PREVENTION OF FRACTURE IN SHIP STRUCTURE

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An Interagency Advisory Committee

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SYMPOSIUM AND WORKSHOP ON THE PREVENTION OF FRACTURE IN SHIP STRUCTURE

The Ship Structure Committee was formed over 50 years ago to investigate the catastrophic failures of the Liberty Ships during World War II. Since then, much has been learned about the causes of ship structure fractures. However, in the past decade there have been tragic losses of bulk carriers worldwide and early fatigue failures of the Trans-Alaska Pipeline and Gulf to Japan service tankers. From these occurrences it is apparent that what is known is not being used in practice.

In order to address this problem the Ship Structure Committee commissioned the Marine Board of the National Research Council to convene a workshop. This workshop gathered together technical persons from the design, construction, operation and regulatory fields of the marine industry. They were tasked to develop methodology to prevent these failures during the design stages. The workshop consisted of two days of invited papers and workshop sessions leading to summary papers on the areas of design, loads, fatigue and fracture, fabrication and repair, and inspection. This report presents the presented papers and findings of that workshop. The industry is highly encouraged to set in place the recommendations in this report in order to develop and maintain the next generation of ships as safer, less polluting, with a lower life cycle cost, and higher reliability.

J. C. CARD
Rear Admiral, U.S. Coast Guard
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16. Abstract

Even though fractures in ship structures have been researched for over 40 years ther continue to be bulk carriers lost at sea with no trace and unexpected fatigue failure in deep draft tank ships. It has been found on many occasions that failures, while unexpected, could have been prevented had currently known information been used in the design, construction, maintenance, and inspection of the ship. The Ship Structure Committee recognized this situation and requested the Marine Board of the National Research Council to convene a workshop to document the methods and educate the industry. This report includes all elements of the Symposium and Workshop on the Prevention of Fracture in Ship Structures held in March, 1995. Durine it invited background papers were presented covering History and Background, Defining the Problem, and Current Practices in Other Industries. Then a series of technical papers on the state-of-the-art in areas of Design, Fatigue and Fracture, Reliability, Inspection, Loads and Materials and Fabrication were given to have the workshop members enter discussions on a level playing field. The first section of this report are the findings of the workshops with the final recommendations to the industry. The papers presented in the symposium follow that.

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PREVENTION OF FRACTURE IN SHIP STRUCTURE

ABSTRACT

Reducing fracture in ship structure involves five interrelated technology areas: design, loads, fatigue and fracture, construction and repair, and inspection. At the request of the interagency Ship Structure Committee (SSC), the National Research Council Marine Board's Committee on Marine Structures (CMS) convened the Symposium and Workshop on Prevention of Fracture in Marine Structures in Washington, D.C. on March 30–31, 1995. This event brought together experts in the fields of fatigue, fracture, and reliability of marine structure and the researchers, designers, fabricators, and operators of these marine structures and provided a forum for addressing causes and remedies for the rash of fracture-related failures that are occurring in ships. Papers were presented on topics including a summary description of failures over the past 5 years, lessons learned, ongoing and completed research in this area, and results of related studies.

Technology is available for reducing fracture in ship structure, and, although there are areas that need improvement, the most important need is to apply current knowledge more effectively to all aspects of ship design, operation, inspection, maintenance, and repair. Action is required by regulators, designers, fabricators, maintainers, owners, and operators of ships to reduce the fracture-related failures in ships. The CMS recommends that specific action be taken in the areas of design, loads, inspection and repair, and communications, as described below. Classification societies also need to develop formal fatigue design criteria. Ship designers, fabricators, and maintainers need to consider the prevention of fracture in more detail; owners and operators of ships need to demand the application of this technology.

NOTICE: The project that is the subject of this report was approved by the Governing Board of the National Research Council, whose members are drawn from the councils of the National Academy of Sciences, the National Academy of Engineering, and the Institute of Medicine. The members of the panel responsible for the report were chosen for their special competencies and with regard for appropriate balance.

This report has been reviewed by a group other than the authors, according to procedures for workshop proceedings, approved by a Report Review Committee consisting of members of the National Academy of Sciences, the National Academy of Engineering, and the Institute of Medicine.

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MARINE BOARD

Preface

The National Research Council Marine Board's Committee on Marine Structures (CMS) convened a symposium and workshop on prevention of fracture in marine structure on March 30–31, 1995, in Washington, D.C. This event provided a forum and the means for bringing together experts in the fields of fatigue, fracture, and reliability of marine structure and the researchers, designers, fabricators, and operators of these marine structures to address causes and remedies for the rash of fracture-related failures occurring in ships. The symposium and workshop, which were planned by the CMS, were designated as the first priority in the committee's annual report, *Marine Structures Research Recommendations for FY 1994*.

If the reliability of ship structure is to improve, the results of numerous past studies need to be implemented in ship design, maintenance, and repair. The CMS and the interagency Ship Structure Committee (SSC) saw that this objective could be encouraged by convening a focused symposium and workshop that brought together ship owners, ship operators, structural designers, shipbuilders, hydrodynamicists, metallurgists, and naval architects with the primary goal of developing changes to the design process. Because prudent design must include knowledge of future operating conditions, the CMS and the SSC believed that the issues of inspection periodicity and repair criteria should be addressed at the symposium. Deficiencies in current technology that prevent a fully rational approach to fracture assessment in ship structure also needed to be identified. The diversity of backgrounds and experience that needed to be brought together to focus on ship fracture problems made this symposium and workshop a good candidate for a National Research Council project.

The objectives of the symposium and workshop were (1) to review the state of the art in preventing fracture in marine structure and (2) to initiate the process of transferring research results into ship design, maintenance, and repair practices. The first objective was accomplished by a 1-day symposium of invited papers on fatigue, fracture, and reliability of marine structure. Paper topics included a summary description of failures over the past 5 years, lessons learned, ongoing and completed research in this area, and results of related studies. The second objective was addressed by following the symposium with a 1-day workshop, during which researchers on ship fracture discussed ship fracture issues with designers, fabricators, and operators of marine structures. Recent advances, obvious shortcomings, and needed advances in technology and in the practice of ship design, maintenance, and repair were identified in the workshop.

During the workshop, five individual sessions were held in the areas of design, loads, fatigue and fracture, fabrication and repair, and inspection. Each workshop session had a leader to guide the discussions and a recorder to capture the comments made by the workshop participants.

Biographies of the authors of the invited papers and the workshop leaders, the agenda for the workshop, and a listing of the workshop participants are included as Appendices A, B, and C to this report. Papers presented at the symposium and workshop are provided in Part II of this report.

To assist the CMS, a fracture symposium planning group was organized by the CMS and the SSC and led by CMS chair John Landes. This group of technical experts participated in the planning and conduct of the symposium and workshop, gathered the information from the workshop, and organized that material for review by the CMS.

The fracture symposium planning group met in Irvine, California, on May 26, 1995, to collect the results of the workshop. The group synthesized the ideas developed in the five individual workshop sessions and organized them under four categories: design; loads; inspection and repair; and a new category, communications, which was not addressed as a separate topic during the workshop sessions. The group added this category because better communications are needed among all of the individuals involved in ship design, construction, operation, and maintenance, as well as among technical specialists in areas such as fatigue and fracture mechanics and hydrodynamics.

The fracture symposium planning group provided its input to the CMS, which met on September 12, 1995, to consider the planning group's suggestions. The CMS developed the conclusions and recommendations that are reported in Part I, Chapter 4. The planning group did not participate in the discussions that led up to these conclusions and recommendations. The views expressed by the workshop participants and the opinions of the individual members of the planning group were considered by the CMS, but the conclusions and recommendations are those of the CMS.

WORKSHOP AND SYMPOSIUM ON THE PREVENTION OF FRACTURE IN SHIP STRUCTURE

Part I: Summary Report

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Part II: Presented Papers

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IV. TECHNICAL SUBJECTS

EXECUTIVE SUMMARY

At the request of the interagency Ship Structure Committee (SSC), the National Research Council Marine Board's Committee on Marine Structures (CMS) convened the Symposium and Workshop on Prevention of Fracture in Marine Structures in Washington, D.C. on March 30–31, 1995. This event brought together experts in the fields of fatigue, fracture, and reliability of marine structure and the researchers, designers, fabricators, and operators of these marine structures and provided a forum for addressing causes and remedies for the rash of fracture-related failures that are occurring in ships. Biographies of the authors of invited papers and those of workshop leaders are provided in Appendices A and B of this report; a list of the workshop participants is included as Appendix C. Paper topics included a summary description of failures over the past 5 years, lessons learned, ongoing and completed research in this area, and results of related studies. The papers presented at the symposium and workshop comprise Part II of the report.

Reducing fracture in ship structure involves five interrelated technology areas: design, loads, fatigue and fracture, construction and repair, and inspection. During the event, participants identified a number of issues in these areas. These issues formed the basis for the conclusions and recommendations of the CMS, which are summarized below and detailed in Part I, Chapter 4.

CONCLUSIONS

Technology is available for reducing fracture in ship structure, and, although there are areas that need improvement, the most important need is to apply current knowledge more effectively to all aspects of ship design, operation, inspection, maintenance, and repair. The conclusions of the CMS for these four areas are as follows:

Conclusion 1. Design—Design methods for fatigue resistance are available, but historically these methods have not been used. Continued application and enhancement of known design methods should be pursued.

Conclusion 2. Loads—Better definitions of loads and procedures for using them in design are needed. Improvements are needed in predicting high-frequency responses and in combining these responses with low-frequency responses.

Conclusion 3. Inspection and Repair—Better tools are needed for inspection and repair to increase understanding of the likelihood and the consequences of cracks on the in-service condition of ships. The industry needs to know the impact on fatigue strength of fabrication standards and the effect of higher-strength steel.

Conclusion 4. Communication—Improved communication is needed to enhance dissemination of fatigue design information. Reduction of fracture would result from better dissemination of information about available design procedures that identify critical areas and good design practice for structural details. For instance, the application of fracture mechanics to ship structure analysis (to better understand and reduce fatigue failure) is more recent relative to its use in the aeronautics and nuclear power industries. Practicing naval architects and undergraduate students of naval architecture need sufficient information and procedures to permit use of fracture mechanics in the analyses of ship structure in all phases of their life cycles.

RECOMMENDATIONS

Action is required by regulators, designers, fabricators, maintainers, owners, and operators of ships to reduce the fracture-related failures in ships. The CMS recommends that specific action be taken in the areas of design, loads, inspection and repair, and communications, as described below. Many of these recommendations, especially those calling for additional research, will be further developed by the CMS in its biennial research recommendations to the SSC. However, classification societies also need to develop formal fatigue design criteria. Ship designers, fabricators, and maintainers need to consider the prevention of fracture in more detail; owners and operators of ships need to demand the application of this technology.

Design

Recommendation 1. Develop a ship-detail guidebook/standard for designers that addresses newer details and fatigue predictions. Producibility should be considered in addition to weight and fatigue performance.

Recommendation 2. Expand simplified fatigue analysis methods, such as SAFEHULL for tankers, to include transverse structure, more complex details, and other ship types. This formalization of fatigue design procedures provides a method for improving ship structure design in a way that most structural designers can understand easily.

Recommendation 3. Gather additional data on the fatigue strength of various large-scale ship details, including repair details, and the validity of linear cumulative-damage prediction (Miner's Rule) under high-cycle, variable-amplitude loadings, such as slamming. Development of this information requires a program of large-scale testing to validate current assumptions on fatigue-life prediction, including the definition of failure.

Recommendation 4. Design ship structure to include access for service inspections. Revise structural design criteria to include different tolerance and fabrication standards for higher-strength steels. This change in structural design practice would require better communication among designers, fabricators, and inspectors.

Loads

Recommendation 1. Develop a rigorous approach for combining high- and low-frequency response. Such an approach should be suitable for use in all phases of fatigue analysis, yet it should not be overly conservative or depend on the performance of model tests.

Recommendation 2. Compute loads using mechanics- and geometry-based simulations. All nonlinearities should be included in such efforts.

Recommendation 3. Develop a relatively inexpensive, easily operated hull-stress monitoring system that provides real-time feedback to the operators. Through the use of such a system, a ship's service experience can be linked with anticipated fatigue failures.

Recommendation 4. Quantify the degree of uncertainty in load predictions. This information is necessary if probabilistic methods are to be used in predicting the risk of structural failure.

Inspection

Recommendation 1. Develop guidelines for fitness-for-purpose assessments, including structural behavior, based on in-service history and fatigue analysis. These guidelines would include the effect of individual fatigue failures on overall ship hull structural strength and would provide a basis for determining the need for immediate repair of cracks in structure.

Recommendation 2. Develop new inspection tools and improve existing tools. Any revisions or newly developed guides should be premised on the outputs from life-cycle databases.

Recommendation 3. Quantify the fatigue life of temporary repairs, including the use of both wet-backed welds and doublers, as permanent repairs based on service history and fatigue analysis. Current regulations limit the use of such repairs, but the use of fatigue analysis could provide for extension of the service life of such repairs and reduce maintenance costs.

Communications

Recommendation 1. Develop a cradle-to-grave ship structural integrity database system that includes hull-stress monitoring, inspection, and repair data. Owners and operators should take advantage of available fatigue design/analysis information in developing inspection plans.

Recommendation 2. Develop and promulgate a manual/library, sorted by levels of fatigue strength of predictable standard details for fatigue resistance.

Recommendation 3. Educate and train future structural designers at the undergraduate level in fatigue and fracture design methods. Most practicing designers of ship structure need fatigue and fracture training.

Recommendation 4. Use concurrent engineering in ship design to include designers, owners, operators, fabricators, and surveyors/inspectors in the process. In-service review of findings should include all of these groups.

1

Introduction

BACKGROUND

The subject of fatigue and fracture of ship structure has been a continuing concern of both the Committee on Marine Structures (CMS) and the interagency Ship Structure Committee (SSC). The CMS and the SSC were both formed during the 1940s to address the fractures that occurred on welded ships during World War II. More than 100 reports on the subject of fatigue and fracture have been issued by the SSC since its beginning in 1946, with about 10 such reports issued in the last 10 years. In spite of this ongoing study of the problems of fatigue and fracture of ship structure, the problems continue to occur. Although there have been no catastrophic failures due to fatigue and fracture of ship structure, more than 80 serious structural failures from sources other than collisions or groundings occurred in the 69 tankers that called on the port of Valdez, Alaska, between 1984 and 1988 (USCG, 1990).[1] Lloyds Register reports that 70 bulk carriers of over 20,000 tons were lost because of structural failure between 1980 and 1994. Of particular concern were the 12 ships lost in 1990 and the 13 lost in 1991 (Lloyds, 1995). Many studies have been conducted, but the continuing occurrence of structural failure problems may indicate that more emphasis is needed on transferring the results of research into design, construction, inspection, and repair of ships. Further research may be needed. The evaluation of the need for research should include feedback from the users of existing research.

The definition of the term "failure," when applied to ship structure, is subject to many interpretations. A seemingly insignificant crack under the right circumstances could propagate across the entire hull, leading to loss of the ship. Such an event would be a catastrophic failure that resulted from an initially minor failure. In other circumstances, a similar crack may never propagate, but, because it is part of the oiltight envelope of the hull, can lead to pollution of the water so that the structurally minor failure becomes an environmentally major failure.

There have been many recent failures in many different ship types in many different operating routes around the world, including total structural failure of bulk carriers and the weld cracking of structural members of Alaska trade tankers. These failures have resulted in extensive

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[1] These failures were categorized as U.S. Coast Guard Class 1 and 2. At the time of the survey, Class 1 was defined as "a crack in the ship's watertight or oiltight boundary over 10 feet long." Class 2 was defined as "any crack in that boundary less than 10 feet long or a crack over 10 feet long in internal structure." Since then, the definition of Class 1 has been expanded to include any fracture in the watertight or oiltight boundary of a ship's hull.

and intense investigations as to their causes. Information provided in documents such as *Tanker Spills: Prevention by Design* (NRC, 1991) and *Report of the Tanker Safety Study Group* (U.S. Coast Guard, 1989), as well as in the numerous SSC reports and studies currently underway that address strength and reliability of marine structure, are all germane to the study of fracture of existing ships. A list of these reports is contained in the symposium paper by Card and Palermo. Knowledge of ongoing marine-oriented efforts applied to past failure experience provides the technical basis for applying fracture-prevention principles to ship design, as well as to maintenance and repair.

ISSUES

Reduction of fracture in ship structure involves the five interrelated technology areas of design, loads, fatigue and fracture, construction and repair, and inspection. It is essential to address the design of ships because it is the process through which ship structure is defined and its ability (or inability) to withstand service loads without the occurrence of fatigue or fracture is established—either explicitly or implicitly. To better understand the ability of the structure to withstand service loads without fracture, the nature of these loads must be understood. The application of fracture mechanics, including fatigue analysis, is the basis for determining the ability of structure to withstand loads without fatigue or fracture failure.

Following design of the structure, it must be fabricated by a shipyard, but the final product is not always the same as what was designed. Both the way in which a ship's structure is fabricated and the tolerances reflected in the final structure have an effect on the structure's strength. These characteristics must be understood in order to assess ship structure strength. Likewise, if damage occurs, repair is necessary to restore strength, but straightforward reconstruction to original conditions may not be sufficient to prevent future failure. Finally, minor failures must be detected by inspection if they are to be repaired before they become major failures.

Design

Preventing fracture in ship structure begins in the design phase. During the design the configuration of the structure, scantlings, and materials of construction are determined. The ability (or inability) of the structure to withstand service loads without the occurrence of fatigue or fracture is also established. Past practice in the design of ship structure was to rely on established design criteria that had already produced ship structure that was considered to be satisfactory for service. Until the inclusion of the SAFEHULL criteria in the American Bureau of Shipping (ABS) rules for 1996, there was no explicit use of fatigue and fracture analysis to ensure the adequacy of structure. However, ships were never free from cracks in the past, and now ships have changed in ways that make the former design criteria insufficient. For example, the use of higher-strength steel construction and more extensive structural analysis have led to reduced scantlings and higher stress in ship structure.

The workshop session in design addressed the design process, seeking the procedural changes needed to prevent fracture and methods for implementing such changes. Issues addressed in the workshop session included the development of standard structural details for which fatigue life is known and development of simplified fatigue analysis procedures for use in the design process. Design of structure for ease of in-service inspection was also addressed in the workshop session.

Loads

Knowledge of the loads imposed on structure from the encounter of the ship with ocean waves, the motion of the ship in the waves, and the ensuing inertia loads from cargo and other sources of loading is necessary for any structural analysis, including analysis of fatigue. The primary loading on ship structure is from the encounter of the ship with ocean waves. The amplitude of the loads varies greatly due to the statistical distribution of the waves and changes in the course, speed, and deadweight condition of a ship.

Determination of loads on ship structure is based on hydrodynamics. The hydrodynamicist must know how the structural analyst or designer is to use the load predictions produced, the accuracy required in load predictions, and the importance of phasing between different types of oscillatory loads that occur concurrently. These and other issues were addressed in the workshop session on loads.

Fatigue and Fracture

Repeated loading can result in the initiation of a crack in the structure from fatigue. Continued growth of a fatigue-initiated crack or of a crack arising from fabrication defects, damage in service, fatigue initiation, or similar sources can lead to rapid fracture and total failure of the structure. Because ships today are constructed of materials that resist rapid fracture from small defects, the initiation and propagation of cracks from fatigue is the more common cause of ship loss. For this reason, more emphasis is placed on analyzing ship structure in fatigue analysis than in fracture analysis.

The "S–N curve" approach is generally used for fatigue analysis. In this method, a structural detail is tested at repeated applications of a certain amplitude of applied stress (S) until failure occurs after N load cycles. Repeated testing of other samples of the same structural detail at various stress amplitudes then produces an S–N curve for that structural detail. Such S–N curves show an inverse relationship between the stress amplitude and the number of cycles to failure. Because the experimental basis of the S-N curve is to repeat testing until crack initiation in the test section (which usually fails soon after crack initiation), structural design based on the S–N approach implies that crack initiation during service will not occur because of structural fatigue.

S–N curves for structural details cannot be applied directly to ship structure because they are based on constant-amplitude testing. The most accepted method of accounting for variability in loading is linear cumulative fatigue damage, or Miner's Rule. Two assumptions of Miner's

Rule are that (a) the cumulative-damage relation has the same form regardless of stress level, and (b) the total fatigue damage is independent of the order of application of stress levels.

The S–N approach applies only for the prediction of crack initiation. To predict crack growth after initiation, or to predict crack initiation when a known defect exists in the structure, a fracture mechanics approach is used. The analysis of fracture in determining the acceptability of structure to withstand service conditions is approached using three different philosophies. The first of these, safe-life design, is based on ensuring that crack initiation never occurs during the anticipated service life. The second, damage-tolerant design, provides for residual strength, time for crack growth, and detection of damage before catastrophic failure occurs. The third philosophy, fail-safe design, provides for structural redundancies. These redundancies ensure that if there is a failure in part of the structure, the remaining structure is able to sustain the loads without total collapse.

These three philosophies of design to prevent failure occur in the context of three different situations: initial design, where the effect of cracking is considered; fitness-for-service during operation, where the effect of minor cracking is assessed; and life-extension considerations, where the effect of severe structural damage is assessed. Methods of addressing each philosophy under each condition have been developed and have provided reasonable results; however, more extensive application of these methods to ship structure is necessary. Therefore, the workshop session in fatigue and fracture considered the issues of fatigue analysis in the context of a matrix of these three fatigue-life design philosophies at three times during the life of a ship—design, operation, and life extension.

Fabrication and Repair

During the process of fabrication of the structure of a ship, precise conformance to the design configuration is not possible, and a certain amount of misalignment of members, distortion of structure, and imperfection in welding occur. The effect of these anomalies of fabrication on fatigue life must be assessed, and the tolerance must be redefined or the design must be changed to accommodate anticipated defects of construction.

When cracking of ship structure occurs during service, decisions must be made about repairs. In some circumstances, immediate repairs may not be necessary. If repairs are to be made, the decision required is whether the structure is to be restored to its original condition or if a design change is necessary to prevent future failure. Usually only a short time is available to make these decisions; therefore, some guidelines are necessary. These guidelines are available, but they need to be updated to reflect changes that have occurred in ship construction over the years. The workshop session on fabrication and repair addressed these issues of adequate guidelines for repair of structure, as well as mechanisms to provide for better communication among designers, operators, and repairers.

Inspection

Inspection of ship structure is important to ensure the safety and integrity of a ship. Minor failures need to be detected while they can still be repaired before resulting in a major failure. Inspection can be an extremely difficult task, especially in tankers, because of the size of the ships and poor inspection conditions, such as access, lighting, temperature extremes, and time constraints. The difficulties of inspection can be taken into account during the design process by recognizing that cracks may not be detected until they reach a certain size and by designing the structure so that it can be inspected more easily. Issues for inspection include identifying ways to inform inspectors about areas that are most likely to crack and have a high consequence of failure and determining the probability of detecting cracks in operating ships.

2

Symposium Presentations

INTRODUCTION

Addressing the problem of fracture in ship structures requires information on the background of the problem; an understanding of the problem from the perspective of owners/operators, classification societies, ship fabricators, and structural analysts; and an understanding of how other industries approach similar problems. Technical knowledge is needed in the areas of hull structural design, principles of fatigue and fracture analysis, structural reliability, inspection of ship structure, loads, and materials and fabrication. The Fracture Symposium Planning Group invited experts in the field to prepare technical papers in all of these subjects and provided them with specific charges about the information to be contained in their papers. After preparing draft papers, the authors met with the planning group to review the papers and ensure that all subjects were adequately discussed and overlaps in similar areas were minimized. The intent of the symposium and workshop was to encourage experts versed in ship design and maintenance to think more about fracture prevention and to encourage those versed in fracture prevention to think more about the problems of ship structure.

The summaries presented in the sections that follow in this chapter provide brief highlights of the more substantive papers presented at the symposium. The papers are presented in Part II of this report in their entirety.

HISTORY/BACKGROUND

Rear Admiral James C. Card and Peter M. Palermo provided a framework for the symposium and workshop with a paper entitled *Safelife for Ships*. In the past ship structural design was premised on seaway loadings while underway and on cargo loadings while pierside. For the past 10 to 20 years, ships have been experiencing structural failures under loadings that were considerably less than the design loadings associated with hull failure. Repeated loadings in areas of stress concentrations lead to fatigue cracks that, if left unrepaired, can grow to a critical length and result in catastrophic crack growth. Over the past 10 years a number of large oceangoing ships have either sunk or experienced significant structural failures attributed to a combination of human error and seaway loadings. The results of these human errors include inadequate maintenance and/or poor detail design or fabrication practices.

Technological advances in certain design and fabrication areas have led to the fatigue cracks associated with many of today's ship designs, particularly in tankers in the Trans-Alaska Pipeline Service (TAPS) trade. Solving or codifying methodologies for design should provide solutions to the problem. Attention must also be paid to the practices of fabrication and inspection when developing a strategy to minimize the risks associated with fatigue and fracture of ships. Properly considered and implemented, the findings of these workshops should provide a real factor in minimizing, if not eliminating, oil spills from tankers and the loss of life associated with ship fractures in bulk carriers and tankers.

DEFINING THE PROBLEM

Owners/Operators

David Sucharski of ARCO Marine presented a paper, *Crude Oil Tanker Hull Structure Fracturing—An Operator's Perspective*, in response to the following charge:

> *Describe, from the viewpoint of an owner and operator, the fracture problem in ship structures, its effects on ship safety and operation, and strategies for future avoidance. Describe past experience, lessons learned, and strategies for reducing or avoiding occurrence of fractures in ships, including design, inspection, and maintenance and repair strategies. The effect of these strategies on life-cycle costs and ship safety should be addressed.*

Fracture of hull structure has been a continuous problem since the first metal ships were built 150 years ago. This tendency to fracture has continued over the years and, even though the use of higher-strength steel in ship construction has often been identified as an important cause for the current cracking in tankers, service records show that most fractures continue to occur in the mild-steel components of these ships. The use of performance documentation, more rigorous and complete engineering, and active onboard stress management is integrating the efforts of naval architects, shipbuilders, and shipboard operators, making them equal participants in seeking the successful performance of a ship's structure over its lifetime.

The performance records of four classes of crude oil tankers operating in the TAPS trade show similar locations for cracking within the same class but different locations of cracking between the classes. This pattern of cracking suggests that better initial design can lead to better performance. Continued operation of ships that are prone to cracking is aided by the development of critical-area inspection plans that ensure the most efficient application of owner and regulatory inspection and maintenance resources. Extensive fatigue analysis of ship structure combined with voyage planning can reduce the incidence of continued fracturing in service.

Classification Societies

Donald Liu of the ABS presented a paper, *Local Cracking in Ships—Causes, Consequences, and Control*, in response to the following charge:

> *Address the view of classification societies on the problem of cracking of existing ships. What problems do the classification societies find in tankers, bulk carriers, and other standard ship types? What solutions and treatments to fracture problems on existing ships have they used? What is the difference between designing a vessel to be good or safe compared with designing ships to be fracture free throughout their life cycle? Provide insight to the effect of fracture prevention design on classification, Protection and Indemnity Club considerations, and regulatory bodies. Describe the approach that they have been taking in addressing the problem not only from their own viewpoint but, more importantly, from the viewpoint of the International Association of Classification Societies (IACS). Be aware that owners, operators, and shipbuilders will also be speaking from their own viewpoint, but they may not wish to discuss some past problems, as it may be admitting some blame. The classification society viewpoint should not have that same concern.*

Typical interrelated causes of cracking on oceangoing vessels, primarily tankers and bulk carriers, are high local stresses, extensive use of higher-strength steels, inadequate treatment of dynamic loads, adverse operational factors, and controllable structural degradation. Consequences of this cracking include possible structural failure, pollution, and increased maintenance costs. For existing vessels, solutions to these cracking problems range from repairs based on structural analysis to service feedback, control of corrosion, and enhanced surveys. For new vessels, improved design procedures should be used that specifically address dynamic loads, load combinations, and explicit fatigue design and account for operation- and maintenance-related factors that affect structural reliability.

Ship Fabrication

Hidetoshi Sueoka of Mitsubishi Heavy Industries presented a paper, *Ship Fabrication—From A Shipbuilder's Viewpoint*, in response to the following charge:

> *Address changes in construction methods and practices to improve the fracture resistance of ship designs. This should include a description of how interaction between the owner or operator and shipyards has changed with regard to design and producibility in the area of domestic and international sales. Specific topics may include analysis methods, design for short-list bidding, and design considerations for higher-strength steel.*

There are increased efforts today by many governments and regulatory bodies to increase the reliability and safety of ships and provide for more rigorous protection of the marine environment. However, the depressed market for ships has led shipbuilders to offer reduced prices. The demand placed on shipbuilders is to ensure greater quality ships at competitive costs. These problems are being addressed by developing standards of quality and increasing shipyard productivity. While these changes in ship reliability are important, there are other changes occurring in ship design. Tankers are now required to have double hulls, and higher-strength steel is being used in construction. Improved structural analysis methods are being used in design. Shipyards are changing the details of the structure to improve reliability and increasing the control of construction tolerances.

Analysis and Repair

Maxwell C. Cheung of MCA Engineers presented a paper, *Cost Effective Analysis for Tanker Structural Repairs*, in response to the following charge:

> *Discuss past experience and lessons learned regarding the occurrence of fractures in ships. The paper should discuss the important aspects and factors affecting the fracture problem and the methodologies used for analyzing and finding solutions to the problem.*

Under the pressure of an operating schedule, many structural problems detected during unscheduled inspections receive temporary "vee-and-weld" repairs or simple, localized reinforcement. Prudent owners will take the precaution of analyzing the cause of class (design) failures and will develop appropriate repair plans synchronized with scheduled repair availability. There are three optional levels of analysis: local stress analysis, extended stress analysis, and spectral fatigue analysis. Local stress analysis involves the development of a relatively simple local model of the affected structure that is analyzed with simple assumptions for boundary conditions and applied loads. Extended stress analysis uses a "telescoping" technique of global, intermediate, and local finite-element models to transfer hull-girder loads and reactions into an accurate set of loads for a local structural-detail model. A spectral fatigue analysis includes structural modeling information on anticipated weather and on material properties.

Current Practices in Other Industries

David O. Harris of Engineering Mechanics Technology, Inc., presented a paper, *Fatigue and Fracture Control in the Aerospace and Power Generation Industries*, in response to the following charge:

> *Describe the current practices in other industries used to predict, control, and design for prevention of fatigue and fracture. The industries of interest include bridge construction, offshore structures, nuclear power generation plants, and aircraft and space structures. Describe current analysis and design practices, standardization codes, and inspection and repair methods.*

In the power-generation and aerospace industries the control of fatigue and fracture is based on fracture mechanics, concentrating on the growth of a dominant crack. The fracture-mechanics approach requires computation of detailed stresses; but, with proper information, this approach can account for the effects of geometry on crack growth and final instability. Fracture-control procedures for safety-critical structures in space applications call for a fracture-mechanics analysis of components using an initial crack of the size that would be found with a 90 percent probability (at the 95 percent confidence level); that is, a crack of the size that is referred to as the nondestructive examination size. A component passes the fracture-control requirements if a flaw of this size can be demonstrated to survive four lifetimes.

Damage-tolerance analysis is used to ensure structural safety throughout the life of aircraft structures. The analysis includes (1) identifying all critical areas in the structure, (2) analyzing fatigue-crack growth under the assumption that flaws exist in the structure at the beginning of the service life, (3) performing periodic nondestructive inspections of areas found to be critical in the fatigue-crack-growth analysis, and (4) investigating the effects of accidental damage that might occur. The calculated fatigue life is made with an assumed initial radius corner flaw of 0.005 inch.

The commercial nuclear power industry has used fracture mechanics extensively in the analysis of the nuclear reactor pressure vessel and piping to predict defect behavior in components. In-service nondestructive inspection is used to detect cracks before they grow to produce a failure. If a crack is found, its future behavior is analyzed using procedures for defining the crack size for the fracture-mechanics analysis in terms of the defect indications. Very extensive materials testing programs that have spanned several decades provide the information required for material properties that are used in these analyses.

Fracture mechanics with a probabilistic approach has been used for fracture control purposes for offshore structures and bridges. Fracture mechanics considerations also play a major role in the analysis and control of failures in the chemical and petroleum industries, where the term "fitness-for-service" is used. Assessments concerning the safety implications of defects found in service are a major concern in these industries, and fracture mechanics is heavily relied upon in such assessments.

Stephen Maddox of The Welding Institute also presented a paper in response to the "Current Practices in Other Industries" charge. This paper is entitled *An Overview of Fatigue and Fracture Design Procedures Used in Construction Industries.*

The fatigue behavior of welded joints is well characterized, and reasonably comprehensive international design rules exist for many structures. Most fatigue design rules are of type 1; that is, they are based on data obtained from constant amplitude fatigue tests on particular weld details. An alternative approach incorporated in some rules (type 2), notably those used for designing pressure vessels, is based on fatigue test data obtained from polished specimens of the material of interest. These data are used in conjunction with fatigue strength reduction factors that take account of structural discontinuities in the structure, including welds.

Most of the development that has resulted in modern fatigue design rules for welded joints originated from bridge research, and several countries have comprehensive fatigue design rules for steel bridges. Offshore structures situated in the North Sea are designed using comprehensive fatigue design rules, notably those published by the United Kingdom's Department of Energy.

