Millennium Class Tanker Structural Design – From Owner Experience to Shipyard Launching Ways

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\textbf{ABSTRACT}

Three 125,000 DWT double hull tankers are currently under construction for ARCO Marine, Inc. at Litton Avondale Industries in New Orleans, LA. These Millennium Class tankers are being built to transport crude oil from Valdez, AK to Cherry Point, WA. The design satisfies the requirements of OPA 90 and incorporates a unique structural design philosophy intended to enhance the structural performance of the vessel. This paper will illustrate how the Owner’s experience with previous vessels in Gulf of Alaska trade is reflected in the structural design of the new ships. The human elements of safety, inspection and maintenance are discussed and the influence of these factors on the structural arrangement is highlighted. In concert with these human factors are structural design improvements that have been implemented to specifically address fatigue and stress cracking with the intent of reducing repair requirements. These topics are presented in a discussion that follows the structural design from concept, through design and analysis and into the construction of the vessels. Lessons learned throughout the process are presented.

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FOREWORD

The goal of this paper is to broadly illustrate the process by which the structure for the Millennium Class tankers was conceived, designed and constructed. Design and construction of the vessels are addressed from both a technical and a non-technical viewpoint. The project has been an exercise in philosophy, engineering and project management, and it is our hope to provide an insight to the successes and pitfalls on a first-of-class ship construction project.

INTRODUCTION

ARCO Marine, Inc. (AMI) is a wholly owned subsidiary of the Atlantic Richfield Company (ARCO). AMI is the marine operating company responsible for carrying ARCO’s crude oil produced on Alaska’s North Slope to ARCO’s refineries on the U.S. West Coast. Crude oil is loaded at the terminal in Valdez, Alaska and transported to refineries in Cherry Point, Washington and Los Angeles, California.

AMI currently operates a fleet of six tankers (five crude, one product) ranging in size from 50,000 DWT to 265,000 DWT. In 1998 and 1999, the Oil Pollution Act of 1990 (OPA '90) forced the retirement of three AMI 120,000 DWT vessels. These single hulled ships moved crude oil from Valdez to Cherry Point since the opening of the Trans-Alaska Pipeline Service (TAPS) in 1978. OPA ’90 will also require that AMI retire two 265,000 DWT vessels in the year 2000.

In the early 1990’s AMI began to consider options that would replace tonnage lost due to OPA ’90. AMI explored all options including retrofitting existing tankers with double hulls, fitting completely new forebodies, and building new vessels. Ultimately the decision was made to construct new vessels, the Millennium Class Tankers.

The full effort towards preliminary design of the Millennium Class tankers commenced in 1996. In July 1997 a contract was signed with Avondale Industries Shipyard Division, now Litton Avondale Industries. Detailed design commenced immediately and the keel was laid in May 1998. The first ship, the ARCO ENDEAVOUR, is scheduled to deliver in the fourth quarter of 2000. The contract currently provides for a total of three Millennium Class tankers.

Decision for Newbuilding

Maintaining the size of the AMI fleet necessitated plans for replacement of tonnage phased out by OPA ’90. Initial exploration into possibilities for obtaining double-hulled vessels yielded three options; retrofit of double hull to an existing ship, replacement of a complete forebody, or new building. Retrofit of a double hull into existing vessels was quickly dismissed. Cargo capacity lost in retrofitting an inner hull to an exiting 120,000 DWT vessel was impractical. It was also understood that the outer hull, while in good structural condition, would remain as part of the hull after 20+ years of Alaskan trade.

Forebody replacement presented a more promising option than retrofit as it provided completely new structure in the cargo block. However, while more desirable from a structural standpoint, the loss in tonnage from the previous 120,000 DWT vessels would still be incurred. Lengthening of the vessels was considered but forward visibility limitations and an increased length to depth ratio limited the feasibility of this additional modification.

The forebody option would also have required the retention of an inefficient steam propulsion plant. As a major retrofit, forebody replacement would necessitate installation of new cargo control and gauging systems as well as new deck machinery. Along the same lines, it would be desirable to upgrade the navigational systems to more modern bridge equipment. In the end, with safety, efficiency and economic consideration given careful attention the forebody option was dismissed and the decision was made to pursue new-build ships.

Design Team

AMI embarked in pursuit of a new design by bringing together an experienced and highly qualified structural design team. From the early stages AMI began to work closely with a number of consultants to ensure that the base of experience was extended as widely as possible. The main consultants, John J. McMullen Associates, Inc. (JJMA), MCA Engineers, Inc. (MCA), and Herbert Engineering Corp. served as collaborative design consultants throughout the initial phases of the project.

The intent of this team approach was to draw from unique skills that each consultant brought to the table. JJMA served as the primary consultant and provided broad-based naval architecture, marine engineering and cost estimating expertise as well as recent international experience with double-hull tanker design programs. MCA contributed with a strong background in structural design analysis and significant historical experience with vessels in the existing AMI and Alaskan tanker fleet. Herbert Engineering provided expertise in specific issues relating to stability, subdivision, and other factors such as ballast exchange. This team was responsible for the general development of the Contract Specification and continued to work together throughout the design and construction phase.

In preparation for contract signing, AMI chose to approach the final stages of preliminary design as a teaming effort with the consultants and Avondale Industries Shipyard Division, now Litton-Avondale...
Fig. 1 Inboard Profile, Outboard Profile, Bow and Stern Views

Shipyard. At the time of these meetings the concept and preliminary design had been developed to clearly represent Owner requirements. The goal of teaming with the Shipyard was to ensure Owner requirements and Shipyard capabilities were compatible and fully incorporated in the detail design.

STRUCTURAL DESIGN PHILOSOPHY

A number of key issues relating to the structural design were identified following the decision to pursue construction of new ships. Safety, reliability, proven design, necessary fatigue life and maintainability were all cited as issues to be addressed in the design process. Strong consideration was given to the fact that the trade routes served by the AMI crude oil tankers are among the most severe operating environments in the world, and require a significant amount of time spent in the waters of the Gulf of Alaska. Features specific to AMI were also to be incorporated into the design based on 20 years of experience transporting crude oil in Gulf of Alaska waters.

The design centered on a belief that designing for prevention is the safest and most economical approach in addressing the long-term operation of tankers. In short, the primary theme of process was “an ounce of prevention is worth a pound of cure.” The overall intent was not to build a ship that just met the Rules of Class. The intent was to construct a vessel that would meet or exceed the Rules as required, to a degree that the vessel became uniquely suited to meet the Owner’s need for extended service in a severe environment. Most importantly, it was felt that future problems could be avoided by turning to past experience as a learning tool. Past experience with ship structure became a key factor in development of the detail design [1-4].

