to 3783 kN, point (e) in Figure 4.41, the buckled transverse floor/outer hull plate configuration displaced and re-established a stable post-buckled configuration. At this point, the rate of increase in the average shear stress of the transverse floor outer panel increased and was linear up to a total applied load of 5718 kN, point (f) in Figure 4.40. This behaviour was consistent with the increased rate of transverse displacement at the longitudinal girder/transverse floor junction given by the total applied load-transverse displacement curves for LVDTs #10 and #14 in Figure 4.17. (Note: the longitudinal and transverse strains given for the central panel of the transverse floor indicated that this section had yielded to at least mid-height at a total applied load of 5718 kN, as illustrated in Figures 4.38 (a) and (b) and Figures 4.39 (a) and (b), respectively, and in Figure 4.42 (c).)

The strains in the inner and outer plates of the outside hull indicated that the inelastic membrane action had become more uniformly distributed over the loaded area and was still contained within the longitudinal girders, as illustrated in Figure 4.42 (a).

Figure 4.44: Undeformed and Buckled Shape of Transverse Floor

Significant portions of these plates had strain hardened. The outer hull plates outside the longitudinal girders continued to transfer load to the reaction frame along the full length of the loaded area in elastic membrane action. The vertical reaction force from the membrane action from the outside hull was transmitted through the thickness of the elastomer causing enhanced bond and deformation of the elastomer which increased its stiffness and strength along the length of the longitudinal girders. As a result, the contribution of the elastomer in carrying the flexural loads in the longitudinal direction along with the longitudinal girders increased. This is reflected in the flexural strain distributions in Figures 4.36, 4.37 and 4.43, which illustrates the migration of the neutral axis toward the centroid of the longitudinal girder web as the total applied load was increased.

At a total applied load of 5718 kN, point (f) in Figure 4.41, the longitudinal butt weld of the outer plate of the outside hull ruptured, as illustrated in Figures 4.45 (a) and (b). The fracture was 1400 mm long, 15 mm wide and was centered 220 mm from the centerline axis toward the transverse floor, as illustrated in Figure 4.42 (a). This weld failure was not anticipated, as the weld should have had greater strength than the surrounding plate. Since the outer panels of the outer plate outside hull were still elastic, failure was attributed to imperfections or a poor weld. The applied transverse load decreased with increasing displacement of the actuator heads, as illustrated by the total applied load-transverse displacement curves for LVDTs #1 through #4 in Figures 4.12 through 4.15, respectively. Measurements were taken and the applied load was reduced to zero. The ruptured weld was closed and repaired by welding a lap-plate overtop. The lap-plate was 25.4 mm thick, 200 mm wide, and ran the full length of the test specimen, with

(a) Fractured Butt Weld

(b) Close-up

(c) Lap-Plate Repair

Figure 4.45: Fracture and Repair of Outer Plate Outside Hull Butt Weld

a 22 mm fillet weld down both sides. The repair of the butt weld is illustrated in Figures 4.42 (a) and 4.45 (c).

Once repaired, the load was reapplied and the total applied load-displacement curves followed the unloading portion of the curves back to a total applied load of 5881 kN, point (g) in Figure 4.41. The effect of the premature butt weld failure was an increase in the transverse displacements of the outer hull plates and in the lateral displacements of the longitudinal girders. All other displacements and strains were relatively unaffected. Beyond the load of 5881 kN and prior to the buckling of the longitudinal girders at a load of 7388 kN, point (h) in Figure 4.41, the slopes of the total applied load-transverse displacement curves of the outer plate of the outside hull indicated a slightly stiffer response, while the slopes of the total applied load-lateral displacement curves of the longitudinal girders indicated a slightly less stiff response.

At a total applied load of 7327 kN, a longitudinal crack was first noticed in the center (loaded) panel of the transverse floor along the floor/inner plate of the outside hull fillet weld in a HAZ and was approximately 75-100 mm in length. The crack is illustrated in Figure 4.44 (b).

The next behavioural event was a result of buckling of longitudinal girders #1 and #2 at a total applied load of 7388 kN, point (h) in Figures 4.41 and 4.42 (c). This was clearly indicated by a general softening and change in stiffness in all the total applied load-transverse displacement curves and by characteristic total applied load-lateral displacement curves for the longitudinal girders, given in Figures 4.26 and 4.27, where the displacements increased without any increase in the total applied load. The general buckling curves for both longitudinal girders were essentially identical and are illustrated

in Figure 4.28. This was anticipated since the geometry was symmetric about the longitudinal centerline. Apparently, the rupture of the outer plate of the outside hull and subsequent repair with the lap-plate did not adversely affected the behaviour of the longitudinal girders. The original undeformed and the buckled shape of one of the longitudinal girders is illustrated in Figures 4.46 (a) and (b), respectively.

At this total applied load level, the shear stress in the transverse floor outer panels dropped sharply from 83 MPa to 71 MPa, as illustrated by Figure 4.40. The shear stress was still elastic. This drop reflected a diminishing flexural contribution and an increased contribution of the outer panels of the outer hull plates in membrane action as the total applied load did not decrease.

Flexural load and transverse vertical load (vertical component of the membrane force at the outer hull/longitudinal girder junction) were still being carried by the longitudinal girders into the inner hull plate, since the displacements of the inner hull had only diminished slightly, as illustrated in Figures 4.21 through 4.25. Based on the relaxation of the transverse displacements and strains of the inner hull plate, it was estimated that there was a 20 to 30 percent decrease in the transverse force from the longitudinal girders to the inner hull.

Subsequent to buckling of the longitudinal girders, the load-carrying capacity of the double hull structure began to increase at a total applied load of 7485 kN, point (i) in Figure 4.41. This was attributed to the development of inelastic membrane action across the full loaded width of the outer hull plates on the stable deformed configuration since the longitudinal girders no longer confined the inelastic membrane action to the central

(a) Undeformed Shape

(b) Buckled Shape

Figure 4.46: Undeformed and Buckled Shape Longitudinal Girder

outer hull plate loaded panel, as illustrated in Figure 4.42 (b). This was coupled with an increased transverse displacement centrally and along the longitudinal girders.

Note, as the hull plate/longitudinal girder junction was displaced vertically due to the applied loads, the lateral displacements of the longitudinal girders was amplified. Due to the severe web distortions, the lateral LVDTs slid off the vertical mark along the deformed girder plate and were measuring lateral displacements of some other (lower) portion of the girder web. To correct the problem, the lateral LVDTs had several manual corrections made during post-buckling of the girders. These manifested themselves as “jumps” in the lateral displacement curves, as illustrated for LVDTs #27 through #32 in Figures 4.26, 4.27 and 4.28. As such, the lateral displacements after buckling are of little or no value due to the inherent inaccuracies that resulted from the measurement system used. The vertical positions of the LVDTs were fixed relative to the reaction frame and for both the large vertical and lateral displacements of a distorted web, the vertical position of the LVDTs would have to be changed at each load step to correctly follow the web.

The cover plate that attached the inner plate of the outside hull to the reaction frame buckled upward on the north side at a total applied load of 8003 kN, point (j) in Figure 4.41, and as illustrated in Figure 4.47 (a). The separation began at the transverse floor and spread away a distance of approximately 1650 mm, as illustrated in Figure 4.42 (b). This was followed shortly by two shear-tension ruptures of the outer plate of the outside hull along the edge of the composite plate on the north side at the centerline (approximately 100 mm in from the reaction frame and each 300 mm long, as shown in Figure 4.42 (b)), at a total applied load of 8201 kN, point (k) in Figure 4.41, and as

(a) Cover Plate Failure of Inner Plate Inside Hull

(b) Shear Tension Failure of Outer Plate Outside Hull

Figure 4.47: Failure of Outer Hull

illustrated in Figures 4.47 (b) and (c). Failure along the north side may have been premature and as a result of the earlier butt weld rupture at a load of 5718 kN. When the butt weld ruptured, the membrane force in the outer plate of the outside hull was relieved and portions of that load were transferred to the inner plate of the outside hull. This may have caused the full penetration groove weld joining the cover plate to the reaction frame to crack at its root, going unnoticed. With the buckling of the longitudinal girders and the corresponding shift in the load carrying mechanism to the outer panels through membrane action, the cracked weld gradually ruptured and the separated cover plate displaced upward as the edge rotated. With the loss of the inner plate of the outside hull, the outer plate was subjected to a sudden increase in load and failed in shear-tension at the edge of the composite plate.

Based on subsequent numerical simulations performed by Ferro, (1998) this failure mode could have occurred at a total applied load of 9762 kN and the general overall behaviour would not have changed. In a continuous structure, this failure may not have occurred, as the longitudinal girders, represented by the reaction frame, would have buckled and the inelastic membrane action spread to the adjacent panels.

Recognizing that further load application through actuators #2, #3 and #4 would only open up the rupture, additional load was applied to the transverse floor actuator only, up to an approximate load of 3000 kN, in an attempt to promote local crack propagation and localized shear-tension failure around the load points. The corresponding total applied load was approximately 8466 kN. However, since no additional information was being obtained, the test was stopped.

A close-up of the deformed outer hull plate around one of the load points is shown in Figure 4.48. The plate deformations were smooth in and around the edges and corners of the load plate, indicating that the strains were dissipated over a larger area. The composite structural laminate precluded localized concentrations of strains, which would be the source of shear-tension failures and ruptures of the outer hull plate.

Figure 4.48: Deformed Hull Plate at Load Point

The final deformed shape along the length and transversely through the centerline of the test specimen is illustrated in Figures 4.49 (a) and (b) and Figures 4.50 (a) and (b), respectively. The corresponding deformed shape from the finite element analysis at this load and the von Mises stresses are illustrated in Figure 4.51, (Ferro, 1998).

(a) Deformed Configuration Based on LVDT Measurements

(b) Corresponding Photo of Deformed Configuration

Figure 4.49: Final Deformed Shape of Large-Scale Test Specimen – Side View

(a) Deformed Configuration Based on LVDT Measurements

(b) Corresponding Photo of Deformed Configuration

Figure 4.50: Final Deformed Shape of Large-Scale Test Specimen – End View

Figure 4.51: Large-Scale Test Specimen von Mises Stresses at Load = 8197 kN (after Ferro, 1998)

Based on the large-scale test evidence, the design configuration of the test hull yielded the maximum possible strain energy absorption prior to failure. Key behavioural characteristics that achieved this were the development of inelastic membrane action of the loaded hull plate, followed by buckling of the longitudinal girders and the progressive development of membrane action away from the loaded area. Inherent in the incorporation of the composite structural laminate was its resistance to localized shear-tension failures and its ability to dissipate strains at hard-spots which would be the source of premature rupture and preclude the full development of membrane action. Based on numerical analysis and test observations, inelastic behaviour of the inner hull can only be achieved when contact is made with it by the load source.

Chapter 5

Summary, Conclusions, Recommendations and Future Work

5.1 Summary and Conclusions

The *Oil Pollution Act of 1990* (OPA 90) was introduced by the American Congress to reduce the probability of oil outflow from tankers subject to accidental or extreme load events by legislating the use of double hull construction for all new oil tankers entering U.S. waters. OPA 90 provides a phase-in schedule for existing single hull tankers to be replaced as retired. The Canadian Coast Guard also implemented similar regulations through the standard, *Standards for Double Hull Construction of Oil Tankers (TP 11710)*, for tankers operating in Canadian waters.

Double hull tanker technology for oil tankers is in its infancy and a number of design configurations have been introduced, each of which has its own merits and deficiencies. These designs include: the conventional double hull (CDH), the advanced double hull (ADH), and variations of the ADH, such as the MarC Guardian curved plate concept.

The CDH design is characterized by its construction that consists of a complex series of longitudinally and transversely framed elements that separate the inner and outer hulls. As a result of its complexity, this structure is much more difficult to design and

analyze, to inspect and to maintain. There are a large number of structural intersections that lead to fatigue problems, crack initiation and hard points that create localized stress and strain concentrations that could possibly lead to premature rupture. Stiff transverse floors are a source of rupture and provide a path for crack propagation to the inner hull in grounding events, thereby breaching the oil tight barrier.

The ADH design offers a simplified structural layout providing unidirectional framing and unstiffened hull plating. This design is easier to analyze and design since there is a reduction in the number of elements, which lends itself to automated welding and to modular construction, thereby reducing fabrication costs. Eliminating the transverse flooring elements reduces the number of structural intersections which, in turn, lessens the likelihood of potential fatigue cracks at these fatigue prone details, and it also reduces the number of hard points which are a source of premature hull plate rupture. The simpler construction reduces surface area and allows for better painting and coating application. This same simplicity makes inspection and maintenance less costly.

Cited disadvantages of the ADH design alternative are: (a) the extent of longitudinal damage is greater for comparative grounding events; (b) the tight spacing of the horizontal stringers and longitudinal girders yields short, unsupported spans of the hull plate panel, which can lead to early rupture and reduced global energy absorption capacity for collisions and grounding events; and (c) the ADH designs may not be practical for tankers larger than 40,000 to 50,000 DWT as the longitudinal girder spacing becomes too small.

The MarC Guardian is an ADH variation that incorporates unstiffened curved hull plates to increase the buckling capacity of the hull plate over equivalent flat plates for in-

plane loads due to hogging and sagging moments. This results in larger spacing between girders, which reduces the number of subdivisions. Hence, it increases accessibility, makes inspection and maintenance easier, and allows for better coating application. Cited disadvantages include: (a) possible adverse effects on propulsion efficiency; (b) reduced hull strength for dry docking; and (c) reduced energy absorption capacity for groundings and collisions relative to other designs.