The basic principles of the fatigue rules for offshore structures are the same as those in the bridge rules; indeed, where appropriate, identical rules are presented. One exception is at the intersection point of tubular members in offshore structures, where the "hot-spot" stress approach is used. In this approach there are agreed-upon definitions of the hot-spot stress that are based on the stress distribution in the vicinity of a weld. Fatigue design rules for pressure vessels throughout the world are based on the type 2 approach, although alternative type 1 rules are being developed for new codes in Europe.

There are two new European fatigue design rules for welded aluminum that are type 1 rules similar to those for steel structures. Many industries also adopt available rules, particularly type 1 rules, because these are closely related to fatigue test data obtained from structural details of the types that might be used in any fabricated structure.

TECHNICAL SUBJECTS

Hull Structural Design

Mark Debbink of Newport News Shipbuilding presented a paper, *Hull Structural Design for a 40,000 DWT Double-Hull Products Carrier* in response to the following charge:

> *Provide a global to local overview of how state-of-the-art analytical methods and software are used to evaluate structural designs for fatigue resistance. Global design should discuss the selection of key design parameters, such as web spacing, longitudinal spacing, profile selection, and material type. Local design should consider the design of critical structural details for reduction of stress concentrations and for producibility.*

The structural configuration of a recently designed 40,000-dwt products carrier was based upon marketing research, with the structure optimized to suit construction facilities, double-hull regulatory requirements, and owner/operator concerns for inspection and maintenance. Additionally, personnel from the design engineering, production control, and industrial engineering disciplines formed a concurrent design team.

Fatigue and Fracture

Harold Reemsnyder of Bethlehem Steel Corporation presented a paper, *Fatigue and Fracture of Ship Structures,* in response to the following charge:

> *Present the technology available to analyze the fracture and fatigue behavior of ship structures. Cover basic assumptions, parameters, and approaches. Discuss new technology that may have potential applications for ship structures. Suggest how this may be implemented in solving design and maintenance problems.*

Two state-of-the-art approaches to fatigue life and residual strength assessment have received extensive experimental corroboration and wide application. The first is the component-test, life-assessment model combined with the Miner-Palmgren cumulative damage model and applied to weldment fatigue design criteria. The second is the failure assessment diagram used to assess the residual strength and structural integrity of cracked elements. Current weldment fatigue design criteria that are adequate for analysis of ship structures, such as British Standard 7608 (BSI, 1993) and the ABS Guide for Fatigue Strength Assessment of Tankers (ABS, 1993), consider the geometry and quality of the weldment, the nominal stress range, limited life-improvement techniques, and the effects of a corrosive environment. The concept of hot-spot stress is especially useful in cases of complex geometry, where the simple nominal stress range for standard categories of classes of structural details is impossible to compute or where a particular structural detail does not match with those shown as categories or classes in design criteria. The engineering tool currently receiving attention for assessing the residual strength of fracture-critical members is the failure assessment diagram (FAD). To date this approach has been used primarily in the electric power industry in Great Britain, as the R6 criteria, and in the United States, as both the FAD and the deformation plasticity failure assessment diagram (DPFAD). The R6, FAD, and DPFAD approaches all use the J–integral criteria for crack-driving force and resistance (i.e., toughness). In addition, the British Standards Institution addresses failure assessment of welded structures with FADs, using crack-tip opening displacement to express crack-driving force and toughness, in their published document (PD) 6493 (BSI, 1991).

Reliability

Paul Wirsching of the University of Arizona and *Alaa Mansour* of the University of California at Berkeley presented a paper, *Reliability in Fatigue and Fracture Analyses of Ship Structures,* in response to the following charge:

> *Describe reliability methods available for fatigue and fracture analysis during design of ship structures. Indicate the benefits and limitations of reliability methods.*

In spite of the uncertainties that exist in the fatigue analysis process, engineers must make decisions regarding the integrity of components with respect to fatigue. A probabilistic and statistical approach using recent developments in probabilistic design theory is relevant. There are several methods of making a probabilistic analysis, including (1) the direct evaluation of the probability-of-failure integral, (2) normal and lognormal formats, (3) mean value first-order, second-moment; (4) Hasofer-Lind generalized safety index, (5) first-order reliability methods; (6) second-order reliability methods; (7) advanced mean value; (8) direct Monte Carlo simulation; importance sampling, domain-restricted sampling, adaptive sampling, and directional sampling. These methods have been used to develop design codes, as well as to perform fatigue and fracture reliability and maintainability analyses.

Inspection

George M. Williams and *Stephen E. Sharpe* of the U.S. Coast Guard presented a paper, *Deep Draft Ship Inspection—Factors that Affect Fatigue and Fracture Prevention,* in response to the following charge.

> *Discuss inspection methods currently used in the construction and in-service periods of ships. Emphasize the difficulties and limitations in the process. Identify the aspects of inspection that the ship structural designer and a fracture mechanician would need to appreciate. These aspects include weld qualification procedure; visual inspection; time to inspect and area to cover; interferences, such as insulation, coatings, sludge, and rust; frequency of inspections; critical area inspection plans; and variances seen between operating companies in self-inspection.*

It is necessary to detect a failure early, before it causes catastrophic harm. The metallurgist and ship designer need to understand the difficulties associated with the inspection of ship structures and the capabilities of inspectors in order to plan for the prevention of failures. There are many variables that enter the inspection equation, including the inspector, the environment of the inspection, fatigue of the inspector, available time, coatings for tanks, and the degree of cleanliness of the structure. As part of an effort to make inspection less costly, to reduce the regulatory impact on inspections without reducing safety, and to focus on methods to improve the inspection process, the U.S. Coast Guard now requires the operators of certain vessels to record and track the occurrence of structural failures and to develop repair methodologies that address the root causes of the failures rather than just the symptoms. An additional approach that is being developed is "safety partnerships" or a "streamlined inspection process," by which a company with a good maintenance and operation history can work with the local U.S. Coast Guard Officer in Charge of Marine Inspection to obtain permission to self-inspect some aspects of their own ships; thereby reducing most of the U.S. Coast Guard inspections to random checks rather than verification of 100 percent compliance. This approach will take much of the labor out of efforts that achieve little return on investment and will free up resources of the U.S. Coast Guard, allowing more time to be spent on real problem vessels.

Loads

Youl-Nan Chen and *Yung-Sup Shin* of the ABS presented a paper, *Consideration of Loads for Fatigue Assessment of Ship Structures*, in response to the following charge:

> *Describe the appropriate means of computing the loads imposed on ship structure to be used in fatigue analysis. Loads would include global hull-girder bending as well as local loads, such as side-shell loading.*

Only moderate to low values of the stress range are important in a fatigue analysis; therefore, linearity can be assumed in the loads. A spectral crack-initiation fatigue analysis using unit wave height, stress-range transfer functions, and superposition can be used as can linear-elastic fracture mechanics for crack-propagation analysis. A spectral fatigue analysis requires data on the joint probability of wave parameters. These wave data are commonly presented in a wave scatter diagram. The selection of wave data for fatigue analysis depends on the anticipated areas of operation. From the classification society's standpoint, most large vessels are normally classed for unrestricted service. On that basis, the nominal North Atlantic wave environment is usually selected. For a site-specific application or for a trade route known to be more severe than the North Atlantic, the wave scatter diagram for a specific wave environment or trade route should be used.

The inclusion of wave-period variation is particularly important for ships for which motion and loads are strongly frequency dependent. Each sea state is typically represented by a mathematical spectrum, such as the Bretschneider (for open ocean conditions) or the JONSWAP (for fetch-limited cases).

Materials and Fabrication

Paul Blomquist of Bath Iron Works presented a paper, *Current and Future Directions in U.S. Shipbuilding*, in response to the following charge:

> *Present an overview of how materials and fabrication procedures are controlled to ensure that specified properties are met in the final ship hull structure. Emphasis should be placed on the control of weld joint geometry details, dimensional accuracy, and erection methods, including the accommodations of distortion and residual stresses. The talk should be focused on the entire subject and not cover specific, detailed accomplishments by the author, examples may be used to illustrate an occasional point.*

All ship fabrication in the United States, be it commercial or naval, requires that welders and welding procedures be qualified prior to use by subjecting all welds to 100 percent visual inspection and performing nondestructive testing of welds in critical areas or components. One of the areas of greatest attention in shipyard improvement programs is developing the ability to cut all plate to the final dimension at the initial numerically controlled burning operation. The cost of correcting welding-induced distortion can be considerable, and many efforts have been made to characterize and control the elements that cause it. A method frequently used for correcting weld-induced distortion is restraint, which produces high levels of residual stress. A method that is currently being explored is presetting. In this method, a part is set up for welding in an initial position that is offset from the actual dimension by the amount of expected weld shrinkage. Distortion is also controlled by regulating the welding sequence and by increasing the use of semi-automatic and mechanized welding processes and equipment that offer generally lower heat input and smaller welds and improved uniformity, consistency, and predictability as compared with manual welding.

U.S. shipbuilders are looking to robotic welding to reduce labor hours. However, inaccuracy is a major obstacle to large-scale implementation of robotic welding. Another problem is the lack of conceptual design standards that favor automation. Increased automation will be the single most likely change to occur in American shipbuilding in the next few years. Large-scale integration of panel fabrication and robotic welding equipment is planned or underway in several major U.S. shipyards. This degree of automation will make possible a level of uniformity and quality that heretofore has been unachievable and will have a positive impact on control of distortion and residual stresses.

REFERENCES

ABS. 1993. Guide for Fatigue Strength Assessment of Tankers. New York: American Bureau of Shipping.

BS 6493. 1991. Guidance on Methods for Assessing the Acceptability of Flaws in Fusion Welded Structures, 2nd Edition. London: British Standards Institution.

BS 7608. 1993. Code of Practice for Fatigue Design and Assessment of Steel Structures. London: British Standards Institution.

3

Workshop Sessions

INTRODUCTION

Consideration of the total process of prevention of fracture in ship structure leads to an examination of five elements, all of which are interrelated. Each of these elements—design, loads, fatigue and fracture, fabrication and repair, and inspection—was the focus of à separate workshop session. Workshop participants examined these elements and their effect on prevention of fracture in ship structure and discussed solutions for specific areas of concern. The workshop sessions developed to address each of these areas are detailed in the following sections.

DESIGN TO PREVENT FRACTURE

Background

Fatigue cracking is the cause of nearly all fracture-related problems experienced in the world's ships at this time. For this reason, the focus of the discussion of the workshop session on design was on designing to avoid fatigue failures; design to avoid first-passage hull fracture was discussed only briefly. Participants in the session included representatives from ship-design agents, classification societies, ship owners, shipbuilders, government agencies, and universities. (See Appendix C for a listing of workshop participants.)

The fundamental constraints on ensuring that a ship will be free of fatigue fractures are the time and cost required to accomplish this goal. These constraints apply during the design process and continue through fabrication of the ship structure. During a ship's life cycle, the constraints apply to inspection and maintenance of the ship structure. Design approaches must be simple and direct, that is, easy to understand and use, so that they can be applied cost effectively in a competitive design environment. The resulting structural configurations must be both practical and cost effective.

Economic considerations are also an essential element of the design process because they allow the fabrication and life-cycle costs of alternative configurations to be analyzed and understood prior to decision making. The total engineering costs for the first of a new class of commercial ships typically do not exceed about 10 percent of the total cost. Thus, it is cost effective to spend somewhat more on engineering in order to achieve much greater reductions in fabrication and life-cycle costs. The value of improved design engineering was illustrated in the symposium paper by Sucharski that included an example in which several labor-months of effort were expended to analyze and redesign structural details of an existing ship to prevent the recurrence of fractures that could have been avoided by better attention to detail during initial design. For example, standard details can be used to avoid "reinventing the wheel" and to save design time and effort. Standard load-prediction algorithms can also be used to achieve this objective. However, it is important to note that structural designs are normally tailored to suit the facilities and techniques of the building yard. Thus, there are apparent constraints on the utilization of standard structural details.

Objectives

The Fracture Symposium Planning Group assigned the following objectives to the participants in the workshop session on design.

- Develop a plan to implement appropriate methods of fatigue and fracture analysis in all design phases.
- Consider design to prevent fracture from maximum lifetime load and fatigue cracking over the life of the ship.
- Consider the effects of the following:
  - fracture arrest
  - new classification society fatigue assessment procedures
  - material selection
  - design of critical details
  - design for inspection, maintenance, and corrosion prevention
  - economics of design, fabrication, and life-cycle performance.
- Identify needed research.

Design to Avoid Fatigue Failures

Useful analysis techniques have recently been developed that predict fatigue crack initiation and growth. One of the predictive techniques for crack initiation uses an S–N type of analysis; fracture mechanics techniques are used to analyze crack growth. Initial defects in fabrication may be large enough to result in unacceptable crack growth. Fracture mechanics techniques are also used to establish fabrication criteria. During the design process, the designer's objective is to avoid crack initiation; therefore, S–N-based analyses are the designer's principal tool. These analyses may range from simplified deterministic methods to very complex spectral fatigue analysis methods. The former methods are most commonly used in the design of marine structures. Reliability-based design for fatigue, for example, can be based on S–N curves, using a linear cumulative-damage approach.

Most fatigue cracking occurs at the intersections of side-shell longitudinals and supporting transverse structure in tankers and bulk carriers. The oscillatory hydrodynamic pressure resulting from wave action along the hull side is an important causal factor in this type of fatigue cracking.

The nature of structural details—including weld details at intersections subjected to large oscillatory stresses—is critically important to the fatigue life of ship structure. These structural details, and the associated "hot-spot" stress amplifications that result in high stress ranges, are the principal determinants of fatigue life. In fact, fatigue life can be vastly improved by improving weld details alone.

Continuity of structure is another key consideration in reducing the number and magnitude of stress hot spots and, therefore, potential fatigue problems. Reducing peak stresses and the associated stress range is especially important from the fatigue standpoint since fatigue life is decreased as the cube of the oscillatory stress increases. This effect must be considered when high-strength steel is used because the stresses are permitted to rise to take advantage of the greater material strength. Regardless of the type of steel used, the designer must solve for and deal with cyclic stresses.

Streamlined, performance-based methods are needed to determine the following:

- design situations—for example, ship type, structural configuration, joint locations, and the like—that require consideration of fatigue during the structural design
- adequacy of existing or proposed structural details

Once developed, these methods must be tested based on past experience (calibrated) and must be accepted by a broad spectrum of the industry. The design procedure must be conservative, must reflect a good explanatory model, and must be updated frequently as additional knowledge is gained. In addition, acceptance criteria should be established by checking procedures against successful and unsuccessful ship designs. The Hull Structure Committee of the Society of Naval Architects and Marine Engineers or the interagency Ship Structure Committee could be the vehicle to achieve the necessary user friendly and broadly accepted techniques.

As stated previously, S–N-based analyses are the designer's principal tool. There are two different methods of developing fatigue predictions using S–N curves. One method, the "hot-spot" stress approach, is first to use the stress in the member times a stress concentration factor to represent the stress at the toe of a weld (but not to account for "notch effects") and then to use a single nominal S–N curve to assess adequacy. In other words, with this method, the ship is designed for a nominal S–N curve and fatigue life, and critical details are checked against the curve to ensure that they are below it. In this approach, different configurations have different stress concentration factors, but all details are "normalized" back to a single S–N curve. The second approach is to use S–N curves that were developed by testing actual details, together with the nominal stress, which would correspond to the stress used in the tests. In this method, the stress is not adjusted for notch effects, nor is it adjusted for any of the normal geometric stress increases due to the detail configuration. The problem with this approach is that when finite element analysis is used to determine the stress at a particular location, the structural engineer must decide how close is "close-enough" to the detail to capture the test-based "nominal" stress without getting so close that the geometrical stress risers already built into the S–N curve are picked up.

Realistically, it is too costly and time consuming to analyze each complex detail in a typical ship design to ensure that it is below the nominal S–N curve. Ship structural designers usually base scantlings on an allowable stress selected to avoid fatigue problems. It is common practice in the ocean engineering world to use a zone system to identify the degree of fatigue criticality of the various component details in a structure. These zones are then used to select appropriate structural details from a library of details sorted for acceptability by zone. The zones are also used to focus inspections on critical areas.

Questions have been raised about the validity of linear cumulative damage fatigue analysis (Miner's Rule) for a varying sequence of loads. Issues include the effect of mean stress, the presence of an endurance limit for welded joints, and the effects of very high stress. At the Carderock Division of the Naval Surface Warfare Center, numerous fatigue tests have been conducted on weldments subjected to constant amplitude and a wide variety of random amplitude loadings. (Sarkani et al., 1992, 1994; Kihl, 1994; Kihl et al., 1995). In general, test results support the use of linear cumulative damage theory (Miner's Rule) and an S–N curve that ignores the existence of a constant amplitude endurance limit. However, test results for very high random loadings (containing cycles to yield stress) were found to be nonconservative when compared to predictions made using Miner's Rule and a single-line S–N curve. This observation implies that it may be necessary to adjust the upper (high-stress) portion of the S–N curve to better predict failure lives under such extreme loading conditions.

Companies cannot afford to spend large sums of money doing spectral fatigue analyses of details in the preliminary design phase when they may be negated by later design changes. Therefore, during the preliminary design of ships such as tankers and bulk carriers, where fatigue is an important design consideration, simplified methods are used to check the adequacy of the selected structural details at the intersections of longitudinal stiffeners with transverse support structure. Other details are checked at a later phase of design.

In the preliminary design phase, designers select principal hull dimensions, establish deck and bulkhead locations and the framing system, choose frame and stiffener spacings, and determine the scantlings for a longitudinally effective structure. Achieving structural continuity is a major consideration as the ship's general arrangements are developed. Trade-off studies are performed to select the optimum frame and stiffener spacings and the choice of stiffener type. These selections can have a large impact on local stresses and the resulting details.

To limit the likelihood of fatigue failures, uncertainties in fatigue strength, loading, and prediction models need to be considered when developing partial safety factors for design.

Fatigue analysis methods developed by the civil engineering sector can be applied to marine structures; however, care must be taken. Marine structures typically undergo much larger stress variations than do land-based civil engineering structures, and they are unique in terms of fatigue details, residual stresses, stress concentrations, environment, and the loads experienced.

Current State of the Art

The workshop participants discussed the current state of the art with regard to fatigue analysis in the design of marine structures. An SSC report published in June 1992, "Fatigue Technology Assessment and Strategies for Fatigue Avoidance in Marine Structures", SSC-367, presents a good overview of the then-current state of the art for fatigue analysis related to the marine industry (Capanoglu, 1992). However, since this report was published, there have been significant advances in the development of fatigue analysis design tools that are useful to designers of marine structures.

Several classification societies have recently introduced user-friendly, PC-based software packages for the fatigue analysis of structural details. For example, the ABS fatigue assessment procedure was developed as an element of the SAFEHULL structural design system, which was initially focused on tankers and recently broadened to embrace bulk carriers. The procedure developed by Lloyd's Register, termed fatigue design assessment (FDA), is similar to the ABS SAFEHULL approach.

A broad overview of a classification societies design approach for tankers is presented in the ABS publication *Guide for Dynamic-Based Design and Evaluation of Tanker Structures* (ABS, 1993a). A more-detailed discussion of a fatigue-strength–assessment approach for tankers is presented in another ABS publication, *Guide for Fatigue Strength Assessment of Tankers* (ABS, 1993b). The procedure discussed in these publications is a designer-oriented approach to fatigue-strength assessment that may be used in place of much more elaborate spectral fatigue analysis methods for the analysis of certain fatigue-critical structural details. The procedure reflects an assumed extent of corrosion, the Palmgren-Miner linear damage model, S–N curve methodologies, a long-term North Atlantic Ocean sea environment, and a 20-year vessel service life. The procedure also addresses connections for longitudinal members, which experience has shown to be the most critical class of connection details for tankers. The number of well-documented fractures in this class of connections was also sufficient for calibrating the proposed acceptance criteria against service experience before adoption.

Problems/Shortfalls with the Current Approach

There are several problems with the current approach for design to avoid fatigue fractures, including the following:

- Currently used analysis methods do not provide adequate guidance on how to deal with production details such as limber holes and temporary tack welds.
- Current fatigue design methods do not account for redundancy (e.g., fail-safe systems) or the lack of same in establishing safety factors.
- User-friendly assessment techniques are generally not available for critical details in transverse structures, such as at the inboard toe of web frames, at their intersection with floors.
- Low-stress, high-cycle fatigue is not adequately addressed by current methods.
- Current simplified assessment techniques cannot be used to analyze complex structural details that consist of several elements and intersections.

Design/Production Interface

Fatigue crack avoidance is a matter of attention to details, not only during design but also during production. Thus, the interface between design and production is critically important. The ship-development process must be managed such that satisfactory details are defined in their entirety and produced without unanticipated deviations. The term "satisfactory" should be taken to mean producible and cost-effective structural details that have been proved by analysis to possess adequate fatigue life. In the past, during the ship design and production process, the definition of structural details has too often been left to draftsmen who had no in-depth knowledge of fatigue design principles, and production personnel have been permitted to modify the details provided to them by the designers— generally to solve a real or perceived problem.

Clearly, neither of these situations is acceptable where fatigue critical details are concerned. Knowledgeable engineers must completely define details that are both acceptable fatigue-wise and buildable. They should also incorporate inputs and suggestions from the production department of the shipbuilder to the extent possible. The production department must then create the details to the design and specifications that are developed. If this cannot be done, the production department must consult with the designers so that fully acceptable modifications are agreed to before construction. No longer should a draftsman or a shipyard worker be permitted to add a "rat hole" in the highest-stress region of a critical detail (e.g., include a scupper in the high-stress region of a fashion plate) or be allowed to make other such changes without the prior knowledge and concurrence of the structural engineer. Situations like this—all too common in the past—must be avoided in the future. In brief, more "up front" engineering must be performed before construction starts, then strict quality controls must be imposed. Examples of the quality controls in place at some shipyards today are given in the symposium papers by Sueoka and Debbink.

Design Loads

In the analysis of the fatigue life of structural details, the loads imposed on the detail are an input. At present, there are acceptable methods to predict these loads by predicting the hydrodynamic loads on the hull and using a finite-element model of the ship's structure to predict the stresses or loads imposed on the specific detail being analyzed. In studies of the fatigue life of ship structure that were conducted the past, there was, perhaps, an overemphasis on hull bending loads. More recently, it has been recognized that wave hydrodynamic-pressure oscillations on the side hull are very important from a fatigue standpoint. The most difficult technical problem to be resolved is the prediction of subtle secondary and tertiary loads on structural details. One risk involved in this type of prediction is that it is common not to consider all loads during fatigue design, especially dynamic loads; therefore, the resulting design details can be nonconservative. Another problem to be resolved is how best to combine in-plane stresses with lateral loads. The prediction of primary linear loads and load combinations is well understood even though the maximum lifetime loads that are predicted by existing methods sometimes predict stresses that exceed the yield strength of the structure, a situation that suggests that the methods are overly conservative.

The biggest obstacle to resolving these problems is that it is too time consuming and difficult for the structural analyst or an associated hydrodynamicist to make the necessary predictions for selected details in the real-world design environment. The job can be done, the raw data exist, but the designer simply can not afford to make the predictions "from scratch" for every design. Additional guidance and the practical methods for working-level engineers to estimate the loads they require for fatigue analyses are needed.

The classification societies' design systems contain simplified load-prediction algorithms for specific cases for tankers and bulk carriers. Eight to 10 load combinations are addressed by one system, and include proper phase accounting. Such algorithms are also needed for other tanker and bulk carrier cases and for other ship types and configurations. Ultimately, standardized, user-friendly methods for load predictions appropriate to the several design phases need to be developed. The classification societies should, perhaps, coordinate such efforts.

Sloshing loads on bulkheads also need to be addressed. One classification society addresses such loads for solid bulkheads, but the method used does not apply to non-tight swash bulkheads. A method for predicting hydrodynamic loads on swash bulkheads needs to be incorporated into a guide for their design. Tests performed at Southwest Research Institute have shown that, as long as the development of standing and/or traveling waves is prevented by using proper tank dimensions and swash bulkheads, sloshing pressures are very small—nothing more, really, than random contact without organization or significant mass. The tests performed at the Southwest Research Institute, as well as a summary of their results, are detailed in Balint, 1992.

More-sensitive load predictions are needed to define predicted fatigue life more accurately. At present, the statistical distribution of loads about the mean expectation are ignored. It is important to address the effects of short-term extremes. Typical assumptions on the amount of time that a ship is exposed to extreme storm conditions may be less than actually occur to ships in service. If these assumptions are false, then the fatigue life predictions are nonconservative.

Commercial ships are commonly designed for unrestricted service. However, it should be possible to analyze the effects of route changes on expected fatigue life. Some classification society procedures use a standard environment for unrestricted service; therefore, they are not sensitive to route changes. However, the fatigue-design analysis methods of other societies can be used to produce route-specific predictions.

Consideration should be given to collecting an operational database during a ship's life and using it periodically to predict the remaining ship fatigue life. Perhaps the ship's logs or data from a modest instrumentation package could be used to collect data for this database, which could also be used for conducting periodic condition assessments. The ability to correlate hull girder strains with accelerations on tankers has recently been achieved and is now being performed routinely by some operators. Using these techniques, it should be possible in the future to link a ship's service experience with anticipated fatigue failures. Another technique that has recently come into its own is weather routing using timely weather predictions and satellite communications. This technique could be used effectively to extend the fatigue life of a ship by judicious course selections.

Stress Analysis

For stress analysis, standardized definitions need to be established for various stress terms, such as mean stress, variable stress, residual stress, and stress concentrations. With detailed finite-element analysis, there is uncertainty as to the point in the finite-element model at which the "nominal" stress to be used with the selected S–N curve will be calculated. This is especially true for transverse members because methods for predicting detailed stresses for them are not as well established as they are for longitudinal members. Standardized methods for determining the stress-concentration factors associated with specific details are needed. Finite-element methods are commonly used for this purpose. Also, the nominal stresses to be used in fatigue calculations must be defined.

The effect of mean stress on fatigue behavior is not fully understood. There is limited experimental evidence of the total shakeout of residual weld stresses in marine structures due to both the high stresses in localized areas and the large stress variations. There have been a few demonstrations on small specimens, but more studies are needed. ABS takes a conservative view and assumes that residual stresses are present, an assumption that may be overly conservative.

Design of Critical Details

The current vague, arbitrary process for the design of details must change. A standardized, well-understood process for the design of fatigue details is needed, along with a practical tool to predict the fatigue lives of candidate designs. These procedures and tools should be incorporated into a guide for the design of details.

The problem with attempting to standardize the details themselves is that individual shipyards have their own techniques and capabilities and thus have their own tailored details. The details selected for a particular design also vary depending on whether the design is weight- or volume-critical and on the required fatigue performance level. Each shipyard should be encouraged to develop its own set of "fatigue-friendly" details that could be offered to customers and that would enhance shipyard competitiveness.

The SSC report, "Fatigue Characterization of Fabricated Ship Details for Design," SSC-318, contains many details that are helpful to designers and includes data on the in-service performance of the details discussed (Munse et al., 1983). At present, however, there is no information in the public domain concerning the in-service performance of double-hull tanker details. It would be very desirable to publish a library of candidate details sorted by zone, that is, by performance levels in a fatigue environment. The designer would then have a "shopping list" of options available for any application and for any required performance level (degree of criticality).

The following factors should be kept in mind when developing these options:

- The lowest-cost, most-producible details that provide the necessary fatigue life in a specific application are desired.
- For some mature commercial ship structures, such as single-hull tankers, only refinements of fatigue details are needed.
- Fatigue-strength data (in the form of S–N curves) on the various details used in ship structure are available in the literature.

Design for Inspection, Maintenance, and Corrosion Prevention

Access for inspection and maintenance must be addressed during design. Ladders, catwalks, and similar means of access are themselves subject to inspection and maintenance, which is costly. Recently, stainless steel was used in a Finnish-built ship to prolong the life of those items that are structurally independent and to reduce the cost of maintaining them. A guide to designing methods of access is provided by the Tanker Structures Cooperative Forum's *Guidance Manual for Inspection and Condition Assessment of Tanker Structures* (TSCF, 1986). These guidelines include standards on the size and location of access openings.

Relative to access, consideration must be given to how the double-bottom tanks will be "mucked out" in double-hull tankers. Tank-ventilation requirements for inspection and maintenance must also be considered. Chevron has done a great deal of work studying tank ventilation within the outer hulls and bottom of double-hull tankers, including performing model air-flow studies (Chevron, 1993).

Modern coatings for corrosion prevention are effective; however, they are very expensive. Tanker owners at the workshop reported that a two-coat epoxy system with a 5- to 10-year life costs $4.00 to $4.75 per square foot. For a double-hull tanker with fully coated cargo and ballast tanks, the coating cost, including application, is about equal to the cost of the steel used in construction. Corrosion starts on sharp edges; thus, the number of rat holes and the total length of free-plate edges must be minimized. Corrugated bulkheads are a help in this regard, as are bulb angles, which are not manufactured by steel mills in the United States and must be imported from Europe. On a recent product tanker design, using bulb angles rather than conventional angles reduced the coating area but added about 40 long tons in steel weight. The net construction cost delta for that tanker was essentially zero, but cargo deadweight was reduced by 40 tons.

Low-Stress, High-Cycle Fatigue

Even with low stress levels, high-cycle oscillatory loadings can cause fatigue problems. Rudder-stock fatigue fracture, which has been a problem on some tankers in the TAPS trade, can be analyzed only through approximations because there are no accurate methods of determining loads. The oscillatory forces are very low, but the number of cycles is very high. Apparently, the natural frequency of the rudder is approximately equal to the propeller-blade passing frequency. Beam seas on the generally North–South route are also thought to have played a role in creating unusual loads that lead to rudder-stock fatigue fracture. For these and related reasons, further investigation on rudder failures, and particularly rudder load-estimation procedures, is necessary.

Education and Training

Most practicing designers of ship structure need training in fatigue and fracture analysis. They must understand the basic phenomena, how to recognize situations where fatigue and fracture are concerns, what design tools are available, and how to use them. At present, many practicing designers of ship structure are not sensitive to these issues and do not have the knowledge they need to reflect fatigue and fracture considerations properly in their design products. To correct this situation, education and training that focus on practicing designers of marine structures are required. The SSC and the Society of Naval Architects and Marine Engineers (SNAME) could play leading roles in this regard. In addition, undergraduate curricula in naval architecture should include fatigue and fracture design methods. The dissemination of design information should include both the S–N approach for crack initiation as well as the fracture mechanics approach of crack growth analysis. The S–N approach reflects a long history of application in the field of materials engineering and has been applied extensively for the analysis of ship structure. The application of fracture mechanics to ship structure analysis is more recent relative to its use in the aeronautics and nuclear power industries. However, such applications are expanding in efforts to understand better and reduce fatigue failure in ship structure.

In brief, current fatigue and fracture technology must be conveyed in an understandable and useful form to practicing naval architects. Only then will large gains be made in the ongoing struggle against these problems. Research will continue to be needed to improve the existing technology, but the existing technology should routinely be used by designers.

Design to Avoid First-Passage Fracture

Designers of ship structure must limit the risk of hull fracture failure due to a maximum lifetime load, that is, first-passage failure in the absence of fatigue effects due to time in service. The maximum load is likely to occur when the ship is operating at sustained speed in high seas and experiences a severe slam. Structural continuity is extremely important to limit this risk, and transitions must be carefully developed.

First-passage fracture is not a significant concern for conventional commercial ships with reasonably continuous structure. The materials and scantlings arrived at based on traditional considerations are adequate to avoid this phenomenon. First-passage fracture is a greater concern for slender, high-speed ships and for ships with unusual hull and structural configurations. Large ships operating at high speed are generally in the supercritical regime, where motions and accelerations are reduced. This can lull the Master into a false sense of security until the ship suddenly encounters a steep "rogue wave." Today there is increasing interest in very–high–speed ships to transport passengers and high-value cargoes. The designers of these ships will certainly need to address the risk of first-passage fracture.

Load prediction is an obstacle for structural designers who wish to mitigate the risk of first-passage fracture. Linear strip theory is adequate for low to moderate seas, but the prediction of extreme loads in extreme seas is not possible today. This is a highly nonlinear phenomenon that includes violent bow flare and bottom slamming for which no suitable theory has been developed that can be used in design computations. In addition, the necessary statistical methods and data are not available to predict the risk. There are also problems inherent in being able to combine slam loads with wave-induced loads properly. The development of appropriate load combinations—with the proper consideration of phase angles—is needed, as is a probabilistic treatment of loads. In particular, the hydrodynamic load distribution along the hull can be very important and should be carefully considered when developing load combinations.

Additional research is needed to better understand the propagation of unstable cracks. This research includes development of the crack-growth-rate predictive tools that are needed to aid in determining the necessity for repair when a crack appears. In addition, fracture-arrest techniques, which have traditionally employed a strake of notch-tough material, need to be investigated further, and improved fracture-arrest methods and definitive acceptance criteria should be developed.

Findings in Design

There are many deficiencies in current structural design practice and in fatigue and fracture analysis procedures that cause ships to be prone to fracture. These are summarized below.

Constraints

- Fatigue design approaches must be easy to understand and use so that they can be applied cost effectively by practicing structural engineers in a competitive design environment.
- Economic considerations must be an essential element of the structural design process, including the design and selection of fatigue details.

Approach

- Continuity of structure to reduce hot-spot stresses, together with improved weld details, would, without other measures, vastly improve fatigue life.
- Streamlined performance-based methods are needed to identify design situations that require consideration of fatigue and to assess the adequacy of proposed solutions.
- A reliability-based approach for fatigue and fracture design is needed.
- The classification societies have recently introduced the formal consideration of fatigue into the structural design process and have developed simplified assessment methods applicable to the most common problem areas. These simplified methods need to be supplemented by specific, user-friendly procedures to address transverse structural details, more complex details, and other ship types.
- An improved understanding of low-stress, high-cycle fatigue must be achieved.
- Knowledgeable structural engineers must design or approve all fatigue-critical details in a new ship design, and strict quality controls must be implemented to prevent ad hoc waterfront changes.