Learning from Experience

Contributing greatly to AMI’s understanding of fatigue cracking and monitoring of ship structure was the United States Coast Guard’s Critical Area Inspection Program (CAIP) [5]. Existing AMI vessels had been operating under this program since its inception in 1991. As a result, AMI had developed a clear plan for the “management” of structure on existing vessels.
Significant amounts of data have been collected and AMI continues to support a proactive, rather than reactive, response to fatigue issues in the structure of its existing ships.

Historically, AMI’s database of experience with structural details extends back through 1987. Locations, lengths and types of fractures were documented. Patterns of structural fatigue cracking were clearly identified and problems were remedied with the development of structural fixes through finite element analysis. Participation in this program has undeniably led to greater understanding of designing for fatigue in ship structure.

Also helpful in determining potential problem areas were the various diverse experiences of the design team members. This experience combined with informational resources published by the industry led to a better understanding of the past performance of a variety of structural arrangements. Particularly noteworthy among published data are publications by the Tanker Structural Cooperative Forum (TSCF) [6,7], Oil Companies International Marine Forum (OCIMF) [8] and Lloyd’s Register [9].

“Ground Rules”

For the Millennium Class tankers, the structural design was to be developed based on extensive engineering. AMI required that the ships be designed for a 30-year fatigue life. This included not only compliance with ABS’ SafeHull A and B programs, but also finite element analysis, structural fatigue analysis and compliance with ABS’ Dynamic Loading Approach (DLA) [10,11]. During concept design preliminary structural details were studied extensively in areas historically known to experience structural fatigue cracking. This preliminary analysis was completed prior to contract signing.

Based on AMI’s Alaskan operating experiences, a number of design criteria were established for the structure. The vessels were to be built primarily with mild steel, with little or no reduction in scantlings. Where high strength steel was used margins above regulatory allowable minimums were typically applied. Critical structural areas were studied extensively by finite element analysis, dynamic load analysis, and spectral fatigue analysis. This conservative approach led to a number of decisions that resulted in a rugged yet functional design.
Addressing Fatigue in Structural Design

One of the most significant conceptual efforts made in the design was to address fatigue in structural details. Clearly, it is acknowledged that design and analysis cannot yield a zero crack condition. However, due diligence and attention to known problems can yield a greater understanding of those structural details to avoid and those details to incorporate. In some cases, details that performed poorly in previous vessels (e.g. mushroom shaped cutouts at erection joints) were redesigned for acceptable performance in new vessels.

The trade route to be traveled by the Millennium Class tankers presents a somewhat unique fatigue life profile. Nearly all of the at-sea operation of the vessels will be in North Pacific waters. Port time spent during loading and discharge is minimal and each ship will average a round trip about every 10.5 days. To illustrate the exclusivity of trade and frequency of voyages, the 120,000 DWT vessels recently retired each completed over 600 voyages to Valdez in approximately 20 years of operation.

CONCEPT DESIGN

Size of Vessel

The decisive commitment to build new vessels turned the focus towards clear definition of a design philosophy under which the vessels would be built [12]. Size of the vessel was the first issue to be addressed. The primary trade route for the proposed Millennium Class tankers is to be between Valdez, Alaska and Cherry Point, Washington. Because this trade extends into Puget Sound, Washington a deadweight limit of 125,000 DWT is mandated. This restriction, imposed by U.S. Coast Guard regulations excludes tankers of greater than 125,000 DWT from delivering oil to U.S. ports in Puget Sound, Washington [13].

While USCG regulations will ultimately govern the deadweight of ships entering into Puget Sound the overall cargo capacity of the Millennium Class tankers was set at one million barrels. This corresponds to roughly 137,000 DWT of Alaskan North Slope crude oil. The choice was made to proceed with a vessel of this size in an effort to enhance its overall trading versatility during the planned 30-year service life. Thus, the vessels have been designed to a scantling deadweight of 137,000 MT but will operate at a deadweight of 125,000 LT when delivering crude oil into the Puget Sound. Table 1 provides further Principal Characteristic information.

Cargo Block Structural Configuration

The overall vessel and tank configurations are shown in Figures 1 and 2. The cargo tanks are arranged six tanks long by two tanks across with two slop tanks located just aft of the No. 6 cargo tanks. The ballast tank arrangement includes six pairs of J-tanks with the No. 6 tanks extending aft beneath the slop tanks. It was found that this arrangement provided an optimum operational configuration in terms of intact and damaged stability.

The Midship Section is shown in Figure 3. Transverse web frame spacing is 3.96 meters and typical longitudinal spacing is 862 millimeters. Web frame structure is balanced on either side of the centerline bulkhead with longitudinal stiffening on the starboard side of the bulkhead. Main longitudinal girders are the centerline vertical keel and girders port and starboard beneath the lower hopper radius. Two horizontal stringers support the transverse bulkheads in the cargo block and are shown in Figure 4. Five longitudinal stringers are included in the ballast tank wing walls and the second and fourth stringers coincide with the cargo tank horizontal stringers.

Of special note is that a continuous, watertight, full height, centerline bulkhead runs below the Upper Deck (strength deck) from the collision bulkhead to the transom. The sole exception to the tightness of this bulkhead is found at the athwartship access in the cargo pump room. Otherwise, the centerline bulkhead divides the cargo block into the port and starboard ballast and cargo tanks and the engine room machinery spaces into two completely separate and redundant engine rooms.

The after portion of the centerline bulkhead, watertight and insulated to an A-60 standard, is also continuous in the machinery spaces and engine room casing above the Upper Deck and continues upward to the top of the stack. Three sliding, watertight A-60 doors, normally closed, are found in the machinery spaces and are available for athwartship access. The continuity of the centerline bulkhead continues forward of the collision bulkhead to the stem but is not watertight in this area.

Table 1 Principal Characteristics

<table>
<thead>
<tr>
<th>Characteristics</th>
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<tbody>
<tr>
<td>Length, Overall</td>
<td>272.69 m</td>
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<tr>
<td>Length, Between Perpendiculars</td>
<td>258.16 m</td>
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<tr>
<td>Beam, Molded</td>
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<td>Depth, Molded (at Centerline)</td>
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<td>16.31 m</td>
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<td>17.50 m</td>
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<td>160,778 MT</td>
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<td>Displacement, Scantling</td>
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<tr>
<td>Cargo Capacity, Scantling</td>
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<tr>
<td>Net Tonnage</td>
<td>36,299 MT</td>
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<tr>
<td>Block Coefficient, Design Draft</td>
<td>0.830</td>
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<tr>
<td>Design Speed</td>
<td>16.5 knots</td>
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</table>
"Hopper" Design

A significant research effort was made during the preliminary design to explore possible variations on the "hopper" design. The "hopper" is the area of sloped plating at the outboard lower corner of the cargo tanks. It was known that the most critical area of design in this configuration is found in the lower hopper transition.