Even with the advances in navigational technology, groundings and collisions will continue to occur and the development of double hull technology must be advanced to minimize or, if possible, eliminate the possibility of oil outflow. It would be simple to modify existing double hull technology to incorporate wider double sides, deeper double bottoms, smaller cargo tanks, increased scantlings or structural members, and improved mechanical properties of materials that would enhance the crashworthiness and minimize oil outflow. However, such design alternatives would not necessarily solve the problem, nor would they be economical and, therefore, would not necessarily be realistic. There exists a need for the development of an innovative, practical, and cost-effective design to minimize or mitigate oil outflow in the case of accidental or extreme load events.

The Unidirectional Double Hull Sandwich Plate System (UDHSPS) integrates a composite structural laminate into the hulls and builds on the advantages of the ADH design concept, which addresses the major problems associated with the all-steel double hull systems. The proposed laminate consists of a steel-polyurethane elastomer-steel sandwich. The UDHSPS design eliminates the longitudinal stiffeners of the inner and outer hull plates, providing a simple, less complex structural system composed of flat surfaces. The structural system improves hull integrity by maximizing the energy

absorption potential of the plating during an accidental or extreme loading event, while offering equivalent in-plane loading capacity to a stiffened panel during normal operating conditions and sufficient residual strength for rescue operations. The composite structural laminate gives equivalent in-plane load capacity without decreasing the longitudinal girder spacing, as required for the ADH. The design reduces the number of structural intersections characteristic to the CDH and ADH designs, thereby decreasing the number of fatigue-prone details. The steel previously used for the stiffening elements of the outer and inner hull plates has been reallocated to the hull plating, thereby increasing the energy absorption characteristics of the plating. Hard points at longitudinal girder/transverse floor and hull plate/stiffener intersections are diminished due to the core material which dissipates the localized strains over a larger field. The structural plastic core material, with appropriate construction and fatigue detailing, provides an effective crack arrest barrier for crack propagation into the inner hull. The flat surfaces of this system lend themselves to better coating application and less corrosion-related problems. The elimination of stiffeners and the reduction of structural intersections give increased accessibility for improved inspection and maintenance.

Based on a comparative study of a 40,000 DWT CDH product oil tanker and a similarly sized UDHSPS tanker, an 11.6 percent and 12.7 percent cost and overall weight savings per cargo tank were noticed with the UDHSPS design, respectively. By utilizing identical double hull modules, the design will lend itself to automated construction and welding of the components. There will be a decrease in coating costs since there is initially less steel and the steel-elastomer interface does not require treatment or coatings. Several other advantages of using a composite structural laminate may include insulation

to the cargo tanks, which would reduce the in-transit heating costs required to maintain oil temperatures, and sound and vibration damping due to machinery.

From the above discussion, it is apparent that the UDHSPS design concept offers many structural advantages as well as cost-saving advantages, making it a viable alternative to existing technology. The next step was to verify the claims of the technology through experimental and analytical studies on an equivalent UDHSPS design for a typical baseline 40,000 DWT product oil tanker structure designed according to either ABS Rules and Regulations or Lloyd's Register.

An equivalent UDHSPS large-scale double hull test section measuring 2600 mm by 5000 mm in plan was modelled after the keel of a CDH baseline structure and incorporated two 700 mm deep longitudinal girders, a transverse floor, a composite structural laminate outer hull, and a single steel plate inner hull. The test geometry was selected to maximize the energy absorption through the development of inelastic membrane action of the outer hull between the longitudinal girders, the promotion of inelastic distortion of the transverse floor and longitudinal girder webs, and the subsequent development of inelastic membrane action of the outer hull panels prior to rupture of the adjacent outer hull when subjected to transverse loads. The test provided data and observations related to behaviour that confirmed the design philosophy. (Note: due to the nature of the quasi-static transverse loading, the crack arrest properties of the elastomer core could not be demonstrated.)

In addition, the test also confirmed that the elastomer had sufficient bond capacity and provided sufficient ductility around hard points (load points, transverse floor/longitudinal girder/outer hull and longitudinal girder/outer hull junctions) to dissipate the strain fields such that shear-tension failure and rupture of the outer hull plate did not occur. The longitudinal centerline spacer butt weld detail was sufficiently ductile not to rupture. As part of the large-scale test, the casting of the elastomer confirmed that existing technology for reaction injection molding (RIM) could be used successfully for this type of horizontal plate construction.

5.2 Recommendations and Future Work

For this technology to be readily accepted by shipowners and shipbuilders, it must be demonstrated to the appropriate Classification Societies that the UDHSPS is structurally equivalent through experimental testing and numerical simulations. Subsequently, design equations, a design philosophy and methods for evaluating the energy absorption capacity and residual post-accidental or post-extreme load event capacity will be developed. All new forms of ship construction must also meet the requirements for fire regulations. Finally, shipbuilders will address the remaining production engineering issues of integrating the design into current shipyard practice, repair and maintenance issues to determine the economic feasibility of this technology.

To develop the design equations which characterize the behaviour of the composite structural laminate, a series of tests have been designed and are currently being

conducted by Braun, (1999). The tests consists of in-plane compression tests, flexural tests, corresponding fatigue tests and shear-bond tests. The test, general description and objectives are listed in Table 5.1. The structural characteristics of the elastomer are based on strength specifications developed by Ferro, (1998) to give equivalent in-plane capacity for typical stiffened plate panels of a CDH design.

Table 5.1: Composite Structural Laminate Tests to Characterize Behaviour

Test Type	General Description and Objectives
Column Buckling	<ul style="list-style-type: none"> • A schematic diagram of a typical column buckling test is illustrated in Figure 5.1. • A series of tests with varying slenderness ratios will be conducted to determine the in-plane buckling capacity for a range of behaviours ranging from yield to inelastic buckling to elastic buckling. • A series of exploratory fatigue tests will be conducted to establish the degradation of bond and fatigue life of the elastomer core and the steel sandwich plating under stress ranges anticipated under normal operating loads.
Flexural Beam	<ul style="list-style-type: none"> • A schematic diagram of a typical flexural beam test is illustrated in Figure 5.2. • A series of statically loaded beams will be tested to establish the flexural capacity of the composite structural laminate. • A series of exploratory fatigue tests will be conducted to establish the degradation of bond and fatigue life of the elastomer core and the steel sandwich plating under stress ranges anticipated under transverse loads due to varying hydrostatic pressures.
Shear-Bond	<ul style="list-style-type: none"> • A schematic diagram of a typical shear-bond test is illustrated in Figure 3.22. • A series of shear-bond tests will be tested to establish the bond capacity of the elastomer and includes the possible degradation of bond due to welding effects.

The UDHSPS design must be tested to determine whether or not the composite structural laminate meets the requirements of the *International Convention for the Safety of Life at Sea (SOLAS)*, 1974, and subsequent amendments, and the *Torremolinos International Convention for the Safety of Fishing Vessels*, 1977. The objectives of the

Figure 5.1: Composite Structural Laminate Column Buckling Test Specimen

Figure 5.2: Composite Structural Laminate Flexural Beam Test Specimen

SOLAS requirements are to restrict new materials from becoming a fire source and to limit the levels of smoke and toxic combustion products so that the crew can continue to operate the vessel, fight the fire, and abandon ship, if necessary. The requirements stipulate the fire performance criteria, fire load curves, fire test specimen dimensions and the measurements that must be made to establish its acceptability. A fire test specimen, as illustrated in Figure 5.3, was fabricated and will be tested by Braun, (1999) in accordance with the SOLAS requirements at the National Research Council (NRC) Fire Laboratory in Ottawa. As part of this test, any off-gases emitted by the polyurethane elastomer will be analyzed to determine the level of toxicity. Subsequent to the fire test, sections will be cut from the specimen to observe the degradation, if any, of the composite structural laminate and to determine the residual strength of the section.

Figure 5.3: Composite Structural Laminate Fire Test Specimen

Production engineering issues that need to be addressed by the shipbuilder are: (a) handling and welding (weld distortions) of large, thin steel plates; (b) repair and maintenance related to fatigue and corrosion problems; (c) welding of subsequent steel plating to the composite structural laminate plating; and (d) quality control of the polyurethane elastomer casting for full-scale double hull tanker modules in a shipyard environment. Specifically, the latter must address the effects of temperature and humidity on the polyurethane elastomer quality; casting large quantities of elastomer for a series of elements of varying orientation with the time constraints associated with modular construction (gel and set times for the elastomer); and quality assurance of the elastomer (constant elastomer thickness, full cavities without overflow or spillage, and consistent elastomer properties).

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Appendix A

Detailed Cost Algorithm for Double Hull Tanker Structures

A. Detailed Cost Algorithm for Double Hull Tanker Structures

A.1 Introduction

A detailed cost analysis was performed on a typical \$40 million US 7-tank all-steel 40,000 DWT CDH tanker and on the proposed UDHSPS tanker concept as part of a feasibility study. The study assessed the proportion of costs associated with the hull structure, propulsion system, machinery, paint and tank coatings, hull outfit and electrical components of the CDH tanker. The hull structure and the cost of painting and tank coatings were separated from the total, since either vessel shared these costs. The analysis then estimated the cost of raw material handling and preparation; plate cutting and forming; fabrication and erection unit joining; and painting and coating application. A quantity take-off was completed for a typical cargo tank of the all-steel CDH for steel weight and surface area, number of pieces, weld lengths and volumes, and the number of structural intersections with the total cost per cargo tank calculated and compared against the industry standard cost. Once the algorithm was calibrated, it was then used to determine the cost per cargo tank of the UDHSPS design with new cost values for the raw elastomer material and for placement of the material. The following sections outline the cost algorithms and provide a summary of the cost for the CDH and UDHSPS designs. Detailed quantity take-offs for a typical cargo tank of the CDH and UDHSPS are given in Appendices B and C, respectively.

A.2 Cost of Double Hull Tanker Structure and Coatings

The approximate cost of a North American built 40,000 DWT double hull tanker is approximately \$40 million US (New Ships, Feb. 1996). The total cost is subdivided into several major groupings: hull structure, propulsion system, machinery, paint and tank coatings, hull outfit, and electrical components and is listed in Table A.1.

Table A.1: Major Groupings of a Double Hull Tanker

Major Groupings	Percentage of Acquisition Cost	Cost (\$ M US)	Source
Hull Structure (bow, stern and tanks)	45	18	MARAD (1988)
Propulsion System	25	10	Marine Log (09/95)
Machinery (including piping)	10	4	Bond (1996)
Paint and Tank Coatings	10	4	Marine Log (09/95)
Hull Outfit (Accommodations)	5	2	Bond (1996)
Electrical/Electronics	5	2	Bond (1996)
TOTAL	100	40	

The hull structure consists of the bow, stern and the cargo tanks, of which the bow and stern are shared components; for the comparative study, only the cargo tank region and all related components are used. It is assumed that 75 percent of the hull structure cost and 90 percent of the coating costs are associated with the cargo tank region and assuming a seven cargo tank tanker, the cost figures for the hull structure and the paint and coatings are shown in Table A.2.

Table A.2: Cargo Tank Region Hull and Coating Costs

Cost Component	Cost (\$ M US)	Cost per Length of Tank Including Overhead Costs (\$ M US)
Hull Structure	13.5	1.93
Painting and Coating	3.6	0.51
TOTAL	17.1	2.44

A.3 Cost Algorithm for Hull Structure and Coatings

The building process for the hull structure, up to the completion of erection unit joining, consists of raw material handling and preparation; plate cutting and forming; structural assembly fabrication; and erection unit joining. Each process unit cost value is given in Table A.3. The cost algorithm attributes material and labour costs associated with each process to that process. The labour component is a function of the amount of material being worked and the ease with which the labour function can be performed which, in turn, is a function of accessibility, crowdedness, size of structural component or whether the building process is uninterrupted. The ease with which the labour function can be performed is represented by a complexity factor.

Table A.3: Building Processes and Unit of Measurement for Cost Influence

Building Process	Sub-Process	Unit of Measurement
Raw Material Handling and Preparation	Sorting and Storage	Tonnes of steel
	Sandblasting	Surface area
	Priming	Surface area
Plate Cutting and Forming	Handling	Number of pieces and average size (weight) of pieces
	Cutting	Plate thickness and number of pieces
Fabrication	Handling	Number of pieces
	Set-up	Flat surfaces and number of intersections
	Welding	Automatic or manual welding, weld type, length and cross-sectional area of weld metal, and accessibility
	Intersection Reinforcement	Number of intersections
	Quality Assurance	Length of weld and surface area
	Paint and Coating Application	Surface area and accessibility
Erection Unit Joining	Unit Handling and Alignment	Tonnes of steel
	Unit Joining	Length of butt weld at joint
	Paint and Coating Application	Surface area and accessibility

A.3.1 Raw Material Handling and Preparation Process

The sub-processes involved in the raw material handling and preparation process include the receipt and handling of steel plates into the shipyard; sorting; preparation for storage; and storage. Materials purchased or expended include steel plate and primer. The input variables for this phase to the cost algorithm are:

- (a) weight of steel (tonnes);
- (b) the number of plates; and

- (c) surface area of the plates to be primed (m^2).

The unit costs associated with each of the listed items are:

- (a) steel = \$620 per tonne (Sikora et al., 1995 and McCallum, 1995);
- (b) surface preparation and protection = \$11 per m^2 (McCallum, 1995); and
- (c) stockyard handling = \$30 per tonne (McCallum, 1995).

A scrap rate of 10 percent is assumed in the cost algorithm.