Design Loads

- Simplified methods of predicting the loads on fatigue-critical details are required. The methods developed by the classification societies need to be supplemented to address a broader variety of situations.
- A method for predicting hydrodynamic loads on non-tight swash bulkheads is required. The method should be incorporated into a guide for the design of swash bulkheads.
- More-sensitive load predictions are required to predict fatigue life more accurately. The statistical distribution of loads about the mean expectation cannot be ignored as at present. Also, the designer should be able to predict the effect of route changes on fatigue life.
- Consideration should be given to making periodic evaluations of the service life remaining in a ship's hull. The remaining service life can be predicted if operational data are routinely collected in service and fatigue and corrosion assessments are made.
- Modern weather routing and hull-response monitoring techniques could be used effectively to extend the fatigue lives of ships.

Stress Analysis

- Standard definitions of stress terms must be established.
- Improved methods of predicting stresses in transverse structure and the stress-concentration factors associated with specific details are needed.

Design of Critical Details

- A guide for the design of fatigue details that reflects a standardized design approach and provides a practical tool to predict the fatigue lives of candidate designs is needed.
- A library of standard details, sorted by zone (fatigue performance level), would be very useful.
- A set of "fatigue-friendly" details, developed by each shipyard, should be encouraged.

Design for Inspection, Maintenance, and Corrosion Prevention

- The number and lengths of sharp edges of openings in and edges of ship structure need to be minimized because corrosion begins on these areas.
- Design for inspection and maintenance needs to be considered, including provision of suitable access (openings, ladders, catwalks, etc.) and ventilation.

Low-Stress, High-Cycle Fatigue

- Appropriate methods are needed to predict rudder-stock fatigue life.

Education and Training

- Short continuing education courses in fatigue and fracture are needed for designers.
- Fatigue and fracture concepts are needed in undergraduate structures courses in naval architecture, ocean engineering, civil engineering, and mechanical engineering curricula.
- Fatigue and fracture concepts are needed in graduate courses in ship structure.
- A symposium, technical forum, design workshop, or convocation should be convened to discuss the issues related to fatigue and fracture concepts.
- A discussion of recent developments in fatigue and fracture, including design methods, should be included in the update of the SNAME reference book, *Ship Design and Construction*, which is currently being revised (Taggart, 1980).
- An update and wider distribution of the SSC report, "Fatigue Technology Assessment and Strategies for Fatigue Avoidance in Marine Structures," SSC-367 (Capanoglu, 1992) should be considered.

Fracture

- The ability to predict extreme loads in extreme seas is needed.
- Better tools to predict unstable crack growth are needed.
- Improved fracture-arrest methods and acceptance criteria are needed.

LOADS FOR FATIGUE AND FRACTURE ANALYSIS

Background

The commonly agreed upon sources of dynamic loads associated with ship fracture are primary hull-girder bending moments and shears, including those that are wave induced and those that are the result of hull-girder vibration associated with springing and whipping; and local and out-of-plane pressures, including those that are wave induced or are the result of direct hull slamming, topside impact of green water, and liquid cargo sloshing. When addressing these load issues, the following questions arise.

- How are loads predicted today?
- What is wrong with how the loads are predicted?

- How should loads be predicted, both now and in the future?
- What needs to be done in terms of procedures, administration, and research to make a change?

These questions formed the general guidance for the workshop session on loads. Participants in the session included representatives from ship-design agents, classification societies, ship owners, universities, and government agencies and research laboratories.

Objectives

The Fracture Symposium Planning Group assigned the following objectives to the participants in the workshop session on loads.

- Develop a plan to implement the appropriate methods for developing lifetime load spectra to be used in various phases of ship structural design for the prevention of fracture, both from fatigue and from first-passage failure.
- Identify the statistics needed to characterize the loads. The statistics of the loads need to be expressed in a format suitable for reliability-based design.
- Recommend needed research for improvement of loads technology relative to fracture.

Current Load-Prediction Capabilities

Although there are many unsettled load issues related to first passage failures, the major emphasis of the workshop session was on fatigue. Accordingly, nearly all discussions concentrated on the "lifetime load spectrum."

The concept of a "lifetime load spectrum" is not entirely clear. Details differ from source to source, but the lifetime load spectrum is much more akin to a probability distribution of stress-reversal events. The population of events to which this probability distribution refers is all events that may occur in a ship's lifetime. Some naval architects are of the opinion that the word "spectrum" should not be used in this context. They believe that the notion of a "spectrum" should be reserved within naval architecture to express the distribution of frequency components. What appears to be needed in fatigue analysis is a "lifetime histogram of stress reversals" or a "long-term distribution of stress reversals." Either definition expresses the technical content of the term "lifetime load spectrum" without introducing unreasonable levels of confusion to those oriented toward hydrodynamics.

During the symposium, a paper that described a methodology for addressing dynamic loads and their application to the fatigue analysis of commercial ships was presented by Chen and Shin of ABS. Their focus was on the use of spectral fatigue methods and linear elastic-fracture mechanics. In their synthesis, the lifetime load spectrum assumes the conventional short term of a scatter diagram, and a semi-empirical allowance is included to account for the effects of vibration.

The paper concludes that the estimated fatigue life is not overly sensitive to the accuracy of the allowance for vibration. The paper addresses the influence of nonlinear rolling on some of the load components and proposes an approach to one of the more vexing current problems—the pressures on the side shell in way of the mean waterline. The paper also proposes an approach to the combination of global and out-of-plane loads and makes the important point that, in many practical applications, the various load effects must be converted into stresses. In effect, we do not have one universal "lifetime load spectrum." In practice, a given "load spectrum" may have stress units—what is a "response" at one stage of a procedure may well become a "load" at the next.

An invited discussion during the workshop session, by Mr. William Hay of Carderock Division, Naval Surface Warfare Center, highlighted the fatigue analysis methodology currently used by the U.S. Navy. This approach relies heavily on the development of a load-exceedance curve from a model test or from full-scale data to account for both ordinary wave-induced and slam-induced whipping loads. The effects of local and out-of-plane pressure loads are not taken into account in this analysis. The Navy's load-exceedance curve is effectively the same as the "lifetime load spectrum."

Conceptually, there are far more similarities than differences between the ABS and Navy approaches. For low-frequency stress variations, both approaches rely on a linear random synthesis of the ship's lifetime mission profile. Both are at least somewhat empirical in the matter of the allowance for the effects of vibration. The major differences are in how the allowance is carried out and in how much empiricism is involved.

So long as the fatigue-life computation is not inordinately sensitive to minor errors in the load spectrum, it would appear that vigorous engineering, as opposed to an excess of rigor, should carry the day. There are clearly some weak points. Arguably, the worst of the weak spots was addressed in the charge to the group—the uncertainty measures pertinent to the lifetime load spectrum may not have been defined, and it is by no means certain that good quantitative measures are available.

Because fatigue is a local problem, the further implication to some naval architects is that to be useful in fatigue analysis the units of the long-term distribution must be, or must readily convert to, stress units. Physical loads, as understood by hydrodynamicists, are merely an intermediate step in the process. No serious connection can be made between ship dynamics and fatigue without the stress analyst.

By whatever name, the "long-term distribution of stress reversals" implies a sort of stationary statistical process over the ship life and an independence of the result upon the sequence of high and low stress reversals. The sequence of magnitudes of stress time–history has long been considered to be important. Presumably, a severe storm early in a ship's life may have the effect of providing stress relief, which may in turn enhance fatigue life. Hydrodynamicists do not have the means to quantify this effect.

Finally, it is understood that in the long-term distribution of stress reversals, the stress-reversal events are normally defined as if the event of interest is approximately equivalent to the maximum range of stress variation during a single wave-encounter cycle. Under this condition, any vibratory response of the local structure may augment the stress range, but it does not usually contribute to the number of stress-range events per wave encounter. The effect is total elimination of the effect of additional vibratory cycles on the fatigue life of the structure. To cite an extreme example, fatigue cracks in plating near the propeller are more related to blade rate than to any combination of wave-encounter frequencies.

Limitations of Current Load-Prediction Methods

There are a number of major technical limitations related to current procedures for developing lifetime load spectra. First, the methods for combining low- and high-frequency responses—for example, ordinary wave-induced loads plus slam-induced whipping—are far from rigorous. The basic approach is to add a predicted maximum slam-induced whipping response distribution to the predicted maximum wave-induced load distribution and account for some average phasing between the two responses. This procedure is likely to be conservative, although the degree of conservatism is unknown. Another limitation is that the process of performing model tests in order to develop a database for high-frequency responses is far too costly and too time consuming.

Pressure variations close to the waterline are the main cause of fatigue problems on the side shell. However, there is no universally accepted procedure for predicting side-shell pressures. This is also a major limitation. In the view of the workshop participants, linear theory will overestimate the pressure range.

For practical reasons we are forced to use linear models, both in the prediction of hydrodynamic loads and in the development of a lifetime spectrum. The synthesis approach relies on the assumption of linear superposition. Nonlinear, three-dimensional, time-domain simulations can be used for predicting ship loads and motions. However, the ratio of computer time to the real time that is being simulated is of such magnitude—50:1 with the simplest code on a fast workstation—that one can only expect to use such methods for selected conditions. Although frequency domain programs can be used to determine a good portion of the lifetime load spectrum, such a method uses an average value for roll damping. This will impact calculations for both lateral loads and pressure.

Yet another limitation is in estimating the uncertainty in predictions of vertical, lateral, or torsional moments as well as side-shell and bottom pressures. The degree of uncertainty varies with each type of load. Standard measures of uncertainty in lifetime load spectra do not appear to be available. It is generally agreed that the levels of uncertainty are likely to vary quantitatively as follows:

- Vertical shears and moments. The resulting predictions based on current methods are probably fair.
- Lateral shears and moments. Results are not nearly as well known as vertical response, and predictions are likely to be less accurate than those for loads occurring in the vertical plane.
- Torsional moments. In most cases torsion is not of great concern. As a result, not much effort has been put into the development of an analytic prediction capability; therefore, uncertainty is likely to be very high.

- Pressures. The need to determine pressure along the side shell is a relatively new issue that is sometimes attributed to unintended consequences of the increased use of lower-scantling, high-strength steels. The degree of uncertainty is unknown.

How Should We Perform Predictions?

Several capabilities need to be developed or refined in order to more accurately predict load spectra. These capabilities include the development of physics- and geometry-based time-domain simulation capabilities. Such simulation tools should have the ability to predict vibratory responses. In addition, the development of a lifetime load spectrum should account for some nonlinearities. At a minimum, these nonlinearities should include roll response, treatment of transient vibratory response as a nonlinear threshold problem, and pressure variation close to mean water level.

Other aspects of loads prediction that should be addressed are (1) modeling loads for a ship caught in a storm for an extended period of time and the effect of these loads on the fatigue life of the ship; (2) the effect of the route of the ship on the fatigue load spectrum and the need to develop route-specific load spectra; (3) uncertainty in the probability distribution of the load spectrum; and (4) the feasibility of using data acquisition systems to develop extended load records in service.

Findings in Loads

There is a great need for a more rigorous method for developing lifetime load spectra, including the following:

- Develop a rigorous approach for combining low- and high-frequency responses.
- Develop an experimental database for side-shell pressure distribution, both model and full scale, using data acquisition systems.
- Use hull-stress monitoring data acquisition systems to develop extended load records in service.
- Assess the impact that inaccuracies in roll prediction have on the development of fatigue damage.
- Perform a systematic series of experiments that address:
  - influence of ship rigidity on vibratory response
  - variation in hull geometry, including bow flare and bulbs, and its impact on vertical and lateral loads.
- Perform uncertainty analyses on the entire procedure, including the effects of extended exposure to storms and ship routes.

FATIGUE AND FRACTURE OF SHIP STRUCTURE

Background

The classic brittle failures of the early welded Liberty ships and T2 tankers during World War II motivated considerable work aimed at avoidance of such failures. Early solutions consisted of design and fabrication changes, which were followed by changes in materials specifications. In the intervening half century, the engineering discipline of fracture mechanics has been developed to address the broad issue of fatigue and fracture in various types of structures. The physical basis of many common fracture and subcritical cracking mechanisms is now well understood and is documented in the literature. Also, characterizing parameters, based on laboratory measurements of material resistance to fatigue or fracture, have been developed to predict the conditions for failure of engineering structures more accurately. Many industries have adopted fracture mechanics procedures into their construction standards and materials requirements. These requirements include, among others, (1) American Petroleum Institute (API) Standard API RP-2A (API, 1989); (2) British Standards Institution (BSI) Published Document BSPD 6493 and BS 7608 (BSI, 1991, 1993); (3) American Society of Mechanical Engineers (ASME) Standard ASME Section XI (ASME BPVC, 1994); (4) Central Electricity Governing Board (CEGB) Procedure CEGB-R6 (Milne et al., 1996); (5) American Welding Society (AWS) Standard D1.1 (AWS, 1992); and (6) American Association of State Highway Transportation Officials bridge code (AASHTO, 1993). The existence of these established requirements for other industries indicates that similar documents can be developed for ship structure and that, where there are similarities, these existing documents can form the basis for the requirements for ship structure.

It is ironic that the American shipbuilding industry, which provided the initial motivation for many developments in fracture mechanics, has yet to fully embrace fracture mechanics concepts in their construction codes and material requirements. Although all relevant issues cannot be fully addressed, sufficient information and procedures are available to permit the use of fracture mechanics in the design phase of ship structure. This application of fracture mechanics will minimize the potential for both fatigue and catastrophic brittle fractures.

The workshop session on fatigue and fracture explored the immediate adoption of established procedures and identified areas in which these procedures could be refined to improve their relevance and applicability to the control of fracture and fatigue failures in ship structure. Participants in the session included representatives from ship-design agencies, ship owners, research organizations, classification societies, universities, and government agencies.

Objectives

The Fracture Symposium Planning Group assigned the following objectives to the participants in the workshop session on fatigue and fracture.

- Develop a plan to implement the relevant technology in fatigue and fracture analysis to solve the basic problems in that area for ship structures. This includes improving design, inspection procedures, and maintenance procedures to prevent failures.

- Consider the economic impact of recommendations on analysis, design, fabrication, inspection, and maintenance.
- The technology described in "Fatigue and Fracture of Ship Structures" (see Part II) should be used to address the problems presented in the "Defining the Problem" session of the symposium.
- Fatigue and fracture resistance need to be expressed in a format suitable for reliability-based design.

Fatigue and Fracture Control

Fatigue and fracture control in ship structure is a very broad topic. The participants in the workshop session developed a matrix and divided the issue into nine categories among three design philosophies and three phases of a ship's service life. The three design philosophies are safe life, damage tolerance, and fail-safe. The three phases of a ship's life considered are initial design, fitness-for-service, and life extension. Life extension decisions must be made when extensive deterioration has occurred to the structure, either from corrosion, structural damage, extensive cracking, or a combination of these factors. The matrix of these three philosophies and three phases defines the nine categories. Each of these categories is addressed individually in the following discussion.

Safe Life

Safe-life components are those whose failure would result in loss of use of the structure. Safe-life components must never have cracks that exceed a maximum allowable size during the anticipated service life. To achieve this result, the computed or tested life of a structure or component is decreased by a life-reduction factor to obtain the anticipated service life. The life-reduction factor is expressed as a number greater than unity and is defined as follows:

$$\text{Life-Reduction Factor} = \frac{\text{Life Obtained from Fatigue Tests}}{\text{Anticipated Service Life}}$$

The life-reduction factors used by the Federal Aviation Administration range from 2.3 to 3; the U.S. Air Force uses 4. Safe-life design is applied where fatigue is a safety problem. Application of a safe-life approach requires knowledge of the fatigue performance of the structure, including environmental and other loading effects, that are not well known for ship structure.

Damage Tolerance

A damage-tolerant structure has the ability to sustain anticipated loadings in the presence of fatigue, corrosion, or accidental damage until such damage is detected through inspections or malfunctions and is repaired. A damage-tolerant design includes three distinct elements that are of equal importance in achieving the desired level of structural safety. These elements are residual strength (i.e., allowable damage), crack growth (i.e., damage), and damage detection (i.e., inspection). The Federal Aviation Administration, for example, requires demonstration—by analysis, tests, or both—that catastrophic failure or excess structural deformation is not possible after full or partial failure of a principal structural element.

Fail Safe

A fail-safe structure will support loads with any single member failure or with partial damage to a large part of the structure. The residual strength of the fail-safe structure is adequate to prevent severe damage, excessive vibration, and similar types of problems. Based on this definition, a fail-safe structure is seen to be one that can contain the damage. Fail-safe design is applied in cases where fatigue is a maintenance problem. Application of a fail-safe approach requires the structure to have multiple load paths that facilitate load transfer between members. Materials used to fabricate a fail-safe structure should be tough and should experience a slow rate of fatigue-crack propagation.

Specific Findings in Fatigue and Fracture

The state-of-the-art in fracture mechanics is such that specific procedures now exist that should be incorporated into the initial design, fitness for service, and life-extension phases of ship structure operation. The workshop participants' findings in each of the nine categories considered are outlined in the following sections.

Initial Design—Safe Life (Category 1)

This fatigue consideration generally uses S–N curves to predict the behavior of various details used in welded construction. ABS has recently developed guidance and published it in *Guide for Fatigue Strength Assessment of Tankers* (ABS, 1993b). This guide should be endorsed and used during the design stage of ship structure. The ABS SAFEHULL design system is also useful for initial design. Phase A of SAFEHULL provides overall guidance but does not account for the actual loading conditions that a particular ship may experience or the actual stress conditions for a particular structural detail within a ship. Phase B of SAFEHULL, although not as comprehensive as the ABS fatigue guide, is also useful, as are procedures such as those described by Sucharski and Cheung (1993) and Payer and Fricke (1994).

Fitness-for-Service—Safe Life (Category 2)

General fitness-for-service procedures are well documented in *Guidance on Methods for Assessing the Acceptability of Flaws in Fusion Welded Structures* (BSI, 1991). In addition, a simplified *Fracture Mechanics Methodology for Fracture Control in Oil Tankers* (Rolfe et al., 1993) has been incorporated as an option by the U.S. Coast Guard in their critical-area inspection plans. The fitness-for-service procedures that are currently available should be used. Either of the two fitness-for-service procedures mentioned above would be an appropriate starting point for the necessary inspections.

Evaluation of the safe life of ship structure containing a known crack should be made using a fracture mechanics analysis of crack growth during service. This analysis should account for the mode of failure, changing conditions in the structure during crack growth, and the loading sequence, including mean stress and stress range.

Life Extension—Safe Life (Category 3)

The primary difference between this category and the previous category is the degree of damage. This category refers to significant damage of the ship structure, such as large fatigue cracks or loss of several members of large (several feet) portions of the hull structure. The methodologies recommended in category 2 should also be used to evaluate the suitability for life extension in category 3.

Initial Design—Damage Tolerance of the Primary Structure (Category 4)

This category refers to that level of initial damage, such as weld defects, misalignment, or out of fairness, that can be tolerated based on known levels of fracture toughness and assumed fatigue loadings. Obviously, if the initial damage is severe, it should be repaired. Currently, the level of acceptable initial damage is established using workmanship standards that are based primarily on historical evidence. However, a designer can apply fracture mechanics to determine if the existing materials, quality of fabrication, and assumed loadings are such that any initial damage (or quality of fabrication) can be tolerated throughout the life of a structure.

Fitness for Service or Life Extension—Damage Tolerance of Primary Structure (Categories 5–6)

These two categories can be dealt with using the methods described in categories 2 and 3. Actually, categories 1, 5, and 6 are more likely occurrences than categories 2, 3, or 4.

Fail Safe (Categories 7–9)

These categories are restricted to secondary (noncritical) members. It should be noted that cracks in these members can spread to primary members or influence the loading in primary members. Cracking in secondary members is probably a major cost factor in the continuing inspection and repair of ship structure. Accordingly, the influence of cracks in these members on the overall structural integrity of ship structures requires quantification. Development of guidelines regarding the need for inspection and repair of secondary structure could result in major cost savings in view of the high incidence of cracking in secondary structure and the low criticality of these cracks (relative to cracks in primary structure).

Assessments in these categories can be dealt with using the methods described in categories 2 and 3. However, realistic assessment of crack criticality will depend on an accurate assessment of the off-loading of secondary structure to redundant load paths, which occurs as cracking progresses. If this off-loading is not accounted for, application of category 2 and 3 analyses will unnecessarily indicate the need to repair many cracks.

Overall Findings in Fatigue and Fracture

In addition to the specific findings, there are overall findings for the nine categories, general methodologies, coordination of professions, input data, and miscellaneous findings.

Categories

The workshop surfaced technical issues in all nine categories that could be grouped into four categories, as noted below. In the view of the CMS, these areas suggest subjects of needed research.

- Initial Design—Safe Life (Category 1). SAFEHULL or equivalent methods of analysis should be applied to tankers. Similar guidelines are needed for bulkers.
- Fitness for Service—Safe Life (Category 2); Life Extension—Safe Life (Category 3); Fitness for Service or Life Extension–Damage Tolerance of Primary Structure (Categories 5–6). The U.S. Coast Guard's *Fracture Mechanics Methodology for Fracture Control in Oil Tankers* (Rolfe et al., 1993) should be used for simplified assessment, and British Standards Institution's *Guidance on Methods for Assessing the Acceptability of Flaws in Fusion Welded Structures*, PD 6493 (BSI, 1991), should be used for detailed assessment.
- Initial Design—Damage Tolerance of the Primary Structure (Category 4). An interdisciplinary group that includes designers, operators, fracture mechanists, classification societies, and the U.S. Coast Guard needs to be convened to revise workmanship standards based on currently available information.
- Fail Safe (Categories 7–9). The methodologies detailed in the publications described in categories 2 and 3 above need to be applied. However, realistic

assessment of crack criticality will depend on accurate assessment of the off-loading of secondary structure to redundant load paths that occur as cracking progresses. If this off-loading is not accounted for, application of category 2/3 analysis will unnecessarily indicate the need to repair many cracks. Guidelines for inspection and repair of secondary structures based on category 2/3 analysis should be developed as these guidelines could result in major operational cost savings.

General Methodologies

In view of the fact that service loadings are not well established, any fatigue and fracture analyses developed for the shipbuilding community should be relatively simple to use so that they become widely adopted throughout the profession. Experience has shown that if fatigue and fracture analyses procedures become too complex, such as those in level 3, as discussed in section 2 of PD 6493, it is very difficult to implement them during the design phase of any kind of welded structures. Studies should be made to determine the feasibility of adopting the S–N fatigue approach to the hot-spot stress approach of API RP-2A.

Coordination of Professions

Operators and Fracture Experts. The concept of voyage planning to minimize the effect of heavy sea states on fatigue-crack propagation in ship structure has been adopted by several companies and should be explored as a methodology for proactively decreasing the fatigue loading of ship structure.

Designers, Inspectors, and Operators. During the design phase, the designer should provide for access to critical areas that will require inspection during the life of a structure.

Designers and Fabricators. Designers and fabricators should work together to minimize the occurrence of those details throughout the ship structure that are particularly susceptible to fatigue damage.

Input Data

Fatigue monitoring devices or indicators should be installed on ships to measure the actual fatigue damage occurring during service. Knowing the actual loads would be a definite improvement. Also, because there are numerous databases on ship structural failures, it would be desirable to collect them into a common format. Fracture mechanics toughness data, expressed in terms of the stress-intensity factor (K), J–integral (J), or crack-tip opening displacement for various ship steels and weldments would also be highly desirable. At present almost all toughness information is in the form of Charpy V-notch impact properties. Finally, the accuracy and ease with which service loadings for ship structures are calculated requires improvement.

Miscellaneous

Techniques for improving post-weld fatigue, which might avoid unnecessary repairs, should be considered. Also fatigue considerations should be taught at the undergraduate level in university curricula.

Fracture-arrest considerations often can be accomplished through design rather than by using materials with greater toughness. Workshop participants suggested that the use of out-of-plane crack arrestors rather than in-plane crack arrestors should be considered. In-plane crack arrestors are plates of higher toughness. An example of an out-of-plane crack arrestor is a wide flange beam inserted where the crack would not only have to rip through the flange but also through the web and the other flange before it continued to propagate.

It is desirable to have a better indication of the actual loadings of ship structures, whether by improved analytical techniques or by closer coordination between analytical techniques and actual measurements of stresses on ships.

Inspection needs to focus on critical areas because of the time and complexity involved. Accordingly, guidance regarding the critical areas within a ship should be developed to assist inspectors.

FABRICATION AND REPAIR TO AVOID FRACTURE

Background

Misalignment of members, distortion of structure, and imperfections in welding occur during ship structure fabrication. These fabrication anomalies affect fatigue life; therefore, the tolerance must be redefined, or the design must be changed to accommodate anticipated defects of construction.

When structure fails in service, it is necessary to decide if any repair to the structure is needed, if the structure is to be restored to its original condition, or if a design change is necessary to prevent future failure. Guidelines are needed to assist in making these decisions in what is usually a short time frame.

Discussion in the workshop session on fabrication and repair centered on current standard practices in fabrication, preventive maintenance, and repair. Among the issues discussed were problems inherent in current practices; immediate and long-term improvements that could be made in current practices; and the procedures, practices, administration, regulation, and research needed to accomplish these changes. Workshop participants included representatives from ship owners, shipbuilders, classification societies, designers, universities, and government agencies and research laboratories.

Objectives

The Fracture Symposium Planning Group assigned the following objectives to the participants in the workshop session on fabrication and repair.

- Address aspects of fabrication, preventive maintenance, and repair that affect fatigue and fracture.
- Emphasize good detailing and the economics of various details.
- Address fabrication issues, including materials, corrosion protection, tolerances, alignment, and cutting.
- Address repair issues, including evaluating and fixing the cause of the cracking and evaluating various repair options.
- Identify additional research and changes in practice that could improve aspects of fabrication, preventive maintenance, and repair relevant to fatigue and fracture.
- Identify the procedures and practices, administration and regulations, and research needed to make a change.

Presentations

During the workshop, Young-Min Lee from Samsung Heavy Industries, presented the paper *Catalogue of Builder's Practice for Fabrication.* (See Part II.) His remarks summarized the current state of fabrication practices that could affect fatigue and fracture. He also discussed material, welding, details (misalignment and gaps), cutting, fabrication of built-up members, assembly block dimension tolerance, and deformation (unfairness).

Mr. Lee stated that owners and classification societies are permitting less pitting and flaking (scaling) of steel plate when it is received from the steel mill. They are also more concerned about tolerances for alignment of structure because of a greater concern for fatigue. At present, the tolerances for alignment are the same for mild steel and for high-tensile steel. Lloyd's now has special alignment tolerances for high-tensile steel, and ABS is considering issuing new tolerances. Mr. Lee indicated that tolerances should be tighter for high-tensile steel because there is greater potential for fatigue problems.

Kuniaki Ishida from Ishikawajima-Harima Heavy Industries also presented a paper, *Substantial Repairs and Maintenance of Hull Structures.* (See Part II.) His remarks emphasized substantial repairs—repairs to fix the cause of the damage and to restore the structure. One example of this type of repair is the rounding off corners on edges to get better coating adhesion. Mr. Ishida also provided a number of examples of what he termed "substantial repairs," including the use of cargo-hatch corner reinforcement to prevent wire chafing from the unloading grabs on a bulk carrier and the addition of small dressing beads to the fillet weld around the toe of a bracket. Mr. Ishida noted that these dressing beads are made with thin, gas metal arc welding wires or shielded metal arc welding. He also discussed the performance of preventive maintenance, such as reinforcement, in the affected areas of transverse bulkheads in wing-water ballast tanks that exhibit slight shear buckling due to corrosion wastage on a bulk carrier. He explained that these buckles could not be observed after the cargo tank was discharged and showed how they could propagate to adjacent areas if not they were not repaired.

David J. Witmer of BP Oil Shipping Company presented *Repair Rules of Thumb*. (See Part II.) Mr. Witmer's remarks summarized current U.S. practices (that are based on experience rather than analysis) for performing repairs to fractured ship structure, including the procedure for the repair of cracks in longitudinal stiffeners. He stated that the current practice for repairing a longitudinal stiffener—if the crack length has not exceeded half the depth of the stiffener—is to drill a stop hole, vee-grind out the crack, and weld the groove. (Generally, these cracks begin at the top of the flange and propagate down into the web.) If the crack is longer, part of the stiffener is usually replaced with an insert. He said that, although the rationale for this criterion is unknown, it is presumed that there would be concern with the ultimate strength if the repair weld extended greater than half the depth. There are no clear guidelines as to the extent of the insert. For example, if the crack is adjacent to a transverse bulkhead, there is no guideline that indicates how close the butt weld of the insert should be to the bulkhead. Mr. Witmer also stated that a perfectly good oil-tight bulkhead should not be disturbed for repair of a longitudinal stiffener.

Workshop participants viewed illustrations of a repair of a side-shell longitudinal that showed where a stop hole was drilled, the crack in the web was vee-ground out, and the groove was welded. Another illustration showed an insert plate that was welded into the face plate with the attached tripping bracket extended using a "soft toe" to relieve the hot-spot stress and prevent the recurrence of fatigue cracking that had begun at the toe of the tripping bracket. The participants agreed that the insert plate was one of the best repair options and that the new vee and weld would not last long before cracking again if the bracket toe was not modified.

Examples of repairs to deck and shell inserts were also presented at the workshop. In the 1970s, the whole plate between existing welding butts or seams was removed and replaced for fractures in shell or deck plating; however, it is now more common to use "postage-stamp" type inserts. There are no consistent guidelines on the minimum size of these inserts (e.g., 300 mm by 300 mm or 450 mm by 450 mm), and both rectangular and circular inserts are used. Circular inserts are not allowed by some U.S. Coast Guard inspectors, although 150-mm-diameter deck penetrations for sounding pipes are permitted. A number of workshop participants reported that U.S. Coast Guard and U. S. Navy ships have used circular inserts for decades with no problems, and representatives from ABS reported that they allow circular inserts, following the guidance in Military Standard 1689 for combatant ships. One oil shipping company has three ships with rectangular deck inserts between deck stiffeners that were made in the 1970s and are still sound—this type of insert would not be allowed today unless it had rounded corners.

The use of bolted doublers is generally accepted as a temporary repair. This is a suitable permanent repair for bridges and could be considered as a permanent repair for bulk carriers, although corrosion could be a problem between the plates. Bolted doublers are not suitable for permanent repairs on tankers because oil and gas could be entrapped between the plates. Many of these issues were discussed in a recent SSC report, SSC-361, "Hull Strapping of Ships" (Basar and Hulla, 1990).

Wet-backed welding is not generally accepted as a temporary repair. There is a concern that the rapid cooling rates that occur during wet-backed welding cause weld cracking. Vacuum boxes are often required on the wet side to keep it dry and to retain heat, thus reducing cooling rates. Most workshop participants agreed that wet-backed welding, if done carefully, can be an adequate temporary repair and could be considered as permanent. ABS has approved procedures for wet-backed welding that depend on water temperature and materials. Preheat is often required, and insulation may be used on the backside (wet side) to keep the heat in. One Japanese shipyard accepts wet-backed welding if the plate thickness is greater than 13.0 mm. Based on the results of extensive laboratory tests in ice cold water, one oil company allows for wet-backed welding, without the use of either insulation or vacuum boxes, if the remaining plate thickness is greater than 6.0 mm and if both torch-drying of the welding area prior to welding and the use of small-diameter welding rods are mandated. Another workshop participant reported that from the heat-flow point of view, plates that are 15 mm or thicker would not experience any difference in weld cooling rates relative to dry welding for a weld heat input corresponding to a shielded metal arc welding 5/16-in. (or smaller) fillet weld. The above discussion relates to mild steel structure, and does not generally apply to high-tensile steel or to areas requiring higher toughness.

Mr. Witmer noted that, in general, the shipping industry is not aware of the benefits that improved welds offer. He also noted that there are data indicating that peening actually decreased fatigue life in one case; however, this is contrary to more than a decade of positive results in bridges and offshore structures. One oil company has reported that they have peened certain details on a tanker and gained 1.5 years of subsequent service without cracking. In addition, over 14 years ago, Lehigh University peened weld repairs of cracks on bridges, and there has been no subsequent reinitiation of the cracks. An example of how the value of peening can be evaluated during fatigue analysis of structural repairs is provided in Gallion, 1993.

A review of peening and other weld-improvement methods was made for the National Cooperative Highway Research Program (Keating and Fisher, 1986). This review found that peening gives better results than weld-toe grinding and tungsten inert-gas remelting of the weld toe. Weld-toe grinding is limited and is intended only to remove weld-toe defects; weld-contour grinding is more extensive and is intended to reduce the stress concentration of the weld. Weld-contour grinding is effective and is used extensively on offshore structures; however, offshore structures have fewer critical welds than ships. Several Japanese shipbuilders are using small weld-dressing beads to achieve contour without grinding, although a comparison of these techniques has not been made. An oil company reported that the dressing beads are considered acceptable in less critical welds but that critical welds should be ground.