Options for the connection at the lower hopper included use of a cruciform weldment, a cruciform casting, continuation of the inner bottom plating past the hopper to the side shell, or a radius hopper plate supported by a longitudinal girder. The final decision was to incorporate a radius plate at the corners of the hopper. Rationale for this decision included finite element analysis, past structural experience documented in the TSCF and Lloyd's publications, ability to hold construction tolerances, and overall stiffening arrangements.

Forebody and Afterbody Structure

The innerbottom plating and the five-stringers in the ballast tanks transition into the flats in the engine room. These levels also extend forward into the bow thruster and forepeak areas. The tank top structure carries aft as the flats of the Pump Room and the lower Engine Room. The major stringer at 9840 mm above baseline (ABL) transitions into the upper Engine Room and Purifier Rooms. The major stringer at 17120 mm ABL carries aft and forms the machinery flat for much of the auxiliary machinery, the Engine Room workshop and a large spare parts storage area. Finally, the ballast tank stringer at 20760 mm ABL slopes upward approximately one meter and forms the flats in the port and starboard Machinery Control Rooms.

HUMAN FACTORS – SAFETY, INSPECTION, AND MAINTENANCE

Special attention was given to the arrangement of structure as it pertains to the human element in design. Specific issues that were addressed in the cargo block included emergency removal of personnel, means of standard access and egress from tanks, ability to adequately ventilate tanks, and access for tank inspection...
and maintenance. In the machinery room consideration was given to equipment removal and maintenance.

Fig. 4 Transverse Bulkhead Stringer at Midship

Safe Access and Ventilation

Features included in the ballast tank design allow for safe access and proper ventilation during tank entry procedures. IMO and SOLAS guidelines for access in tanks were rigorously applied as were ASTM standard practices regarding ladder and platform construction [14-16]. Inclined ladders are located at the forward outboard corners of the ballast tanks and provide primary access. A vertical access trunk with a vertical ladder is located at the aft inboard corner of the tank. Finally, at the aft outboard corner of the tank is a second 600 mm x 800 mm clear opening for personnel and stretcher removal. This opening extends through each stinger level to the bottom of the tank.

Cross-ventilation is to be provided by locating portable fans at the vertical trunk or the personnel removal hatches. An additional provision for ventilation is included with the attachment of a ventilation fan to a branch line on the ballast main that may be used to ventilate back through the suction/fill lines into the tanks. Bolted tank cleaning openings are provided in every other frame bay to facilitate cleaning of sediment from the tanks or, in the event of a leakage from the cargo tanks into the ballast tanks, the cleaning of oil from the tank via portable tank washing machines.

Access into cargo tanks is provided via a hatch at the aft end of the tank. A vertical ladder provides access down to the first intermediate platform. This ladder is hinged so that it may be moved clear of a 600 mm x 800 mm clear drop opening to the bottom of the tank. Inclined ladders extend from the intermediate platform down to both horizontal stringers and to the bottom of the tank. Handrails are provided on each stringer as required by IMO guidance.

Finally, a bolted manhole is located in the forward and aft tight bulkhead of each cargo and ballast tank in the cargo block. The manhole consists of a 24 inch pipe with a 150# bolted blind flange that is hinged for handling. The manholes, which would be opened only during shipyard repair period, provide two additional means of access and egress from each tank.

Cleaning of Cargo Tanks

Cleaning of cargo tanks is benefited by the inclusion of most major longitudinal structure within the ballast tanks. The result is “clean” bulkheads in the cargo tanks. Longitudinal structure in the tanks is found at the upper deck and the starboard side of the centerline bulkhead, which carries the centerline longitudinal structure. The aft bulkheads have two horizontal stringers and carry the vertical transverse bulkhead. This arrangement effectively results in four clean surfaces in the port cargo tanks and three clean surfaces in the starboard cargo tanks.

It is worthy of note than the IMO standard for coverage was enhanced in the ship specification to facilitate safer tank entry and to assist in cleanup for shipyard repairs [17]. This higher standard required direct impingement from the crude oil wash machines on 95% of the primary and the secondary structure (i.e. bulkhead longitudinals and vertical bulkhead stiffeners). Three top and two bottom machines in the starboard side tanks and three top and one bottom machine on the port side tanks were used to meet these criteria.

Access and Maintenance in Machinery Spaces

Special efforts were made to include provisions in the design that facilitated maintenance and removal of equipment in the engine room and machinery spaces. Clear vertical accesses to the exterior are provided through the decks on the starboard side of the Pump Room access trunk and in both Engine Rooms. A ten-ton stores crane located on the exterior “B”-Deck services the port side Engine Room access. A five-ton air hoist services the starboard side Engine Room access. Underdeck bridge cranes service each of the two main engines and are capable of transporting equipment to common lifting points at the vertical accesses.

An extensive monorail system in each Engine Room is capable of reaching most auxiliary equipment and transporting it to the port side Engine Room Workshop for servicing or to the vertical accesses for removal from the ship. Bolted equipment removal plates on the 17120 mm ABL machinery room flat allow access to the cargo pump motors located on the 9840 mm ABL engine room flat. Again, the monorail system allows for removal of these eight-ton motors from the port side of the ship. A watertight, A-60 insulated and normally-closed bolted equipment removal plate (BERP) is located in the Engine
Room Workshop to facilitate the athwartship movement of large equipment across the ship.

PRELIMINARY DESIGN

The overall scantling arrangement and Midship Section were developed and analyzed using ABS SafeHull and separate coarse meshed finite element models of the midship section. The separate finite element analysis telescoped from global to intermediate and then to general local models.

Nine Critical Details Addressed

The main objective in analyzing critical structural details was to identify and implement any potentially large structural changes prior to turnover of the design to

Fig. 5 Nine Local Details in Preliminary Analysis (Top row, l to r: Lower Hopper, Upper Hopper, Stringer at Centerline Bulkhead; Middle Row, l to r: Bottom Longitudinal, Transverse Web Frame at Tank Top, Double Bottom at Transverse Bulkhead; Bottom Row, l to r: Horizontal Stringer at Transverse Bulkhead, Inner and Outer Shell at Transverse Bulkhead, Inner and Outer Shell at Web Frame)
the Shipyard. Further, the owner also wanted to have timely and complete control in selecting the sensitive structural details to be used repeatedly throughout the cargo block. This is again a reflection of extensive experience with existing ships operating in the harsh TAPS trade environment, which has demonstrated that the design of the local details ultimately determines the probability of premature fractures. Concurrent with concept design, numerous design variations were analyzed in common structural details to optimize and improve their structural behavior.