The raw material handling and surface preparation process (RMHP) cost is given in Equation (A.1) as:

$$RMHP\$ = [(Tonnes\ of\ Steel) * 1.10 * 620] + [(Steel\ Surface\ Area) * 1.10 * 11] \quad (A.1)$$

A.3.2 Plate Cutting and Forming Process

This process involves cutting of the plates into component pieces and shaping the pieces that are required to fit together into the vessel assembly units. The input variables are:

- (a) weight of steel (tonnes);
- (b) number of component pieces; and
- (c) length of butt and fillet welds (m).

Edge preparation is done at the time of cutting by the cutting machine. Edges that require two bevels, typical of butt welds, take longer to cut and, therefore, are more expensive. The unit costs associated with each of the listed items are:

- (a) shop handling of large pieces = \$19 per tonne (McCallum, 1995);
- (b) shop handling = \$11 per piece (McCallum, 1995);
- (c) cutting edges for butt weld = \$2 per m (McCallum, 1995);
- (d) cutting edges for fillet weld = \$1 per m (McCallum, 1995); and
- (e) forming = \$4 per piece (McCallum, 1995).

The plate cutting and forming (PCF) process cost is given by Equation (A.2) as:

$$PCF\$ = [(Tonnes\ of\ Steel) * 1.10 * 19] + [(Number\ of\ Pieces) * 1.10 * 15] + [(Meters\ of\ Butt\ Weld) * 2] + [(Meters\ of\ Fillet\ Weld) * 1] \quad (A.2)$$

A.3.3 Fabrication of Assembly and Erection Units and Erection Unit Joining Processes

This process involves welding the component pieces together into assembly units which are then welded together to form erection units. The erection units are then joined in the drydock. The component pieces must be handled, positioned for welding and then welded together. At points of structural intersections, reinforcements may be required to strengthen the joint. Reinforcements are assumed to be required at each bottom longitudinal and floor intersection and at each side longitudinal and bulkhead intersection. The input variables are:

- (a) weight of steel (tonnes);

- (b) number of component pieces;
- (c) length of butt and fillet weld (m);
- (d) weighted average size based on length of butt welds (m);
- (e) weighted average size based on length of fillet welds (m);
- (f) surface area (m^2);
- (g) number of intersections; and
- (h) amount of structural congestion.

The unit costs associated with each of the listed items are:

- (a) weld material = \$0.05 per cm^3 of weld (McCallum, 1995);
- (b) paint and coating materials = \$12 per m^2 of surface area (McCallum, 1995);
- (c) UDHSPS elastomer material = \$3.72 per kg (private communication with BASF, Canada Ltd.);
- (d) assembly fit-up = \$90 per piece (McCallum, 1995);
- (e) welding = \$30 per piece (McCallum, 1995);
- (f) butt welding = \$0.60 per cm^3 of weld (McCallum, 1995);
- (g) fillet welding = \$0.60 per cm^3 of weld (McCallum, 1995);
- (h) intersection reinforcement = \$30 per intersection (1 hour labour assumed);
- (i) UDHSPS elastomer placement = \$1 per kg (assumed);
- (j) paint and coating application = \$1 per m^2 of surface area; and
- (k) quality assurance of welds = \$0.30 per m of weld (McCallum, 1995).

It is during this phase of construction that cost differences between the all-steel tanker and the UDHSPS design concept become more difficult, since the cost is influenced by accessibility, congestion, shape, etc., which become difficult to quantify. As such, complexity factors are introduced into the cost algorithm. For each of the items modified by a complexity factor, there comes a point at which the influence ceases to affect cost. For human accessibility, this point is when the spacing between adjacent pieces of vertically oriented structure is approximately 1000 mm. Table A.4 describes the items incorporated into the complexity factors. The mathematical form of the equations describing the complexity factors were empirically developed.

The following equations define the complexity factors:

$$CF1 = \frac{1}{e^{\left[\frac{x-600}{x}\right]^{\frac{300}{x}}}} \quad (A.3)$$

where x = distance between parallel welds within a confined space

if $x > 1000$ mm, then $CF1 = 0.9$

$$CF2 = \frac{2 * AC + SO}{3} \quad (A.4)$$

where AC = human access related term = $\left[\frac{A^2 + B^2}{2}\right]^{0.10}$

$$A = \text{term related to horizontal clearance} = \frac{1}{e^{\left[\frac{x-1000}{1000}\right]^{\frac{1000}{x}}}}$$

x = horizontal distance between the inner and outer hull stiffening elements in side or horizontal distance between girders in bottom

$$B = \text{term related to vertical clearance} = \frac{1}{e^{\left[\frac{y-1500}{1500}\right]^2}}$$

y = vertical distance (headroom)

$$SO = \text{surface orientation term} = \left[\frac{25}{\left(\frac{\text{Surface Area}}{\text{Number of Pieces}} \right)} \right]^{0.10}$$

Table A.4: Description of Complexity Factors for Fabrication Labour Costs

Complexity Factor	Influencing Item	Influencing Factor
CF1 – Welding	Access	<ul style="list-style-type: none"> - Welding machine requires at least 400 mm separation between adjacent structural components - Manual stick welding is four times more expensive than machine welding - Weld efficiency is a function of uninterrupted weld length
CF2 – Painting	Access	- Vertical access over 1500 mm and horizontal access over 1000 mm (clear transverse distance between the inner and outer hull stiffening elements)
	Surface Orientation	- A 10 percent reduction in the ratio of number of structural pieces to surface area is assumed to result in a 5 percent reduction in the complexity factor
CF3 – Quality Assurance	Access	- Access for quality assurance is similar to that for painting

The total fabrication (Fab) cost is given by Equation (A.5) using a total elastomer cost of \$4.72 per kilogram:

$$\begin{aligned}
 \text{Fab\$} = & [(\# \text{ of Pieces}) * 120] + [(\text{Surface Area}) * 12] \\
 & + [(\text{Volume of Butt Weld}) * 0.65 * CF1] \\
 & + [(\text{Volume of Fillet Weld}) * 0.65 * CF1] + [(\# \text{ of Intersections}) * 30] \\
 & + [(\text{Surface Area}) * 1 * CF2] + [(\text{Length of Weld}) * 0.30 * CF2] \\
 & + [(\text{Weight of Elastomer}) * 4.72]
 \end{aligned} \tag{A.5}$$

A.4 Summary of Detailed Cost Analysis

This section contains a summary of the detailed cost analysis conducted on a baseline conventional double hull tanker and the proposed Unidirectional Double Hull Sandwich Plate System.

A.4.1 All-Steel Conventional Double Hull Tanker

A cost analysis was performed on a baseline 40,000 DWT all-steel conventional double hull tanker, as shown in Figures A.1 and A.2, and detailed in Appendix B. The basic design parameters for the conventional double hull tanker are outlined in Table A.5. Table A.6 tabulates the scantlings determined through Lloyd's Register (Ferro, 1998) for the baseline all-steel tanker. The final cost parameters are then tabulated in Table A.7, with a final cost per cargo tank value for which the baseline was evaluated against the calibrated value of \$2.44 million as found in Table A.2.

Figure A.1: Isometric View of Conventional Double Hull Tanker

Figure A.2: Cross-Section of Conventional Double Hull Tanker

Table A.5: Basic Design Parameters for the Baseline Conventional Double Hull Tanker

Design Parameters	Value
Deadweight, DWT	40,000 t
Length between Perpendiculars, <i>LBP</i>	180.0 m
Rule Length (97% of LBP), <i>L</i>	174.6 m
Molded Breadth, <i>B</i>	32.0 m
Molded Depth, <i>D</i>	20.0 m
Molded Draft, <i>d</i>	12.0 m
Block Coefficient, <i>C_b</i>	0.85
Centerline Bulkhead	None
Double Hull Bottom Shell Separation	2.0 m
Double Hull Side Shell Separation	2.0 m
Longitudinal Girder Spacing	3.0 m
Stringer Spacing	3.0 m
Transverse Floor and Deck Spacing	3.6 m
Transverse Bulkhead Spacing	18.0 m

Table A.6: Strake Scantlings for 40,000 DWT Conventional Double Hull Tanker

Element	Value
Keel Plate Thickness	15.0 mm
Outer Bottom Plate Thickness	13.0 mm
Inner Bottom Plate Thickness	14.0 mm
Bilge Plate Thickness	13.0 mm
Side Shell Plate Thickness	12.5 mm
Sheerstrake Plate Thickness	12.5 mm
Strength Deck Plate Thickness	11.5 to 12.0 mm
Centerline Girder Thickness	14.0 mm
Side Girder Thickness	12.0 mm
Longitudinal Bulkhead Plate Thickness	10.0 to 12.5 mm
Stringer Thickness	10.0 mm
Transverse Floor Thickness	12.0 mm
Transverse Deck Plate Thickness	12.0 mm
Transverse Bulkhead Plate Thickness	12.0 mm
Transverse Bulkhead Stool Plate Thickness	12.0 mm
Bottom Shell Stiffeners	400 x 25 flatbar
Bilge Stiffeners	400 x 25 flatbar
Inner Bottom Stiffeners	400 x 27 flatbar
Side Shell and Sheerstrake Stiffeners	400 x 25 to 300 x 20 flatbar
Strength Deck Stiffeners	250 x 23 to 250 x 19 flatbar
Longitudinal Bulkhead Stiffeners	400 x 25 to 300 x 20 flatbar
Transverse Bulkhead Stiffeners	400 x 25 flatbar

Table A.7: Cost Parameters for 40,000 DWT Conventional Double Hull Tanker

Cost Parameter per Cargo Tank	Value
Total Steel Weight	859.3 t
Surface Area	15,105 m ²
Number of Pieces	1,384
Butt Weld Length	2,674 m
Fillet Weld Length	15,159 m
Number of Intersections	5,020
Butt Weld Volume	538,600 cm ³
Fillet Weld Volume	1,232,000 cm ³
Raw Material Handling and Preparation Cost	\$ 0.80 million
Plate Cutting and Forming Cost	\$ 0.06 million
Fabrication Cost	\$ 1.63 million
Total Cost	\$ 2.49 million
Target Cost for Calibration	\$ 2.44 million

A.4.2 Unidirectional Double Hull Sandwich Plate System Tanker

A detailed cost analysis was performed on a 40,000 DWT Unidirectional Double Hull Sandwich Plate System tanker, as shown in Figures A.3 and A.4, and illustrated in Appendix C. The basic design parameters are outlined in Table A.5, except for the transverse floors which will only be situated under the stools of the transverse bulkhead. Table A.8 tabulates the scantlings determined for the tanker. The final cost parameters are then tabulated in Table A.9, with a final cost per cargo tank value for the UDHSPS tanker versus that for the all-steel CDH tanker.

Figure A.3: Isometric View of UDHSPS Tanker

Figure A.4: Cross-Section of UDHSPS Tanker

Table A.8: Strake Scantlings for 40,000 DWT UDHSPS Tanker

Element	Value
Keel Plate Thickness	7.5 mm
Outer Bottom Plate Thickness	6.5 mm
Inner Bottom Plate Thickness	7.0 mm
Bilge Plate Thickness	6.5 mm
Side Shell Plate Thickness	6.25 mm
Sheerstrake Plate Thickness	6.25 mm
Strength Deck Plate Thickness	5.8 to 6.0 mm
Centerline Girder Thickness	14.0 mm
Side Girder Thickness	12.0 mm
Longitudinal Bulkhead Plate Thickness	5.0 to 6.3 mm
Stringer Thickness	10.0 mm
Transverse Floor Thickness	6.0 mm
Transverse Deck Plate Thickness	6.0 mm
Transverse Bulkhead Plate Thickness	6.0 mm
Transverse Bulkhead Stool Plate Thickness	6.0 mm
Transverse Bulkhead, Stool and Floor Vertical Framing Plate Thickness	12.0 – 14.0 mm
Transverse Bulkhead Horizontal Framing Plate Thickness	10.0 mm

Table A.9: Cost Parameters for 40,000 DWT UDHSPS Tanker

Cost Parameter per Cargo Tank	Value
Total Steel Weight	515.8 t
Total Elastomer Weight	234.1 t
Total Weight	749.9 t
Surface Area	19,306 m ²
Number of Pieces	738
Butt Weld Length	4,511 m
Fillet Weld Length	4,833 m
Number of Intersections	292
Butt Weld Volume	172,700 cm ³
Fillet Weld Volume	47,500 cm ³
Raw Material Handling and Preparation Cost	\$ 0.60 million
Plate Cutting and Forming Cost	\$ 0.04 million
Fabrication Cost	\$ 0.46 million
Elastomer Cost (Including Placement)	\$1.10 million
Total Cost	\$ 2.20 million
Total Cost for All-Steel CDH	\$ 2.49 million

Appendix B

Costing Spreadsheet for an All-Steel 40,000 DWT Conventional Double Hull Tanker

B. Cost Analysis for 40,000 DWT All-Steel CDH Tanker

Design Parameters

Length of Tank = 18000 mm

Density of Steel = 7850 kg/m³

Using 2 Plates per Cargo Length (were specified) L = 9000 mm

Note: All dollar values are in US currency.