Recent experience on pneumatic peening from Lehigh and Trondheim universities was summarized in several papers at the 1993 Offshore Mechanics and Arctic Engineering conference in Glasgow, Scotland, and at a joint AWS/Welding Institute of Canada conference on fatigue that was held in Toronto, Canada, in 1994. Pneumatic peening is cumbersome and uncomfortable for the operator, and ABS is working with Paton Welding Institute in the Ukraine to field test an ultrasonic peening device and expose the industry to a method that works at least as well as pneumatic peening. It was generally agreed by the participants that thermal stress relief is expensive and not very effective. The SSC has begun a research project on weld-improvement techniques, Weld Detail Fatigue Life Improvement Techniques, SR-1379.

Discussion

Steel Grade and Strength

There was agreement from fatigue experts that there is no difference in the fatigue strength of welded details fabricated from different grades (A to E) and strength levels (mild steel and high-tensile steel) of steel. However, scantlings are generally reduced when high-tensile steel is used, which means that the stress ranges from service loading increase. Therefore, more fatigue problems have occurred in ships built with extensive use of high-tensile steel. Tighter tolerances on alignment might be required for ships using high-tensile steel.

Thermo-Mechanical-Controlled Processed Steel

Thermo-mechanical-controlled processed steel is increasingly used because of its enhanced weldability and fracture toughness relative to conventional structural steel. The fatigue strength of this steel is no different than other steels in the as-welded condition. Concern has been expressed about the changes in properties that could occur from repair welding and cutting on this steel. The SSC has begun a project, "Optimized Design Parameters for Welded TMCP Steels," SR-1358, to develop static, fatigue, and fracture-strength requirements for TMCP steels and weldments.

Plate Thickness

It was reported by a workshop participant that, in some cases, shipbuilders order steel to the required thickness minus the tolerance allowed by the classification society in such a way that the average thickness is below the nominal specified thickness. This practice bypasses the intent of the thickness tolerances of classification societies. The classification rules must be enforced in a way that will preserve the intent of the classification societies.

Recently the IACS tightened the unified requirements for the under-thickness tolerance, which is now -0.3 mm. In naval combatant ships, excessive thickness can be a problem due to weight-control considerations of the vessel. Bath Iron Works reported that, although they used to pay a premium for close-tolerance steel, they now have several mills competing for their business and, therefore, receive a better price on plates that have a thickness tolerance of only plus or minus 1 percent. It was reported that if plate began with minus tolerance, the corrosion would reach the allowable wastage limit sooner than that of the nominal specified thickness; therefore, minus tolerances are disadvantageous to the owner.

Rat Holes and Cut Edges

Cutting is perceived to be a significant problem, especially cut edges of rat holes; and flame-cut edges of rat holes are still handcut in the field. All types of edges have defects that affect coating adhesion and that must be ground where necessary. The AWS has developed plastic models that can be compared with cut edges to determine the roughness, and one oil company has supplemental requirements that specify the grinding of any flame-cut edges. It was reported that the fatigue strength of a rat hole is strongly influenced by its shape, particularly by the angle at which the rat hole intercepts the plating. Elliptical rat holes have been used for high-tensile steel to increase fatigue strength by a factor of two. This resulted in a stress concentration factor that was also lowered by a factor of two.

Fitup-Alignment Tolerances

Misalignment causes stress concentrations that can lead to premature fatigue cracking. Mr. Ishida presented examples in which misalignment had caused cracks; he also provided some repair solutions. Formulae to determine the stress concentration factor due to a variety of types of misalignment can be found in *Guidance on Methods for Assessing the Acceptability of Flaws in Fusion Welded Structures* (BSI, 1991). These formulae can be used to determine fitness for purpose of misaligned joints, based on the expected stress ranges and the fatigue strength of the joint. ASTM Committee F-25 developed standard F-1053, which provides information on misalignment tolerances; however, the standard does not discuss the effect of misalignment on fatigue. Japanese standards were developed 30 years ago based on mild steel, but it is not clear if these standards would be applicable to the high-tensile steel used at higher stress ranges.

"Guidelines for Inspection and Maintenance of Double-Hull Tankers," an SSC report that is currently under development, will identify fatigue-prone areas of misalignment in double-hull structures. It is also expected that the report will address methods for improving alignment tolerance and will consider current experience.

The Carderock Division of the Naval Surface Warfare Center has conducted some experiments with misalignment on cruciform-type joints. ABS reported that a large amount of data on this subject could also be obtained from the Russians and Ukrainians. In addition, ABS is beginning a literature review on the effect of misalignment. A figure in the paper that Mr. Sueoka presented at the symposium showed experimental data on the influence of misalignment for one type of detail. Additional analytical and experimental research is needed to rationalize the misalignment tolerances. As mentioned previously, tighter tolerances are required for high-tensile steel because it is used at higher stress ranges.

Deformation and Fairness

Fairness affects the potential for local buckling of panels between stiffeners. Repeated local buckling (referred to as oil-can deformation or panting) can lead to fatigue failures, including the loss of side-shell plating, which has recently occurred in several bulk carriers. The deformation tolerances are normally greater on thinner plates; however, use of thinner plates is undesirable because they are more susceptible to local buckling. Perhaps two criteria, one for critical areas and the other for noncritical areas, should be required. Because the stresses are higher, tighter tolerances may be required for high-tensile steel.

Weld Quality and Workmanship

Exceptionally poor weld quality is sometimes responsible for fatigue cracking. However, in most cases, fatigue cracks originate on the very small (about 0.1 mm deep) discontinuities that always are present at the weld toe. The weld improvement methods discussed above are intended to suppress cracking at these discontinuities. Stress ranges and geometric stress concentrations, not the weld defects, cause fatigue cracking. Defects such as the weld-bead shape and undercut can affect the geometric stress concentration, but these are considered to be adequately covered by present workmanship standards. Heat-affected-zone properties may affect final fracture, but they have little influence on fatigue. Generally, cracks are observed to grow out of the heat-affected-zone and into the base metal before final fracture. Therefore, heat-affected-zone properties are not a significant concern.

During fabrication, visual inspection for undercut and weld profile is considered adequate. Detailed inspection of butt welds using nondestructive test methods is perceived to be important and should be encouraged. Results from nondestructive testing can be used better to explain potential problems that could arise from poor welding practices and, in turn, enable welders to produce higher quality welds.

Post-Inspection Analysis

It is very important to identify the cause of service fatigue-cracking problems. Finite-element methods are increasingly being used for these analyses. One oil company is performing a global finite-element method analysis for all vessels in its fleet; another oil company has completed a fatigue analysis on one of its tanker classes.

Recently several classification societies developed guides for fatigue strength assessment that provide a design-oriented approach to fatigue strength assessment of structural details. Such an approach could also be used to evaluate proposed repair methods for some types of structural details. As a part of its SAFEHULL design procedure, ABS has developed a fatigue assessment guide that is available in a software program for personal computers. Also, Lloyd's Register of Shipping included a fatigue assessment procedure in the SHIPRIGHT program, and Det Norske Veritas provides a fatigue assessment guide in the NAUTICUS hull-design system. Extensive research on fatigue assessment of critical structural details has also been carried out at the University of California at Berkeley since 1991.

Good designs, proven in service, could be benchmarked in order to feed back into new-building design or repair practices, such as those in the Tanker Structure Cooperative Forum's guidance manuals. These manuals have proven to be beneficial to the shipping industry. Fatigue-strength-analysis guidelines for repairs are also needed to assess repair options.

Guidelines for fitness-for-purpose assessment could be developed for ship structure. The BSI's PD 6493 is a good starting point, but this document does not address structural behavior such as load shedding. However, an SSC project, "Evaluation of Ductile Fracture Models for Ship Structural Details," SR-1349, is currently addressing this issue.

Impact of Corrosion on Fatigue Strength

In general, corrosion in structural members either increases the working stress or redistributes the load to adjacent members. Therefore, the susceptibility to fatigue cracking increases as the corrosion continues to progress. The University of California at Berkeley has completed a research project on fatigue strength associated with corrosion problems in ship structure. The SSC has begun the project, "Strength Assessment of Pitted Plate Panels," SR-1356, but this project focuses on strength rather than fatigue. The Tanker Structure Cooperative Forum also performed a pilot-test program to assess the influence of pitting intensity on the residual strength of pitted plate.

Repair Methods

It is important to make a distinction between two types of cracking in order to understand the appropriateness of certain types of repairs. Some cracking is due to distortion. This type of cracking is essentially displacement controlled. The cracking that occurs in the transverse-web frame cutouts around longitudinals is an example of distortion-induced cracking. Typically, smaller distortion-induced cracks can be drill-stopped and left in place without rewelding. The stress due to distortion that caused the cracking is usually relieved by the cracking itself. Rewelding these cracks restores the driving force and ensures the recurrence of cracking; however, "non-repair" decisions currently are not allowed in the shipping industry.

The other type of cracking that affects ship structure, such as the cracking in the face plate and web of longitudinal stiffeners, is due to primary or secondary loads. Cracks that are load induced must be repaired, or an alternative load path, such as doublers, must be provided. Doublers can often provide a satisfactory permanent repair, although they are considered temporary unless they are accompanied by design modifications.

Drilling stop holes is a good option only for through-thickness cracks; surface and part-through cracks cannot be repaired with hole drilling. Lehigh University has developed guidelines for determining the appropriate size of the hole. The U.S. Coast Guard Navigation Vessel Inspection Circular (NVIC) 7-68, *Notes on Inspection and Repair of Steel Hulls* (USCG, 1968), contains some outdated guidelines on stop-hole drilling. For example, the guidelines recommend drilling the stop hole 2 in. beyond the end of the crack. These guidelines need to be revised, especially to reflect the research in the maintenance and repair of ship structure that has occurred since 1968.

Restoration of the structure to the original (as-built) condition is an option only if limited intended service life is required. Design modification or reinforcement (beyond restoration) is needed to prevent cracking. For example, additional collar plates can be effective in reducing the stresses at the intersection of longitudinals with transverse frames, as indicated in the symposium paper by Cheung. Clear guidelines are needed to determine the most effective option in a limited time frame.

Findings in Fabrication and Repair

The methods used for fabrication and repair play important roles in the prevention of fractures in ship structure. The following needs in research should be addressed, and changes in practice should be made:

- The current fabrication standards were developed based on the use of mild steel, but many new ships are constructed of high-strength steel. A publication that summarizes fabrication standards and their impact on fatigue strength and explicitly addresses the effect of high-tensile steel is needed for the industry. In the long term, more experimental data and in-service experience data are needed on the effect of fabrication standards on fatigue strength.
- U. S. Coast Guard NVIC 7-68 has been used successfully by the industry since it was published in 1968; however, some provisions of that document are not suitable for current practices. NVIC 7-68 should be updated to include recent industry experience and research.
- Guidelines for acceptance of temporary and permanent repairs are needed. User experience and experimental data together with analyses, should be acquired and reported. These data can be used to demonstrate the long-term endurance of some repairs that are currently allowed only as temporary repairs and to qualify them as permanent. Guidelines based on this information could then be incorporated into a revised U.S. Coast Guard NVIC 7-68.
- General guidelines for fatigue-strength analyses used in performing repairs to cracked structures should be developed. These guidelines should provide guidance on fatigue-cracking diagnosis and repair methods. In the long term, more data from experiments and experience are required to characterize the fatigue strength of repair methods.
- Guidelines for fitness-for-purpose assessments of critical structural details of ships should be developed. This guidance should include load-shedding effects. BSI 6493 is a good starting point.
- The further enhancement of hull-stress monitoring systems that could provide real-time ship performance data and guidance to the operators and enhance prevention of fatigue fractures should be investigated for future applications.
- There are a few organizations and forums in the international arena, such as the International Ship and Offshore Structures Congress and the Tanker Structure Cooperative Forum, that deal with the ship structural issues. However, in the United States, there is a lack of an information feedback mechanism in the industry to deal with these issues. Reinventing the wheel or repeating the same mistake are common occurrences in the design, fabrication, operation, inspection, repair, and maintenance of ship structure. Therefore, a centralized organization or forum composed of the administrations, classification societies, owners/operators, shipyards, designers, and academic institutions should be formed. The mission of such a forum would be to provide feedback to the industry. Three organizations, the SSC, the National Shipbuilding Research Program, and the SNAME Hull

Structure Committee, represent good mechanisms for this transfer of knowledge, and their functions could be expanded to make them a center of an information feedback process. Such an effort should be supported by the industry.

INSPECTION FOR FRACTURES

Background

After all parties involved in the shipbuilding process have completed their work, after loads have been analyzed, after steel has been cut and welded, after crew and cargo are on board and the ship is in operation, it is the inspection process that plays a major role in ensuring the safety and integrity of the vessel. Yet inspection may be the least understood aspect of a ship's life cycle. The ability to detect defects—cracks, corrosion, and deflection—depends on a wide variety of factors related to the design of the ship, its operation and inspection procedures, and personnel. Control over these factors does not rest with a single authority—some factors are influenced by designers; others by owner/operators; the inspectors, themselves; and classification societies and regulatory agencies. Improvements in the quality of inspection, therefore, may require the cooperation of many parties.

Inspection, particularly of tankers, but also of bulk carriers and other large vessels, takes place under conditions that are at best poor and at worst dangerous. These conditions are summarized in the symposium paper by Williams and Sharpe. Among the factors mentioned as contributing to less-than-ideal conditions were the large area that must be inspected; the lack of easy access; poor lighting, temperature extremes, and time pressure; and, in some cases, the limited experience of the inspector. Efforts to improve the quality of inspection have typically focused on these factors (see, for example, Holzman, 1992; Goodwin and McClave, 1993; and Allen et al., 1993). However, a broader view of opportunities for improvement can be taken, starting with the decisions made and the information stored during design and fabrication and continuing through the inspection process itself to post-inspection data archival, retrieval, and use.

In looking at changes to current practice, it is important to keep in mind that, in general, the more resources applied to inspection, the better the result. If a few weeks could be spent inspecting a particular weld, all of the defects—or at least those detectable given the inspection method used—would surely be found. However, in a world of constrained resources, there is a need to balance the goals of safety and quality of inspection against the constraints of time and budget. Technological improvements in inspection may affect this balance, as may improvements in other aspects of the overall inspection process.

Participants in the workshop session on inspection included representatives of the U.S. Coast Guard (several with significant inspection experience), the U.S. Navy, owner/operators, private inspectors, and university faculties. The participants benefitted from representation of different perspectives; especially, the input of those participants with inspection experience.

Objectives

The Fracture Symposium Planning Group assigned the following objectives to the participants in the workshop session on inspection:

- Develop changes to be made in the inspection of hulls during fabrication, operation, and periodic maintenance and repair to drastically reduce the incidence and severity of fatigue failures. Considerations should include inspection techniques, quality of inspection, critical areas, inspection frequency; the "repair, replace, or leave as-is" decision; and economic consequences of inspection.
- Identify measures that can be taken in design and fabrication to facilitate inspection.
- Identify needed research for improvement of inspection technology relative to fatigue and fracture.

Current Inspection Practice

Inspections are carried out by different groups, including owner/operators, classification societies, and regulatory agencies, such as the U.S. Coast Guard. Each group may have different motivations and requirements for conducting an inspection; therefore, they may use different methods and achieve different outcomes.

The basic approach to detecting cracks is to perform a visual inspection. An inspector—armed with a flashlight, camera, pen, and paper; a chipping hammer; chalk or spray paint; and protective clothing and equipment—walks through the ship looking for defects. Access to the structure may be solely by "walking the bottom" or may include climbing without restraint (in most cases officially prohibited but done nonetheless) or climbing with fall safety devices, portable staging, fixed staging, or rafting. Prior to the inspection, the inspector should have reviewed the history of the vessel, previous inspection and repair reports, and the critical area inspection plan, if one exists. Upon finding a defect, the inspector marks it with chalk or spray paint and notes its location and nature. After exiting the space, the inspector transcribes this information to a more permanent storage media, typically the inspection report or, increasingly, to a database of defects, as well. Depending on the purpose of the inspection, this report may be used to satisfy classification or regulatory requirements or to plan for subsequent repairs.

The Williams and Sharpe paper presented at the symposium provides an excellent overview of current inspection practices; a more detailed description can be found in Holzman, 1992. Efforts to improve the inspection process are ongoing, both in research and implementation. For example, the past several years have seen an increased interest in the use of portable staging that is suspended from the ship's structure. During the course of the symposium and workshop, one owner/operator described a modified approach to rafting of tankers. Typically, rafting is accomplished by filling the tank to each of several discrete levels in turn, with rafting allowed only at these levels. In the new approach, rafting takes place as the tank is drained. This provides continuous access to the structure, rather than access only at discrete levels.

Three types of factors affect inspections—inspector factors, vessel factors, and environmental factors. Inspector factors include the inspectors' experience; knowledge of the vessel's history, including critical areas; preparation, training, and fatigue; point of view (owner versus regulatory agency versus classification society); and the number of inspectors. Vessel factors include the vessel's type, size, access, coatings, location (e.g., shipyard versus 60 miles offshore), load conditions, operational history (i.e., cargo, route, and conditions), and the location of the inspection site within the vessel. The environmental factors are the purpose of the inspection (e.g., close up, repair, annual, special, etc.); the interval between inspections; the planning that proceeds the inspection; the inspection method and procedure used; the report format and cooperation of crew; the time available, access, lighting, and cleanliness; ship motion, temperature, and air quality; and equipment or tools. More detailed descriptions of many of these factors can be found in Ayyub and White, 1992 and in Demsetz et al., 1995.

Opportunities for Improvement

The inspection process can be viewed as having four phases—planning the inspection, carrying out the inspection, documenting inspection results, and using inspection results. Current research, development, and implementation efforts focus primarily on the second and third phases and address specific technologies that could improve inspection practice. Improvements in these phases are to a large extent internal to the inspection process, although there is certainly interaction with the design phase through efforts to design ships so that there is better access for inspection. Workshop participants, for the most part, were aware of these efforts and supported them. There was general acceptance of the idea that new applications of technology could improve the inspection process and facilitate documentation of inspection results. However, the bulk of this workshop session focused on opportunities for improvement in the first phase (planning) and in the fourth phase (use of inspection results). These phases extend beyond the traditional boundaries of inspection.

With respect to planning the inspection, there is a strong need for better information about the probability and consequences of failure. Such information would allow inspection efforts to be focused where they provide the greatest benefit. Determining the probability and consequences of failure requires the integration of knowledge about design, fabrication, operation, and inspection. If the probability and consequences of failure were known, critical or "at risk" locations, details, defects, and defect sizes could be defined and would become the focus of the inspection effort. As used here, the notion of critical or "at risk" encompasses not only the likelihood that a defect will occur but its consequences as well. Such risk-based inspection can be performed in a decision-analysis framework that considers both the likelihood and consequences of structural failure. Additional issues include an understanding of crack growth rates (see Rolfe et al., 1993; Witmer and Lewis, 1994) and an understanding of structural redundancy. There is a related need to determine realistic estimates of detectable crack lengths, particularly in critical or "at risk" areas. The SSC has initiated a project, "Optimal Strategies for Inspection of Ships for Fatigue and/or Corrosion Damage," SR-1365, that addresses this area.

With respect to the use of inspection results, there is a strong need for better use of the data gathered in inspections and for an aggregation of the data available from different organizations—especially from different owners. This would support efforts to understand the likelihood and consequences of failure and would help document the impact of design and repair decisions. It is important that such an effort have strong industry endorsement and that the aggregated data be stored and maintained by a suitable independent entity, such as the University of Michigan's Transportation Research Institute.

Finally, the current focus on inspection has been reactive; unanticipated problems, particularly with the TAPS vessels, has focused attention on the inspection process. With the introduction of double-hull vessels, there is an opportunity to take a proactive approach and to take steps to improve the inspection process before unanticipated problems arise.

New and improved methods and technologies of inspection are important and can help reduce the incidence and severity of fatigue failures. However, the biggest opportunities for improvement lie in the links between design, fabrication, and inspection. Inspection should be driven by knowledge of the probability and consequences of failure. A technical obstacle to obtaining this knowledge is the lack of a consistent format to store information. Potentially more difficult to overcome are the institutional obstacles to developing an industry-wide database of vessel performance. As a starting point, much could be accomplished by developing a standard representation of a ship's inspection history. In addition, better dissemination of previous research should provide significant benefits at modest cost.

Findings in Inspection

Within the broad areas defined above, there are a number of specific issues that need to be addressed. These issues can be categorized as follows: design/analysis, fabrication/assembly, in-service inspection, and industry wide. Addressing some of these issues will require only minor modifications to existing methods; others may require significant resources during development and/or in implementation. In some cases, the desired information is available but is not being transferred to the appropriate party. In other cases, designers and/or fabricators are unable or reluctant to provide the desired information unless specifically required to do so by contract or regulatory agencies. Cost/benefit analyses should be carried out before research funds are allocated to address these issues.

Design/Analysis Issues

- Design procedures should include the identification of critical or "at risk" areas in terms of both the probability and consequences of failure.
- Tolerable and critical flaw sizes should be determined for "at risk" areas.
- Failure modes should be postulated. Structures should then be analyzed to identify "bellwether" cracks that would provide advance warning of a problem.
- Output of the design phase should include a computer-assisted design (CAD) drawing or other representation of the structure that indicates critical areas, critical crack sizes, and predicted service life between detection and criticality.
- Good design practice for details should be followed. The report, SSC-379, "Improved Ship Hull Structural Details Relative to Fatigue," provides guidance in this area and should be more widely disseminated (Stambaugh et al., 1994).

- Designers should continue to consider access and inspectability in making design decisions.

Fabrication/Assembly Issues

- The CAD drawing or other representation that is produced by the design process should be annotated with information from the fabrication/assembly process, including details that required rework or were difficult to fabricate. The intent here is to identify potential locations of "rogue" flaws.
- A post-assembly inspection should be carried out, with particular attention given to alignment of members.

In-Service Inspection Issues

- Probability of detection curves should be developed to provide a realistic estimate of detectable crack size. Major factors affecting the probability of detection, such as the experience of the inspector, the means of access and lighting, and the location of the defect, should be taken into account.
- New technologies to improve inspection performance should continue to be developed.
- The benefits and costs of new technologies,such as the use of acoustic emission, should be investigated, including their effect on probability of detection and on safety.
- As the knowledge of critical or "at risk" areas is improved, inspection procedures should be refined to focus on these areas while also detecting "rogue" or unexpected problems.
- Guidelines for inspector training and certification should be developed, recognizing that different organizations may have different needs.

Industry-wide Issues

- Institutional barriers to the sharing of information should be addressed. Some means of distilling, filtering, and aggregating ship defect histories should be developed so that liability issues do not prevent owners from pooling their information. These issues have been overcome in the airline industry, in part due to the higher visibility of failure. The benefits of shared information, such as prevention of failure, increased reliability, and reduced life-cycle costs, should be quantified to the extent possible.
- A standard representation of a ship's inspection history should be developed. Initial input for this representation should be provided in the design phase and should include an indication of critical or "at risk" areas. The record should be

supplemented with input from fabrication. The record should be updated with each inspection and each repair and should be made available to inspectors prior to each inspection. Currently, there are several different database systems that have been developed to track defects. At the very least, a protocol should be designed to allow data represented in each of these systems to be combined in a single database. Relevant information can be found in SSC-379, Ship Structural Details Relative to Fatigue (Stambaugh et al., 1994).

- Inspection frequency and content should be modified in light of critical or "at risk" areas, realistic assessments of probability of detection, and realistic assessments of crack-growth rate.
- Improved dissemination of, access to, and use of previous research—for example, the body of work reported by the SSC—should be promoted. New means of dissemination should be considered, including CD-ROMs, access over the Internet, and the like. Special attention should be given to improving indexing, abstracting, and search capabilities.

REFERENCES

AASHTO. Standard Specification for Highway Bridges. 1993. Washington, D.C.: American Association of State Highway Transportation Officials.

ABS. 1993a. Guide for Dynamic Based Design and Evaluation of Tanker Structures. New York: American Bureau of Shipping.

ABS. 1993b. Guide for Fatigue Strength Assessment of Tankers. New York: American Bureau of Shipping.

Allen, S.J., B. Macesker, and D. Mazurek. 1993. Innovative technologies for Coast Guard marine inspectors. Pp O-1–O-17 in *Ship Structures Symposium '93* held November 16–17, 1993 in Arlington, Virginia. Jersey City, New Jersey: The Society of Naval Architects and Marine Engineers.

API. 1989. API Recommended Practice for Planning, Designing, and Constructing Fixed Offshore Platforms. API RP 2A, 18th Edition. Washington: American Petroleum Institute.

ASME. 1994. Rules for In-service Inspection of Nuclear Power Plant Components. Section XI, Division 1, Boiler and Pressure Vessel Code. New York: American Society of Mechanical Engineers.

AWS. 1992. Structural Welding Code—Steel, 13th Edition. ANSI/AWS D1.1–92. Miami, Florida: American Welding Society.

Ayyub, B.M., and G.J. White. 1992. A Plan for the Evaluation of Alternative Tank-Vessel Visual Inspection Practices. Avery Point, Connecticut: U.S. Coast Guard R&D Center.

Balint, S. 1992. Design loads for sloshing in TLP pontoon tanks. Civil Engineering in the Oceans. 5:99–113.

Basar, N.S., and R.B. Hulla. 1993. Hull Strapping of Ships. Report SSC-361. Washington, D.C.: Ship Structure Committee.

BSI. 1991. Guidance on Methods for Assessing the Acceptability of Flaws in Fusion Welded Structures, BSI 6493, 2nd Edition. London: British Standards Institution.

BSI. 1993. Code of Practice for Fatigue Design and Assessment of Steel Structures, BSI 7608. London: British Standards Institution.

Capanoglu, C.C. 1992. Fatigue Technology Assessment and Strategies for Fatigue Avoidance in Marine Structures. Report SSC-367. Washington, D.C.: Ship Structure Committee.

Chevron. 1993. Inerting and Gas Freeing Tests of Double-Hull Water Ballast Tanks. Presented to Tanker Structures Cooperative Forum, TSCF Work Group Meeting #17, October 6–7, 1993.

Demsetz, L.A., R. Cario, and R. Schulte-Strathaus. 1995. Inspection of Marine Structures. Draft final report submitted to the Maritime Administration. Project DTMA91-93-G-00040 and SSC-SR-1355.

Gallion, K.A. 1995. Structural Maintenance Project, vol. 3—Repair Management System. Report SSC-386. Washington, D.C.: Ship Structure Committee.

Goodwin, M.J., and E.F. McClave. 1993. Innovative Inspection Techniques, vol. 1. Avery Point, Connecticut: U.S. Coast Guard Research and Development Center.

Holzman, R. 1992. Advancements in Tankership Internal Structural Inspection Techniques. M.S. thesis. University of California, Berkeley.

Keating, P.B., and J.W. Fisher. 1986. Evaluation of Fatigue Tests and Design Criteria on Welded Details. Report NCHRP-286. Washington, D.C.: National Cooperative Highway Research Program.

Kihl, D.P. 1994. Axial fatigue behavior of advanced double hull combatant weld details. Presented to The Advanced (Unidirectional) Double Hull Symposium at the National Institute of Standards and Technology, Gaithersburg, Maryland, October 25–26, 1994.

Kihl, D.P., S. Sarkani, and J.E. Beach. 1995. Stochastic fatigue damage accumulation under broadband loadings. International Journal of Fatigue. 17(5):321–328.

Milne, I., R.A. Ainsworth, A.R. Dowling, and A.T. Stewart. 1996. Assessment of the Integrity of Structures Containing Defects. R/H/R6 – Revision 3. Leatherhead, Surrey, United Kingdom: Central Electricity Generating Board.

Munse, W.H., T.W. Wilbur, M.L. Tellalian, K. Nicoll, and K. Wilson. 1983. Fatigue Characterization of Fabricated Ship Details for Design. Report SSC-318. Washington, D.C.: Ship Structure Committee.

NRC (National Research Council). 1991. Tanker Spills: Prevention by Design. Washington, D.C.: National Academy Press.

Payer, H.G., and W. Fricke. 1994. Rational dimensioning and analysis of complex ship structures. Society of Naval Architects and Marine Engineers Transactions. 102: 395–418.

Rolfe, S.T., A.E. Henn, and K.T. Hays. 1993. Fracture mechanics methodology for fracture control in oil tankers. Pp F-1–F-11 in *Ship Structures Symposium '93* held November 16–17, 1993 in Arlington, Virginia. Jersey City, New Jersey: The Society of Naval Architects and Marine Engineers.

Sarkani, S., D.P. Kihl, and J.E. Beach. 1992. Fatigue of welded cruciforms subjected to narrowband loadings. Journal of Engineering Mechanics. 118(2):296.

Sarkani, S., D.P. Kihl, and J.E. Beach. Fatigue of welded joints under narrowband non-Gaussian loadings. Probabilistic Engineering Mechanics. 9(3):179–190.

Stambaugh, K.A., F. Lawrence, and S. Dimitriakis. 1994. Improved Ship Hull Structural Details Relative to Fatigue. Report SSC-379. Washington, D.C.: Ship Structure Committee.

Sucharski, D., and M. Cheung. 1993. Application of spectral fatigue analysis to the ARCO 190 MDWT tankers. Pp M-1–M-14 in *Ship Structures Symposium '93*. Held November 16–17, 1993 in Arlington, Virginia. Jersey City, New Jersey: The Society of Naval Architects and Marine Engineers.

Taggart, R. 1980. Ship Design and Construction. Jersey City, New Jersey: The Society of Naval Architects and Marine Engineers.

TSCF. 1986. Guidance Manual for the Inspection and Condition Assessment of Tanker Structures. International Chamber of Shipping and the Oil Companies' International Marine Forum, Tanker Structures Cooperative Forum. ISBN 0 900 886 978. London: Witherby & Co. Ltd.

U.S. Coast Guard. 1968. Notes on Inspection and Repair of Steel Hulls. Navigation Inspection Circular NVIC 7-68. Washington, D.C.: U.S. Coast Guard.

U.S. Coast Guard. 1989. Report of the Tanker Safety Study Group. Washington, D.C.: Washington, D.C.

Witmer, D.J., and J.W. Lewis. 1994. Operational and scientific hull structure monitoring on TAPS trade tankers. Transactions of the Society of Naval Architects and Marine Engineers. 100:501–533.

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Conclusions and Recommendations

CONCLUSIONS

The technology is available to reduce fracture in ship structure, and, although there are areas that need improvement, the most important need is to apply current knowledge more effectively to all aspects of ship design, operation, inspection, maintenance, and repair. The conclusions of the CMS for these four areas are as follows:

- **Conclusion 1.** Design—*Design* methods for fatigue resistance are available, but historically, these methods have not been used. Continued application and enhancement of known design fatigue methods should be pursued. Any operational failures should be linked back to initial analyses and design as important lessons learned, and those details to be avoided in the future should be indicated.
- **Conclusion 2.** Loads—A better definition of *loads* and how to apply them is needed. The methods for combining low- and high-frequency responses are not sufficiently rigorous. The current approach of adding a predicted maximum slam-induced, whipping-response distribution to the predicted maximum wave-induced load distribution and accounting for some average phasing between the two responses is likely to be conservative. The process of performing model tests in order to develop a database for high-frequency responses is far too costly and time consuming.
- **Conclusion 3.** Inspection and Repair—Better tools are needed for *inspection and repair* to increase understanding of the likelihood and the consequences of cracks on the in-service condition of ships. The impact on fatigue strength of fabrication standards and the effect of higher-strength steel have not been made widely available to the industry.
- **Conclusion 4.** Communications—Improved *communication* for better dissemination of fatigue design information is needed. Design procedures are available but are not being used to identify critical or "at risk" areas in terms of both the probability and consequences of failure. The good design practice for structural details that has been developed in several technical reports, such as the SSC report, "Improved Ship Hull Structural Details Relative to Fatigue," SSC-379 (SSC, 1994), should be more widely disseminated. Current fatigue and fracture technology is not being conveyed in an understandable and useful form to

practicing naval architects or to undergraduate students of naval architecture. Sufficient information and procedures are available to permit applications of fracture mechanics in the analysis of ship structure in all life-cycle phases, but most practicing naval architects are unaware of them.

The dissemination of design information should include both the S–N approach for crack initiation and the fracture-mechanics approach of crack growth analysis. The S–N approach reflects a long history of application in the field of materials engineering and has been applied extensively for the analysis of ship structure. The application of fracture mechanics to ship structure analysis is more recent relative to its use in the aeronautics and nuclear power industries. However, such applications are expanding in efforts to better understand and reduce fatigue failure in ship structure.

RECOMMENDATIONS

Action is required by regulators, designers, fabricators, maintainers, owners, and operators of ships to reduce the fracture-related failures in ships. The CMS recommends that specific action be taken in the areas of design, loads, inspection and repair, and communications, as described below. Many of these recommendations, especially those calling for additional research, will be further developed by the CMS in its biennial research recommendations to the SSC. However, classification societies also need to develop formal fatigue design criteria. Ship designers, fabricators, and maintainers need to consider the prevention of fracture in more detail; owners and operators of ships need to demand the application of this technology.

Design

- **Recommendation 1.** Develop a ship-detail guidebook/standard for designers that addresses newer details and fatigue predictions. Producibility should be considered in addition to weight and fatigue performance. Such a guide would facilitate the application of fatigue information in design, reduce design time and cost, enhance ship producibility, and reduce maintenance costs.
- **Recommendation 2.** Expand simplified fatigue analysis methods, such as SAFEHULL for tankers, to include transverse structure, more complex details, and other ship types. This formalization of fatigue design procedures provides a method of improving ship structure design in a way that most structural designers can understand easily.
- **Recommendation 3.** Gather additional data on the fatigue strength of various large-scale ship details, including repair details and the validity of linear cumulative-damage prediction (Miner's Rule) under high-cycle,

variable-amplitude loadings, such as slamming. Development of this information requires a program of large-scale testing to validate current assumptions on fatigue-life prediction, including the definition of failure.