Nine local details found in the midship section were selected and analyzed using detailed solid element mesh. The mesh size was on the order of the plate thickness in the critical regions. The nine details are shown in Figure 5 and cover typical problematic construction details. The models were analyzed using eight initial design load conditions. These consisted of four loading conditions: Full Cargo, Ballast and two “Checkerboard” loading conditions. The Full Cargo and Ballast loading conditions were analyzed in still water, head-sea design hog and head-sea design sag waves, while the two Checkerboard conditions were analyzed in still water only. The design wave used had a length equal to the ship’s LBP and a wave height of 8.05 meters.

Results of Preliminary Analysis

The preliminary analysis revealed the existence of high general stresses in the two horizontal stringers when subjected to the checkerboard loading conditions. The stresses were too high to be eliminated by local reinforcement, and the need for modest changes to the stringer design was identified. The modification required additional and symmetric transition structure aft of the transverse bulkhead. Forward of the transverse bulkhead the stringer transition was softened at the centerline and longitudinal bulkheads and extended forward by one additional frame.

The results also included the identification of two alternative side shell longitudinal to frame connections. The first involved the use of an offset panel stiffener with a sophisticated web frame collar design. The second involved the use of soft-toed brackets butt-welded to the bulb-section of the side shell longitudinal on both sides of the frame. The latter of the two details was chosen for use on the vessel based on construction preference.

Other recommendations included improvements to the lower hopper design, including rearrangement of the hopper longitudinal, a reduction of the hopper radius and increased web frame thickness in way of the hopper. It was found that the radius plating in the hopper design, when subjected to a sagging condition and without proper support, tended to “shrink” similar to the radial contraction of a pipe subject to stretching. The deformation in the hopper plating was restricted at the web frame thereby creating high stresses in the welded connection of the web frame to the hopper plating.

Also noteworthy in the findings was the need for additional support to improve the effectiveness of the curved sections of the web frame faceplates. Without additional bracket and flatbar support the curved section of the web frame faceplates become ineffective, resulting in unacceptable stresses in the web frame and flange at these locations.

Use of Mild vs. High Tensile Steel

The rationale for the decision to tend towards use of mild steel was based on historical problems encountered with the liberal usage of high tensile steel in ships. In the 1970’s and 1980’s the movement of the shipbuilding industry to look towards high strength steel as a means for reducing the overall weight, and therefore cost, of the vessel created many problems. Credit was taken in hull scantlings as high tensile steel allowed higher allowable stresses. In many cases, particularly for ships operating in less severe trade routes, this was acceptable. However, vessels frequently subjected to a harsh environment resulting in moderate or high cyclic stresses, developed fatigue cracks in the structural transition details.

In a distinct effort to avoid chronic fatigue problems it was decided that mild steel (ABS Grade A) would be used for all structure except in areas which specifically required high tensile steel. If high tensile steel were to be used, a reduction in scantling would be allowed, providing there was a minimum ten percent allowance above the ABS requirements, including corrosion allowance. For the upper deck, where the inherently low neutral axis of double hull construction results in mild steel plate thickness well above 25 mm, the deck is constructed of ABS Grade DH plate. However, by using Grade DH plate, ABS rules allow for a plate thickness of 19.5 mm, including corrosion allowance, but 22 mm plate was used. A second example of the use of high tensile steel is in the horizontal bulkhead stringers, which are subjected to high static stress but relatively low cyclic stress.

Figure 6 illustrates the application of these criteria to the design. ABS Rule minimum scantlings are shown in parentheses with the as-built plate scantlings shown outside the parentheses.

Fixed Range for Scantlings

For structure in the cargo block, in fuel oil tanks, and in miscellaneous ballast tanks a “rule of thumb” was set for determination of minimum and maximum scantlings. Structural steel was limited to a minimum of 12 mm in thickness and was to be no greater than 25 mm in thickness. In the case of the minimum scantling,
experience showed that steel less than 12 mm proved sensitive to corrosion. For the maximum scantling, steel greater than 25 mm reduced the benefit of the steel’s ductile behavior and thereby reduced the inherent redundancy of the structure [21]. Thick plates create a 3-D stress field at a potential crack tip, making the steel more brittle. From a maintenance standpoint, the relatively limited availability of steels thicker than 25 mm can problematic in repair yards.

These decisions were made based on past experience in vessel operation and with extensive preliminary finite element analyses complete. Ultimately, as design progressed, it was found that in only a very limited number of cases would material less than 12 mm have been of benefit or met the fatigue life requirements.

**Finalization of Preliminary Design**

The preliminary phase in the design process was closed with JJMA as the lead technical coordinator. This effort included modifying and updating the concept design Midship Section and Scantling Plans. Modifications were based on the findings of the structural finite element analyses and included additional changes desired by the Owner. The finalization of this phase also included extensive review of the complete concept design with all consultants present. This combined effort allowed all participants to benefit from the extensive experience of the group as a whole.

Prior to contract a fully developed set of lines, a Midship Section and a General Arrangement were turned over to Avondale. An overall teaming effort was initiated at this point in time. The Midship Section and a preliminary Scantling Plan were modified slightly and tailored to meet specific requirements demanded by Avondale’s production scheme. These modifications were small in nature and generally were the result of plate straking and erection butt location. All modifications were completed with the approval of AMI and the consulting team.

**SafeHull**

Following contract signing, work quickly advanced on design issues related to Class approval of the structural arrangements. These activities included the running of ABS’ SafeHull Phase A and Phase B. Because much initial work had been completed on the structural design, SafeHull served almost exclusively as a verification tool rather than as a design tool. The design was based on good engineering practices and did not rely on SafeHull as a tool to find minimum allowable scantlings.

In Phase A, the rule-based initial scantlings were verified and shown to satisfy the SafeHull prescribed load strength and fatigue requirements. In Phase B, strength
assessment of global and local structure was verified using SafeHull imposed loads. Here, global structure was addressed as a coarse mesh, three-tank model. Local structure was addressed as two-dimensional fine mesh models.

Limited modification to the structure was necessary based on the results of SafeHull. Plate scantlings at the toes of the cargo tank horizontal stringers required some increase, as did the local longitudinal bulkhead in way of these toes. Panel breakers were necessary in some instances to meet small panel buckling criteria. All necessary modifications were made and the resultant changes to structure were turned over to MCA for incorporation into the DLA analysis.

**DYNAMIC LOAD APPROACH (DLA)**

**Analytic Approach**

Concurrent with the Shipyard effort to run SafeHull, MCA Engineers commenced work on the ABS' prescribed Dynamic Load Approach (DLA). The DLA analysis was based upon standard ABS procedure, but using MCA-developed finite element models and SPLASH-generated wave-induced pressure profiles. SPLASH is a 3-D panel code for computing inviscid, irrotational potential flows [18]. This procedure included the following steps:

1. **Selection of four worst-case loading conditions.** Loading cases selected and shown in Table 2 were:
   - Alaskan Crude, typical full load condition.
   - Heavy Ballast, typical ballast condition
   - Arrival in San Francisco, a partial loaded condition
   - Checkerboard #6 SWB, a checkerboard loading condition per ABS.