Strake Scantlings

Bottom Shell Plate Keel (2 Plates per Cargo Length)

2	0	0	900	0	15.0	900	32.40	0.2430	1.9076
4	900	0	3000	0	13.0	2100	75.60	0.4914	3.8575
6	3000	0	6000	0	13.0	3000	108.00	0.7020	5.5107
8	6000	0	9000	0	13.0	3000	108.00	0.7020	5.5107
10	9000	0	12000	0	13.0	3000	108.00	0.7020	5.5107
12	12000	0	14000	0	13.0	2000	72.00	0.4680	3.6738
14	0	0	-900	0	15.0	900	32.40	0.2430	1.9076
16	-900	0	-3000	0	13.0	2100	75.60	0.4914	3.8575
18	-3000	0	-6000	0	13.0	3000	108.00	0.7020	5.5107
20	-6000	0	-9000	0	13.0	3000	108.00	0.7020	5.5107
22	-9000	0	-12000	0	13.0	3000	108.00	0.7020	5.5107
24	-12000	0	-14000	0	13.0	2000	72.00	0.4680	3.6738

24 Total Number of Pieces per Section

Bilge (2 Plates per Cargo Length)

26	14000	0	15414	585	13.0	1570.80	56.55	0.3676	2.8854
28	15414	585	16000	2000	13.0	1570.80	56.55	0.3676	2.8854
30	-14000	0	-15414	585	13.0	1570.80	56.55	0.3676	2.8854
32	-15414	585	-16000	2000	13.0	1570.80	56.55	0.3676	2.8854

8 Total Number of Pieces per Section

Inner Bottom (2 Plates per Cargo Length)

34	0	2000	3000	2000	14.0	3000	108.00	0.7560	5.9346
36	3000	2000	6000	2000	14.0	3000	108.00	0.7560	5.9346
38	6000	2000	9000	2000	14.0	3000	108.00	0.7560	5.9346
40	9000	2000	12000	2000	14.0	3000	108.00	0.7560	5.9346
42	0	2000	-3000	2000	14.0	3000	108.00	0.7560	5.9346
44	-3000	2000	-6000	2000	14.0	3000	108.00	0.7560	5.9346
46	-6000	2000	-9000	2000	14.0	3000	108.00	0.7560	5.9346
48	-9000	2000	-12000	2000	14.0	3000	108.00	0.7560	5.9346

16 Total Number of Pieces per Section

Centerline Girder (2 Plates per Cargo Length)

50	0	0	0	2000	14.0	2000.00	72.00	0.5040	3.9564
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2 **Total Number of Pieces per Section**

Side Girders (2 Plates per Cargo Length)

52	3000	0	3000	2000	12.0	2000	72.00	0.4320	3.3912
54	6000	0	6000	2000	12.0	2000	72.00	0.4320	3.3912
56	9000	0	9000	2000	12.0	2000	72.00	0.4320	3.3912
58	12000	0	12000	2000	12.0	2000	72.00	0.4320	3.3912
60	-3000	0	-3000	2000	12.0	2000	72.00	0.4320	3.3912
62	-6000	0	-6000	2000	12.0	2000	72.00	0.4320	3.3912
64	-9000	0	-9000	2000	12.0	2000	72.00	0.4320	3.3912
66	-12000	0	-12000	2000	12.0	2000	72.00	0.4320	3.3912

16 Total Number of Pieces per Section

Side Shell and Sheerstrake (2 Plates per Cargo Length)

68	16000	2000	16000	4000	12.5	2000	72.00	0.4500	3.5325
70	16000	4000	16000	7000	12.5	3000	108.00	0.6750	5.2988
72	16000	7000	16000	10000	12.5	3000	108.00	0.6750	5.2988
74	16000	10000	16000	13000	12.5	3000	108.00	0.6750	5.2988
76	16000	13000	16000	16000	12.5	3000	108.00	0.6750	5.2988
78	16000	16000	14000	20000	12.5	5141.59	185.10	1.1569	9.0813
80	-16000	2000	-16000	4000	12.5	2000.00	72.00	0.4500	3.5325
82	-16000	4000	-16000	7000	12.5	3000.00	108.00	0.6750	5.2988
84	-16000	7000	-16000	10000	12.5	3000.00	108.00	0.6750	5.2988
86	-16000	10000	-16000	13000	12.5	3000.00	108.00	0.6750	5.2988
88	-16000	13000	-16000	16000	12.5	3000.00	108.00	0.6750	5.2988
90	-16000	16000	-14000	20000	12.5	5141.59	185.10	1.1569	9.0813

24 Total Number of Pieces per Section

Strength Deck (2 Plates per Cargo Length)

Element	Start		End		Thickness (mm)	Width (mm)	Area (mm ²)	Volume (mm ³)	Weight (kg)
	Y	Z	Y	Z					
92	0	21500	2977	21128	12.0	3000.15	108.01	0.6480	5.0871
94	2977	21128	5954	20756	12.0	3000.15	108.01	0.6480	5.0871
96	5954	20756	8931	20384	11.5	3000.15	108.01	0.6210	4.8751
98	8931	20384	11908	20013	11.5	3000.03	108.00	0.6210	4.8749
100	11908	20013	14000	20000	11.5	2092.04	75.31	0.4331	3.3995
102	0	21500	2977	21128	12.0	3000.15	108.01	0.6480	5.0871
104	2977	21128	5954	20756	12.0	3000.15	108.01	0.6480	5.0871
106	5954	20756	8931	20384	11.5	3000.15	108.01	0.6210	4.8751
108	8931	20384	11908	20013	11.5	3000.03	108.00	0.6210	4.8749
110	11908	20013	14000	20000	11.5	2092.04	75.31	0.4331	3.3995

20 Total Number of Pieces per Section

Longitudinal Bulkhead (2 Plates per Cargo Length)

112	12000	2000	13999	4002	12.5	2829.14	101.85	0.6366	4.9970
114	13999	4002	13999	7000	12.5	2998.00	107.93	0.6746	5.2952
116	13999	7000	13999	10000	11.5	3000.00	108.00	0.6210	4.8749
118	13999	10000	13999	13000	11.0	3000.00	108.00	0.5940	4.6629
120	13999	13000	13996	16003	10.5	3003.00	108.11	0.5676	4.4554
122	13996	16003	12000	18002	10.0	2824.89	101.70	0.5085	3.9916
124	12000	18002	12000	20001	10.0	1999.00	71.96	0.3598	2.8246
126	-12000	2000	-13999	4002	12.5	2829.14	101.85	0.6366	4.9970
128	-13999	4002	-13999	7000	12.5	2998.00	107.93	0.6746	5.2952
130	-13999	7000	-13999	10000	11.5	3000.00	108.00	0.6210	4.8749
132	-13999	10000	-13999	13000	11.0	3000.00	108.00	0.5940	4.6629
134	-13999	13000	-13996	16003	10.5	3003.00	108.11	0.5676	4.4554
136	-13996	16003	-12000	18002	10.0	2824.89	101.70	0.5085	3.9916
138	-12000	18002	-12000	20001	10.0	1999.00	71.96	0.3598	2.8246

28 Total Number of Pieces per Section

Stringers (2 Plates per Cargo Length)

140	16000	4000	13999	4000	10.0	2001	72.04	0.3602	2.8274
142	16000	7000	13999	7000	10.0	2001	72.04	0.3602	2.8274
144	16000	10000	13999	10000	10.0	2001	72.04	0.3602	2.8274
146	16000	13000	13999	13000	10.0	2001	72.04	0.3602	2.8274
148	16000	16000	13999	16000	10.0	2001	72.04	0.3602	2.8274
150	-16000	4000	-13999	4000	10.0	2001	72.04	0.3602	2.8274
152	-16000	7000	-13999	7000	10.0	2001	72.04	0.3602	2.8274
154	-16000	10000	-13999	10000	10.0	2001	72.04	0.3602	2.8274
156	-16000	13000	-13999	13000	10.0	2001	72.04	0.3602	2.8274
158	-16000	16000	-13999	16000	10.0	2001	72.04	0.3602	2.8274

20 Total Number of Pieces per Section

Floors and Transverse Deck Elements (5 Floors Per Tank)

Height = 2000 mm

163	0	0	3000	0	12.0	3000.00	60.00	0.3600	2.826
168	3000	0	6000	0	12.0	3000.00	60.00	0.3600	2.826
173	6000	0	9000	0	12.0	3000.00	60.00	0.3600	2.826
178	9000	0	12000	0	12.0	3000.00	60.00	0.3600	2.826
183	12000	0	16000	0	12.0	N/A	131.42	0.7885	6.190
188	16000	4000	16000	7000	12.0	3000.00	60.00	0.3600	2.826
193	16000	7000	16000	10000	12.0	3000.00	60.00	0.3600	2.826
198	16000	10000	16000	13000	12.0	3000.00	60.00	0.3600	2.826
203	16000	13000	16000	16000	12.0	3000.00	60.00	0.3600	2.826
208	16000	16000	14000	20000	12.0	N/A	131.42	0.7885	6.190
213	0	18000	3000	18000	12.0	3000.00	65.63	0.3938	3.091
218	3000	18000	6000	18000	12.0	3000.00	76.88	0.4613	3.621

Floors and Transverse Deck Elements (Cont'd)**Height = 2000 mm**

Element	Dimensions				Height	Area	Weight	Volume	Mass
	X	Y	Z	T					
223	6000	18000	9000	18000	12.0	3000.00	88.13	0.5288	4.151
228	9000	18000	12000	18000	12.0	3000.00	99.38	0.5963	4.681
233	0	0	-3000	0	12.0	3000.00	60.00	0.3600	2.826
238	-3000	0	-6000	0	12.0	3000.00	60.00	0.3600	2.826
243	-6000	0	-9000	0	12.0	3000.00	60.00	0.3600	2.826
248	-9000	0	-12000	0	12.0	3000.00	60.00	0.3600	2.826
253	-12000	0	-16000	0	12.0	N/A	131.42	0.7885	6.190
258	-16000	4000	-16000	7000	12.0	3000.00	60.00	0.3600	2.826
263	-16000	7000	-16000	10000	12.0	3000.00	60.00	0.3600	2.826
268	-16000	10000	-16000	13000	12.0	3000.00	60.00	0.3600	2.826
273	-16000	13000	-16000	16000	12.0	3000.00	60.00	0.3600	2.826
278	-16000	16000	-14000	20000	12.0	N/A	131.42	0.7885	6.190
283	0	18000	-3000	18000	12.0	3000.00	65.63	0.3938	3.091
288	-3000	18000	-6000	18000	12.0	3000.00	76.88	0.4613	3.621
293	-6000	18000	-9000	18000	12.0	3000.00	88.13	0.5288	4.151
298	-9000	18000	-12000	18000	12.0	3000.00	99.38	0.5963	4.681

140 Total Number of Pieces per Section**Transverse Bulkhead (1 per Cargo Tank)****L = 3000 mm**

Element	Dimensions				Height	Area	Weight	Volume	Mass
	X	Y	Z	T					
306	0	4000	3000	4000	12.0	3000	144.00	0.864	6.782
314	3000	4000	6000	4000	12.0	3000	144.00	0.864	6.782
322	6000	4000	9000	4000	12.0	3000	144.00	0.864	6.782
330	9000	4000	12000	4000	12.0	3000	144.00	0.864	6.782
338	12000	4000	14000	4000	12.0	2000	96.00	0.576	4.522
346	0	4000	-3000	4000	12.0	3000	144.00	0.864	6.782
354	-3000	4000	-6000	4000	12.0	3000	144.00	0.864	6.782
362	-6000	4000	-9000	4000	12.0	3000	144.00	0.864	6.782
370	-9000	4000	-12000	4000	12.0	3000	144.00	0.864	6.782
378	-12000	4000	-14000	4000	12.0	2000	96.00	0.576	4.522

80 Total Number of Pieces per Section**Stools (4 per Transverse Bulkhead)****L = 2385 mm**

Element	Dimensions				Height	Area	Weight	Volume	Mass
	X	Y	Z	T					
382	0	4000	3000	4000	12.0	3000	57.25	0.343	2.696
386	3000	4000	6000	4000	12.0	3000	57.25	0.343	2.696
390	6000	4000	9000	4000	12.0	3000	57.25	0.343	2.696
394	9000	4000	12000	4000	12.0	3000	57.25	0.343	2.696
398	12000	4000	14000	4000	12.0	2000	19.08	0.114	0.899
402	0	4000	-3000	4000	12.0	3000	57.25	0.343	2.696
406	-3000	4000	-6000	4000	12.0	3000	57.25	0.343	2.696
410	-6000	4000	-9000	4000	12.0	3000	57.25	0.343	2.696
414	-9000	4000	-12000	4000	12.0	3000	57.25	0.343	2.696
418	-12000	4000	-14000	4000	12.0	2000	19.08	0.114	0.899

40 Total Number of Pieces per Section

Stiffener Scantlings (2 Plates per Cargo Length)