- **Recommendation 4.** Design ship structure to include access for service inspections. Revise structural design criteria to include different tolerance and fabrication standards for higher-strength steels. This change in structural design practice would require better communication among designers, fabricators, and inspectors. Attention to inspection of structure during design will minimize the effects of of time and budget constraints against the goals of safety and quality of inspection. The higher stress levels associated with the use of higher-strength steel have not been reflected in fabrication standards that were originally developed for mild-steel structures.

Loads

- **Recommendation 1.** Develop a rigorous approach for combining high- and low-frequency response. Such an approach should be suitable for use in all phases of fatigue analysis, yet it should not be overly conservative or depend on the performance of model tests.
- **Recommendation 2.** Compute loads using mechanics- and geometry-based simulations. All nonlinearities should be included in such efforts. These computations would include hull vibratory responses, roll response, and pressure variation close to the waterline.
- **Recommendation 3.** Develop a relatively inexpensive, easily operated hull-stress monitoring system that provides real-time feedback to the operators. Through the use of such a system, a ship's service experience can be linked with anticipated fatigue failures. When such a system is combined with weather routing, the service life of a ship can be extended.
- **Recommendation 4.** Quantify the degree of uncertainty in load predictions. This information is necessary if probabilistic methods are to be used in predicting the risk of structural failure.

Inspection

- **Recommendation 1.** Develop guidelines for fitness-for-purpose assessments, including structural behavior, based on in-service history and fatigue analysis. These guidelines would include the effect of individual fatigue failures on overall ship hull structural strength and would provide a basis for determining the need for immediate repair of cracks in structure.

- **Recommendation 2.** Develop new, and improve existing, inspection tools. U.S. Coast Guard Navigation and Inspection Circular NVIC 7-68, which is used as a guide for inspection and repair of steel hulls, should be revised. Any revisions or newly developed guides should be premised on the outputs from life-cycle databases.
- **Recommendation 3.** Quantify the fatigue life of temporary repairs, including the use of both wet-backed welds and doublers as permanent repairs based on service history and fatigue analysis. Current regulations limit the use of such repairs, but the use of fatigue analysis could provide for extension of the service life of such repairs and reduce maintenance costs.

Communications

- **Recommendation 1.** Develop a cradle-to-grave ship structural integrity database system that includes hull-stress monitoring, inspection, and repair data. Owners and operators should take advantage of available fatigue design/analysis information in developing inspection plans.
- **Recommendation 2.** Develop and promulgate a manual/library, sorted by levels of fatigue strength of predictable standard details for fatigue resistance.
- **Recommendation 3.** Educate and train future structural designers at the undergraduate level in fatigue and fracture design methods. Most practicing designers of ship structure need fatigue and fracture training. They must understand the basic phenomena, how to recognize situations in which fatigue and fracture are concerns, what design tools are available, and how to use them. To correct this situation, education and training that focuses on practicing designers of marine structures are required. The SSC as well as the Society of Naval Architects and Marine Engineers—in both the Hull Structure Committee and the Education Committee—should play leading roles in this regard. However, this process must be an integral part of an engineer's education, and undergraduate curricula in naval architecture, as well as in related fields, such as civil and mechanical engineering, should include fatigue and fracture design methods.

In brief, current fatigue and fracture technology must be conveyed in an understandable and useful form to practicing naval architects. Only then will large gains be made in the ongoing struggle against these problems. This basic education would be the beginning of greater understanding of this vital technology in all phases of ship life cycles.

- **Recommendation 4.** Use concurrent engineering in ship design to include designers, owners, operators, fabricators, and surveyors/inspectors in the process. In-service review of findings should include all of these groups. Prevention of fracture in ship structure was examined during the workshop sessions in the separate categories of design, loads, fatigue and fracture, fabrication and repair, and inspection; however, because of the interrelationship among these areas, they must be considered as a whole. In the total context of ship design, construction, operation, and repair, the message must be conveyed that good practices can significantly reduce the incidence of fracture.

REFERENCE

Stambaugh, K.A., F. Lawrence, and S. Dimitriakis. 1994. Improved Ship Hull Structural Details Relative to Fatigue. Report SSC-379. Washington, D.C.: Ship Structure Committee.

APPENDIX A

Biographies

AUTHORS

James C. Card is the chief of the Office of Marine Safety, Security, and Environmental Protection of the United States Coast Guard. Prior assignments include Commander, Eighth Coast Guard District, New Orleans; chief, Merchant Vessel Inspection and Documentation Division; commanding officer, Marine Safety Office/Group, Los Angeles/Long Beach, California; chief of operations, 11th Coast Guard District, Long Beach, California; chief of staff, 13th Coast Guard District, Seattle, Washington; and several other tours ashore and afloat. He has received two Legion of Merit awards, three Meritorious Service awards, and a Coast Guard Commendation Medal. He is the chair of the interagency Ship Structure Committee and heads the U.S. delegation to the Maritime Safety and Marine Environmental Protection committees of the International Maritime Organization. He is a member of the Society of Naval Architects and Marine Engineers, the American Society of Naval Engineers, the Royal Institution of Naval Architects, the American Bureau of Shipping, Det Norske Veritas, International Cargo Bureau, Marine Index Bureau, Marine Engineering Council of Underwriters Laboratories, and the Sealift Committee of the National Defense Transportation Association. Rear Admiral Card is a graduate of the United States Coast Guard Academy, received an M.S in naval architecture and in mechanical engineering from the Massachusetts Institute of Technology, and is a graduate of the Industrial College of the Armed Forces.

Y. N. Chen is with the Research and Development Group of the American Bureau of Shipping. He formerly taught at the Polytechnic Institute of Brooklyn. His research interests include nonlinear shell theory, structural dynamics, elasticity and plasticity, probabilistic mechanics, reliability, fatigue, and fracture. Dr. Chen received a B.S. in mechanical engineering from the National Taiwan University and a Ph.D from the Polytechnic Institute of Brooklyn.

Maxwell C. Cheung is president of MCA Engineering and specializes in structural mechanics of dynamics and stability. Prior to founding MCA Engineering in 1972, he was with Global Marine. He is a member of the Society of Naval Architects and Marine Engineers and is a member of the nominating committee. He was formerly chair of the Los Angeles Metropolitan Section and received the Outstanding Section award. Dr. Cheung received a Ph.D. from the California Institute of Technology in 1969.

Mark D. Debbink is the structural supervisor for New Product Engineering of Newport News Shipbuilding. He was formerly a First Class Pilot (Great Lakes) with American Steam Ship Company. Mr. Debbink received a B.S in civil engineering from Michigan State University, and an MBA from the College of William and Mary.

Paul A. Blomquist is a research assistant at the Applied Research Laboratory of Pennsylvania State University. Prior to that, he was senior welding engineer at Bath Iron Works. Previous positions were with the Electric Boat Division of General Dynamics Corporation and other positions concerned with welding in the heavy construction industry. He is a member of the Welding Panel (SP-7) of the National Shipbuilding Research Program, and is a member of the Technical Advisory Board of the Navy Joining Center. Mr. Blomquist received a B.S. in applied science from Charter Oak College in Connecticut.

David O. Harris is a principal engineer and vice president at Engineering Mechanics Technology, Inc. He has worked in fracture mechanics for the last 30 years, including development of analysis tools for probabilistic fracture mechanics. He is the vice chair of the American Society of Mechanical Engineers Research Committee on Risk-Based Technology, and a consultant on fracture mechanics to the Advisory Committee on Reactor Safeguards. Dr. Maddox received a B.S. in mechanical engineering from the University of Washington and a Ph.D. in applied mechanics from Stanford University.

Donald Liu is a senior vice president of the American Bureau of Shipping where he has held progressively responsible positions for 25 years. He is a member of the Society of Naval Architects and Marine Engineers, from which he received the Captain Joseph H. Linnard Prize. He is also a member of the American Welding Society. Dr. Liu received a B.S. in nautical studies from the U.S. Merchant Marine Academy, a B.S. and M.S. in naval architecture and marine engineering from the Massachusetts Institute of Technology, and a Ph.D. in mechanical engineering from the Arizona State University.

Stephen J. Maddox is the technology manager–fatigue at The Welding Institute (TWI), where he specializes in fatigue of welded structures. He was formerly with Southhampton University. He is the TWI representative on several national and international code-writing committees, and is chair of Commission XIII of the International Institute of Welding. Dr. Maddox received a Ph.D. from London University.

Alaa E. Mansour is a professor in the Naval Architecture and Offshore Engineering Department at the University of California at Berkeley. Previous positions were at the Massachusetts Institute of Technology; M. Rosenblatt and Son, Inc.; John J. Mc Mullen Associates, Inc.; and the Suez Canal Authority. His research interests are in sea loads, finite element analysis of marine structures, probabilistic structural mechanics, reliability methods, and ultimate strength of marine structures. He has served on the Committee on Sea Floor Engineering and the Ship Research Committee of the National Academy of Sciences, the International Ship and Offshore Structures Congress, the Stress Analysis and Strength of Structural Elements panel of the Society of Naval Architects and Marine Engineers, and the Structural Stability Research Council. He is also a member of Sigma Xi and the American Association of University Professors. Dr. Mansour received a B.S. in mechanical engineering from the University of Cairo, and an M.E. and Ph.D. from the University of California at Berkeley.

Peter M. Palermo is a consultant in naval architecture. Previous positions include technical director at J.J. Henry, Inc. (Crystal City office); vice president, Engineering Group, CASDE Corporation; assistant deputy commander and technical director, Ship Design and Engineering Directorate of the Naval Sea Systems Command (NAVSEA); and he worked at the David Taylor Research Center. He has lectured on ship design, maintenance, and overhaul in the United States as well as in England, France, Italy, Japan, and Egypt. Mr. Palermo received a B.S. in civil engineering from Manhattan College and has completed graduate courses in applied mechanics at the University of Maryland.

Harold S. Reemsnyder is a senior research consultant for fatigue and fracture in the Research Department of Bethlehem Steel Corporation. He has taught at Lehigh University and at Carnegie-Mellon University. He is a Registered Professional Engineer in Pennsylvania and a Chartered Welding Engineer in the United Kingdom. Awards received include the Award of Merit and the Fatigue Achievement Award of the American Society of Testing and Materials, the Davis Medal of the American Welding Society, and the Bradley Stoughton Award of the American Society for Metals. He participated in the United States-Union of Soviet Socialist Republics Fatigue and Fracture Technology Exchange Program, is a member of ASTM Committee E-8 on Fatigue and Fracture, the Material Work Group of the Committee on Marine Structures of the National Academy of Sciences, and the Fatigue and Design Evaluation Committee of the Society of Automotive Engineers. Professional associations also include being a Fellow of the British Welding Institute, a Fellow of the American Society of Testing and Materials, a member of the International Institute of Welding, and a member Sigma Xi. Dr. Reemsnyder received a B.S. and M.S. in civil engineering from Carnegie-Mellon University and a Ph.D. in civil engineering from Lehigh University.

Stephen E. Sharpe is the executive director of the interagency Ship Structure Committee. Previous assignments with the U. S. Coast Guard include chief of inspections for the Marine Safety Office in Baltimore, marine inspector at the port of New York, and chief engineer of the USCG cutter *Duane*. He is a member of the Society of Naval Architects and Marine Engineers and ASTM committee F-25 on shipbuilding standards. Commander Sharpe graduated from the U.S. Coast Guard Academy and received an M.S. in naval architecture and in mechanical engineering from the University of Michigan.

Yung S. Shin is the principal engineer responsible for the hydrodynamics program in the Research and Development Department of the American Bureau of Shipping. He is a member of the Society of Naval Architects and Marine Engineers, for which he is the chair of the Technical Panel on Seakeeping. He was also the chair of the SWATH Working Group in the Netherlands Ship Model Basin Cooperative Research for Ships. Dr. Shin received a Ph.D. in naval architecture from the University of Michigan.

David Sucharski is a consulting naval architect. He was formerly the manager of technical services for ARCO Marine, Inc., where he had assignments in offshore and Arctic oil exploration, and manager, safety and environmental protection. Previous employment was with Global Marine in offshore design and operations assignments and in the Central Technical Division of Bethlehem Steel Corporation. He is a technical advisor to the U.S. delegation to the International Maritime Organization subcommittee on Safety and Load Lines and chairs the participants committee for the Massachusetts Institute of Technology joint industry project on "Grounding and Collision Protection for Tankers." He is a member of the Society of Naval Architects and Marine Engineers, for which he is a member of the Committee on Naval Architecture and West Coast regional vice president. Mr. Sucharski received a B.S.E. in naval architecture from the University of Michigan.

Hidetoshi Sueoka is the manager–structure designing for the Ship and Ocean Engineering Department of Mitsubishi Heavy Industries. Previous assignments with Mitsubishi included development of fast-speed ships and hull structural designing. Mr. Sueoka graduated from the University of Tokyo with a bachelor's degree in naval architecture and ocean engineering.

George M. Williams is chief of the Merchant Vessel Inspection and Documentation Division of the U.S. Coast Guard. Previous tours of duty with the U.S. Coast Guard include commanding officer of the marine safety offices in Miami, Florida, and San Juan, Puerto Rico; and several other tours ashore and afloat. He was an assistant professor in ship design at the U.S. Coast Guard Academy. Captain Williams is a graduate of the U.S. Coast Guard Academy and received an M.S. in naval architecture and marine engineering and an M.S. mechanical engineering from the University of Michigan.

Paul H. Wirsching is a professor of Aerospace and Mechanical Engineering at the University of Arizona. His research interests are in reliability engineering, probabilistic design, and fatigue and fracture reliability, especially in the area of marine structures. He a member of the American Society of Civil Engineers for which he is chair of the Technical Administrative Committee on Structural Safety and Reliability, and was formerly the chair of the Committee on Fatigue and Fracture Reliability, and member of the Committee on Offshore Structural Reliability and the Committee on Practical Reliability Concepts. He was chair of the Design Work Group of the Committee on Marine Structures of the National Academy of Sciences, and a member of the Design Philosophy committee of the International Ship and Offshore Structures Congress. He is also a member of the American Society of Mechanical Engineers, for which he was the chair of the Southern Arizona Section. Dr. Wirsching received a Ph.D in civil engineering and in mathematics and statistics from the University of New Mexico.

WORKSHOP LEADERS

John Dalzell is a research naval architect at the Carderock Division of the Naval Surface Warfare Center. Previous positions include the Davidson Laboratory (formerly the Experimental Towing

Tank), Hydronautics, Inc., Southwest Research Institute, and a brief tour of Navy active duty. He has served as the NAVSEA Research Professor at the U.S. Naval Academy and he is a Fellow of the Society of Naval Architects and Marine Engineers, for which he was a contributor to the 1986 edition of "Principles of Naval Architecture." He was awarded the society's Davidson Medal in 1990. Mr Dalzell graduated from Webb Institute in 1954.

Laura Demsetz is an assistant professor in the Department of Civil and Environmental Engineering at the University of California at Berkeley, where she teaches courses in construction engineering and management. Her research interests include construction equipment and automation, inspection of constructed facilities, building control systems, construction productivity, and human factors. Dr. Demsetz received a B.S. in mechanical engineering from the University of California at Berkeley and an M.S. in mechanical engineering and a Ph.D. in civil engineering from the Massachusetts Institute of Technology.

Rong T. Huang is head of the Design and Construction Division of Chevron Shipping company, where he has held succedingly responsible positions for more than 20 years. He is a member of the Society of Naval Architects and Marine Engineers, and the National Association of Corrosion Engineers. Mr. Huang received a M.S. in naval architecture from the University of California at Berkeley and a B.S. in naval architecture from Taiwan Maritime College.

Peter A. Gale is chief naval architect for John J. Mc Mullen Associates, Inc. He was previously a professor of naval architecture at Webb Institute. Earlier, he was with the Naval Sea Systems Command, where held several key Senior Executive positions, including Deputy Director of the Design Group and the Hull Engineering Group. He is a member of several professional societies, including the Society of Naval Architects and Marine Engineers, and the American Society of Naval Engineers. Mr. Gale received an M.S. in nautical engineering from Stevens Institute of Technology and a B.S. in naval architecture and marine engineering from Webb Institute of Naval Architecture.

Stanley T. Rolfe, NAE, is chair of the Department of Civil Engineering of the University of Kansas, where he has been involved in a comprehensive experimental and analytical research program in fracture mechanics for more that 20 years. He was previously the Division Chief of the Material Behavior Division at U.S. Steel's Applied Research Laboratory. He is a member of numerous professional societies, including the American Society of Civil Engineers, for which chaired the Technical Committee on Fracture and Structural Fatigue, and the American Society for Testing and Materials, for which he is a member of the Committee on Fracture Testing of Materials. He is a member of the National Academy of Engineering and served on several committees, including the Committee on Applications of Fracture Mechanics Analysis Techniques to Marine Systems and the National Research Council Project Advisory Committee. Dr. Rolfe received a Ph.D, M.S., and B.S. in civil engineering from the University of Illinois.

APPENDIX B

Workshop Agenda

SYMPOSIUM AND WORKSHOP
THE PREVENTION OF FRACTURE IN SHIP STRUCTURE

March 30–31, 1995
Washington, D.C.

Location:	*Hotel:*
National Academy of Sciences	State Plaza Hotel
2101 Constitution Avenue, N.W.	2117 E Street, N.W.
Washington, DC 20418	Washington, DC 20037
202-334-3119;	202-861-8200
FAX 202-334-3789	800-424-2859
	202-659-8601(FAX)

THURSDAY, MARCH 30, 1995

0800 Continental Breakfast in Lecture Room Lobby

Lecture Room

0830	Welcome, Introductory Remarks	John Landes University of Tennessee
0835	**A. History/Background**	RADM James C. Card United States Coast Guard
		Peter M. Palermo Committee on Marine Structures
0850	**B. Defining the Problem**	
0850	B.1 Owner/Operators Viewpoint	David Sucharski ARCO Marine
0915	B.2 Classification Societies Viewpoint	Donald Liu American Bureau of Shipping

Fracture Symposium Agenda
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THURSDAY, MARCH 30, 1995 (Continued)

0940	B.3 Ship Fabrication	Hidetoshi Sueoka Mitsubishi Heavy Industries
1015	**BREAK**	
1030	B.5 Analysis and Repair	Maxwell C. Cheung MCA Engineers
1110	**C. Other Industries' Approaches**	
1110	C.1 Other Industries' Approach	David O. Harris EngineeringMechanicsTechnology, Inc.
1135	C.2 Other Industries' Approach	Stephen Maddox The Welding Institute
1200	LUNCH (BUFFET)	
1300	**D. Technical Subjects**	
1300	D.1 Hull Structural Design	Mark Debbink Newport News Shipbuilding
1330	D.2 Fatigue and Fracture	Harold Reemsnyder Bethlehem Steel Corporation
1415	D.3 Reliability	Paul Wirsching University of Arizona Alaa Mansour University of California, Berkeley
1445	D.4 Inspection	Stephen E. Sharpe United States Coast Guard
1530	BREAK	
1545	D.5 Loads	Youl-Nan Chen Yung-Sup Shin American Bureau of Shipping

Fracture Symposium Agenda
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FRIDAY, MARCH 31, 1995

1615	D.6 Materials and Fabrication	Paul Blomquist Bath Iron Works
1645	CHARGE TO WORKSHOPS	John Landes
1700	ADJOURN FOR DAY	
1800	RECEPTION	
0800	Continental Breakfast in Lecture Room Foyer	
0830	Workshop Sessions	
	1. Fracture and Fatigue *Room NAS 150*	Workshop Session Leader: Stanley T. Rolfe University of Kansas Recorder: Mark Kirk Edison Welding Institute
	2. Design *Room NAS 180*	Workshop Session Leader: Peter A. Gale J.J. McMullen Associates Recorder: Bilal Ayyub University of Maryland
	3. Loads *Room NAS 250*	Workshop Session Leader: John Dalzell Carderock Division, Naval Surface Warfare Center Recorder: Allen Engle Naval Sea Systems Command
	4. Fabrication and Repair *Room NAS 280*	Workshop Session Leader: Rong T. Huang Chevron Shipping Company Recorder: Robert Dexter Lehigh University

Fracture Symposium Agenda
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THURSDAY, MARCH 30, 1995 (Continued)

5. Inspection
Board Room

Workshop Session Leader:
Laura Demsetz
University of California
at Berkeley

Recorder:
Kurt Hansen
U.S. Coast Guard Research
and Development Center

1200 LUNCH (BUFFET)

1200 Leaders and Recorders meet to compare results

1300 Workgroups reconvene in breakout rooms
(Work Group 4 meet in Lecture Room)

1400 Joint Meeting of all Workgroups

1530 ADJOURN

APPENDIX C

Workshop Participants

Workshop Session 1. *Room NAS 150*
Fatigue and Fracture

Workshop Leader—Stanley T. Rolfe, University of Kansas, Lawrence
Recorder: Mark Kirk, Edison Welding Institute, Columbus, Ohio

Akira Akiyama, American Bureau of Shipping, New York
Roshdy Barsoum, Office of Naval Research, Arlington, Virginia
Joseph Bloom, Babcox and Wilcox, Alliance, Ohio
Youl-Nan Chen, American Bureau of Shipping, New York
Maxwell Cheung, MCA Engineers, Costa Mesa, California
John M. Cushing, U.S. Coast Guard, Washington, D.C.
Nash Gifford, Carderock Division, Naval Surface Warfare Center, Carderock, Maryland
Jacques Giovanola, SRI International, Menlo Park, California
Thomas Gustafsson, Chalmer University of Technology, Goteborg, Sweden
William Hanzelek, American Bureau of Shipping, Houston, Texas
David Harris, Engineering Mechanics Technology, Inc., San Jose, California
Jerry W. Hottel, BP Oil Shipping Company, Cleveland, Ohio
Thomas Hu, Defense Research Establishment, Atlantic, Halifax, Nova Scotia, Canada
Andrea Iflander, U.S. Coast Guard, Governors Island, New York
Hajime Kawano, Mitsubishi Heavy Industries, Nagasaki, Japan
David Kihl, Carderock Division, Naval Surface Warfare Center, Carderock, Maryland
John Kokarakis, Failure Analysis Associates, Menlo Park, California
Stephen Maddox, The Welding Institute, Abbington, United Kingdom
Jack Morrison, Defense Research Establishment Pacific, Victoria, British Columbia, Canada
Hans Paetzold, Instut fur Schiffbau der Universitat, Hamburg, Germany
John F. Porter, Defense Research Establishment, Atlantic, Halifax, Nova Scotia, Canada
Yapa Rajapakse, Office of Naval Research, Arlington, Virginia
Harold Reemsnyder, Bethlehem Steel Corporation, Bethlehem, Pennsylvania
Walter G. Reuter, Idaho National Energy Laboratory, Idaho Falls, Idaho
Gavin Rudgley, Ministry of Defense, Foxhill, Bath, United Kingdom
Bruce Sommers, Lehigh University, Bethlehem, Pennsylvania
Yoichi Sumi, Yokohama National University, Yokohama, Japan

Art Symmes, John J. McMullen Associates, Arlington, Virginia
William Tyson, Canada Centre for Minerals and Energy Technology, Ottawa, Ontario, Canada
Eugene A. Van Rynbach, Sea-Land Service, Elizabeth, New Jersey
Chao-Cheng Wu, United Ship Design and Development Center, Keelung, Taiwan
Dayan Xiao, Lehigh University, Bethlehem, Pennsylvania
Richard Yee, Fleet Technology Ltd., Kanata, Ontario, Canada
Nicholas Zettlemoyer, Exxon Production Research Company, Houston, Texas

Workshop Session 2. *Room NAS 180*
Design

Workshop Leader—Peter A. Gale, J.J. McMullen Associates, Inc., Washington, D.C.
Recorder: Bilal M. Ayyub, University of Maryland, College Park
Akira Akiyama, American Bureau of Shipping, New York
Steve Balint, Shell Oil Company, Houston, Texas
Jeffrey Beach, Carderock Division, Naval Surface Warfare Center, Carderock, Maryland
Capt. Alan Brown, USN, Massachusetts Institute of Technology, Cambridge
Mark D. Debbink, Newport News Shipbuilding, Newport News, Virginia
Paul E. Hess, Carderock Division, Naval Surface Warfare Center, Carderock, Maryland
Owen Hughes, Virginia Polytechnic Institute, Blacksburg
Roy E. Johnson, Mobil Shipping and Transportation Co., Fairfax, Virginia
Chandru (Jack) Kalro, Military Sealift Command, Washington, D.C.
Vitaly Kozliakov, Odessa State Maritime University, Odessa, Ukraine
Ray Kramer, John J. McMullen Associates, Inc., Arlington, Virginia
Chao H. Lin, U.S. Maritime Administration, Washington, D.C.
Walter Maclean, U.S. Merchant Marine Academy, Kings Point, New York
Alaa Mansour, University of California, Berkeley
Robert Michiels, Avondale Industries, New Orleans, Louisiana
Ryszard Piskorski, P&P Engineering, Horsham, Pennsylvania
John D. Ryan, National Center for Excellence in Metalworking Technology, Johnstown, Pennsylvania
Frank Scotto, American Bureau of Shipping, Paramus, New Jersey
David Sucharski, Arco Marine Inc., Long Beach, California
Anil Thayamballi, American Bureau of Shipping, New York
Anders Ulfvarson, Chalmers University of Technology, Goteborg, Sweden
David Weong, National Steel & Shipbuilding Co., San Diego, California
Paul Wirsching, University of Arizona, Tucson
Dave Wood, Gibbs & Cox, Inc., Arlington, Virginia
Peter Zink, Herbert Engineering Corp., San Francisco, California

Workshop Session 3. *Room NAS 250*

Loads

Workshop Leader—John Dalzell, Carderock Division, Naval Surface Warfare Center, Carderock, Maryland
Recorder: Allen Engle, Naval Sea System Command, Arlington, Virginia

Participants:

Youl-Nan Chen, American Bureau of Shipping, New York
William Hay, Carderock Division, Naval Surface Warfare Center, Carderock, Maryland
Greg Hermanski, Institute for Marine Dynamics, St. Johns, Newfoundland, Canada
Paul Kaplan, Hydromechanics, Inc., Delray Beach, Florida
Jui-fang Kuo, Exxon Production Research, Houston, Texas
Carl M. Larsen, University of Michigan, Ann Arbor
Walter Lincoln, U.S. Coast Guard Research and Development Center, Avery Point, Connecticut
Lewis Motter, Carderock Division, Naval Surface Warfare Center, Carderock, Maryland
William Richardson, Carderock Division, Naval Surface Warfare Center, Carderock, Maryland
Nils Salvesen, Science Applications International Corporation, Annapolis, Maryland
Spencer Schilling, Herbert Engineering Corp., San Francisco, California
Yung-Sub Shin, American Bureau of Shipping, New York

Workshop Session 4. *Room NAS 280*

Fabrication and Repair

Workshop Leader—Rong T. Huang, Chevron Shipping Company, San Francisco, California
Recorder: Robert J. Dexter, Lehigh University, Bethlehem, Pennsylvania
Paul Blomquist, Bath Iron Works, Bath, Maine
Erling Elholm, SeaRiver Maritime, Houston, Texas
William Hanzalek, American Bureau of Shipping, Houston, Texas
Kuniaki Ishida, Ishikawajima-Harima Heavy Industries, Yokohama, Japan
Jasbir S. Jaspal, Mobil Shipping and Transportation Co., Fairfax, Virginia
Young-Min Lee, Samsung Heavy Industries, Kyungnam, Korea
Donald Liu, American Bureau of Shipping, New York
Lalit Malik, Fleet Technology, Kanata, Ontario, Canada
Keijiro Miyabe, Ishikawajima-Harima Heavy Industries, San Francisco
Chris Paquette, Naval Sea Systems Command, Arlington, Virginia
Todd Seaman, U.S. Coast Guard, Governers Island, New York
Mike Sieve, Naval Sea Systems Command, Arlington, Virginia
Hidetoshi Sueoka, Mitsubishi Heavy Industries, Yokahama, Japan
Robert J. von Saal, Consultant, Severna Park, Maryland

Archie Wiggs, Carderock Division, Naval Surface Warfare Center, Carderock, Maryland
Robb Wilcox, U.S. Coast Guard, Washington, D.C.
David J. Witmer, BP Oil Shipping Co., Cleveland, Ohio
Chao-Cheng Wu, United Ship Design and Development Center, Keelung, Taiwan

Workshop Session 5. *Board Room*

Inspection

Workshop Leader—Laura Demsetz, University of California, Berkeley
Recorder: Kurt Hansen, U.S. Coast Guard R&D Center, Avery Point, Connecticut

Attendees:

Greg Buie, U.S. Coast Guard, Washington, D.C.
Guido Dei, ARCO Marine, Long Beach, California
Mike Goodwin, Goteberg, Sweden
Kurt Hansen, U.S. Coast Guard R&D Center, Avery Point, Connecticut
Robert Holzman, U.S. Coast Guard, Washington, D.C.
David Knight, Carderock Division, Naval Surface Warfare Center, Carderock, Maryland
Al Moore, U.S. Coast Guard, Washington, D.C.
Peter M. Palermo, Consultant, Alexandria, Virginia
Robert Sedat, U.S. Coast Guard, Washington, D.C.
Stephen E. Sharpe, U.S. Coast Guard, Washington, D.C.
Gary Strait, Ronald Nesbet Associates, Oregon City, Oregon
Thomas Waters, U.S. Coast Guard, Washington, D.C.
Richard Whiteside, BP Oil, Cleveland, Ohio
Nicholas Zettlemoyer, Exxon Production Research, Houston, Texas

PREVENTION OF FRACTURE IN SHIP STRUCTURE

PART II

PRESENTED PAPERS

March 30–31, 1995
Washington, D.C.

Convened by the
COMMITTEE ON MARINE STRUCTURES
Marine Board

Commission on Engineering and Technical Systems
National Research Council

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Safelife for Ships

James C. Card and Peter M. Palermo[1]

ABSTRACT

Over the past 10 years a number of large oceangoing ships have either sunk or experienced significant structural failures attributed to a combination of human error and seaway loadings. The results of these human errors include inadequate maintenance and/or poor detail design or fabrication practices. Technological changes in certain design and fabrication areas have led to the fatigue cracks associated with many of today's ship designs, particularly in tankers in the Trans-Alaska Pipeline Service (TAPS) trade. Solving or codifying methodologies for design should provide solutions to the problem. Attention must also be paid to the practices of fabrication and inspection in developing a strategy to minimize the risks associated with fatigue and fracture of ships. Properly considered and implemented, the findings of this workshop should provide a real factor in minimizing, if not eliminating, oil spills from tankers and the loss of life associated with ship fractures in bulk carriers and tankers.

DISCUSSION

In the past, ship structural design was primarily concerned with seaway loadings while underway and cargo loadings while pier side. Those designs were premised on a finite lifetime of approximately 25 years and on assumed proper and regular maintenance. In the past 10 to 20 years these original design assumptions have been proven to be somewhat unrealistic, at worst, and optimistic, at best. It has been observed that ships have been experiencing structural failures under loadings that were considerably less than the loadings associated with hull failure. Repeated loadings in areas of stress concentrations lead to fatigue cracks that, if left unrepaired, can grow to a critical length and result in catastrophic crack growth. Over the past 10 years a number of large oceangoing ships have either sunk or have experienced significant structural failures attributed to a combination of human error and seaway loadings. The human errors are the result

[1]The comments and opinions expressed in this paper are those of the authors and not necessarily those of the U.S. Coast Guard or the U.S. Department of Transportation.

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of inadequate maintenance and/or poor detail design or fabrication practices. The seaway loading effects are well known to the audience at this workshop. It is the purpose of this workshop to emphasize the need to consider fatigue and fracture consciously, as part of the front-end design process. It should be noted that steps have already been taken by classification societies to improve the design of new tankers and the assessment of existing tankers; both systems are being extended to bulk carriers (ABS, 1994).

When first conceived, it was felt that this workshop was necessary to provide the analytical tools that should be used to solve the cracking problems associated with the Trans-Alaska Pipeline Service (TAPS) tankers. However, it rapidly became obvious that the principles and methodologies were equally applicable to all ship types. It is ironic that technological advances in certain design and fabrication areas have, in all likelihood, led to problems associated with many of today's designs. The advent of high-speed computing capability coupled with advanced finite-element analysis techniques has permitted designs that are much more efficient and, therefore, somewhat less "rugged" than those of yesteryear. Coupled with improved design capabilities is the increasing use of higher-strength steels. Unfortunately, increased material strength does not mean increased fatigue resistance. Thus, our design capabilities permit designs with more efficient distribution of material, that is, thinner scantlings; the use of higher-strength materials permits even thinner hull scantlings. We have come full circle: thinner scantlings if improperly coated and inspected are prone to much faster corrosion effects as compared with earlier designs. Further, inattention to proper details in design and fabrication results in areas of stress concentrations that are much more susceptible to fatigue cracking. Failures of older designs have been attributed to lack of proper maintenance, damage caused by cargo-loading equipment, and material corrosion.