2. **Calculation of the long-term extreme value of the Dominant-Load-Parameters (DLPs).** This was accomplished using the ABS-specified H-family weather spectra and SCORES II-derived Frequency Response Functions. SCORES II is a strip theory ship motion program used by ABS for the DLA analysis [19]. Four Dominant-Load-Parameters were calculated per ABS guidance; hull girder bending moment, roll angle, vertical acceleration forward, and lateral acceleration forward.

3. **Calculation of the equivalent waves for each load case and Dominant-Load-Parameter combination.** These were determined by dividing the long-term Dominant-Load-Parameter extreme values by the SPLASH-generated Frequency Response Functions to determine the equivalent wave heights. The SPLASH-calculated Frequency Response Functions were generated based on the equivalent wave systems (headings and frequencies) derived by the SCORES II results. The external wave pressure and internal tank pressure profiles were then multiplied by this scaling factor to obtain the finite element model loading profile. The results of this phase are shown in Table 3.

**Table 2 - Dominant Load Parameters**

<table>
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<tr>
<th>Case #</th>
<th>Load Case</th>
<th>Wave-induced Bending Moment [MT-m]</th>
<th>Roll [degrees]</th>
<th>Vertical acceleration [m/s²]</th>
<th>Lateral acceleration [m/s²]</th>
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<td>14</td>
<td>125K Alaskan Crude</td>
<td>656350</td>
<td>20.2</td>
<td>5.463</td>
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<td>27</td>
<td>Heavy Ballast</td>
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### Table 3 - Equivalent Wave Data

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4. **Development of numerous finite element models for the DLA analysis.**

The models developed included:

- A 3-D stem-to-stern global model for overall DLA stress assessment (Figure 7).
- A 3-D intermediate model of the amidships structure.
- 3-D detailed general local model of the typical midship structure at frames.
- 3-D detailed general local model of the upper portion of a transverse bulkhead structure including the two stringers (Figure 9 and 11).
- 3-D detailed general local model of the lower portion of a transverse bulkhead (Figure 8).
- A 2-D refined mesh model of the aft centerline bulkhead between Frame 33 and 73 to examine high shear stresses observed in the 3-D global model (Figure 12).

Scantlings were revised as necessary until all structure met the maximum allowable ABS Von Mises stress limit of 95% of yield. Note that since this analysis included detailed 3-D models of the complete midship structures, the general stress field was included in all typical structural components. For example, in addition to seeing the stresses in a 2-D web frame, the general stress levels in the frame brackets and the brackets to longitudinal connections were included. Another example would be the deck stresses; instead of looking only at the nominal deck plating stress the more detailed local models captured the additional stress increase created at the longitudinal-to-bracket connections at frames and bulkheads (Figure 9).

![General Local Model at Transverse Bulkhead](image)

Appropriate judgment was used when evaluating the different stress levels. Judgement factors included consideration of element size, element shape and the modeling of structural discontinuities. Higher peak stresses are expected in a more detailed and finely meshed model. Hence stress results larger than 95% of yield were in some cases considered acceptable in finely meshed models. The structure was also analyzed for buckling without resulting in any significant changes. In all cases, the results were submitted to ABS for specific approval.
The following modifications were shown in the analysis to be effective and were implemented in the design:

- The centerline bulkhead thickness was increased at several locations. Initially large pump-room cutouts in the centerline bulkhead were reduced in size and modified in shape (Figure 10).

- The depth and thickness of forepeak frames and one forepeak stringer were increased.

- The thickness of the lower transverse bulkhead stringers was increased locally at the connection of the toes to the longitudinal wing bulkhead. This modification took place both forward and aft of the bulkheads. The panel stiffener arrangements were modified at both the upper and lower stringers.

- Two additional brackets were included between the vertical bulkhead stiffeners and the sloped hopper plating. The thickness was increased for all four brackets at this attachment.

- The vertical bulkhead stiffeners above the top stringer were modified, as was the attachment to the deck longitudinals. The depth of each stiffener slopes from 693 mm at the upper stringer to 280 mm at the deck longitudinal. The soft-toed bracket attaching the stiffener and the deck longitudinal was made smaller than the original design (Figure 6).

All DLA models were also rerun using element stresses instead of unaveraged nodal stresses, per ABS request. The results showed that all elements in the final global and intermediate models passed the 95%-of-yield criteria. The 3-D general local models showed a limited number of locations (elements) which did exceed the 95%-of-yield criteria. The upper most side shell girder and the double bottom girder exceeded the yield criteria in way of the access openings included in this model (Figure 9). Hull girder bending combined with the modeled cutouts stress concentration drives the stresses in these girders. The nominal stresses in the girders are in fact significantly lower than the nominal stress in the deck, which itself is significantly lower stressed than what is required by the ABS rules.

The Checkerboard load case combined with its corresponding maximum roll conditions created a maximum stress of 105% of yield in the triangular brackets at the base of the transverse bulkhead vertical stiffeners. The checkerboard loading pattern is not a typical condition and combining this in port leak-test condition with a long term extreme roll condition is conservative. The final design of this bracket had passed a detailed finite element analyses using solid elements and was subjected to both a still water checkerboard loading and an extensive fatigue analysis as described below.
one element at the upper horizontal stringer (Figure 11). If such a combination of in port loading condition and extreme wave load were to occur it could lead to yielding of one localized region of the stringer. However, the remainder of the structure can absorb any load that may be shed. Since the normal loading conditions and cyclic loads create only low stresses at this location the design was deemed acceptable without further changes.

The 2-D model of the centerline bulkhead at the pump room showed above yield stresses at the fine meshed regions around the cutouts. These stress levels were deemed acceptable since they were induced by the fine mesh around cutouts rather than by the nominal stresses (Figure 12). Also, beam elements that were used in the 2-D model do not properly represent the beneficial effect of the flange coaming that is used as reinforcement around these cutouts.

No additional changes to the design were implemented nor required due to the evaluation of the element stresses. This is primarily a result of having used the more conservative unaveraged nodal stresses for the initial DLA submittal. Note that using the average nodal stresses, which is the default of many finite element packages and typically used by ABS for illustration purposes, is not a conservative approach. It will in many instances significantly reduce or hide high stress regions.