Bottom Shell	426	1	4	400	25	57.60	0.7200	5.6520
	434	2	4	400	25	57.60	0.7200	5.6520
	442	3	4	400	25	57.60	0.7200	5.6520
	450	4	4	400	25	57.60	0.7200	5.6520
	456	5	3	400	25	43.20	0.5400	4.2390
	464	6	4	400	25	57.60	0.7200	5.6520
	472	7	4	400	25	57.60	0.7200	5.6520
	480	8	4	400	25	57.60	0.7200	5.6520
	488	9	4	400	25	57.60	0.7200	5.6520
	494	10	3	400	25	43.20	0.5400	4.2390
Bilge	502	11	4	400	25	57.60	0.7200	5.6520
	510	12	4	400	25	57.60	0.7200	5.6520
Inner Bottom	518	13	4	400	27	57.60	0.7776	6.1042
	526	14	4	400	27	57.60	0.7776	6.1042
	534	15	4	400	27	57.60	0.7776	6.1042
	542	16	4	400	27	57.60	0.7776	6.1042
	550	17	4	400	27	57.60	0.7776	6.1042
	558	18	4	400	27	57.60	0.7776	6.1042
	566	19	4	400	27	57.60	0.7776	6.1042
	574	20	4	400	27	57.60	0.7776	6.1042
Side Shell	580	21	3	400	25	43.20	0.5400	4.2390
	588	22	4	400	25	57.60	0.7200	5.6520
	596	23	4	350	23	50.40	0.5796	4.5499
	604	24	4	300	25	43.20	0.5400	4.2390
	612	25	4	300	21	43.20	0.4536	3.5608
	626	26	7	300	20	75.60	0.7560	5.9346
	632	27	3	400	25	43.20	0.5400	4.2390
	640	28	4	400	25	57.60	0.7200	5.6520
	648	29	4	350	23	50.40	0.5796	4.5499
	656	30	4	300	25	43.20	0.5400	4.2390
	664	31	4	300	21	43.20	0.4536	3.5608
	678	32	7	300	20	75.60	0.7560	5.9346
Strength Deck	718	33	20	250	19	180.00	1.7100	13.4235
	724	34	3	250	23	27.00	0.3105	2.4374
	762	35	19	250	19	171.00	1.6245	12.7523
	768	36	3	250	23	27.00	0.3105	2.4374
Long BHD	776	37	4	400	25	57.60	0.7200	5.6520
	784	38	4	400	25	57.60	0.1800	1.4130
	792	39	4	350	23	50.40	0.1449	1.1375
	800	40	4	300	25	43.20	0.1350	1.0598
	808	41	4	300	20	43.20	0.1080	0.8478
	816	42	4	300	20	43.20	0.1080	0.8478
	820	43	2	250	21	18.00	0.0945	0.7418
	828	44	4	400	25	57.60	0.1800	1.4130
	836	45	4	400	25	57.60	0.1800	1.4130
	844	46	4	350	23	50.40	0.1449	1.1375
	852	47	4	300	25	43.20	0.1350	1.0598

Stiffener Scantlings (Cont'd)

Long BHD	860	48	4	300	20	43.20	0.1080	0.8478
	868	49	4	300	20	43.20	0.1080	0.8478
	872	50	2	250	21	18.00	0.0945	0.7418
Trans BHD	936	51	8	400	25	132.93	1.6616	13.0436
	1000	52	8	400	25	132.93	1.6616	13.0436
	1064	53	8	400	25	132.93	1.6616	13.0436
	1128	54	8	400	25	132.93	1.6616	13.0436
	1192	55	8	400	25	132.93	1.6616	13.0436
	1256	56	8	400	25	132.93	1.6616	13.0436
	1320	57	8	400	25	132.93	1.6616	13.0436
	1384	58	8	400	25	132.93	1.6616	13.0436
Total Number of Pieces per Section				966				

Quantities for Single Tank

Total Number of Pieces = 1384
 Total Area of Steel = 15104.7 m²
 Total Volume of Steel = 109.46 m³
 Total Weight of Steel = 859.26 t

Quantities for All (7) Tanks

Total Number of Pieces = 9688
 Total Area of Steel = 105733 m²
 Total Volume of Steel = 766.22 m³
 Total Weight of Steel = 6014.8 t

Raw Material Handling and Preparation

The unit cost figures used in the raw material handling and preparation are:

Steel \$620 per tonne
 Surface Preparation and Protection \$11 per square meter
 Stockyard Handling \$30 per tonne

Note: A scrap rate of 10% was assumed.

$$\text{RMHPS} = (\text{Tonnes of Steel}) * 1.10 * 620 + (\text{Surface Area}) * 1.10 * 11$$

RMHPS Per Cargo Tank = 0.797 (in Millions)

Plate Cutting and Forming

Butt and Fillet Weld Calculations

Bottom Shell	2	9000	900.00	19.8	N/A
	4	9000	2100.00	22.2	N/A
	6	9000	3000.00	24.0	N/A
	8	9000	3000.00	24.0	N/A
	10	9000	3000.00	24.0	N/A
	12	9000	2000.00	22.0	N/A
	14	9000	900.00	19.8	N/A
	16	9000	2100.00	22.2	N/A
	18	9000	3000.00	24.0	N/A
	20	9000	3000.00	24.0	N/A
	22	9000	3000.00	24.0	N/A
	24	9000	2000.00	22.0	N/A
Bilge	26	9000	1570.80	21.1	N/A
	28	9000	1570.80	21.1	N/A
	30	9000	1570.80	21.1	N/A
	32	9000	1570.80	21.1	N/A
Inner Bottom	34	9000	3000.00	24.0	N/A
	36	9000	3000.00	24.0	N/A
	38	9000	3000.00	24.0	N/A
	40	9000	3000.00	24.0	N/A
	42	9000	3000.00	24.0	N/A
	44	9000	3000.00	24.0	N/A
	46	9000	3000.00	24.0	N/A
	48	9000	3000.00	24.0	N/A
CL Girder	50	9000	2000.00	4.0	72.0
Side Girders	52	9000	2000.00	4.0	72.0
	54	9000	2000.00	4.0	72.0
	56	9000	2000.00	4.0	72.0
	58	9000	2000.00	4.0	72.0
	60	9000	2000.00	4.0	72.0
	62	9000	2000.00	4.0	72.0
	64	9000	2000.00	4.0	72.0
	66	9000	2000.00	4.0	72.0
S. Shl & Shrstrk	68	9000	2000.00	22.0	N/A
	70	9000	3000.00	24.0	N/A
	72	9000	3000.00	24.0	N/A
	74	9000	3000.00	24.0	N/A
	76	9000	3000.00	24.0	N/A
	78	9000	5141.59	28.3	N/A
	80	9000	2000.00	22.0	N/A
	82	9000	3000.00	24.0	N/A
	84	9000	3000.00	24.0	N/A
	86	9000	3000.00	24.0	N/A
	88	9000	3000.00	24.0	N/A
	90	9000	5141.59	28.3	N/A

Butt and Fillet Weld Calculations (Cont'd)

Strength Deck	92	9000	3000.15	24.0	N/A
	94	9000	3000.15	24.0	N/A
	96	9000	3000.15	24.0	N/A
	98	9000	3000.03	24.0	N/A
	100	9000	2092.04	22.2	N/A
	102	9000	3000.15	24.0	N/A
	104	9000	3000.15	24.0	N/A
	106	9000	3000.15	24.0	N/A
	108	9000	3000.03	24.0	N/A
	110	9000	2092.04	22.2	N/A
Long Blkhd	112	9000	2829.14	23.7	N/A
	114	9000	2998.00	24.0	N/A
	116	9000	3000.00	24.0	N/A
	118	9000	3000.00	24.0	N/A
	120	9000	3003.00	24.0	N/A
	122	9000	2824.89	23.6	N/A
	124	9000	1999.00	22.0	N/A
	126	9000	2829.14	23.7	N/A
	128	9000	2998.00	24.0	N/A
	130	9000	3000.00	24.0	N/A
	132	9000	3000.00	24.0	N/A
	134	9000	3003.00	24.0	N/A
	136	9000	2824.89	23.6	N/A
	138	9000	1999.00	44.0	N/A
Stringers	140	9000	2001.00	4.0	72.0
	142	9000	2001.00	4.0	72.0
	144	9000	2001.00	4.0	72.0
	146	9000	2001.00	4.0	72.0
	148	9000	2001.00	4.0	72.0
	150	9000	2001.00	4.0	72.0
	152	9000	2001.00	4.0	72.0
	154	9000	2001.00	4.0	72.0
	156	9000	2001.00	4.0	72.0
	158	9000	2001.00	4.0	72.0
Floors and Transverse Deck Elements	163	3000.00	2000.00	N/A	100.0
	168	3000.00	2000.00	N/A	100.0
	173	3000.00	2000.00	N/A	100.0
	178	3000.00	2000.00	N/A	100.0
	183	13970.02	2000.00	N/A	139.7
	188	3000.00	2000.00	N/A	100.0
	193	3000.00	2000.00	N/A	100.0
	198	3000.00	2000.00	N/A	100.0
	203	3000.00	2000.00	N/A	100.0
	208	13970.02	2000.00	N/A	139.7
	213	3000.00	2375.00	5.9	60.2
	218	3000.00	2750.00	12.8	60.2
	223	3000.00	3125.00	14.7	60.2
	228	3000.00	3500.00	16.6	60.2

Butt and Fillet Weld Calculations (Cont'd)

Floors and Transverse Deck Elements	233	3000.00	2000.00	N/A	100.0
	238	3000.00	2000.00	N/A	100.0
	243	3000.00	2000.00	N/A	100.0
	248	3000.00	2000.00	N/A	100.0
	253	13970.02	2000.00	N/A	139.7
	258	3000.00	2000.00	N/A	100.0
	263	3000.00	2000.00	N/A	100.0
	268	3000.00	2000.00	N/A	100.0
	273	3000.00	2000.00	N/A	100.0
	278	13970.02	2000.00	N/A	139.7
	283	3000.00	2375.00	5.9	60.2
	288	3000.00	2750.00	12.8	60.2
	293	3000.00	3125.00	14.7	60.2
	298	3000.00	3500.00	16.6	60.2
Transverse BHD	306	3000.00	3000.00	48.0	N/A
	314	3000.00	3000.00	48.0	N/A
	322	3000.00	3000.00	48.0	N/A
	330	3000.00	3000.00	48.0	N/A
	338	3000.00	2000.00	40.0	48.0
	346	3000.00	3000.00	48.0	N/A
	354	3000.00	3000.00	48.0	N/A
	362	3000.00	3000.00	48.0	N/A
	370	3000.00	3000.00	48.0	N/A
	378	3000.00	2000.00	40.0	48.0
Stools	382	2385.37	3000.00	15.5	24.0
	386	2385.37	3000.00	15.5	24.0
	390	2385.37	3000.00	15.5	24.0
	394	2385.37	3000.00	15.5	24.0
	398	2385.37	2000.00	8.8	16.0
	402	2385.37	3000.00	15.5	24.0
	406	2385.37	3000.00	15.5	24.0
	410	2385.37	3000.00	15.5	24.0
	414	2385.37	3000.00	15.5	24.0
	418	2385.37	2000.00	8.8	16.0
Stiffeners Bottom Shell	426	9000	400	3.2	144.0
	434	9000	400	3.2	144.0
	442	9000	400	3.2	144.0
	450	9000	400	3.2	144.0
	456	9000	400	2.4	108.0
	464	9000	400	3.2	144.0
	472	9000	400	3.2	144.0
	480	9000	400	3.2	144.0
	488	9000	400	3.2	144.0
	494	9000	400	2.4	108.0
Bilge	502	9000	400	3.2	144.0
	510	9000	400	3.2	144.0

Butt and Fillet Weld Calculations (Cont'd)

Inner Bottom	518	9000	400	3.2	144.0
	526	9000	400	3.2	144.0
	534	9000	400	3.2	144.0
	542	9000	400	3.2	144.0
	550	9000	400	3.2	144.0
	558	9000	400	3.2	144.0
	566	9000	400	3.2	144.0
	574	9000	400	3.2	144.0
Side Shell	580	9000	400	2.4	108.0
	588	9000	400	3.2	144.0
	596	9000	350	2.8	144.0
	604	9000	300	2.4	144.0
	612	9000	300	2.4	144.0
	626	9000	300	4.2	252.0
	632	9000	400	2.4	108.0
	640	9000	400	3.2	144.0
	648	9000	350	2.8	144.0
	656	9000	300	2.4	144.0
	664	9000	300	2.4	144.0
	678	9000	300	4.2	252.0
Strength Deck	718	9000	250	10.0	720.0
	724	9000	250	1.5	108.0
	762	9000	250	9.5	684.0
	768	9000	250	1.5	108.0
Long BHD	776	9000	400	3.2	144.0
	784	9000	400	3.2	144.0
	792	9000	350	2.8	144.0
	800	9000	300	2.4	144.0
	808	9000	300	2.4	144.0
	816	9000	300	2.4	144.0
	820	9000	250	1.0	72.0
	828	9000	400	3.2	144.0
	836	9000	400	3.2	144.0
	844	9000	350	2.8	144.0
	852	9000	300	2.4	144.0
	860	9000	300	2.4	144.0
	868	9000	300	2.4	144.0
	872	9000	250	1.0	72.0
Trans BHD	936	20770	400	38.4	332.3
	1000	20770	400	38.4	332.3
	1064	20770	400	38.4	332.3
	1128	20770	400	38.4	332.3
	1192	20770	400	38.4	332.3
	1256	20770	400	38.4	332.3
	1320	20770	400	38.4	332.3
	1384	20770	400	38.4	332.3

The unit cost figures used in the plate cutting and forming are:

Shop Handling (Large Pieces)	\$19	per tonne
Shop Handling	\$11	per piece
Cutting (Edges for Butt Weld)	\$2	per meter of butt weld
Cutting (Edges for Fillet Weld)	\$1	per meter of fillet weld
Forming	\$4	per piece

Note: A scrap rate of 10% was assumed.

$$PCFS = (\text{Tonnes of Steel}) * 1.10 * 19 + (\text{Number of Pieces}) * 1.10 * 15 + (\text{Meter of Butt Weld}) * 2 + (\text{Meter of Fillet Weld}) * 1$$

$$PCFS \text{ Per Cargo Tank} = 0.861 \quad (\text{in Millions})$$

Fabrication Costs

Complexity Factor # 1 (Used In Considering Welding Access)

$$CF_1 = \frac{1}{e^{\left(\frac{x-600}{x}\right)^{\frac{300}{x}}}}$$

Note: If $x > 1000$ mm then $CF_1 = 0.9$

$$\begin{aligned} x &= 600 \text{ mm} && \text{distance between parallel welds within a confined space} \\ CF_1 &= 1.000 \end{aligned}$$