The most catastrophic, and at first seemingly inexplicable, casualties were the sinking of approximately 25 bulk carriers and the damage beyond repair of approximately 25 others in the early 1990s. In general these were older ships, serving under third, fourth, or even fifth owners (Alvarez, 1992). Some had their certificates and classification withdrawn because of nonconformance with classification requirements (Lloyds, 1994). They were poorly maintained, exhibited excessive corrosion, and generally failed in two principle ways: sections of shell plating severely weakened by corrosion could have been carried away under high seaway loadings; or corroded sections could have permitted sea water to enter the holds, resulting in liquefaction of cargo. Ensuing sloshing forces on already weakened, heavily corroded shell plating could then lead either to structural failure or even loss of stability. Though not specifically germane to the issue of fatigue and fracture, some important lessons can be learned from these failures. First and foremost is the need for proper maintenance and corrosion protection. Second, these are very large ships, on the order of 1,000 ft long, with beams on the order of 200 ft. In order to save hull weight and maintain the same deadweight, hulls of the later ships were fabricated of higher-strength steel. Thus, for a given design loading, thinner hull plating could be used as compared with ordinary steel. Therefore, the effects of corrosion could be more pronounced.

Major structural failures in the form of hull-structure cracking have been observed in a number of very large tankers, in particular in the TAPS trade (USCG, 1989). Subsequent investigations have indicated a similar propensity for cracking in tankers of the same designs in other trade routes. However, the TAPS trade very large crude carriers (VLCCs) are subjected to probably the most severe sea conditions compared with other seaway routes, including the North

Atlantic. Damage noted on TAPS tankers dramatically reflects the effects of these severe seaway loadings. While northbound in ballast, thus high in the water, the lower portion of the port-side shell evidenced damage. Southbound, fully loaded, thus low in the water, the upper portions of the starboard-side shell evidenced damage. In addition, some tankers are constructed of thinner-plating, higher-strength steels, and, without proper attention to maintenance and corrosion control, they are potential follow-on disasters, similar to the bulk carriers (USCG, 1990). Further, any through-cracking can result in a significant environmental impact, with ensuing large financial penalties. With these thoughts in mind, it is incumbent upon the maritime community to preclude any and all sources of impending failures.

A number of tankers evidenced fatigue cracking very early in their service life, while those of similar design, operating on the same trade routes, did not show any early cracking. These differences in fatigue life can be explained by a number of almost random occurrences, including different fit-up and attention to detail during fabrication or the use of plate material with different strength, fatigue, and fracture properties but still within original specifications. However, the existence of large fatigue cracks after a relatively few years of operation is a cause for concern. Accurate technical data are needed to permit localization of areas where these fatigue cracks initiate. Crack-growth data are also necessary, as well as inspection periodicity and repair methods. The presence of a fatigue crack means the presence of significant local stresses and repetitive cycles of loadings.

The next important consideration is a determination of "critical" crack length for a given hull material in order to assess potential for catastrophic fracture. Cracks can initiate at poor details or poor welds, or they can be present in welds during fabrication and can grow due to sea-way loadings or even due to stress corrosion associated with elevated cathodic potential. Catastrophic crack growth in these VLCCs need not result in the loss of the ship, as in the case of the Liberty ships 50 years ago. However, any hull crack that permits the crude oil to leak to the sea must be considered catastrophic in today's environment. Double-hull or comparable type tankers, while offering an extra level of protection, are not immune to this type of "catastrophic" failure. Thus, for tankers, any cracking that severely weakens the main-hull structure or that permits leakage of crude to the sea is considered catastrophic.

In general, once cracking is detected weld repairs are conducted, and structural modifications are introduced. These approaches may solve the immediate problem, but in the long run they often just move the initiation point for the crack to a slightly different location. The ultimate solution is to lower stress levels caused by load-induced conditions or structural misalignments to the point where fatigue-crack initiation and growth are either precluded or are deferred to much later in the ship's life. Recognizing that elimination of all potential crack-initiation sites or conditions may not be practical, other approaches are available to preclude catastrophic failures. Application of these approaches is the embodiment of a structural integrity concept in design and life-cycle maintenance.

The structural integrity concept includes the development of a stress map of the structure based on verified analytical methods that account for fabrication tolerance limits vice just using past design configurations; use of steel and weld metal with known fatigue and upper-shelf fracture properties; fabrication details rigidly controlled within specification limits; and, finally, an inspection plan that includes method and periodicity for specific areas. While such an approach may seem unduly expensive, long-term savings in ship repairs and inspections and possible litigation should more than offset any initial cost increases.

The interagency Ship Structure Committee (SSC) has sponsored considerable research germane to the subject of fatigue and fracture. Technical thrusts in the area of reliability of ship structures provide guidance for improving the integrity of existing ships and new designs. The fracture phenomenon is usually depicted in three separate but interconnected phases: crack initiation, crack growth, and catastrophic rapid crack growth (fracture). Each is the subject of different study and analysis.

Crack initiation is usually analyzed using an S–N approach. This requires load-prediction capability and validated stress-analysis tools for specific details. Nonlinear loadings, due to slamming and large amplitude waves, must also be considered. Another approach would entail using experimental S–N data from a number of structural details, as provided in SSC reports (Munse et al., 1983; Park and Lawrence, 1988; Stambaugh et al., 1992). With loading history known and S–N data available, it is necessary to develop a statistical projection of service life.

Crack growth is analyzed using fracture mechanics da/dn [*crack growth (da) per stress cycle (dn)*] technology, relating experimental crack-growth data to stress intensity factors. Again, this is also a probabilistic approach. Finally, catastrophic crack growth is generally handled in the design and fabrication stage by using steels with a high notch-toughness rating. Toughness levels are determined by empirical techniques that are not directly related to the actual in-service loadings. Charpy-V, dynamic tear, and Navy-developed explosion bulge tests are usually used to categorize toughness levels.

Solving or codifying the probabilistic methodologies for stress analysis, load determination, crack initiation, growth, and fast fracture should provide a better means of design and solution to the problems of fatigue and fracture. The development of probability-based design codes in other areas has provided the stimulus for developing such codes for ship structures. The SSC has sponsored numerous efforts in this area (Mansour, 1989; Nikolaidis and Kaplan, 1991; Hughes et al., 1994), and they are completed or are coming to a conclusion (Projects SR-1344, "Assessment of Reliability of Existing Ships," and SR-1345, "Probability Based Ship Design: Implementation of Design Guidelines for Ships)."

Steps are being taken to minimize the risk of oil spills due to collisions or groundings. Legislation enacted as the Oil Pollution Act of 1990 (OPA 90) requires tankers operating in U.S. waters to have double-hull or comparable construction. Such requirements are significant and should go a long way toward minimizing the risks associated with low-energy impacts. However, they do little to alleviate the more insidious problem of crude-oil leakage due to cracking or fracture. From a statistical standpoint, the length of welds and number of detail joints that are potential incubation sites for fatigue cracks far exceed the potential for collisions or groundings. Added to this is the further degradation of structural integrity associated with corrosion. Though statistically significant, the potential for collision or grounding pales in comparison with the potential associated with the much larger population of sites for fatigue, fracture, and corrosion. Mandating double hulls or equivalents is an excellent first step. However, such types of construction cannot only be the source of additional corrosion but can also make inspections much more difficult.

Because welded ships have miles of welds and acres of plating to be inspected, the most cost-effective inspections are those conducted during fabrication. Attention to detail in the fabrication stages with regard to fit-up, welding procedures, and weld profiles will go a long way toward minimizing in-service problems. In-service inspections are labor and time intensive. Published data on the inspected areas of a 250,000 DWT pre-MARPOL tanker indicate 750 miles of welds and over 76 acres of plating (NRC, 1991). Based on these numbers it becomes patently obvious that designing-in and fabricating-in quality should be orders of magnitude easier than trying to inspect-in quality.

SUMMARY

Considering the importance of proper design and limitations on inspections, we must now develop the strategy for design, fabrication, and inspection methodologies necessary to minimize the risks associated with fatigue and fracture of ship structures. OPA 90 has provided the mandate for ship configurations. We must now address the issue of making such configurations fabrication- and inspection-friendly. In particular, the technology to be explored and exploited will not be done in a vacuum. Owners' and classification societies' concerns, tempered with economic realities, should all be considered in developing the roadmaps of future research and applications associated with load determinations; design methods; fatigue and fracture considerations; and fabrication, inspection, and repair. Properly considered and implemented, the findings and recommendations of the workshops to be held tomorrow should provide the muscle to make the mandate of OPA 90 a real factor in minimizing, if not eliminating, oil spills in tankers and the loss of life associated with ship fractures for bulkers and tankers.

REFERENCES

ABS (American Bureau of Shipping). 1994. ABS SafeHull System and SafeHull Condition Assessment. New York: American Bureau Shipping.

Alvarez, A. 1992. *New York Times Magazine*. May 17, 1992. Sinking fast: The world's largest ships are going down in record numbers. Vol. 141, Sunday ed. col. 1, p. 36.

Hughes, O., E. Nikolaidis, B. Ayyub, G. White, and P. Hess. 1994. Uncertainty in Strength Models for Marine Structures. Report SSC-375. Washington, D.C.: Ship Structure Committee.

Lloyds Register of Shipping. 1994. Sinking of Iron Antonis, September 6. London, England: Lloyds List.

Mansour, A.E. 1989. An Introduction to Structural Reliability Theory. Report SSC-351. Washington, D.C.: Ship Structure Committee.

Munse, W.H., T.W. Wilbur, M.L. Tellalian, K. Nicoll, and K. Wilson. 1983. Fatigue Characterization of Fabricated Ship Details for Design. Report SSC-318. Washington, D.C.: Ship Structure Committee.

Nikolaidis, E., and P. Kaplan. 1991. Uncertainties in Estimating Loads and Load Effects on Marine Structures. Report SSC-363. Washington, D.C.: Ship Structure Committee.

NRC (National Research Council). 1991. Tanker Spills—Prevention by Design. Marine Board, National Research Council. Washington, D.C.: National Academy Press.

Park, S.K., and F.V. Lawrence. 1988. Fatigue Characterization of Fabricated Ship Details for Design-Phase II. Report SSC-346. Washington, D.C.: Ship Structure Committee.

Stambaugh, K.A., D.H. Leeson, F. Lawrence, C.Y. Hou, and G. Banas. 1992. Reduction of S–N Curves for Ship Details. Report SSC-369. Washington, D.C.: Ship Structure Committee.

USCG (U.S. Coast Guard). 1989. Report of the Tanker Safety Study Group, October 6. Washington, D.C.: USCG.

USCG. 1990. Report on the Trans-Alaska Pipeline Service (TAPS) Tanker Structural Failure Study, June 25. Washington, D.C.: USCG.

Crude Oil Tanker Hull Structure Fracturing: An Operator's Perspective

David Sucharski

ABSTRACT

Dealing with fractures in the structure of tankers has become increasingly important for both regulators and operators in the United States. This paper describes typical fracturing and addresses causes and response strategies from the perspective of an operator. The ships discussed range in size from 90 MDWT to 265 MDWT and have a total of 144 ship years of operating experience in the severe North Pacific environment.

INTRODUCTION

Hull structure has fractured since metal ships were first built about 150 years ago. Centuries of effort by naval architects to minimize leakage in the flexible basket-like wooden ships of the past were replaced by the need to manage the inherent stiffness of iron and steel ships. By early in this century, the importance of a design process that mitigates failure modes was understood and taught for joints in riveted structures—probably a result of cautionary experiences (Murray, 1916). During World War II, welding replaced riveting for the joints in mass-produced ships. Catastrophic fracturing in some of these early welded ships demonstrated that there is more to designing welded structures than was initially recognized, and the envelope of design freedom contracted somewhat for critical details. The differences between welded and riveted joints had compromised important, perhaps subtle, practices essential to the design of failure-resistant joints. During the next two decades, the role of stress concentrations at details, locked in stresses, notch toughness, and the need for crack arrestors were increasingly understood and applied to the design of ships. These steps constituted a return to engineering design characterized by failure anticipation and mitigation, and catastrophic fractures were largely eliminated in welded ships. In the 1960s and 1970s, the size of U.S.–built tankers increased by a factor of about six in the relatively short period of 10 years (Sucharski and Cheung, 1993). At the same time, simplified structural joints and reduced scantlings were adopted for optimized production, finite element analysis (FEA) became routinely available, and higher-strength steel selectively replaced mild steel in the top and bottom flanges of the hull girder. Most of these new ships entered the North Pacific Trans-Alaska Pipeline Service (TAPS) trade, where they experienced environmental conditions about 20 percent more severe than classification society North Atlantic design standards (Lacey and Chen, 1993). Collectively these changes again compromised the design process, resulting in increased frequency and severity of fracturing in many of the new tankers, mainly at the intersection of longitudinal frames and the transverse supporting structure. Analysis of representative fractures suggests that the likely cause of fracturing in these ships is a design process that did not fully anticipate and mitigate failure modes in the structural joints and a detailing process divorced from the original design analysis. Higher-strength steel is often identified as an important cause of the current fracturing, but the fracture record for the eight TAPS tankers considered in this paper shows that most fractures continue to occur in mild-steel components. This finding is true even for ships in this group, with a significant use of higher-strength steel, further suggesting that the use of only mild steel would not by itself have eliminated the current fracturing. Although the fractures in modern tankers are neither as spectacular nor as destructive as those associated with the early welded ships, they have caused a reallocation of ship maintenance and monitoring resources and a rethinking of the structural design process. This has led to the establishment of fracture documentation databases for tankers, the recognition of the importance of fatigue as a cause of fracturing, improvement in the strategy of ship structural design, and the development of active shipboard stress management and voyage planning tools for the bridge team.

These changes began in the mid-1980s, when the trend toward extensive fracturing was first recognized in relatively new tankers. Several operators responded with formal documentation of fracturing using computer-assisted relational databases. This documenting continues to give essential guidance for efficiently focusing analytical and maintenance resources and also gives valuable insight for improving design. Conventional elastic analysis of the fractures, including the use of FEA, showed that most of these fractures occurred at stresses much less than yield. This result pointed to fatigue as the fracture-initiation mechanism and led to the application of spectral fatigue analysis to several classes of tankers. The results of these analyses correlated well with existing fractures and helped confirm that fatigue was the main source of fracturing. Based on these findings, a transition began toward design based on both experiential anticipation and analytic prediction of failure, which assigns performance limits for elastic strength, fatigue life, and fracture-growth rate. Recognizing the importance of fatigue has also led to the continuous and active management of cumulative fatigue damage by the ship's bridge team throughout the vessel's life with onboard voyage planning. Active hull-stress management greatly expands the ability of the bridge team to contribute to successful structural performance beyond their historic role of ensuring that the ship is loaded within the designer's expectations based on a static stress numeral. Although recently introduced, voyage planning has already demonstrated the ability to reduce significantly the exposure of tankers to severe seas without penalizing transit time.

Often communication among the design, construction, and operation segments in a tanker's life has been limited. Today the use of performance documentation, more rigorous and complete engineering, and active onboard stress management is integrating the efforts of naval architects, builders, and shipboard bridge teams and making them equal participants in the lifetime successful performance of a ship's structure. The following sections describe the fracturing experience of a representative set of tankers and the characteristics and function of an operational fracture-mitigation and -prevention process.

THE FRACTURE RECORD

This section presents data on the characteristics, frequency, and severity of fracturing found in four classes of crude oil tankers operated almost exclusively in the North Pacific Ocean between Alaska and the U.S. west coast in the TAPS trade. The eight ships of these four classes are the products of three different U.S. shipyards, represent a wide size range, include both mild- and higher-strength steel construction, and have a cumulative service life of about 144 ship years. Table 1 summarizes the relevant characteristics of these ships. This section begins with an overview that summarizes the incidence of fracturing and its severity versus the group as a whole, the different classes, and the age of the ships. The overview shows that (1) nearly all of the fractures are small, especially with respect to the structure involved; (2) at least two-thirds of the fractures occur at the intersection of longitudinal frames and significant transverse structure (bulkheads and web frames); (3) different components fail in different classes, suggesting that better design can lead to better performance; and (4) eventually the incidence of fracturing markedly increases for all of the classes, but this change occurs later in life for some classes than others. The overview is followed by a description and assessment of typical fractures. Several representative fractures are described including repair and modification strategies. These fracture case studies show that failure is the result of inadequate joint design (discontinuous load path and discontinuous strength and stiffness properties) and the common use of details that introduce severe stress risers (free-plate edges and notches). Longitudinal, vertical, and transverse distributions of the fractures by frequency and severity are presented for each ship class. The fracture distributions further show the propensity for fracturing to occur at the intersection of the longitudinals and transverse structure. Despite concerns about the role of higher-strength steel in fatigue fracturing in recent tankers, most fractures in the classes with higher-strength steel (A, B, and D) occurred in mild-steel components. The problem seems to be more related to design than material.

Overview

An average of about 27 fractures per ship year have been recorded for the 4 tanker classes throughout their service life. Of these, 99.9 percent have been small localized fractures in the internal structure (U.S. Coast Guard, Class 2 and Class 3). Most of these fractures occur at the connections between longitudinal frames on the bottom or side shell and transverse bulkhead and transverse web frame stiffeners. Eleven Class 1 fractures have occurred involving small ruptures in the oil/water boundary. Two types of Class 1 fractures are found in these ships: very small cracks through the side or bottom shell, which originate as a fracture in a longitudinal; and small fractures in the upper deck (mostly in Class D), which originate at roughly cut tank cleaning openings. Several cracks have occurred in the longitudinal girder of the center vertical keel (CVK) under the upper-deck and under-deck girder of several transverse web frames. Some of these fractures were nearly full depth. All but two of these fractures occurred in Class B ships. All of these fractures are believed to be caused by fatigue.

TABLE 1 Characteristics of Classes A—D

Class	Shipyard	Size (million barrels)	Type	Delivered	Hull Steel
A	a	2	Single Bottom	1977	ABS DH Deck and Bottom Longitudinals and Plate EH Stringer, Sheer and Bilge Strakes AH CVK Flange and Web A and B Elsewhere
B	b	1.5	Double Bottom	1980	ABS AH Longitudinals, DH Deck and Bottom Shell Plate AH CVK Flange and Web A Elsewhere
C	a	1	Single	1974	ABS A and B
D	c	0.6	Partial Double Bottom	1981	ABS Grade AH Longitudinals and DHN Upper Deck and Sheer Plate A and D Elsewhere

NOTES:
ABS: American Bureau of Shipping
AH, DH, EH, DHN: ABS grades of higher-strength steel
CVK: center vertical keel

Figure 1 shows the cumulative frequency versus severity of fractures for the four tanker classes. About 50 percent of the fractures are less than 6 in. long; nearly all (at least 85 percent) are less than 12 in. long, and the occurrence of through fractures (breaking the structural component into two pieces) is rare. The majority of the fractures in these ships occur in structural components that are at least 18 in. deep. The distribution of fracture size in Figure 1 suggests that the practice of inspecting the structure during shipyard periods every 2 to 3 years reliably detects most fractures before they reach critical length.

Figure 2 shows the frequency versus the severity of fractures for each tanker class. There is some variation between the classes, but the pattern in each case is consistent with the cumulative pattern in Figure 1. Comparing the areas of the curves in Figure 2 gives a rough measure of the relative amount of fracturing amongst the classes. Class C is the oldest, followed by Class A, Class B, and Class D. Class C is constructed of mild steel; classes A, B, and D are constructed of higher-strength steel in the deck and (classes A and B) the bottom structure (See Table 1). Noting the similar areas of the curves in Figure 2 and the difference in ages of the

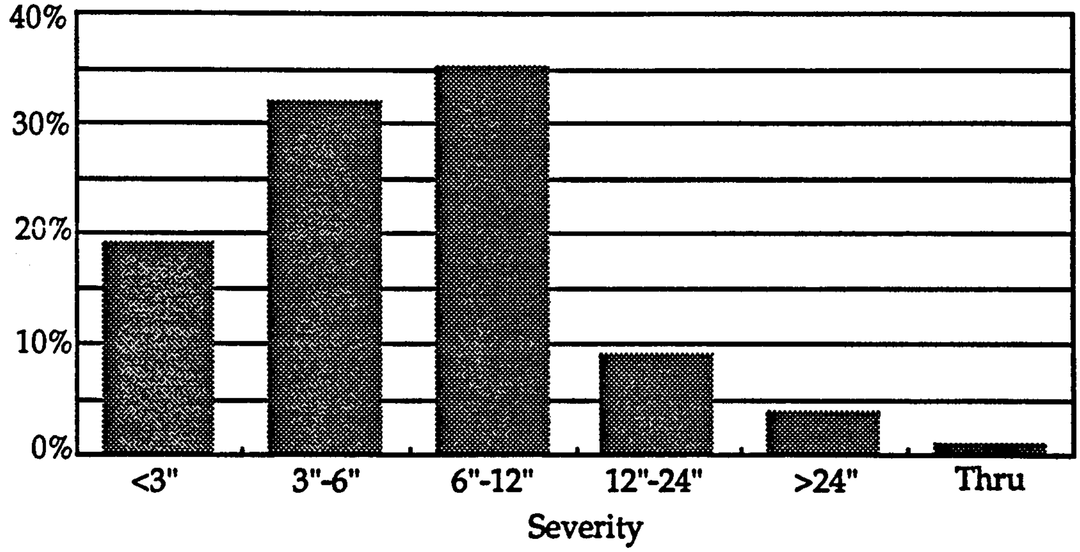


FIGURE 1 Cumulative frequency versus severity for classes A—D.

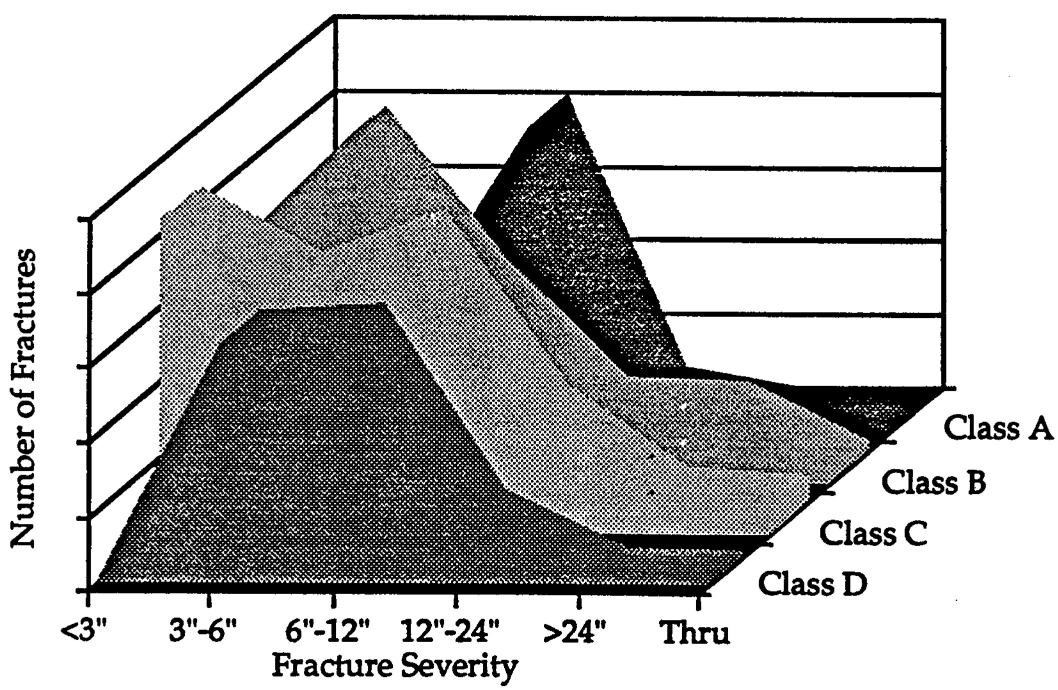


FIGURE 2 Frequency versus severity of fracturing for classes A—D.

ships, the newer classes seem to be fracturing more, or at least earlier, than the older classes. This difference is not fully explained by recent improvements in inspection and documentation since the searching of ships by shipyard teams for revenue-generating fractures to repair is a long-standing part of biannual shipyard periods. Nor is this difference explained by the use of higher-strength steel in the newer ships since most of the fractures have occurred in mild-steel components.

Figure 3 shows the cumulative frequency of fractures versus structure type for the four tanker classes. Longitudinal frames are the most frequently fractured structural component. These fractures occur almost exclusively at the intersection of the longitudinals and the transverse bulkheads and transverse web frames. Usually these fractures occur where the longitudinal is joined via a bracket or a flat-bar tie plate to a bulkhead stiffener and at the connections between longitudinals and web-frame panel stiffeners. At least two-thirds of the fractures in these ships occur in these structural environments. The greater incidence of fracturing of longitudinals, compared to fracturing of the brackets, tie plates, and panel stiffeners at these joints, may be due to the stress field present in the longitudinal from the primary hull-girder bending moment. Most of the shell-plate fractures result from through-fractures of longitudinals at these locations as well.

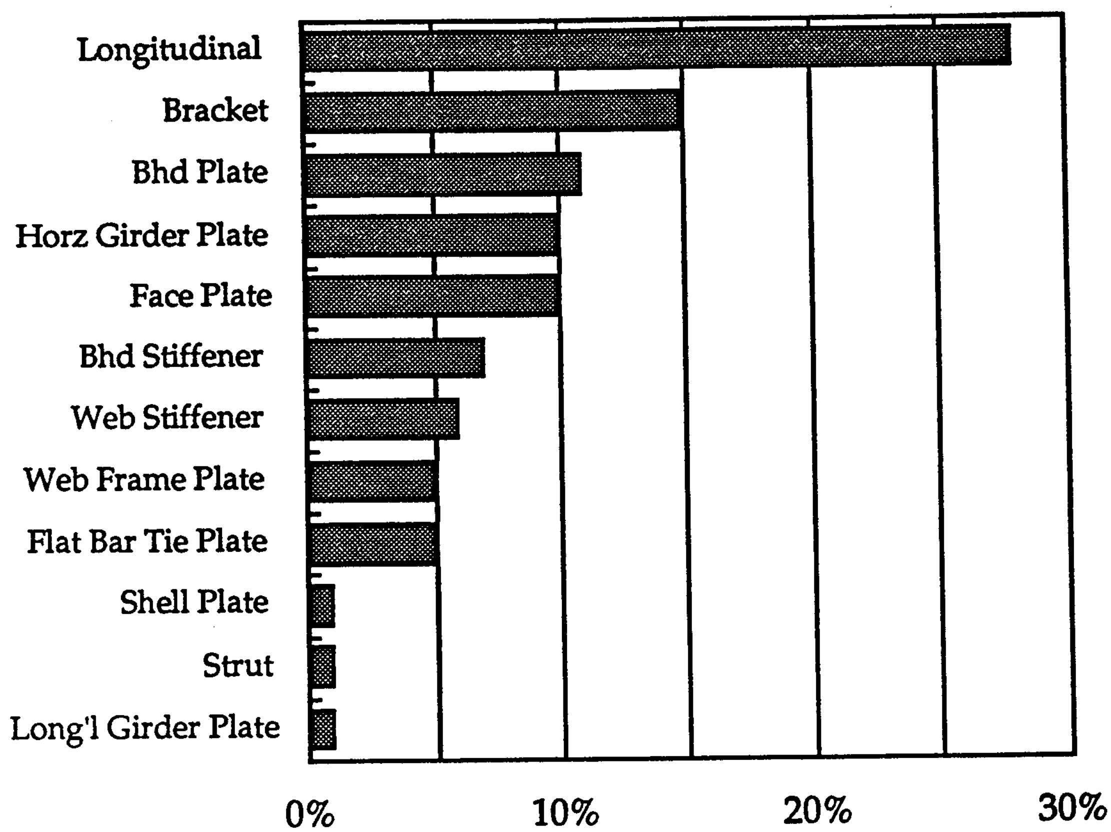


FIGURE 3 Frequency of fracturing versus structural component classes A—D.

Fatigue initiates a fracture in the longitudinal at a stress concentration in a joint near a bulkhead or web frame. The fracture propagates through the longitudinal until it reaches the shell plate, causing a stress riser, which eventually initiates a surface fracture that propagates into and sometimes through the plating.

Figure 4 shows that the distribution of fracture severity for individual structural components is consistent with the general distribution shown in Figure 1. The data in Figure 4

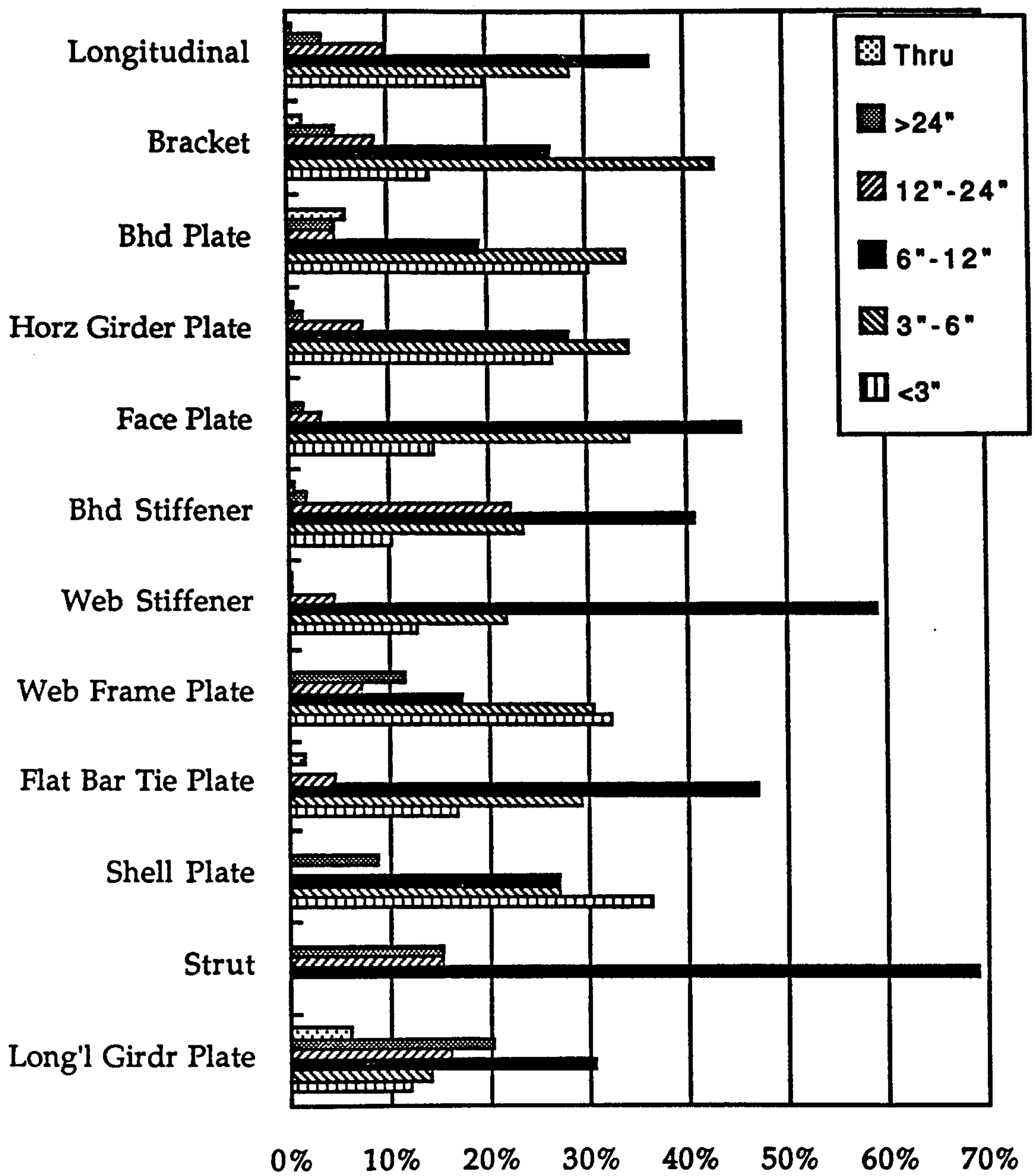


FIGURE 4 Cumulative frequency versus severity and structural component classes A—D.

also suggest that, for most of the structural components in the tankers of these four classes, fractures typically reach a length of about 6 in. during the two to three year interval between shipyard periods. Although the actual growth period is unknown, these data may be useful in establishing an envelope of plausible fracture-growth rates for these structural components.