Importance of Model Test Program

A rigorous model test program was undertaken at SSPA in Gothenburg, Sweden prior to contract signing. As a result the development of a hull form was effectively complete when negotiations with the shipyard began. This proved advantageous as it allowed AMI and the Shipyard to immediately concentrate on structural design and resulting machinery arrangements. Foreseeing this benefit, during model tests, the seakeeping models were instrumented to later aid designers in correlating the DLA motions model to actual measurements. Comparison of the motions and extensive side shell pressure measurements has been documented in other publications [20].

Of note is that there was intent to determine if structural modifications were necessary based on results of a hydrodynamic model test program. The emphasis placed on redundancy in the overall design (e.g. the ship has two rudders, two propellers, two engine rooms etc.) demanded that relatively uncommon features were present in the hull form. For example, modifications were found necessary in the twin skeg arrangement that was chosen for the afterbody configuration. The area between the skegs was modified from a flat bottom to a V-shaped bottom to reduce aft slamming in following seas. Likewise, forepeak structure was later modified to accommodate resultant pressures found in the relatively flared areas of the bow structure.

FATIGUE ANALYSIS

Analytic Approach

The fatigue procedure used on the Millennium Class tankers has been developed by MCA in a continually evolving process that combines theory, finite element analysis techniques, and dynamic 3-dimensional load generation. The strength of the analytical process has been its consistent verification through actual field experience. Identifying, analyzing, and developing effective repairs for structures on a number of AMI’s existing ships has successfully proved the fatigue procedure. The analytical process includes the following key steps:

1. Construction of global, intermediate, general local and local detail models capable of capturing hull girder responses into small details such as rat-holes and bracket toes

2. Generation of dynamic 3-dimensional loads for a matrix of wavelengths, angles, and ship operating conditions using the SPLASH 3-dimensional CFD code

3. Analysis of the finite element models for the matrix of SPLASH loads to develop structural element RAO curves

4. Calculation of the fatigue crack initiation life of detail structures using the stress RAO’s, anticipated ship voyage/weather data, and published S-N fatigue curves

Telescoping Finite Element Analysis

The Millennium Class tanker design was analyzed using a telescoping process that captures global and regional structural performance and applies it to the structural detail of interest. The process allows for
The same Global, Intermediate and General Local finite element models used in the DLA analysis were also used in the fatigue analysis. In addition to the modified versions of six of the nine local models used in the initial design, three new local models were analyzed. These included bottom longitudinal to frame connections in hopper, deck longitudinals at frames and deck longitudinals at bulkheads.

These models are the final step in the telescoping finite element analysis process. They are constructed with general-local plate mesh along the boundaries, which is then transitioned to a mesh of greater density in the area of interest, using 8-noded brick (solid) elements with edge lengths as small as the plate thickness. Weld contours have been simulated using either 6-noded bricks or 8-noded bricks meshed along the weld contour to account for their geometric effect.

**SPLASH-Generated Loads**

The fatigue analysis used for this project is a fully three-dimensional dynamic spectral analysis. The load cases are generated using the three dimensional free-surface computational fluid dynamics code SPLASH.

Unsteady flows are treated in the frequency domain, as a linear small-disturbance, harmonically oscillating perturbation to the steady basis flow. First-order unsteady flow predictions yield unsteady ship forces and motions. Flow solutions are fully 3-D, with six degree-of-freedom ship motions, and arbitrary incident wave-heading angle. The unsteady flow is computed using a steady flow panel model with the hull and free-surface panels fixed, while movement is simulated via transfer of unsteady boundary conditions to the steady time-averaged panel location. Forces and moments are computed by integration of panel pressures over the vehicle surface, plus panel-based waterline integral contributions. The latter account for the oscillating area at the waterline due to ship and free-surface motions.

The SPLASH calculations were performed by South Bay Simulations, resulting in motions and pressure profiles for a matrix of waves. The hull was analyzed with nonlinear roll damping derived from Ship Motion Program (SMP) runs, at an average speed of 15 knots [24]. The analytical matrix (384 spectral fatigue load cases) included the following:

- Eight wavelengths, from 1/4 to 2 times the ship’s length
- Seven wave-angles from ahead to astern in 30° increments. Since “windward” and “leeward” effects are not symmetric, off-centerline models required 12 wave angles (full 360°)

- Two loading conditions - Full Load and Ballast
- The real and imaginary cyclic stress components (two instances in time are used to determine the complete cyclic variation, assuming sinusoidal loads and responses)

**Spectral Fatigue Analysis**

Fatigue can be divided into two stages: an initiation phase (calculated here using Miner’s Rule) and a growth phase (calculated using fracture mechanics), until the crack reaches a critical length and propagates in a brittle mode. Crack propagation analysis was not completed for this effort, as the objective was to minimize or eliminate crack initiation [21]. The crack initiation phase is calculated in the following steps:

1. The finite element models are analyzed for every SPLASH run, starting with the global model and telescoping down to the detail local models. The hydrodynamic pressure fields are mapped onto the finite element models using a bi-linear interpolation program. The 6-DOF calculated accelerations are applied to the light ship weight, and are used to derive the fluctuating internal tank pressures. Once parent model analysis is complete, the boundary reaction forces are transferred to the telescoped model, and pressures and accelerations applied to the applicable contained elements.

2. The real and imaginary stress results are vector-added (square root of the sum of the squares) to calculate the cyclic stress component for a given load condition, wavelength, and heading. By analyzing the finite element models for all loading cases and normalizing to wave height, stress RAO’s are developed for all free surfaces of every solid element in the detailed models.

3. The weather spectra, taken from a hindcast study based on historical hindcast weather data (Navy GSOWM) along zones in the TAPS trade route are combined with the stress RAO’s to produce response spectra. This directional weather data combined with an assumed trip profile of travel between Valdez, Alaska and Cherry Point, Washington three times per month determined the weather spectra.

4. The response spectra are compared with published S-N curves to calculate Cumulative Damage Ratios (CDR’s), an integrated ratio of the actual fatigue cycles at a given stress divided by the allowable number (defined by the S-N curves).

5. The CDR’s are integrated across the ship voyage profiles to obtain a fatigue crack initiation prediction.
SN-Curves

A number of SN curves have been experimentally determined by the United Kingdom Department of Energy, and are used as an industry standard [25]. Curves such as F and F2 apply to calculations based on nominal stress fields. Since the models are built and analyzed to the “hot-spot” detailed level, the appropriate curves are the ‘C’ and “D” curves. The “C” curve is appropriate for parent plate. The “D” curve is more conservative and is used for elements adjacent to welds. It is possible to improve the fatigue performance at the weld toes, by reducing or putting in compression the intrusions or other imperfections, which is an inherent feature of weld toes in steel.