Complexity Factor # 2 (Considering Painting Access, Surface Orientation, and Quality Assurance)

$$CF_2 = \frac{2 * AC + SO}{3}$$

$$CF_2 = 0.965$$

where

$$AC = \left(\frac{A^2 + B^2}{2}\right)^{0.1}$$

$$AC = 0.9039$$

AC = access related term

$$A = \frac{1}{e^{\left(\frac{x-1000}{1000}\right)^{\frac{1000}{x}}}}$$

A = 0.7699	horiz. dist. b/w side shell and long bh stiffener group #2
0.6730	horiz. dist. b/w side shell and long bh stiffener group #3
0.5947	horiz. dist. b/w side shell and long bh stiffener group #4
0.5947	horiz. dist. b/w side shell and long bh stiffener group #5
0.2837	horizontal distance b/w CL girder and side girder #1
0.2837	horizontal distance b/w side girder #1 and side girder #2
0.2837	horizontal distance b/w side girder #2 and side girder #3
0.2837	horizontal distance b/w side girder #3 and side girder #4

$$B = \frac{1}{e^{\frac{(\gamma - 1500)}{1500} \cdot \frac{1300}{1500}}}$$

B =	#NUM!	vert dist b/w inner and outer bottom stiffener group #1
	#NUM!	vert dist b/w inner and outer bottom stiffener group #2
	#NUM!	vert dist b/w inner and outer bottom stiffener group #3
	#NUM!	vert dist b/w inner and outer bottom stiffener group #4
	0.3679	vert distance between stringer #1 and #2
	0.3679	vert distance between stringer #2 and #3
	0.3679	vert distance between stringer #3 and #4
	0.3679	vert distance between stringer #4 and #5

x =	1200 mm	horiz. dist. b/w side shell and long bh stiffener group #2
	1300 mm	horiz. dist. b/w side shell and long bh stiffener group #3
	1400 mm	horiz. dist. b/w side shell and long bh stiffener group #4
	1400 mm	horiz. dist. b/w side shell and long bh stiffener group #5
	3000 mm	horizontal distance b/w CL girder and side girder #1
	3000 mm	horizontal distance b/w side girder #1 and side girder #2
	3000 mm	horizontal distance b/w side girder #2 and side girder #3
	3000 mm	horizontal distance b/w side girder #3 and side girder #4

y =	1200 mm	vert dist b/w inner and outer bottom stiffener group #1
	1200 mm	vert dist b/w inner and outer bottom stiffener group #2
	1200 mm	vert dist b/w inner and outer bottom stiffener group #3
	1200 mm	vert dist b/w inner and outer bottom stiffener group #4
	3000 mm	vert distance between stringer #1 and #2
	3000 mm	vert distance between stringer #2 and #3
	3000 mm	vert distance between stringer #3 and #4
	3000 mm	vert distance between stringer #4 and #5

$$SO = \left(\frac{25}{\frac{\text{Surface Area}}{\text{\# of Pieces}}} \right)^{0.1}$$

SO = surface orientation term

$$SO = 1.0864$$

Volume of Weld

Average Plate Thickness

$$t_p = 20.49 \text{ mm}$$

Fillet Weld Throat Thickness

$$w = t_p * wf * 2^{0.5}$$

$$w = 12.8 \text{ mm}$$

where $wf = 0.44$ weld factor

Area of Fillet Weld

$$A_{fw} = 0.5w^2$$

$$A_{fw} = 81.3 \text{ mm}^2$$

Volume of Fillet Weld

$$V_{fw} = A_{fw} * L_{fw}$$

$$L_{fw} = 15159 \text{ m}$$

$$V_{fw} = 1.232\text{E}+06 \text{ cm}^3$$

Area of Butt Weld

$$A_{bw} = t_p * s + 0.166t_p^2 * 2$$

$$A_{bw} = 201.4 \text{ mm}^2$$

$$\text{where } s = 3 \text{ mm}$$

Volume of Butt Weld

$$V_{bw} = A_{bw} * L_{bw}$$

$$L_{bw} = 2673.7 \text{ m}$$

$$V_{bw} = 5.386\text{E}+05 \text{ cm}^3$$

Number of Intersections (Where Longitudinal and Transverse Cross)**Stiffeners**

$$\text{Each stiffener will intersect 5 floors twice: } 483 * 5 * 2 = 4830$$

Girders

$$\text{Each side girder and the CL girder will intersect 5 floors twice: } 9 * 5 * 2 = 90$$

Stringers

$$\text{Each stringer will intersect 5 floors twice: } 10 * 5 * 2 = 100$$

$$\text{TOTAL NUMBER OF INTERSECTIONS} = 5020$$

The unit cost figures used in the fabrication are:

Weld Material	\$0.05 per cubic centimeter of weld
Paint and Coating Materials	\$12 per square meter of surface area
Assembly Fit-Up	\$90 per piece
Welding	\$30 per piece
Butt Welding	\$0.60 per cubic centimeter of weld
Fillet Welding	\$0.60 per cubic centimeter of weld
Intersection Reinforcement	\$30 per intersection
Paint and Coating Application	\$1 per square meter of surface area
Quality Assurance of Welds	\$0.30 per meter of weld

$$\text{Fab\$} = (\# \text{ of Pieces}) * 120 + (\text{Surface Area}) * 12 + (\text{Volume of Weld}) * 0.65 * CF_1 + (\# \text{ of Intersections}) * 30 \\ + (\text{Surface Area}) * 1 * CF_2 + (\text{Length of Weld}) * 0.3 * CF_2$$

$$\text{Fab\$ Per Cargo Tank} = \$1.63 \text{ (In Millions)}$$

$$\text{TOTAL OVERALL COST PER CARGO TANK} = \$2.49 \text{ (In Millions)}$$

Total Estimated Cost for a 40,000 DWT Double Hull Tanker =	\$40.00 Million
Hull Structure Percentage of Acquisition Price =	45%
Paint and Tank Coatings Percentage of Acquisition Price =	10%
Percentage of Coating Costs Associated with Cargo Tank Region =	90%
Percentage of Hull Structure Cost Associated with Cargo Tank Region =	75%
Estimated Hull Structure Cost per Cargo Tank =	\$1.93 Million
Estimated Painting and Coatings Cost per Cargo Tank =	\$0.51 Million
TOTAL ESTIMATED COST PER CARGO TANK =	\$2.44 Million

Appendix C

Costing Spreadsheet for a 40,000 DWT Unidirectional Double Hull Sandwich Plate System Tanker

C. Cost Analysis for 40,000 DWT UDHSPS Tanker

Design Parameters

Length of Tank = 18000 mm

Density of Steel = 7850 kg/m³

Using 4 Plates per Cargo Length (were specified) L = 9000 mm

Thickness of Outer Hull Elastomer 50 mm

Thickness of Inner Hull and Transverse BHD Elastomer = 50 mm

Density of Elastomer = 1.107 g/cm³

Note: All dollar values are in US currency.

Strake Scantlings

Bottom Shell Plate Keel (4 Plates per Cargo Length)

Plates	0	0	900	0	7.5	900	64.80	0.2430	1.9076	0.81	0.897
4	0	0	900	0	7.5	900	64.80	0.2430	1.9076	0.81	0.897
8	900	0	3000	0	6.5	2100	151.20	0.4914	3.8575	1.89	2.092
12	3000	0	6000	0	6.5	3000	216.00	0.7020	5.5107	2.70	2.989
16	6000	0	9000	0	6.5	3000	216.00	0.7020	5.5107	2.70	2.989
20	9000	0	12000	0	6.5	3000	216.00	0.7020	5.5107	2.70	2.989
24	12000	0	14000	0	6.5	2000	144.00	0.4680	3.6738	1.80	1.993
28	0	0	-900	0	7.5	900	64.80	0.2430	1.9076	0.81	0.897
32	-900	0	-3000	0	6.5	2100	151.20	0.4914	3.8575	1.89	2.092
36	-3000	0	-6000	0	6.5	3000	216.00	0.7020	5.5107	2.70	2.989
40	-6000	0	-9000	0	6.5	3000	216.00	0.7020	5.5107	2.70	2.989
44	-9000	0	-12000	0	6.5	3000	216.00	0.7020	5.5107	2.70	2.989
48	-12000	0	-14000	0	6.5	2000	144.00	0.4680	3.6738	1.80	1.993
48 Total Number of Pieces per section											

Bilge (4 Plates per Cargo Length)

Plates	14000	0	15414	585	6.5	1570.80	113.10	0.3676	2.8854	1.41	1.565
52	14000	0	15414	585	6.5	1570.80	113.10	0.3676	2.8854	1.41	1.565
56	15414	585	16000	2000	6.5	1570.80	113.10	0.3676	2.8854	1.41	1.565
60	-14000	0	-15414	585	6.5	1570.80	113.10	0.3676	2.8854	1.41	1.565
64	-15414	585	-16000	2000	6.5	1570.80	113.10	0.3676	2.8854	1.41	1.565
16 Total Number of Pieces per section											

Inner Bottom (4 Plates per Cargo Length)

Element	Start	End	Length	Area	Volume	Weight	Stress	Strain	Factor	Value
68	0	2000	3000	2000	7.0	3000	216.00	0.7560	5.9346	2.70
72	3000	2000	6000	2000	7.0	3000	216.00	0.7560	5.9346	2.70
76	6000	2000	9000	2000	7.0	3000	216.00	0.7560	5.9346	2.70
80	9000	2000	12000	2000	7.0	3000	216.00	0.7560	5.9346	2.70
84	0	2000	-3000	2000	7.0	3000	216.00	0.7560	5.9346	2.70
88	-3000	2000	-6000	2000	7.0	3000	216.00	0.7560	5.9346	2.70
92	-6000	2000	-9000	2000	7.0	3000	216.00	0.7560	5.9346	2.70
96	-9000	2000	-12000	2000	7.0	3000	216.00	0.7560	5.9346	2.70

32 Total Number of Pieces per section

Centerline Girder (2 Plates per Cargo Length)

Element	Start	End	Length	Area	Volume	Weight	Stress	Strain	Factor	Value
98	0	0	0	2000	14.0	2000.00	72.00	0.50	3.956	

2 Total Number of Pieces per section

Side Girders (2 Plates per Cargo Length)

Element	Start	End	Length	Area	Volume	Weight	Stress	Strain	Factor	Value
100	3000	0	3000	2000	12.0	2000	72.00	0.43	3.391	
102	6000	0	6000	2000	12.0	2000	72.00	0.43	3.391	
104	9000	0	9000	2000	12.0	2000	72.00	0.43	3.391	
106	12000	0	12000	2000	12.0	2000	72.00	0.43	3.391	
108	-3000	0	-3000	2000	12.0	2000	72.00	0.43	3.391	
110	-6000	0	-6000	2000	12.0	2000	72.00	0.43	3.391	
112	-9000	0	-9000	2000	12.0	2000	72.00	0.43	3.391	
114	-12000	0	-12000	2000	12.0	2000	72.00	0.43	3.391	

16 Total Number of Pieces per section

Side Shell and Sheerstrake (4 Plates per Cargo Length)

Element	Start	End	Length	Area	Volume	Weight	Stress	Strain	Factor	Value
118	16000	2000	16000	4000	6.25	2000	144.00	0.4500	3.5325	1.80
122	16000	4000	16000	7000	6.25	3000	216.00	0.6750	5.2988	2.70
126	16000	7000	16000	10000	6.25	3000	216.00	0.6750	5.2988	2.70
130	16000	10000	16000	13000	6.25	3000	216.00	0.6750	5.2988	2.70
134	16000	13000	16000	16000	6.25	3000	216.00	0.6750	5.2988	2.70
138	16000	16000	14000	20000	6.25	5141.59	370.19	1.1569	9.0813	4.63
142	-16000	2000	-16000	4000	6.25	2000.00	144.00	0.4500	3.5325	1.80
146	-16000	4000	-16000	7000	6.25	3000.00	216.00	0.6750	5.2988	2.70
150	-16000	7000	-16000	10000	6.25	3000.00	216.00	0.6750	5.2988	2.70
154	-16000	10000	-16000	13000	6.25	3000.00	216.00	0.6750	5.2988	2.70
158	-16000	13000	-16000	16000	6.25	3000.00	216.00	0.6750	5.2988	2.70
162	-16000	16000	-14000	20000	6.25	5141.59	370.19	1.1569	9.0813	4.63

48 Total Number of Pieces per section

Strength Deck (4 Plates per Cargo Length)

Item	Plates	Plates	Plates	Plates	Plates	Plates	Plates	Plates	Plates	Plates	Plates
166	0	21500	2977	21128	6.0	3000.15	216.01	0.6480	5.0871	2.70	2.989
170	2977	21128	5954	20756	6.0	3000.15	216.01	0.6480	5.0871	2.70	2.989
174	5954	20756	8931	20384	5.8	3000.15	216.01	0.6210	4.8751	2.70	2.989
178	8931	20384	11908	20013	5.8	3000.03	216.00	0.6210	4.8749	2.70	2.989
182	11908	20013	14000	20000	5.8	2092.04	150.63	0.4331	3.3995	1.88	2.084
186	0	21500	2977	21128	6.0	3000.15	216.01	0.6480	5.0871	2.70	2.989
190	2977	21128	5954	20756	6.0	3000.15	216.01	0.6480	5.0871	2.70	2.989
194	5954	20756	8931	20384	5.8	3000.15	216.01	0.6210	4.8751	2.70	2.989
198	8931	20384	11908	20013	5.8	3000.03	216.00	0.6210	4.8749	2.70	2.989
202	11908	20013	14000	20000	5.8	2092.04	150.63	0.4331	3.3995	1.88	2.084