Figure 5 shows the frequency of fractures versus structure component for each of the four tanker classes. The general pattern of Figure 5 is consistent with the distribution in Figure 3, but the specific distributions for the individual classes vary considerably. For example the longitudinals fracture more often in Class A than Class D, but in Class D the brackets and flat-bar tie plates attached to the longitudinals fracture more often than in Class A. These results are

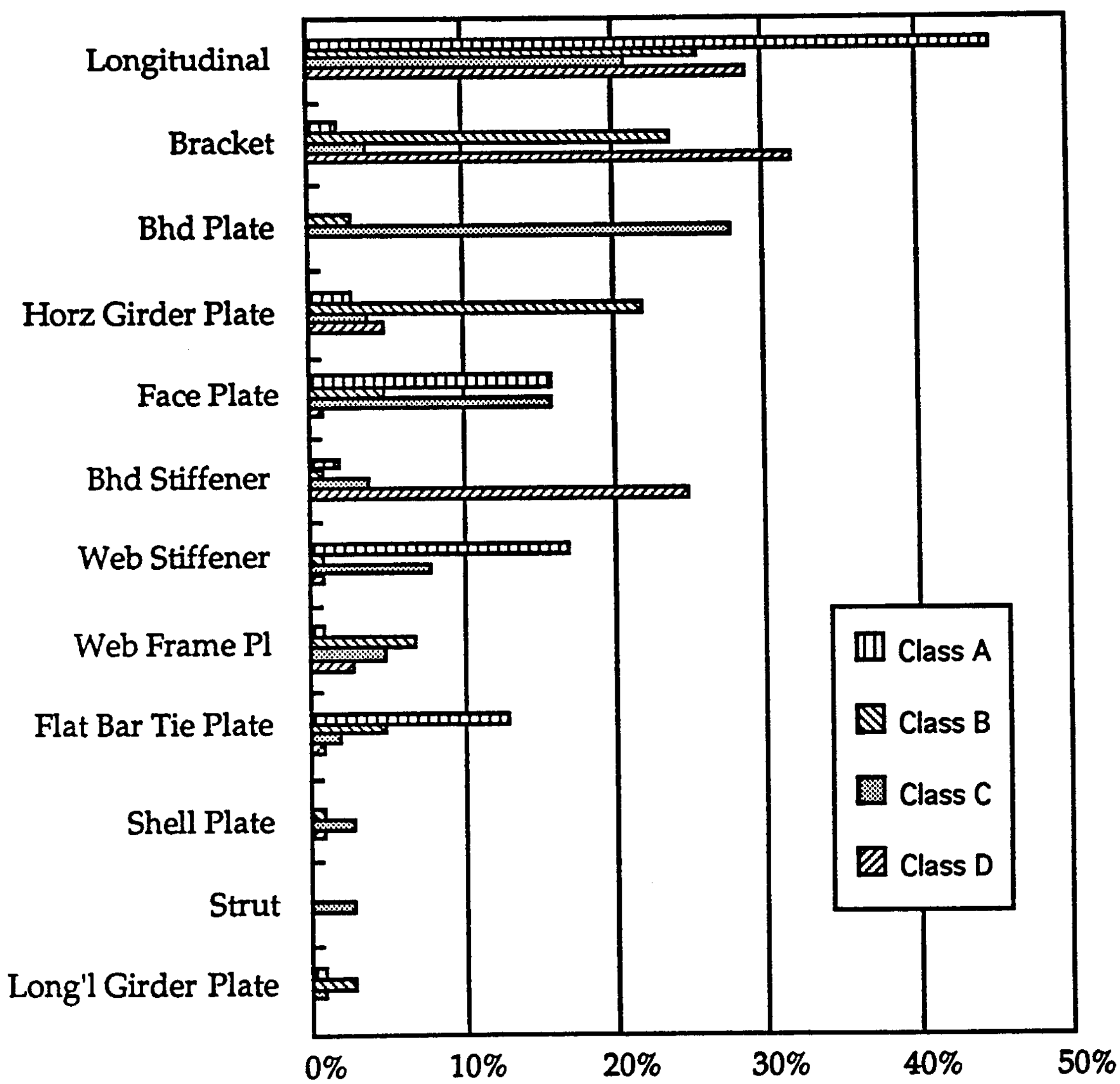


FIGURE 5 Frequency of fracturing versus structural component classes A—D.

significant because they demonstrate how differences between the details in ships may make a difference in the fracture performance, and they encourage a strategy of improved design practices.

Figure 6 shows the rate at which new fractures have been discovered in each class versus the age of the ships. For some tanker classes, the onset of significant fatigue fracturing occurred when the ship was about 10 years old. Other tankers did not experience significant fracturing until they were nearly twice as old. Figure 6 also suggests the possibility that during the 1970s, the design of U.S. tankers may have been increasingly influenced by practices more appropriate for non-U.S. ships,which often have life expectancies of about 15 years. The trend lines in Figure 6 may be influenced by structural modifications, changed trading patterns, and increasingly focused and intense inspections and fracture documentation. For example the rate may have slowed during the past five years for some of the classes because some of the more egregious fracturing problems have been eliminated with structural modifications. Classes A and B operated between Alaska and Panama during their first five to ten years of service and now operate

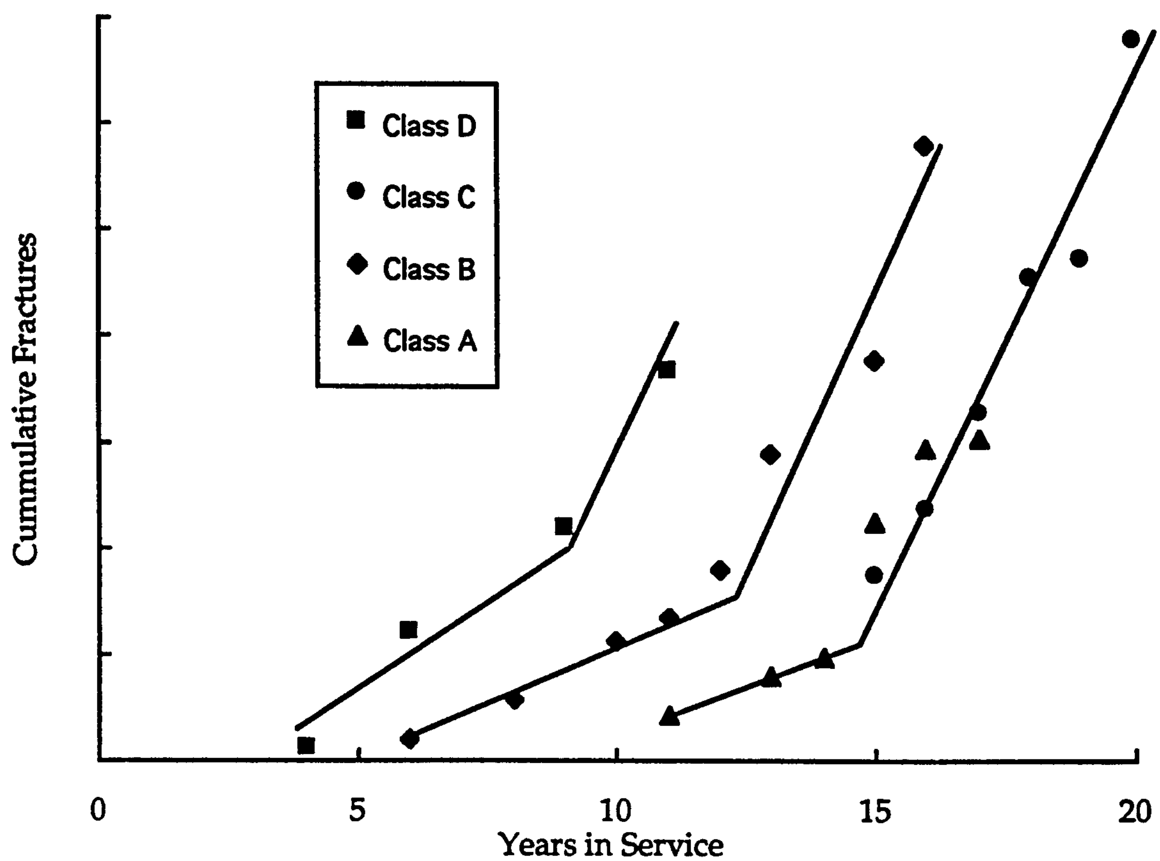


FIGURE 6 Cumulative fractures versus service life for classes A—D.

between Alaska and the U.S. west coast, which exposes them to longer and more frequent encounters with severe seas. Finally, as a practical matter, the more fractures found in a ship, the harder the search; the harder the search, the more fractures found. This typically human behavior may distort the recent record during the past 2 years for all classes. Figure 6 suggests that tankers designed without the benefit of fatigue- and fracture-growth resistance will eventually succumb to an accelerating fracture rate increase as they age. Historically, tankers without ongoing corrosion protection succumbed to wastage and the economics of steel renewal. Today with more effective corrosion measures, tankers designed without effective regard for fatigue may succumb to the economics of fracture repair and structural modification.

Typical Fractures and Modifications

These four tanker classes have mostly experienced small fractures relative to the size of the component involved, and most of these have occurred at joints between longitudinal frames and vertical stiffeners on transverse bulkheads. These fractures typically occur at the intersection of vertical stiffeners and bottom longitudinals and at the flat-bar tie plates between the first vertical stiffener on a transverse bulkhead inboard of the side shell and the side shell longitudinals. A smaller number of fractures have also occurred in the knee brackets of the web frames in classes B and C; the rudders of classes A, C, and D; and the deck of Class D. An understanding of these fractures illustrates the essential importance of rigorous engineering analysis and the anticipation, assessment, and mitigation of failure modes for successful structural design.

Figure 7 shows the connection between a vertical stiffener on an oiltight bulkhead and a bottom longitudinal (ABS Grade A steel) as built in Class C, the fracture location, and the repair/modification applied. An odd feature about this joint is the placing of the bracket on the opposite side of the bulkhead from the bulkhead stiffener in the original design. A bottom longitudinal segment between transverse supports approximates a built-in beam with local bending stress from the uniform hydrostatic loading and a primary stress field resulting from the macro-bending of the hull girder. The longitudinals are asymmetrical built-up profiles; thus, the state of stress in this structure is further complicated by the torsion that results because, in this case, the hydrostatic shear load does not pass through the shear center of the longitudinal. Fatigue damage occurs because the primary and local bending stresses and the torsional shear stress oscillate about a mean value due to the passage of waves. In the original design, the bracket toe and the outer edge of the bulkhead stiffener attached to the longitudinal, where the secondary moment and shear in the longitudinal are about 50 percent of the maximum values or greater. Initially fractures occurred in the longitudinal where the bulkhead stiffener is lapped onto it or at the bottom of the bracket where it ends against the longitudinal. The bracket toe in the original design is above a drainage cutout in the bottom of the longitudinal. Co-locating these two features was another source of trouble that led to through-fractures of the longitudinals in some cases. When fractures occurred where the bulkhead stiffener lapped onto the longitudinal, an unanalyzed field repair added a bracket on the same side of the bulkhead as the bulkhead stiffener at some locations. The resulting joint was much stiffer than the original, and new fractures soon occurred at the top of the new bracket. Subsequently both brackets were removed, and a single bracket was

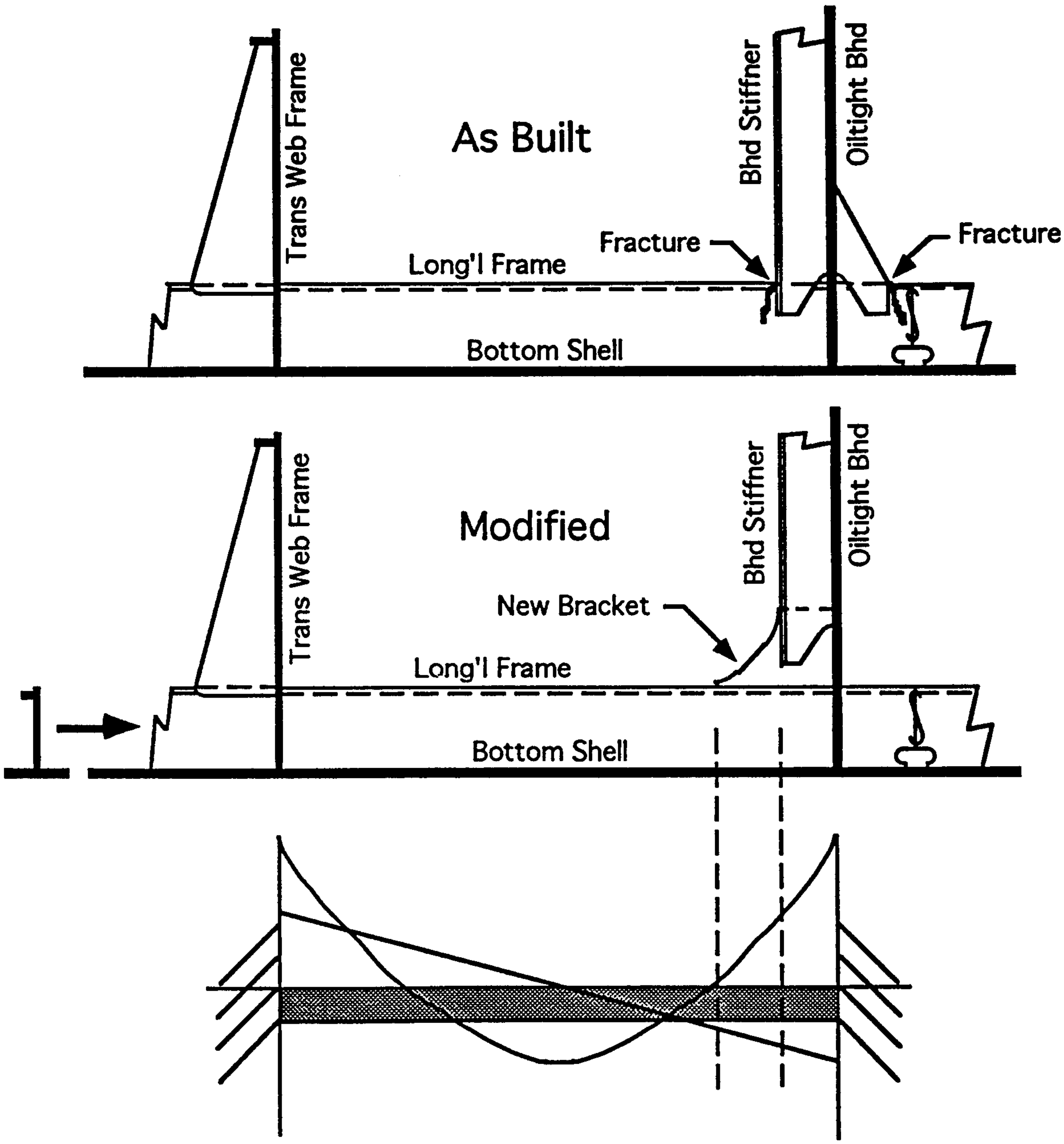


FIGURE 7 Class C bottom longitudinal frame intersection with transverse oiltight bulkhead.

installed connecting the vertical stiffener and the bottom longitudinal. The design concept for the modification focused on locating the toes of the new bracket in regions of low bending and reduced shear stress and ensuring low resistance to torsional displacement of the longitudinal.

The toe of the new bracket is near the point of zero moment for the longitudinal, and a large radius with a nearly vertical run-out was used to minimize the stress at the connection between the bracket and the vertical bulkhead stiffener. This modification has not refractured after about 6 years and may prove successful. The evolution of the joint in Figure 7 shows the importance of joint design that minimizes stress concentrations in identifiable regions of greater stress, particularly under oscillating load—in other words, design that anticipates and mitigates failure. The example of Figure 7 also shows how excessive stiffness may attract load more effectively than it increases the joint's capacity, especially with respect to fatigue. When unanalyzed stiffness increases are combined with significant and increased stiffness discontinuities, it is unlikely that any fatigue performance improvement will result. This situation may be exacerbated when modifications that greatly increase stiffness, and thus the tendency to attract more load, are made selectively only to the joints that have failed. The result can be a few load lightning rods in a structural matrix of less-stiff structure. In this case, any improvement may be negated by the increased load, which the modified detail sees compared to similar but unmodified surrounding joints. For this reason, matrix structural solutions are much more likely to identify the effectiveness of a proposed design or modification correctly. Figure 8 shows the bottom girder of a transverse web in the wing tank of Class C (ABS Grade A steel). Fractures most often occur in the longitudinals joined to the web by the heavy brackets, rather than those joined with a flatbar; further illustrating the need for a structure that balances and distributes stiffness and load-carrying capacity appropriately.

Figure 7 also shows that the web of the longitudinal extends beyond the flange, which is welded onto the side of the web. In this case, the most highly stressed fiber of the beam is the

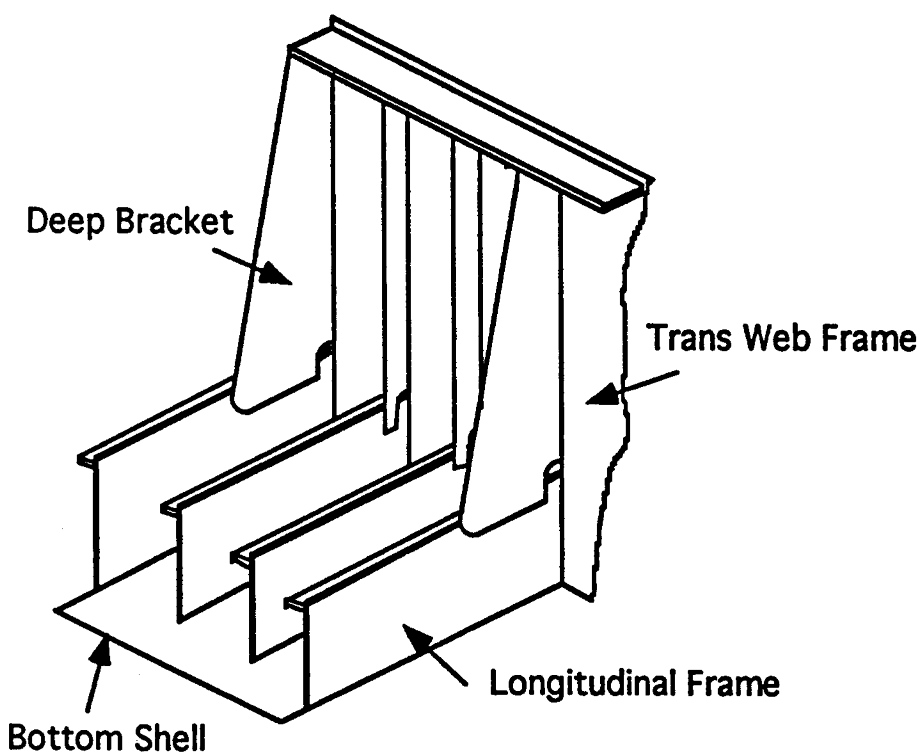


FIGURE 8 Class C longitudinal frame intersection with transverse web frame.

free-plate edge of the web. There is a lifelong disadvantage associated with this longitudinal profile that is noticeable when repairs are needed. Free-plate edges in tension are not good practice, and free-plate edges in tension with welds on them are worse. When longitudinals of this style fracture, they may be repaired by veeing and welding the fracture. The fatigue life of the repair will be less than the original, unfractured longitudinal because the repair weld, which is more susceptible to fatigue than the adjacent steel, is now at the point of maximum stress in the longitudinal. For a ship in middle age or older, such repairs may easily last its remaining life. In some locations, there is evidence of repeated welding repairs on the same longitudinal over the life of the ship. These cases illustrate the need for a careful welding procedure and welding finish to ensure success when welding across the exposed tension edge of such profiles. With the exception of significantly corroded or deformed structure, a carefully made welding repair to the fracture is preferable to cropping and renewing a section of longitudinal. The simple, single-welded repair minimizes the welding across the outer fiber of the longitudinal and generally resets the fatigue clock. Since most of these fractures occur at the intersection with transverse structure, additional connections are involved when cropping and renewing. In this case the amount of welding is further increased; consequently, the risk of weld-related future problems, especially when the work is done under field conditions, is increased. Fatigue fracturing is a very localized event that is almost exclusively associated with welds and/or structural discontinuities. For these reasons, there is little validity in the concept of "tired steel" and little practical benefit from cropping and renewing. The effort would more usefully be spent on better welded joint design, execution, and surface treatment.

Figure 9 shows the flat-bar tie plate that connects the first vertical-stiffener inboard of the side shell on transverse oiltight bulkheads with the side-shell longitudinals, as built and as modified, in the ships of Class B. Typically this tie plate is lapped and fillet-welded onto the longitudinal. Failure of this detail was very widespread in ships of Classes B and C before modification. Fractures occurred at the intersection of the tie plate and the longitudinal and propagated either outboard into the longitudinal or fore and aft into the tie plate. In a few cases, fractures would spread across a longitudinal frame and through the side shell, allowing oil to stain the exterior hull and resulting in an oil-pollution incident, albeit an extremely small one, necessitating an immediate repair. The nature and pattern of these fractures are consistent with that reported for the BP Oil Company 165 MDWT tankers (Witmer and Lewis, 1994), including the somewhat greater concentration of damage on the starboard, or loaded, weather side of the ship in the TAPS trade. The sharp discontinuity where the flat bar laps the longitudinal is the main cause of trouble with the original design. As originally built, this joint is similar in concept to that in Figure 7—severe stress concentration, located within a highly stressed region at the free-plate edge of an asymmetrical longitudinal, under oscillating load. Some classification societies no longer approve a lapped connection at this location. A modification was designed for this joint that relocates the intersection of the tie plate and the longitudinal to a lower stress region and further reduces the stress at the toe weld with a radiused nearly fore and aft runout of the plate. Calculations, including FEA, indicate that the new bracket reduces the stress at the intersection of the tie plate and longitudinal by nearly 60 percent. The modified tie plate has not yet refractured after about 6 years of service, which may suggest that lapped joints can be successful if they are the product of appropriate analysis. This repair/modification affected several hundred locations in each ship and required extensive staging.

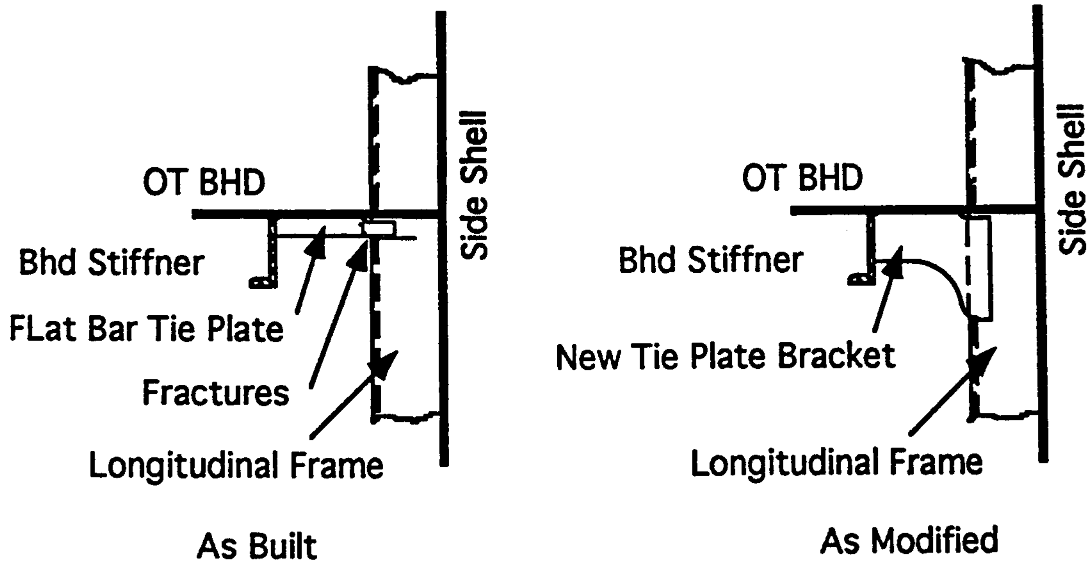


FIGURE 9 Class B side-shell longitudinal frame intersection with transverse oiltight bulkhead.

Figure 10 shows a near-full–depth fracture found at a few locations in the underdeck girder of the CVK (ABS Grade AH steel) in the center cargo tanks of Class B. The CVK is a vertical ring frame that supports the transverse oiltight bulkhead horizontal girder/vertical stiffener system and also provides a shear connection between the deck and bottom flanges of the hull girder. When adjacent center cargo tanks are filled to different levels, the unbalanced hydrostatic load on the bulkheads between the tanks is supported by the vertical legs of the CVK ring frame. These loads are dissipated in shear into the deck and bottom structure via the centerline girders of the CVK. The CVK also functions as a truss, providing a shear connection between the deck and bottom flanges of the hull girder. Without an effective shear connection on the centerline, shear-lag disproportionately concentrates the hull-girder bending stress in the deck and bottom structure between the side shell and the longitudinal bulkheads. In this capacity, the CVK functions as a Vierendeel truss, and large moments develop in the connections between the horizontal and vertical components. In both of these roles, ensuring the integrity of the load path through the connection between the vertical and horizontal components of the CVK is the critical issue affecting the success of the design.

The fractures originated at or near the aft end intersection of the CVK underdeck girder and the transverse web frame girder at the top of the knee bracket and were detected when the ships were about 5 years old. The initial discovery occurred after the crew reported banging sounds coming from a center cargo tank at sea. Figure 10 shows the geometry of the CVK and knee bracket as originally built. The design is characterized by massive and rigid brackets, creating fixed-end connections restraining the ends of the vertical member. The brackets are kept from rotating under load by their attachment to the grillage formed by the horizontal members of the CVK and the transverse web frame girders at the deck and bottom. The moments are resisted by vertical reactions at the fore and aft toes of the brackets. The bracket toes coincide

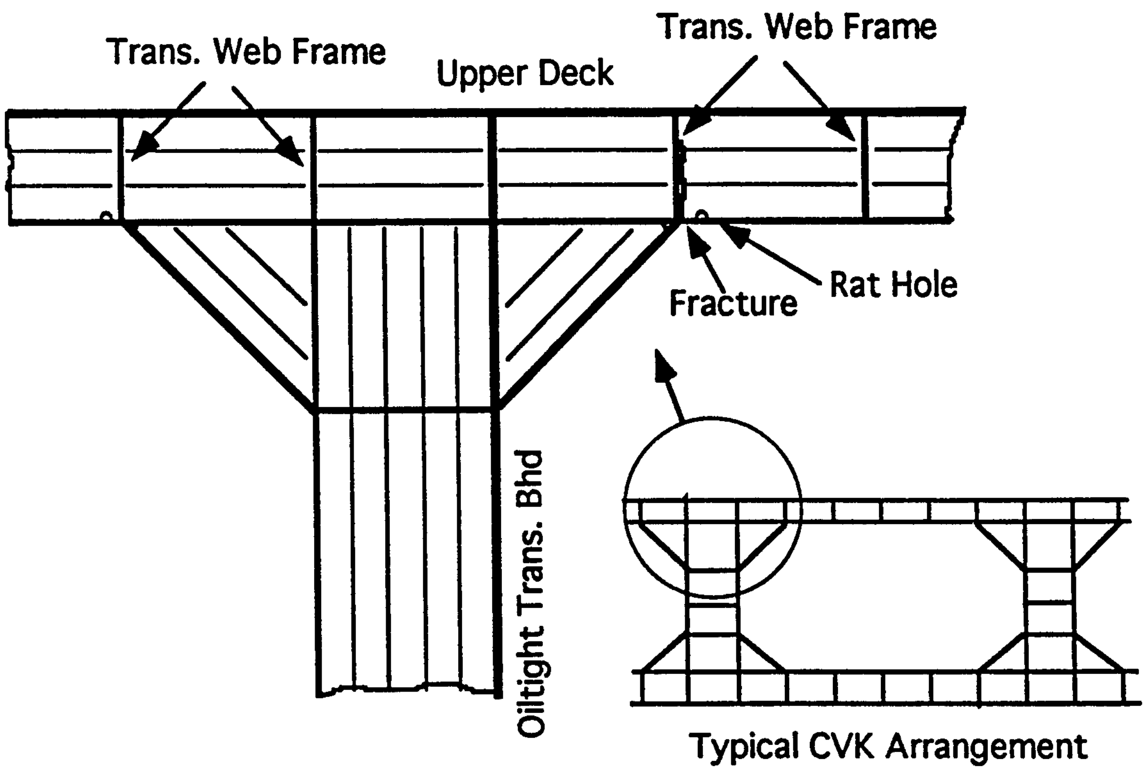


FIGURE 10 Class B CVK at intersection of upper deck and transverse oiltight bulkhead, as built.

with rat holes (i.e., semicircular cutout in the web plate at a butt weld in the flange) in the web plate adjacent to the flange of the underdeck girder forming the top component of the CVK. Following the failures, FEA identified significant stress concentration at the bracket toes, which was compounded by the stress concentration associated with the nearby rat holes in the CVK web plate. These design and detail features compromised the load path between the vertical and horizontal components of the CVK, introducing opportunities for failure. The use of flame-cut details and higher-strength steel also may have increased susceptibility to fatigue. Such details may have notched edges. Their use should be minimized, and the quality of their free edges specified and controlled in highly stressed regions. The use of higher-strength steel may result in greater allowable and actual stress in the structure without a concomitant increase in fatigue resistance. Spectral fatigue analysis, using a rigorously compiled voyage history and hindcasting of enroute sea conditions, identified that the expected life for this detail is about 3 to 5 years. The short life is due to the relatively high stress combined with more than 2 million cycles per year. The maximum stress may have approached yield at the top of the rathole with an L/20 wave in the full-load condition.

Figure 11 shows the detail of Figure 10, as modified to prevent reoccurrence of fracturing. The design strategy aimed at reducing the bracket-toe stress concentration with radiused transitions between the knee brackets and the CVK horizontal girder and removing free-plate

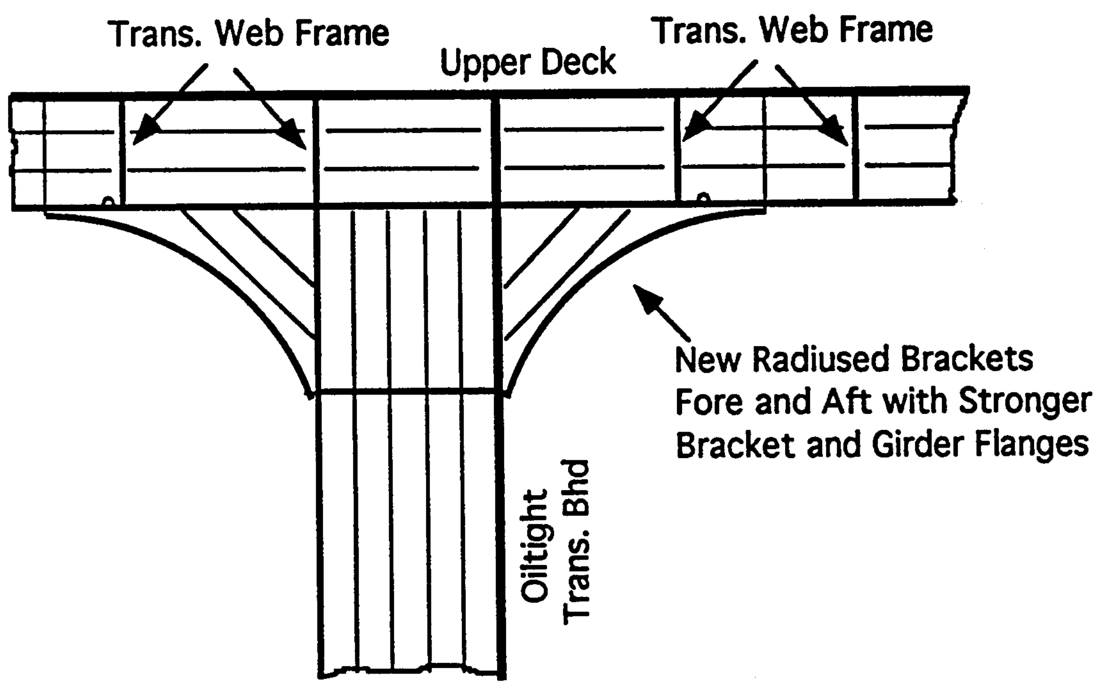


FIGURE 11 Class B modified CVK at intersection of upper deck and transverse oiltight bulkhead.

edges from locations with greater susceptibility to fatigue. The maximum stress at the critical fracture sites at the top of the rat hole and adjacent to the web frame are significantly reduced, and the fatigue life is extended beyond the expected service life of the ship. The modification has been in service for about 8 years, and may prove successful. A better modification would have filled in the rat holes and eliminated their free-plate edges. In 1987, when this modification was planned, the implications of fatigue were not adequately understood to appreciate the benefit of eliminating the rat-hole free edges.

The fractures in the CVK show the value of clearly establishing the load path through the structure, especially at the boundaries between components. Although the magnitude of the loads may be uncertain, the better that their path is identified, the better the chance that failure modes can be anticipated and mitigated. Subtle and seemingly minor aspects of a detail can greatly influence its resistance to fatigue fracturing. The rat holes are only 3 in. in diameter in a web plate that is 9 ft deep, yet their location and execution may have contributed to the fracturing of the CVK. Details like those shown in Figure 10, with straight-line intersections, are departures from earlier design practice characterized by radiused connections that lessened the likelihood of egregious stress concentrations. This change seems likely to have resulted from the need to economize the fabrication process. Economizing is a commendable goal, but significant departures from successful past practice, especially in complex structures, should be entertained cautiously. Similarly it is important that efforts to economize analyses do not result in a design process that inadequately identifies load paths and potential failure modes or one that obscures significant subtleties like the stress at the top of rat holes.