Edison Welding Institute’s experience based on various test data, indicates that the fatigue life at the weld toes can be extended to approximately a “C” curve if burr grinding or hammer peening is used to form a smooth curved area which removes the toe and all undercut [26]. Hence, the fatigue life for these details can be assumed to increase by a factor of ~2.5, making “C” curve fatigue life prediction more appropriate than “D” curve prediction.

Fatigue Results

Modifications were necessary in most of the structural details in order to achieve the desired 30-year fatigue life. Most modifications were minor and included adjustments such as softening of bracket terminations, changing shape and location of erection joint cutouts, changing plate thickness and the use of full collars at a few locations.

More extensive improvements were needed in order to reach the desired fatigue initiation life in the lower hopper and at the deck longitudinal to vertical bulkhead stiffener connection. Even with the improvements implemented from the preliminary design analysis, further improvement were needed to extend the fatigue life in the radius tank top plating at web frames, and in the adjacent web frame cutouts.

In the hopper connection, the desired fatigue life was reached by using a full collar in way of the adjacent longitudinals. A partial longitudinal stiffener was added between the last tank top longitudinal and the girder below the radius hopper. Finally, to ensure that the desired fatigue life was achieved, peening was used on the welds between the curved portion of the tank top plating and web frame (Figure 13 and 14).

A significant stress riser was found in the deck longitudinals where they intersected the vertical bulkhead stiffener. The initial design used a large vertical stiffener in combination with a large bracket at this connection; the final design, which gave us the desired fatigue life, used a tapered vertical bulkhead stiffener in combination with a small soft-toed bracket (Figure 15). The tapered vertical stiffener reduced the longitudinal stiffness transition at the connection with the deck longitudinal while still having the needed stiffness at the upper horizontal stringer.

Fig. 13 Lower Hopper, Tanktop Plating Removed

Fig. 14 Lower Hopper, Zoom of Radius, Plating Removed

Fig. 15 Initial and Final Bulkhead Stiffener to Deck Longitudinal Connection
VIBRATIONS

The shipyard performed preliminary vibratory natural frequency studies on a subcontract basis with Det Norske Veritas (DNV). The studies were based upon a two-dimensional variable beam model for the hull-girder and a two-dimensional beam model with springs for the deckhouse. Simple beam and plating theories were used for the local structural elements.

The studies were performed to determine if there were any resonant frequencies with the 4th and 8th order of the propeller and/or 7th order of the main engine. The studies revealed that, in the after body, several local members and panels were deficient due to the reduced frequencies in the submerged mode (fluid one side or both sides). The corresponding inertia of these members and panels was increased to ensure that fatigue failures would not result from their vibratory response.

The preliminary studies also determined that there was a resonant condition of the deckhouse bridge wings with the propeller 4th order excitation frequency. Because the studies determined that there was a resonant condition with the bridge wings, additional studies were warranted to determine the forced response of the bridge wings with the interaction of the hull.

The final vibratory (natural and forced) responses were determined by a three-dimensional model of the entire vessel including the main engines, with the model consisting of 6,570 elements with 20,190 degrees of freedom. The calculated pressure impulse forces induced from the propeller as well as the imposed main engine forces were applied to the model, for the full load and ballast conditions.

The forced response study determined that the resulting vibratory response for the bridge wing was not in compliance with ISO standard velocity limit of 4.0 mm/sec for a frequency of 8.47 Hz, related to the susceptibility of human exposure. Hence, the vertical-truss support configuration for the wings was amended as well as the support structure between the bridge windows.

In the sloshing analysis, the first mode natural sloshing resonance period (with the vessel’s pitching period) was determined to be at only 18% of the cargo tank filling height. This would result in the corresponding induced pressures being less than the normal required design pressures.

OVERVIEW OF SHIPYARD ACTIVITIES

Plan Review Process

The design review process resulted in a particularly interesting relationship between the Shipyard, AMI, and JJMA. AMI contracted with JJMA to support design review. Drawings were issued from the Shipyard to both AMI and JJMA. JJMA performed initial review of the drawings for compliance with the specification and regulatory body rule. Comments were provided to AMI in the form of detailed comments on the design. AMI in turn reviewed the comments, made appropriate changes as necessary based on Owner preference or experience and submitted the final review comments to the Shipyard. In order to ensure all issues were addressed, weekly meetings were held between AMI and the Shipyard to review design issues. The end result of this process yielded minimal on-site staff for AMI but included the overall expertise of a design firm with extensive design review experience.

Construction Milestones

Keel laying for the ARCO ENDEAVOUR occurred on May 5, 1998 and the ship was launched on December 17, 1999. Keel laying for Hull 2498, the ARCO RESOLUTION, took place on July 12, 1999. Keel laying for the ARCO DISCOVERY, Hull 2499, is scheduled for November 2000.

Analysis in Support of Production

During the detailed design phase and into the construction period, several miscellaneous structural issues were also addressed through the use of finite element analysis during the design and building phase of the project:

• Rudder and Associated Castings - A rudder analysis was performed to verify the adequacy of the rudder design when subjected to hydrodynamic loading due to both ocean waves and ship’s steering. Past experience with cracking at sharp radii at the lower gudgeon led to an expressed emphasis on the rudder design by AMI. Prudent design work by the Shipyard resulted in an acceptable design and utilized a continuous casting between the upper and lower gudgeon.

• Cargo Riser and Drop Pipes - An extensive analysis of the cargo riser and drop pipes in the cargo tanks was performed. The piping system was analyzed for internal thrust loads, cargo sloshing and loads induced by thermal expansion. In the desire to keep the exposed Upper Deck clear of oil carrying pipes to the greatest extent possible all cargo pipes were run through the cargo tanks. At the manifold the cargo fill and discharge lines run vertically from the innerbottom tank top to the Upper Deck and pass through the horizontal stringers. Initially, the riser and drop pipes were to be welded at their penetrations through the stringers. With the ambient steel temperature as low as 40 degrees F (4 degrees C) and the cargo loaded at as much as 105 degrees F (41 degrees C) there proved to be a significant thermal expansion in the pipes. The solution was to
leave the cargo pipes loose through the stringers and anchor them to restrict horizontal motion, allowing vertical expansion. This eliminated “locking” the stringers together as well as pushing up the Upper Deck (Figure 16).

- **Plug Unit Stresses** - An analysis was also performed to identify the amount of locked in stresses created by fitting of the last of the construction block in each transverse band across the ship. These Upper Deck units were plugs placed between the centerline bulkhead and the longitudinal bulkhead structure. In matching of this plug unit with existing structure at the upper web frame erection joint it was found that the sequencing of weld-out on panel stiffeners was critical to maintaining the in-plane alignment of the web frame. The analysis demonstrated how certain panel stiffeners could be freed-up to avoid any adverse effects of the weld-out process.