40 Total Number of Pieces per section

Longitudinal Bulkhead (4 Plates per Cargo Length)

Item	Plates	Plates	Plates	Plates	Plates	Plates	Plates	Plates	Plates	Plates	Plates
206	12000	2000	13999	4002	6.3	2829.14	203.70	0.6366	4.9970	2.55	2.819
210	13999	4002	13999	7000	6.3	2998.00	215.86	0.6746	5.2952	2.70	2.987
214	13999	7000	13999	10000	5.8	3000.00	216.00	0.6210	4.8749	2.70	2.989
218	13999	10000	13999	13000	5.5	3000.00	216.00	0.5940	4.6629	2.70	2.989
222	13999	13000	13996	16003	5.3	3003.00	216.22	0.5676	4.4554	2.70	2.992
226	13996	16003	12000	18002	5.0	2824.89	203.39	0.5085	3.9916	2.54	2.814
230	12000	18002	12000	20001	5.0	1999.00	143.93	0.3598	2.8246	1.80	1.992
234	-12000	2000	-13999	4002	6.3	2829.14	203.70	0.6366	4.9970	2.55	2.819
238	-13999	4002	-13999	7000	6.3	2998.00	215.86	0.6746	5.2952	2.70	2.987
242	-13999	7000	-13999	10000	5.8	3000.00	216.00	0.6210	4.8749	2.70	2.989
246	-13999	10000	-13999	13000	5.5	3000.00	216.00	0.5940	4.6629	2.70	2.989
250	-13999	13000	-13996	16003	5.3	3003.00	216.22	0.5676	4.4554	2.70	2.992
254	-13996	16003	-12000	18002	5.0	2824.89	203.39	0.5085	3.9916	2.54	2.814
258	-12000	18002	-12000	20001	5.0	1999.00	143.93	0.3598	2.8246	1.80	1.992

56 Total Number of Pieces per section

Stringers (2 Plates per Cargo Length)

Item	Plates	Plates	Plates	Plates	Plates	Plates	Plates	Plates	Plates
260	16000	4000	13999	4000	10.0	2001	72.04	0.36	2.827
262	16000	7000	13999	7000	10.0	2001	72.04	0.36	2.827
264	16000	10000	13999	10000	10.0	2001	72.04	0.36	2.827
266	16000	13000	13999	13000	10.0	2001	72.04	0.36	2.827
268	16000	16000	13999	16000	10.0	2001	72.04	0.36	2.827
270	-16000	4000	-13999	4000	10.0	2001	72.04	0.36	2.827
272	-16000	7000	-13999	7000	10.0	2001	72.04	0.36	2.827
274	-16000	10000	-13999	10000	10.0	2001	72.04	0.36	2.827
276	-16000	13000	-13999	13000	10.0	2001	72.04	0.36	2.827
278	-16000	16000	-13999	16000	10.0	2001	72.04	0.36	2.827

20 Total Number of Pieces per section

Floors and Transverse Elements (2 Floors/Tank-2 Plates/Floor) Height = 2000 mm

282	0	0	3000	0	6.0	3000.00	48.00	0.14	1.130	0.60	0.664
286	3000	0	6000	0	6.0	3000.00	48.00	0.14	1.130	0.60	0.664
290	6000	0	9000	0	6.0	3000.00	48.00	0.14	1.130	0.60	0.664
294	9000	0	12000	0	6.0	3000.00	48.00	0.14	1.130	0.60	0.664
298	12000	0	16000	0	6.0	N/A	105.13	0.32	2.476	1.31	1.455
302	16000	4000	16000	7000	6.0	3000.00	48.00	0.14	1.130	0.60	0.664
306	16000	7000	16000	10000	6.0	3000.00	48.00	0.14	1.130	0.60	0.664
310	16000	10000	16000	13000	6.0	3000.00	48.00	0.14	1.130	0.60	0.664
314	16000	13000	16000	16000	6.0	3000.00	48.00	0.14	1.130	0.60	0.664
318	16000	16000	14000	20000	6.0	N/A	105.13	0.32	2.476	1.31	1.455
322	0	18000	3000	18000	6.0	3000.00	79.50	0.24	1.872	0.99	1.100
326	3000	18000	6000	18000	6.0	3000.00	70.50	0.21	1.660	0.88	0.976
330	6000	18000	9000	18000	6.0	3000.00	61.50	0.18	1.448	0.77	0.851
334	9000	18000	12000	18000	6.0	3000.00	52.50	0.16	1.236	0.66	0.726
338	0	0	-3000	0	6.0	3000.00	48.00	0.14	1.130	0.60	0.664
342	-3000	0	-6000	0	6.0	3000.00	48.00	0.14	1.130	0.60	0.664
346	-6000	0	-9000	0	6.0	3000.00	48.00	0.14	1.130	0.60	0.664
350	-9000	0	-12000	0	6.0	3000.00	48.00	0.14	1.130	0.60	0.664
354	-12000	0	-16000	0	6.0	N/A	105.13	0.32	2.476	1.31	1.455
358	-16000	4000	-16000	7000	6.0	3000.00	48.00	0.14	1.130	0.60	0.664
362	-16000	7000	-16000	10000	6.0	3000.00	48.00	0.14	1.130	0.60	0.664
366	-16000	10000	-16000	13000	6.0	3000.00	48.00	0.14	1.130	0.60	0.664
370	-16000	13000	-16000	16000	6.0	3000.00	48.00	0.14	1.130	0.60	0.664
374	-16000	16000	-14000	20000	6.0	N/A	105.13	0.32	2.476	1.31	1.455
378	0	18000	-3000	18000	6.0	3000.00	79.50	0.24	1.872	0.99	1.100
382	-3000	18000	-6000	18000	6.0	3000.00	70.50	0.21	1.660	0.88	0.976
386	-6000	18000	-9000	18000	6.0	3000.00	61.50	0.18	1.448	0.77	0.851
390	-9000	18000	-12000	18000	6.0	3000.00	52.50	0.16	1.236	0.66	0.726

112 Total Number of Pieces per section

Transverse Bulkhead (2 per Tank - 2 Plates per T.B.)

L = 3000 mm

406	0	4000	3000	4000	6.0	3000.00	288.00	0.86	6.78	3.60	3.985
422	3000	4000	6000	4000	6.0	3000.00	288.00	0.86	6.78	3.60	3.985
438	6000	4000	9000	4000	6.0	3000.00	288.00	0.86	6.78	3.60	3.985
454	9000	4000	12000	4000	6.0	3000.00	288.00	0.86	6.78	3.60	3.985
470	12000	4000	14000	4000	6.0	2000.00	192.00	0.58	4.52	2.40	2.657
486	0	4000	-3000	4000	6.0	3000.00	288.00	0.86	6.78	3.60	3.985
502	-3000	4000	-6000	4000	6.0	3000.00	288.00	0.86	6.78	3.60	3.985
518	-6000	4000	-9000	4000	6.0	3000.00	288.00	0.86	6.78	3.60	3.985
534	-9000	4000	-12000	4000	6.0	3000.00	288.00	0.86	6.78	3.60	3.985
550	-12000	4000	-14000	4000	6.0	2000.00	192.00	0.58	4.52	2.40	2.657

160 Total Number of Pieces per section

Stools (4 Per T.B. - 2 Plates per Stool)**L = 2154 mm**

Item	Y	Z	X	Y	Y	Y	Y	Y	Y	Y	Y
558	0	4000	3000	4000	6.0	3000.00	103.40	0.31	2.43	1.29	1.43
566	3000	4000	6000	4000	6.0	3000.00	103.40	0.31	2.43	1.29	1.43
574	6000	4000	9000	4000	6.0	3000.00	103.40	0.31	2.43	1.29	1.43
582	9000	4000	12000	4000	6.0	3000.00	103.40	0.31	2.43	1.29	1.43
590	12000	4000	14000	4000	6.0	2000.00	32.00	0.10	0.75	0.40	0.44
598	0	4000	-3000	4000	6.0	3000.00	103.40	0.31	2.43	1.29	1.43
606	-3000	4000	-6000	4000	6.0	3000.00	103.40	0.31	2.43	1.29	1.43
614	-6000	4000	-9000	4000	6.0	3000.00	103.40	0.31	2.43	1.29	1.43
622	-9000	4000	-12000	4000	6.0	3000.00	103.40	0.31	2.43	1.29	1.43
630	-12000	4000	-14000	4000	6.0	2000.00	32.00	0.10	0.75	0.40	0.44

80 Total Number of Pieces per section**Transverse Bulkhead, Stool and Floor Vertical Framing Elements**

Item	Y	Z	X	Y	Y	Y	Y	Y	Y	Y
636	0	0	0	21500	14.0	896.00	51.31	0.36	2.82	
642	3000	0	3000	21125	12.0	896.00	50.00	0.30	2.35	
648	6000	0	6000	20750	12.0	896.00	48.69	0.29	2.29	
654	9000	0	9000	20375	12.0	896.00	47.38	0.28	2.23	
660	12000	0	12000	20000	12.0	896.00	46.06	0.28	2.17	
666	-3000	0	-3000	21125	12.0	896.00	50.00	0.30	2.35	
672	-6000	0	-6000	20750	12.0	896.00	48.69	0.29	2.29	
678	-9000	0	-9000	20375	12.0	896.00	47.38	0.28	2.23	
684	-12000	0	-12000	20000	12.0	896.00	46.06	0.28	2.17	

54 Total Number of Pieces per section**Transverse Bulkhead Horizontal Framing Elements****L = 3000 mm**

Item	Y	Z	X	Y	Y	Y	Y	Y	Y	Y
689	0	2000	3000	2000	10.0	896.00	26.88	0.13	1.06	
694	3000	2000	6000	2000	10.0	896.00	26.88	0.13	1.06	
699	6000	2000	9000	2000	10.0	896.00	26.88	0.13	1.06	
704	9000	2000	12000	2000	10.0	896.00	26.88	0.13	1.06	
709	0	2000	-3000	2000	10.0	896.00	26.88	0.13	1.06	
714	-3000	2000	-6000	2000	10.0	896.00	26.88	0.13	1.06	
719	-6000	2000	-9000	2000	10.0	896.00	26.88	0.13	1.06	
724	-9000	2000	-12000	2000	10.0	896.00	26.88	0.13	1.06	

40 Total Number of Pieces per section

Deck Girders

726	0	19500	0	21500	14.0	2000.00	57.60	0.40	3.17
728	3000	19125	3000	21125	12.0	2000.00	57.60	0.35	2.71
730	6000	18750	6000	20750	12.0	2000.00	57.60	0.35	2.71
732	9000	18375	9000	20375	12.0	2000.00	57.60	0.35	2.71
734	-3000	19125	-3000	21125	12.0	2000.00	57.60	0.35	2.71
736	-6000	18750	-6000	20750	12.0	2000.00	57.60	0.35	2.71
738	-9000	18375	-9000	20375	12.0	2000.00	57.60	0.35	2.71
14	Total Number of Pieces per section								

Quantities for Single Tank

Total Number of Pieces = 738

Total Area of Steel = 19306 m²Total Volume of Steel = 65.57 m³

Total Weight of Steel = 515.78 t

Volume of Elastomer = 211.45 m³

Weight of Elastomer = 234.07 t

TOTAL WEIGHT = 749.86 t**Quantities for All (7) Tanks**

Total Number of Pieces = 5166

Total Area of Steel = 135143 m²Total Volume of Steel = 458.99 m³

Total Weight of Steel = 3610.46 t

Volume of Elastomer = 1480.15 m³

Weight of Elastomer = 1638.52 t

TOTAL WEIGHT = 5249.0 t**Raw Material Handling and Preparation**

The unit cost figures used in the material handling and preparation are:

Steel	\$620 per tonne
Surface Preparation and Protection	\$11 per square meter
Stockyard Handling	\$30 per tonne

Note: A scrap rate of 10% was assumed.

RMHPS = (Tonnes of Steel) * 1.10 * 620 + (Surface Area) * 1.10 * 11

RMHPS Per Cargo Tank = 0.602 (in Millions)

Plate Cutting and Forming for Single Tank

Butt and Fillet Weld Calculations for Single Tank

Bottom Shell	4	9000	900.00	39.6	N/A
	8	9000	2100.00	44.4	N/A
	12	9000	3000.00	48.0	N/A
	16	9000	3000.00	48.0	N/A
	20	9000	3000.00	48.0	N/A
	24	9000	2000.00	44.0	N/A
	28	9000	900.00	39.6	N/A
	32	9000	2100.00	44.4	N/A
	36	9000	3000.00	48.0	N/A
	40	9000	3000.00	48.0	N/A
	44	9000	3000.00	48.0	N/A
	48	9000	2000.00	44.0	N/A
Bilge	52	9000	1570.80	42.3	N/A
	56	9000	1570.80	42.3	N/A
	60	9000	1570.80	42.3	N/A
	64	9000	1570.80	42.3	N/A
Inner Bottom	68	9000	3000.00	48.0	N/A
	72	9000	3000.00	48.0	N/A
	76	9000	3000.00	48.0	N/A
	80	9000	3000.00	48.0	N/A
	84	9000	3000.00	48.0	N/A
	88	9000	3000.00	48.0	N/A
	92	9000	3000.00	48.0	N/A
	96	9000	3000.00	48.0	N/A
CL Girder	98	9000	2000.00	4.0	72.0
Side Girders	100	9000	2000.00	4.0	72.0
	102	9000	2000.00	4.0	72.0
	104	9000	2000.00	4.0	72.0
	106	9000	2000.00	4.0	72.0
	108	9000	2000.00	4.0	72.0
	110	9000	2000.00	4.0	72.0
	112	9000	2000.00	4.0	72.0
	114	9000	2000.00	4.0	72.0
S. Shl & Shstrk	118	9000	2000.00	44.0	N/A
	122	9000	3000.00	48.0	N/A
	126	9000	3000.00	48.0	N/A
	130	9000	3000.00	48.0	N/A
	134	9000	3000.00	48.0	N/A
	138	9000	5141.59	56.6	N/A
	142	9000	2000.00	44.0	N/A
	146	9000	3000.00	48.0	N/A
	150	9000	3000.00	48.0	N/A
	154	9000	3000.00	48.0	N/A
	158	9000	3000.00	48.0	N/A
	162	9000	5141.59	56.6	N/A

Butt and Fillet Weld Calculations for Single Tank (Cont'd)

Strength Deck	166	9000	3000.15	48.0	N/A
	170	9000	3000.15	48.0	N/A
	174	9000	3000.15	48.0	N/A
	178	9000	3000.03	48.0	N/A
	182	9000	2092.04	44.4	N/A
	186	9000	3000.15	48.0	N/A
	190	9000	3000.15	48.0	N/A
	194	9000	3000.15	48.0	N/A
	198	9000	3000.03	48.0	N/A
	202	9000	2092.04	44.4	N/A
Longitud. BHD	206	9000	2829.14	47.3	N/A
	210	9000	2998.00	48.0	N/A
	214	9000	3000.00	48.0	N/A
	218	9000	3000.00	48.0	N/A
	222	9000	3003.00	48.0	N/A
	226	9000	2824.89	47.3	N/A
	230	9000	1999.00	44.0	N/A
	234	9000	2829.14	47.3	N/A
	238	9000	2998.00	48.0	N/A
	242	9000	3000.00	48.0	N/A
	246	9000	3000.00	48.0	N/A
	250	9000	3003.00	48.0	N/A
	254	9000	2824.89	47.3	N/A
	258	9000	1999.00	44.0	N/A
Stringers	260	9000	2001.00	4.0	72.0
	262	9000	2001.00	4.0	72.0
	264	9000	2001.00	4.0	72.0
	266	9000	2001.00	4.0	72.0
	268	9000	2001.00	4.0	72.0
	270	9000	2001.00	4.0	72.0
	272	9000	2001.00	4.0	72.0
	274	9000	2001.00	4.0	72.0
	276	9000	2001.00	4.0	72.0
	278	9000	2001.00	4.0	72.0
Floors and Trans. Elements	282	3000.00	2000.00	8.0	48.0
	286	3000.00	2000.00	8.0	48.0
	290	3000.00	2000.00	8.0	48.0
	294	3000.00	2000.00	8.0	48.0
	298	13970.02	2000.00	8.0	79.8
	302	3000.00	2000.00	8.0	48.0
	306	3000.00	2000.00	8.0	48.0
	310	3000.00	2000.00	8.0	48.0
	314	3000.00	2000.00	8.0	48.0
	318	13970.02	2000.00	8.0	79.8
	322	3000.00	2375.00	14.8	24.2
	326	3000.00	2750.00	16.3	24.2
	330	3000.00	3125.00	17.8	24.2
	334	3000.00	3500.00	19.3	24.2

Butt and Fillet Weld Calculations for Single Tank (Cont'd)

Floors and Trans. Elements	338	3000.00	2000.00	8.0	48.0
	342	3000.00	2000.00	8.0	48.0
	346	3000.00	2000.00	8.0	48.0
	350	3000.00	2000.00	8.0	48.0
	354	13970.02	2000.00	8.0	79.8
	358	3000.00	2000.00	8.0	48.0
	362	3000.00	2000.00	8.0	48.0
	366	3000.00	2000.00	8.0	48.0
	370	3000.00	2000.00	8.0	48.0
	374	13970.02	2000.00	8.0	79.8
	378	3000.00	2375.00	14.8	24.2
	382	3000.00	2750.00	16.3	24.2
	386	3000.00	3125.00	17.8	24.2
	390	3000.00	3500.00	19.3	24.2
Transverse BHD	406	3000.00	3000.00	96.0	N/A
	422	3000.00	3000.00	96.0	N/A
	438	3000.00	3000.00	96.0	N/A
	454	3000.00	3000.00	96.0	N/A
	470	3000.00	2000.00	80.0	96.0
	486	3000.00	3000.00	96.0	N/A
	502	3000.00	3000.00	96.0	N/A
	518	3000.00	3000.00	96.0	N/A
	534	3000.00	3000.00	96.0	N/A
	550	3000.00	2000.00	80.0	96.0
Stools	558	2154.07	3000.00	35.2	24.0
	566	2154.07	3000.00	35.2	24.0
	574	2154.07	3000.00	35.2	24.0
	582	2154.07	3000.00	35.2	24.0
	590	2154.07	2000.00	29.2	45.3
	598	2154.07	3000.00	35.2	24.0
	606	2154.07	3000.00	35.2	24.0
	614	2154.07	3000.00	35.2	24.0
	622	2154.07	3000.00	35.2	24.0
	630	2154.07	2000.00	29.2	45.3
Transverse BHD Vert Stiff Elem	636	3000.00	896.00	4.5	95.5
	642	3000.00	896.00	4.5	94.0
	648	3000.00	896.00	4.5	92.5
	654	3000.00	896.00	4.5	91.0
	660	3000.00	896.00	4.5	89.5
	666	3000.00	896.00	4.5	94.0
	672	3000.00	896.00	4.5	92.5
	678	3000.00	896.00	4.5	91.0
	684	3000.00	896.00	4.5	89.5

Butt and Fillet Weld Calculations for Single Tank (Cont'd)

Transverse BHD	689	3000.00	896.00	N/A	77.9
Horz Stiff Elem	694	3000.00	896.00	N/A	77.9
	699	3000.00	896.00	N/A	77.9
	704	3000.00	896.00	N/A	77.9
	709	3000.00	896.00	N/A	77.9
	714	3000.00	896.00	N/A	77.9
	719	3000.00	896.00	N/A	77.9
	724	3000.00	896.00	N/A	77.9
Deck Girders	726	7200.00	2000.00	2.0	36.8
	728	7200.00	2000.00	2.0	36.8
	730	7200.00	2000.00	2.0	36.8
	732	7200.00	2000.00	2.0	36.8
	734	7200.00	2000.00	2.0	36.8
	736	7200.00	2000.00	2.0	36.8
	738	7200.00	2000.00	2.0	36.8

The unit cost figures used in the plate cutting and forming are:

Shop Handling (Large Pieces)	\$19 per tonne
Shop Handling	\$11 per piece
Cutting (Edges for Butt Weld)	\$2 per meter of butt weld
Cutting (Edges for Fillet Weld)	\$1 per meter of fillet weld
Forming	\$4 per piece

Note: A scrap rate of 10% was assumed.

$$PCFS = (\text{Tonnes of Steel}) * 1.10 * 19 + (\text{Number of Pieces}) * 1.10 * 15 + (\text{Meter of Butt Weld}) * 2 + (\text{Meter of Fillet Weld}) * 1$$

$$PCFS \text{ Per Cargo Tank} = 0.037 \quad (\text{in Millions})$$

Fabrication Costs

Complexity Factor # 1 (Used In Considering Welding Access)

$$CF_1 = \frac{1}{e^{\left(\frac{x-600}{x}\right)^{\frac{100}{x}}}}$$

Note: If $x > 1000$ mm then $CF_1 = 0.9$

$$x = 1876 \text{ mm} \quad \text{distance between parallel welds within a confined space}$$

$$CF_1 = 0.391 \quad \text{use } 0.90$$

Complexity Factor # 2 (Considering Painting Access, Surface Orientation, and Quality Assurance)

$$CF_2 = \frac{2 * AC + SO}{3}$$

$$CF_2 = 0.930$$

where $AC = \left(\frac{A^2 + B^2}{2} \right)^{0.1}$
 $AC = 0.8970$

AC = access related term

$$A = \frac{1}{e^{\left(\frac{x-1000}{1000} \right)^{\frac{1000}{x}}}}$$

$A = 0.3941$

0.3941

0.3941

0.3941

0.2837

0.2837

0.2837

0.2837

horiz. dist. b/w side shell and long bh stiffener group #2

horiz. dist. b/w side shell and long bh stiffener group #3

horiz. dist. b/w side shell and long bh stiffener group #4

horiz. dist. b/w side shell and long bh stiffener group #5

horizontal distance b/w CL girder and side girder #1

horizontal distance b/w side girder #1 and side girder #2

horizontal distance b/w side girder #2 and side girder #3

horizontal distance b/w side girder #3 and side girder #4

$$B = \frac{1}{e^{\left(\frac{y-1500}{1500} \right)^{\frac{1500}{y}}}}$$

$B = 0.7203$

0.7203

0.7203

0.7203

0.3679

0.3679

0.3679

0.3679

vert dist b/w inner and outer bottom stiffener group #1

vert dist b/w inner and outer bottom stiffener group #2

vert dist b/w inner and outer bottom stiffener group #3

vert dist b/w inner and outer bottom stiffener group #4

vert distance between stringer #1 and #2

vert distance between stringer #2 and #3

vert distance between stringer #3 and #4

vert distance between stringer #4 and #5

$x = 1875 \text{ mm}$

1875 mm

1875 mm

1875 mm

3000 mm

3000 mm

3000 mm

3000 mm

horiz. dist. b/w side shell and long bh

horiz. dist. b/w side shell and long bh

horiz. dist. b/w side shell and long bh

horiz. dist. b/w side shell and long bh

horizontal distance b/w CL girder and side girder #1

horizontal distance b/w side girder #1 and side girder #2

horizontal distance b/w side girder #2 and side girder #3

horizontal distance b/w side girder #3 and side girder #4

$y = 1873 \text{ mm}$

1873 mm

1873 mm

1873 mm

1873 mm

3000 mm

3000 mm

3000 mm

3000 mm

vert dist b/w inner and outer bottom

vert dist b/w inner and outer bottom

vert dist b/w inner and outer bottom

vert dist b/w inner and outer bottom

vert distance between stringer #1 and #2

vert distance between stringer #2 and #3

vert distance between stringer #3 and #4

vert distance between stringer #4 and #5

$$SO = \left(\frac{25}{\frac{\text{Surface Area}}{\# \text{ of Pieces}}} \right)^{0.1}$$

SO = surface orientation term

$$SO = 0.9955$$

Volume of Weld

Average Plate Thickness

$$t_p = 7.12 \text{ mm}$$

Fillet Weld Throat Thickness

$$w = t_p * wf * 2^{0.5}$$

$$w = 4.43 \text{ mm}$$

$$\text{where } wf = 0.44 \text{ weld factor}$$

Area of Fillet Weld

$$A_{fw} = 0.5w^2$$

$$A_{fw} = 9.8 \text{ mm}^2$$

Volume of Fillet Weld

$$V_{fw} = A_{fw} * L_{fw}$$

$$L_{fw} = 4833.3 \text{ m}$$

$$V_{fw} = 4.750E+04 \text{ cm}^3$$

Area of Butt Weld

$$A_{bw} = t_p * s + 0.166t_p^2 * 2$$

$$A_{bw} = 38.29 \text{ mm}^2$$

$$\text{where } s = 3 \text{ mm}$$

Volume of Butt Weld

$$V_{bw} = A_{bw} * L_{bw}$$

$$L_{bw} = 4510.9 \text{ m}$$

$$V_{bw} = 1.727E+05 \text{ cm}^3$$

Number of Intersections (Where Longitudinal and Transverse Cross)

Girders

Each side girder and the CL girder will intersect 2 floors (2 plates/floor) twice:

$$9 * 2 * 2 = 72$$

Deck Girders

Each deck girder with intersect 2 floors:

$$7 * 2 = 14$$

Stringers

Each stringer will intersect 2 floors (2 plates/floor) twice:

$$10 * 2 * 2 * 2 = 80$$

Transverse Bulkhead Vertical/Horizontal Stiffening Elements

Each vertical stiffening element will intersect a horizontal element twice:

$$9 * 7 * 2 = 126$$

TOTAL NUMBER OF INTERSECTION 292

The unit cost figures used in the fabrication are:

Weld Material	\$0.05 per cubic centimeter of weld
Paint and Coating Materials	\$12 per square meter of surface area
Assembly Fit-Up	\$90 per piece
Welding	\$30 per piece
Butt Welding	\$0.60 per cubic centimeter of weld
Fillet Welding	\$0.60 per cubic centimeter of weld
Intersection Reinforcement	\$30 per intersection
Paint and Coating Application	\$1 per square meter of surface area
Quality Assurance of Welds	\$0.30 per meter of weld

$$\text{Fab\$} = (\# \text{ of Pieces}) * 120 + (\text{Surface Area}) * 12 + (\text{Volume of Weld}) * 0.65 * CF_1 + (\# \text{ of Intersections}) * 30 \\ + (\text{Surface Area}) * 1 * CF_2 + (\text{Length of Weld}) * 0.3 * CF_2$$

Fab\$ = \$0.46 (In Millions)

Elastomer Cost

Elastomer Cost = 4.72 \$/kg

Total Weight of Elastomer Per Cargo Tank = 234.07 ton

Total Cost of Elastomer Per Cargo Tank = 1.1 (In Millions)

TOTAL OVERALL COST PER CARGO TANK \$2.20 (In Millions)