- **Erection Joint Cutout Tolerance** - The early finite element analyses proved to be a useful tool later in the construction process. This became apparent in the fabrication of “mushroom”-shaped cutouts that spanned erection joints. Because of minor variations in the amount of neat material on some units or the occasional shift in placement of longitudinals the dimensions of the mushroom cutouts varied. Because of prior knowledge that these mushroom cut details were sensitive to fatigue a design study was undertaken. The desire was not to analyze each detail but to provide a range of dimensional tolerances for the mushroom cutouts. Knowledge of the relative effects of variations in the mushroom...
cutouts gave both Shipyard production personnel and AMI’s steel inspectors a hard and fast guide for cutouts in question during the fabrication process (Figure 17).

![MUSHROOM TOLERANCES](image)

**Fig. 17 Construction Tolerance for “Mushroom” Cutouts**

- **Weld Procedure for Bulb Plates** - The desire by ARCO to use bulb plates provided another unique challenge for the Shipyard. With no previous construction experience using bulb plates there was a need to develop weld procedures and disseminate the information throughout production. Little was found in the way of published guidance relating to the end preparation for butted connection of bulb plates. The ARCO steel inspectors had significant international experience and worked with the Avondale welding engineers to perfect end preparation details suitable for use in production. Samples were fabricated in the Shipyard Welding Laboratory and an approved process was established. Buy-in was achieved between the Owner, the Shipyard and the Classification Society.

**Stages of Construction/ Block Breakdown Photos**

Appendix A provides photos taken during various construction phases of the Millennium Class tankers. The photos provide various perspectives on the structure and the structural details that are included on the vessels.

**LESSONS LEARNED**

The significant scope and unique approach on the Millennium Class Tanker project has offered many lessons to be learned. With the extensive analytical work that was performed and the feedback into the design it is safe to say that few, if any, aspects of the overall design would be changed if the project were repeated. The greatest application of the lessons learned on the project can be used in the processes undertaken to meet design requirements. Important caveats were illustrated in the finding of unexpected analytical results and in the unique production challenges that this ship presents.

**Encourage Extensive Development of Scantling Plan**

One of the most important aspects of the design effort was the translation of the Scantling Plans into the design details. The Midship Section was clearly defined and the detailed design of the cargo block was fairly straightforward. However, when the detail design moved to the forward and aft transition structure, the challenge to achieve continuity was significantly increased. This occurred for a number of reasons. Scantling Plans were developed only to a preliminary extent. Instead, effort was made to quickly support the detailed structural needs of the DLA and fatigue analysis rather than to develop the detailed Scantling Plans. As detailed design progressed it became clear that additional development of the Scantling Plans at the early stages of design would have been beneficial. Most importantly, a full definition of the fore and aft structural transition areas should be clear at the scantling level.

**Explore Local Details Carefully**

Neither SafeHull nor DLA initially identified some problematic structural areas. It was particularly noteworthy that no stress or fatigue problem showed up at the lower hopper in either the SafeHull or the DLA analysis. No problems were identified at the side shell, bottom or deck longitudinal connections. Fatigue was addressed in SafeHull but SafeHull empirical tables predicted much longer life than what the detailed fatigue analysis revealed.

In other cases, findings from the DLA that were not addressed in SafeHull were also of significant interest. A prime example is the strengthening of centerline bulkhead away from midship, particularly in way of cutouts in the pump room. SafeHull and DLA are useful tools in design but it is also necessary to take care of local fatigue-sensitive details.

**Know Your Design Software**

The sensitivity of long-term extreme calculations was noted when comparing extreme values based on output from different ship motion programs. The DLA procedure provided significantly different long-term extreme results even when the basic motion characteristics of the input matched reasonably well with the input of the other ship motion programs. The extreme roll angle calculated by SMP, based on model test-derived damping coefficients, gave an unrealistically high long-term extreme roll angle. The SCORES II program, using a standard damping factor, estimated a maximum roll
angle consistent with what has typically been used on DLA analysis of similar ships.

**Understand Design Impact on Engineering and Production**

One particularly interesting aspect of the Millennium Class tanker design was the impact on engineering support services and Production. As a result of the effort to achieve continuity of structure the overall ratio of parts derived from plate versus parts derived from shapes was significantly different. That is, the exclusive use of backing brackets and web frame stiffeners cut from flat plate was not typical of other vessels. The Millennium Class tankers have a ratio of plate-originated parts to profile-originated parts of approximately 70:30. According to the Litton-Avondale Lofting Department this varies significantly from other vessel designs where the typical distribution is relatively standard at 50:50. This variation resulted in a redistribution of man-hours spent in the lofting effort as well as in the production loading of product lanes. In short, it was necessary to cut more parts cut on plate burning machines.

**Take Advantage of a Team Approach**

The team approach of addressing the structural design of the Millennium class tanker proved to be an excellent situation. Ideas were exchanged openly, fitness of the design was supported by extensive experience and the overall quality of the final product was enhanced. This carries not only to the ship itself but also to all design information that will be maintained throughout the life of the vessel.

Constant communication was necessary to make this approach work. In the end, the Owner obtained significant input on the design and the design requirements were met. The Shipyard benefited from the Owner’s past experience and obtained information on new design techniques from the contractors. The consultants derived benefits from the Owner’s extensive exploration of details and new design requirements.

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**Fig. 18** Diagram Showing Selected Benefits of Team Approach to Millennium Class Tanker Project
CONCLUSION

The effort to achieve a high structural standard for the Millenium Class tankers clearly departs from traditional thought processes. Design has been extended beyond Class Society requirements to incorporate lessons learned as successful marine operators. Throughout design the goal was to improve the safety and structural reliability of the vessels and address the needs of future operators of the ships. While the physical result of these effort are visible now, in the end, time will be the ultimate judge of success.

ACKNOWLEDGEMENTS

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REFERENCES


18. SPLASH, A Free-Surface Panel Code Flow Solver by South Bay Simulations, Inc.

19. SCORES II, Strip Theory Ship Motion Program Used by ABS


24. SMP91, A strip theory shipmotion program, Naval Sea Systems Command, Code: 05H3, Washington, DC


Fig. A1 View Looking Aft Along Transverse Web Frames in Cargo Tank

Fig. A2 Detail of Transverse Web Frame Toe Attachment to Tank Top

Fig. A3 View Looking Aft Along Hopper To Transverse Bulkhead

Fig. A4 Typical “Mushroom” Cutout at Erection Joint in Ballast Tanks
Fig. A5  View of Hopper After Erection, Looking Aft

Fig. A6  Detailed View of Hopper Connection, Looking from Below with Forward Unit Removed
Fig. A7  Photo from Bow Looking Aft

Fig. A8  Photo from Stern Looking Forward (Note Twin Skeg Arrangement Aft)