The use of high-strength steel may have contributed to the early failure of the CVK. This does not mean that the use of mild steel will prevent these kinds of fractures, regardless of the design process. Figure 12, the next example, shows a similar fracture that occurs in the mild-steel transverse web frames of this same class, apparently caused by local details that interfered with the load path through a highly stressed area. Figure 12 shows a near-full–depth fracture that occurred at a few locations at the outboard end of the underdeck girder of the transverse web frames (ABS Grade A steel) in the wing tanks of Class B. Figure 12 shows the first of a series of fractures in this general area of the web frames in the mid-length of the ships, which began after about 3 years of service. At the time, this fracture was not explained. The importance of fatigue as a failure mechanism was not appreciated, and FEA identified maximum stresses insufficient to cause failure. The cause was erroneously assumed to be related to stresses locked in during the construction process. It is now understood that these were fatigue fractures that are

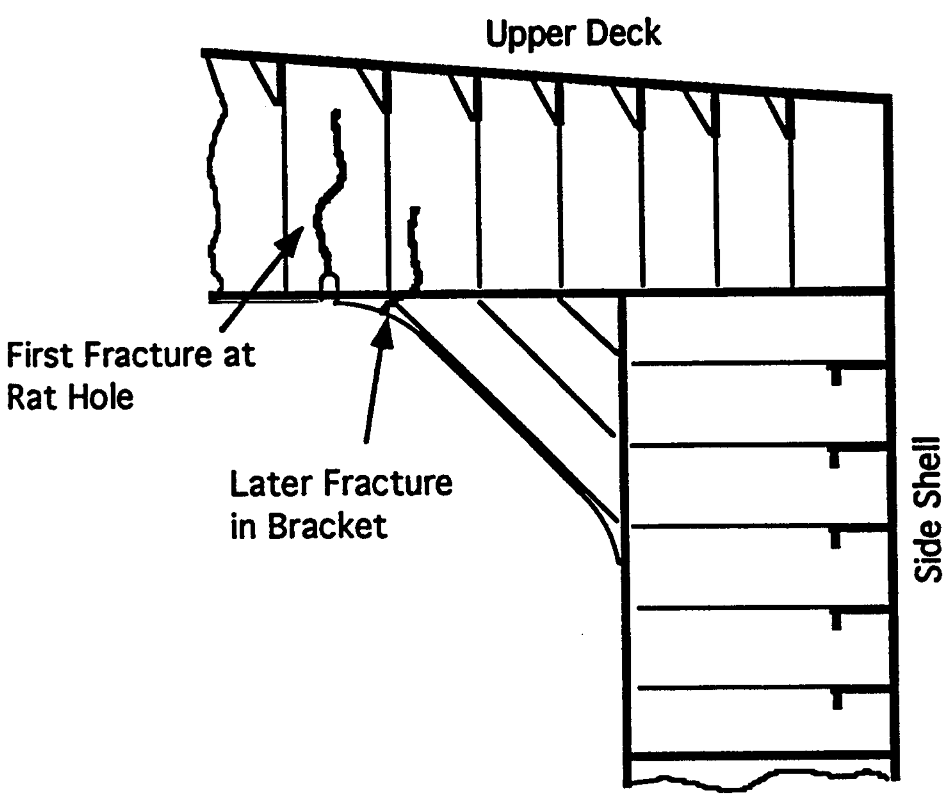


FIGURE 12 Class B transverse web frame at intersection of upper deck and side shell as built.

the result of the design and fabrication of the connection and its details. The abrupt straight line intersections between the bracket and the girder caused large stress concentrations in the web frame at the end of the bracket. The rat hole and its stress concentration were located within the region of intensified stress. During fabrication some of the rat holes may have been left with roughly flame-cut, free-plate edges, and in several locations there was misalignment between the structural components. This situation is similar to that already described for the CVK in this class and produced the same kind of fracturing even earlier in the service life. FEA identified a major stress concentration at the top of the rat hole where the fracture had initiated. Although the actual cause was not understood, the rat holes were filled in with plate, and the flange was widened locally in way of the rat holes in all of the web frames. This modification significantly reduced the stress at the top of the rat hole and seems successful in that fracturing from the rat hole has not reoccurred after about 10 years. Unfortunately, expensive thermal stress relief was also applied during the modification process in an effort to maximize success. This procedure was applied because of the incorrect assumption that the main cause of the fracturing was related to unrelieved construction-induced stresses. This example illustrates that FEA is a powerful tool even when applied somewhat empirically. It also demonstrates the importance of achieving a correct, first principles cause and effect relationship for a fracture as part of the analysis process. In this case, adopting a speculative cause for the fracturing led to the use of an expensive, unnecessary procedure that fortunately seems to have done no harm to the structure. Not to recognize the error of adopting speculative causes increases the danger of less benign mistakes in the future. This experience caused a reassessment, which was an early step toward appreciating the important role of fatigue.

When Class B was about 10 years old, additional fractures were found in the knee bracket at the outboard end of the transverse-web–frame underdeck girder in the wing tanks, near the closed rat holes. Figure 12 also shows these fractures, which initiated as smile cracks in the bracket web plate at the upper end of the lower diagonal flat-bar panel stiffener. A smile crack forms in a plate field at the free end of a panel stiffener and resembles the shape of a U, or smile, as it wraps around and encloses the end of the stiffener. The cracks spread downward to the outer edge of the radiused fashion plate between the bracket and the underdeck girder and upward through the transverse underdeck girder flange and into its web plate. As built, there was a discontinuity in the load path through the bracket between the underdeck transverse and the side-shell vertical girders. The stiffness of the bracket is sufficient to cause the girders to behave partially like beams with built-in end connections. Along the outer edge of the bracket there is a flat-bar panel stiffener that tries to provide continuity through the bracket between the girder flanges. As built, the alignment of the stiffener intersected the girder flange at about 45° and stopped with a steep snipe about 1 in. to 2 in. short of actual intersection.

Fractures initiated in the bracket web plate in this gap as loads attempted to leap from the girder flange to the panel stiffener. This situation was worsened by a deep vertical bracket on the web of the underdeck girder that intersects the girder flange directly above the end of the panel stiffener. In addition to these stress concentrations, the girder web plates extend beyond the flanges. The girder flange is welded onto the side of the girder web about 2 in. from the edge. This places a free-plate edge as the outermost (most stressed) fiber of the girder, increasing the likelihood of fracture initiation. This free edge was further compromised by notching it with a rat hole a few inches inboard of the end of the 45° flat-bar stiffener along the edge of the bracket. Initially an attempt was made on a trial basis to prevent these fractures by adding a flange in way of the radiused bracket and by scalloping the upper end of the panel stiffener. This modification was made without supporting analysis. In about 1 year, fractures were found at the ends of the new flanges. The state of stress in this structure is too complicated to expect anything but serendipitous success from unanalyzed modifications. Subsequent FEA identified that the stress in the gap at the end of the panel stiffener could be reduced by increasing the gap and providing a 30° snipe. As the gap was increased, stresses increased along the outer edge of the radiused fashion plate and at the ends of the new flange, particularly the top end. Extending the flange along the outer edge of the bracket reduced the stresses at the ends. Iterative solutions identified that the maximum stress at the probable fracture initiation sites at the end of the panel stiffener and the ends of the bracket flange could be reduced by about 50 percent. Figure 13 shows the modification adopted.

Functionally, the new design provides a path for load from the girder flange into the 45° bracket panel stiffener, where it can be dissipated into the bracket web. This new path replaces the original path, through the gap between the girder flange and the bracket panel stiffener, and eliminates the free-plate edges of the girder and bracket webs in the region of greatest stress. The new bracket flange is extended inboard under the girder web far enough to allow the girder flange

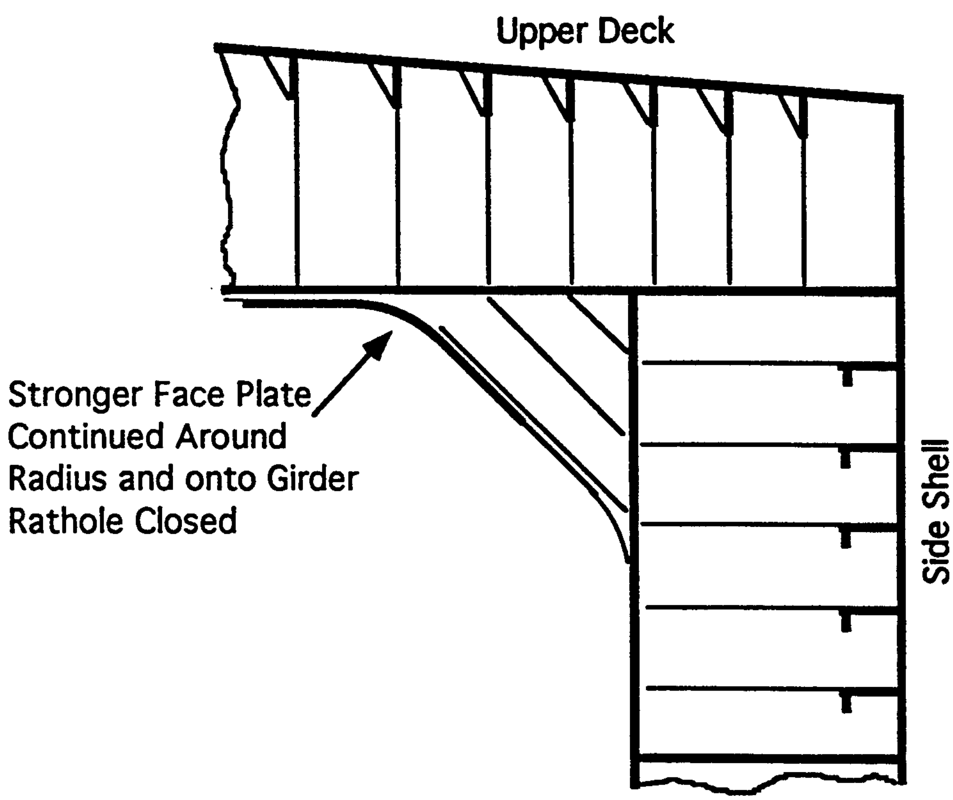


FIGURE 13 Class B modified transverse web frame at intersection of upper deck and side shell.

load to transfer to it through the girder web plate as shear. The bracket flange carries these loads around the radius and transfers them through the bracket web to the panel stiffener as shear. The gap between the top end of the panel stiffener and the girder flange is increased to ensure sufficient flexibility to reduce load transfer at this point. The modification is not optimal, but it is expected to eliminate further fracturing in this area for the remaining life of the class. This example shows how difficult it is to achieve any practical improvement in existing structure, within the limits of practical modifications, and reinforces the importance of sound structural concepts and rigorous engineering in the original design.

Significant fractures may also occur in tanker structure outside of the cargo block. Figure 14 shows a fracture that occurred in the throat of the cutout in the skin plate of the rudder at the pintle-pin casting in ships of Classes A, C, and D. About 10 years ago, a ship of Class C lost approximately the lower half of its rudder due to a fracture of this type. Following that incident, the rudders were modified by replacing the skin plate locally in the area of the

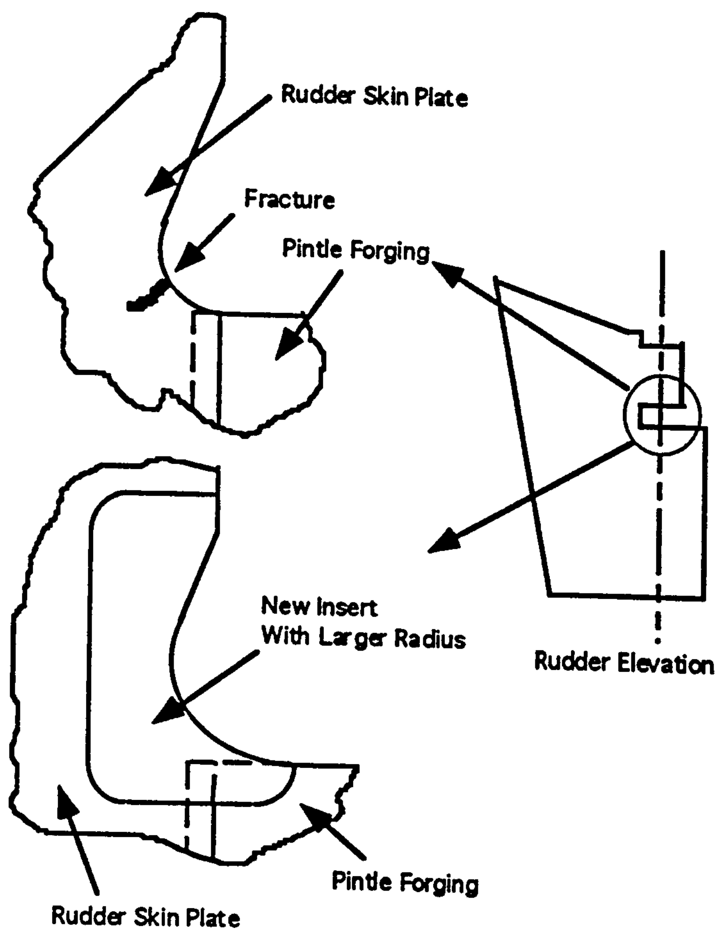


FIGURE 14 Rudder fracture and modification plan.

fracturing with higher-strength steel plate that is about 15 percent thicker than the original plate. Recently new fractures were found in the same location on several ships. These fractures are the result of low-stress, very high-frequency fatigue driven by the propeller discharge. The natural frequency of the immersed rudder for Class A is between 5 Hz and 7 Hz. Propeller blade frequency for these ships is about 6 Hz.

During their service lives, the rudders of these ships have experienced about 7.5×10^7 cycles. Even at modest stresses, fatigue fracturing may occur in welds or free-plate edges. The cutout is intended to prevent a stress concentration where the rudder skin plate is rabbeted into the pintle casting. The more horizontal the runout of the plate into the rabbet, the lower the stress at the top corner. Conversely, the more horizontal the run of plate into the top corner of the rabbet, the tighter the radius and the greater the stress in the throat of the cutout. This design requires that the plate intersect the top corner of the rabbet at an angle above horizontal to allow a softer radius in the throat, but at an angle small enough to avoid inducing fractures at the top of the rabbet. This may be a design that was effective for smaller vessels and that was scaled up to larger vessels unsuccessfully. It is difficult to achieve acceptable stresses at both locations, especially without the use of FEA and fatigue analysis. Figure 14 also shows the modification adopted for Class A. In this case there is adequate clearance between the swung rudder and the rudder horn to allow extension of the skin-plate intersection with the pintle forging. This allowed the designers to maintain the horizontal run out of the skin plate with an increased radius in the throat of the cutout. The stress at the toe of the skin plate is not increased, and the stress in the throat at the fracture site is reduced by about 50 percent. These fractures are another example of the dominant role that subtle detail has in the success of structure and of the need for rigorous analysis.

Several small class one fractures have occurred in the deck of Class D at the tank cleaning openings in the higher-strength steel deck. These fractures are typically less than 2 in. long at the outer surface. Fracturing originates from small notches in the cutouts, which are apparently the result of the flame-cutting process. In this case, a relatively high fluctuating stress field is combined with a free-plate edge; a stress concentration due to the hole and the small notches. The fractures are repaired by drilling a small-diameter stop hole, veeing and welding, and grinding smooth the cutout edge surface. Even with smooth edges, there are some openings in the deck of Class D in which the free-edge plate may have a shorter fatigue life than the expected ship life. Doubler and insert plates were considered as a means of reducing the stress and extending the fatigue life at these locations. An FEA/fatigue analysis shows that, in general, it is difficult to make doubler plates effective because of the stress concentrations at their edges and the discontinuity through the thickness between the doubler and original plate. The doubler tends to benefit the original plate on the side where it is attached but is significantly less effective on the opposite side. Based on analysis of several doubler-plate configurations, doublers may not be as effective as commonly supposed as a means of reducing stress and fatigue. The fractures at the tank cleaning openings in Class D are a cautionary tale about how the creation of ship structure is sometimes characterized by too little attention to detail throughout the process and the unnecessary bad results that follow. The fractures also show how old bromides like doubler plates may be insufficient when applied to a new paradigm of thinner plate, higher stress, and fatigue as a limiting condition.

Fracture Distribution

The distribution of fractures within the ships is roughly similar for each class. Fracturing occurs at the intersection of major structure, mainly the intersection of transverse web frames and bulkheads with the shell structure. Fractures have not been found in plate fields remote from intersections with major structure, except at significant stress concentrations like those associated with tank cleaning openings in the deck. Fractures are more prevalent at the boundaries of tanks that are used for both clean and dirty ballast. This effect is not explained by the greater cleanliness of the ballast tanks, which makes them easier to inspect, as is sometimes suggested. The cleaning of cargo tanks for inspections has become increasingly thorough during the past decade to ensure fire safety if hot work repairs are needed and also to prevent benzene exposure. Since the onset of the U.S. Coast Guard critical area inspection program (CAIP) inspections, small fractures are routinely found in cargo tanks at close range, and larger fractures, such as those in the underdeck girder of the CVK, have been detected from the tank bottom using an ordinary tank light. The probable cause of the concentration of fractures in the ballast tanks is their location. The mid-length is the most highly stressed part of the ship, where fracturing is most likely, and ballast is concentrated there to minimize hogging in the ballast condition. The concentrated ballast loads also affect the global shear distribution and cause unbalanced hydrostatic and sloshing loads on the ballast tank boundaries, which contribute to the increased density of fracturing at these locations. The structure of the ships is repetitive, and similar fractures in similar structure are typical. For this reason, inspections focused in selected mid-length tanks have proven to be a conservative and reliable means of monitoring the overall performance of the cargo block structure. The distribution of fractures for these four classes shows the importance of concentrating analysis and design effort at the connections between transverse web frame stiffeners and shell longitudinals, between transverse bulkhead vertical stiffeners and shell longitudinals, at the connections between web frame and CVK components, and at the free-plate edges associated with cutouts. A modest amount of additional effort in the design and fabrication of these few repetitive details would have prevented most, if not all, of the fractures reported. The general pattern of past fracturing is well documented and understood for several tanker classes, including the four discussed in this paper. Yet it is naive to suppose that future fracturing will necessarily adhere to the established patterns, and analyses are in progress to investigate how these patterns may change as the ships age and what countermeasures should be adopted.

Figure 15 shows the distribution of fractures found in Class A during nearly 40 ship years of service. The propensity for fracturing in the mid-length of the ship at the connections between longitudinals on the shell and longitudinal bulkhead and transverse webs and bulkheads is clearly indicated. The largest concentration of fractures is located within the wing ballast tanks, especially at the transverse boundaries. The horizontal lines of fractures across the transverse view show the location of small fractures found in the cutouts of the horizontal girders on the transverse bulkheads. Fractures near the side and bottom shell and longitudinal bulkheads are in the bracket or tie-plate connections.

Figure 16 shows the distribution of fractures found in Class B during about 30 ship years of service. The fracturing in Class B is somewhat less concentrated in the mid-length of the ship, but it is clearly associated with the wing ballast tanks and the historical practice of carrying dirty

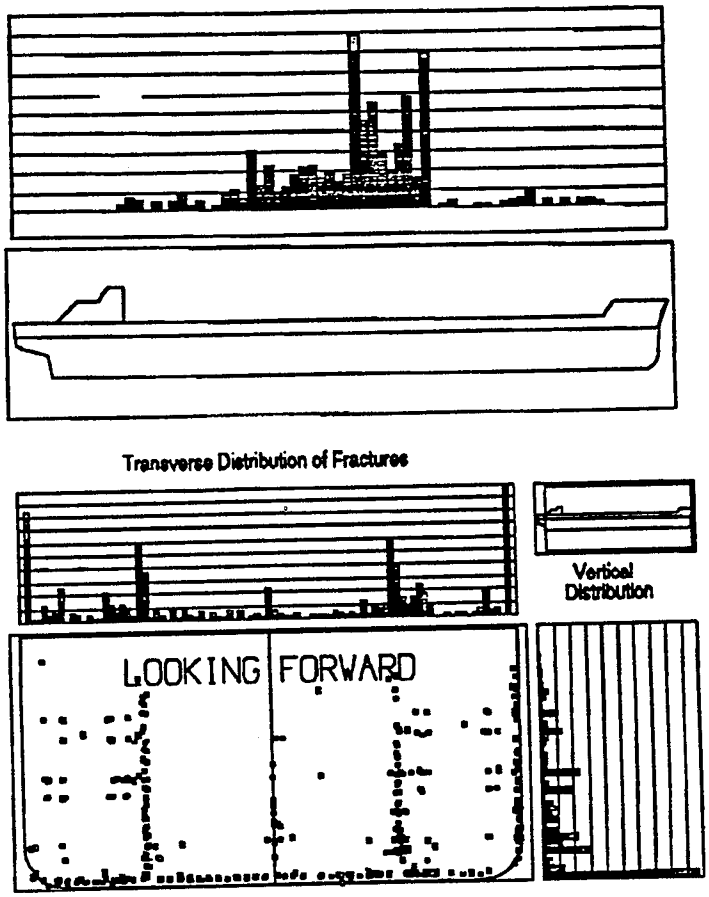


FIGURE 15 Distribution of fractures in Class A.

ballast in the number two wing and number three and four center tanks. The lines of the horizontal girders and innerbottom are evident in the transverse view from the fracture indications. The fractures in these locations are associated with cutouts in the transverse horizontal girders and bracket connections between the bulkhead stiffeners and longitudinals. The fractures at the side shell are mainly in the starboard side and uniformly distributed over the depth of the ship between the innerbottom and the athwartship girder of the transverse web frames. These fractures typically begin in the tie plate that is between the first vertical stiffener on the transverse bulkhead, inboard of the side shell and the side shell longitudinals. A smaller number of similar fractures are found on the port side concentrated at a lower level in the hull. These differences may be connected to the trading pattern of the ships, which exposes the starboard side at loaded draft and the port side at ballast draft to the open expanse of the Pacific Ocean to the west. A less pronounced but similar distribution of fractures near the side shell is evident in all four classes, suggesting a cause that transcends design and fabrication.

The transverse view also shows the fracturing at the top of the knee brackets at the outboard ends of the transverse web frames. Like the fracturing near the side shell, the fracturing in the knee brackets has greater frequency on the starboard side. The fractures indicated along

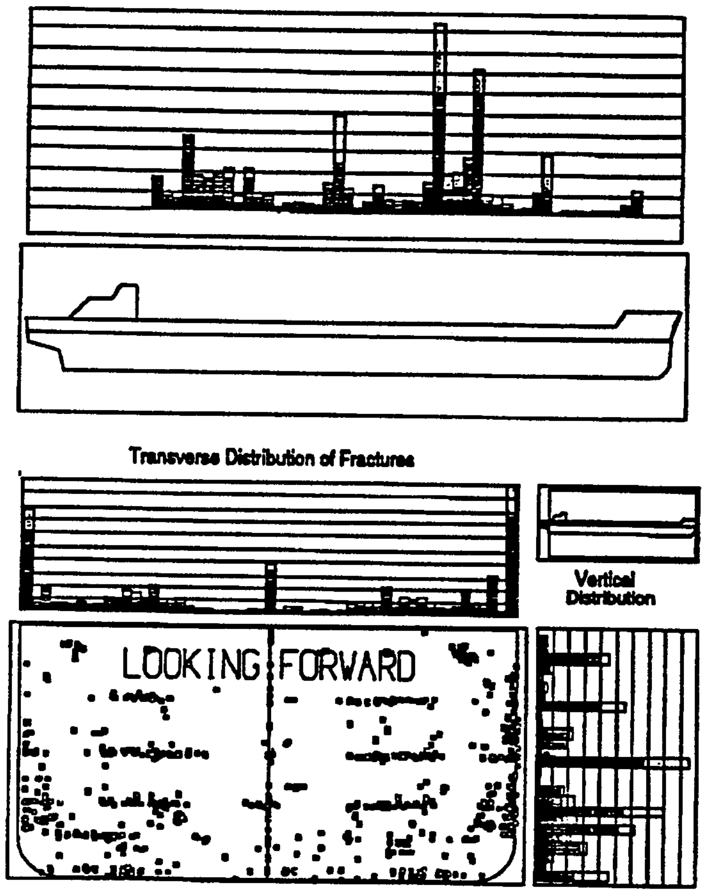


FIGURE 16 Distribution of fractures in Class B.

the bottom occurred in longitudinals at the top of drainage rat holes, again emphasizing the importance of attention to these details during design and construction.

Figure 17 shows the distribution of fractures in Class C during more than 60 ship years of service. The fracturing in these ships is a combination of that found in Class A and Class B, except that fractures in the knee brackets of the transverse webs and the CVK underdeck girder have been rare in this class. The longitudinal distribution shows that fracturing is concentrated in the vicinity of the wing ballast tanks, especially at the ends of these tanks.

The most common fractures in Class C are in the bracket connections between the longitudinals and transverse bulkhead stiffeners in the tie plates between the side shell longitudinals and transverse bulkhead and web stiffeners. Recently, fractures were found in the lapped connection between the transverse web underdeck and side-shell girders at two locations in one ship of this class. These lapped seams gave reliable service for more than 20 years and demonstrate that lapped joints can be successful throughout a normal ship life.

Figure 18 shows the fractures found in Class D during about 14 ship years of service. This ship is a segregated ballast tanker with multiple, clean-ballast wing tanks. The longitudinal view shows the concentration of fractures within the ballast tanks and at their fore and aft boundaries

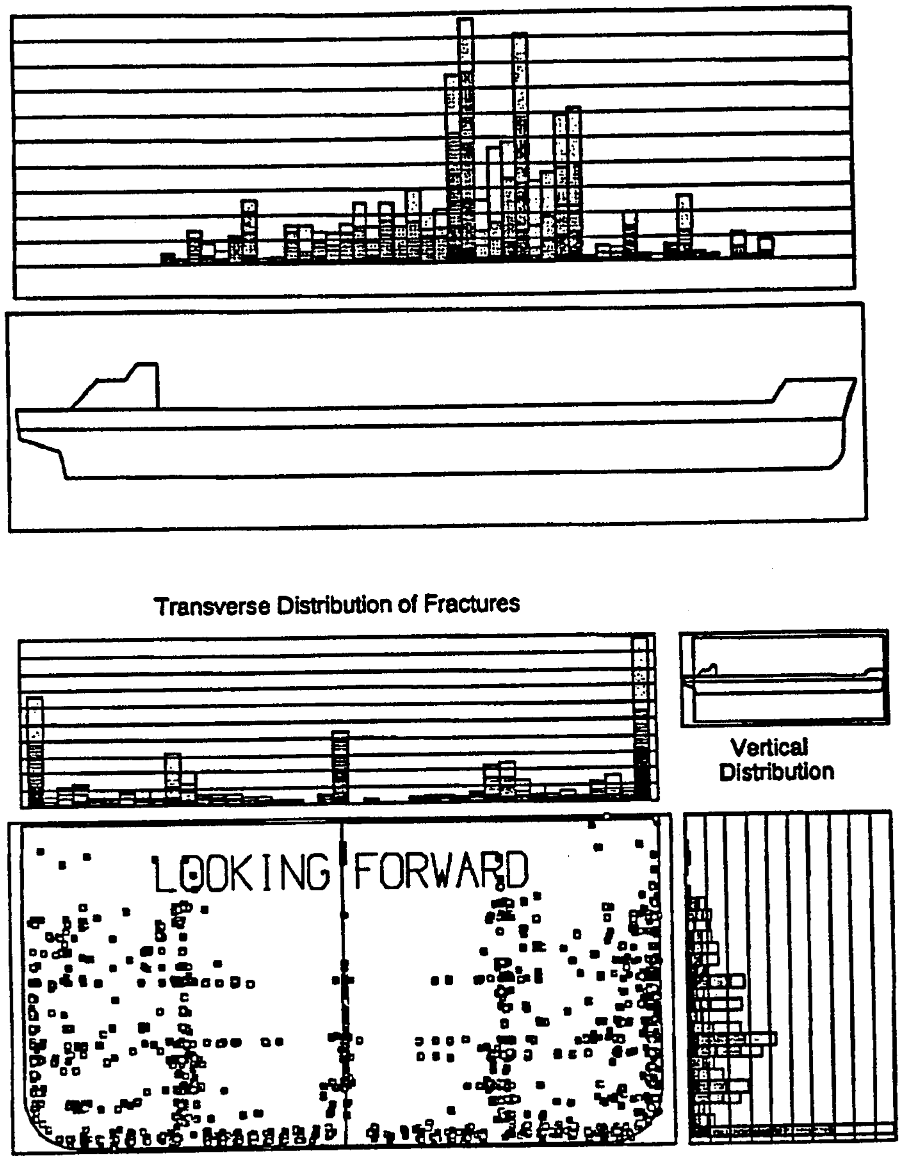


FIGURE 17 Distribution of fractures in Class C.

are similar to the distribution in classes A, B and C. The transverse view shows that the most common fractures occur at the connection between the vertical stiffeners on the transverse bulkheads and the bottom longitudinals and at the connections between the vertical stiffeners and horizontal girders on the transverse bulkheads. The fractures between the vertical bulkhead stiffeners and the bottom longitudinals are found only in the wing tanks because this class has an innerbottom in the center tanks. In the center tanks, the fractures occur at the intersection between the vertical stiffeners and the innerbottom.

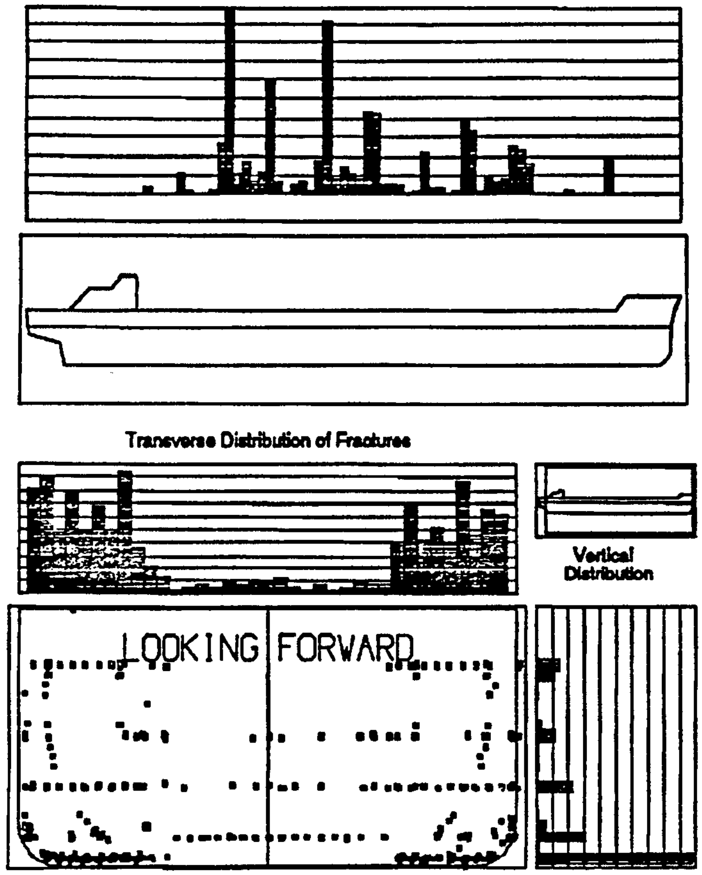


FIGURE 18 Distribution of fractures in Class D.

FRACTURE PREVENTION

Figure 19 shows the ongoing process for minimizing the incidence and severity of fracturing in the eight tankers discussed in this paper. Collateral goals include improving the structural design process by assimilating the performance experience of existing tankers and rationalizing the U.S. Coast Guard CAIP with better understanding of failure modes and risk. Better structural performance in existing tankers and better designs in the future ensure reduced risk and more reliable and efficient tankers. Rationalizing the U.S. Coast Guard CAIP ensures the most effective application of owner and regulatory inspection and maintenance resources. The approach defined in Figure 19 combines experiential fracture anticipation (documentation), analytical fatigue performance prediction spectral fatigue analysis), and mitigating intervention (structural modification and voyage planning).

Fracture prevention and mitigation depend upon four key technologies: documentation of the actual fracture record from the ships, spectral fatigue and fracture-growth–rate analysis, voyage planning, and definition of the sea environment. Documentation provides an experiential basis for anticipating fracture modes and locations, effectively focusing analytical and repair resources, and provides a basis for testing the validity of the fatigue analysis. Spectral fatigue and

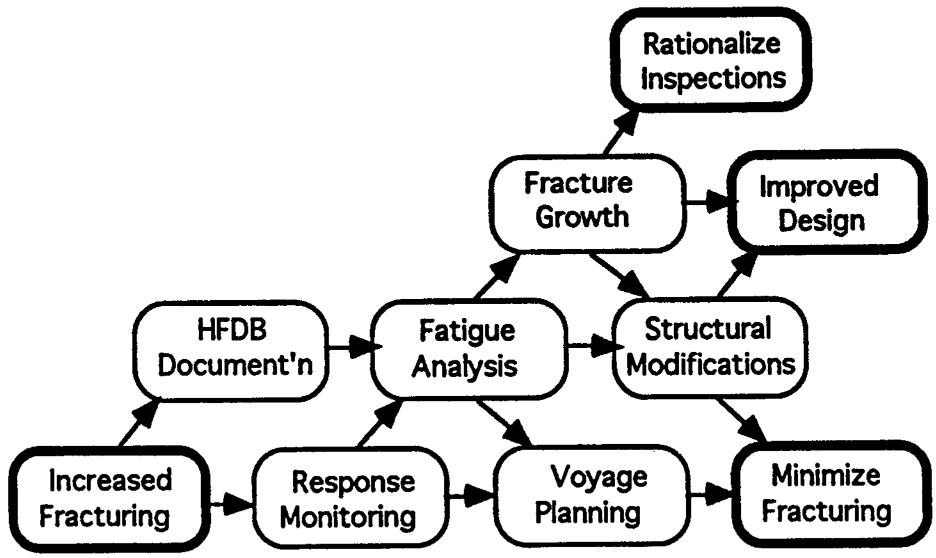


FIGURE 19 Fracture prevention process.

fracture growth analyses are the essential tools for ensuring that structural modifications and new designs have appropriate fatigue life and resistance to fracture from undiscovered flaws. Voyage planning provides the ship's bridge team with a tool for actively minimizing the accumulation of fatigue damage during the ship's routine operation. Definition of the statistical and real-time sea environment is essential to meaningful spectral analyses and voyage planning.

In the late 1980s, anecdotal evidence suggested that both the incidence and severity of fracturing were increasing in TAPS tankers. The hull-fracture database (HFDB) was developed to establish an accurate definition of the fracture record for classes A through D, and an onboard monitoring program was initiated for Class B to gain insight into the implications for fracturing of the motion and stress responses of tankers to the sea. HFDB revealed that the pattern of fracturing is well organized within the structure and is concentrated at a relatively few repetitive connections between the longitudinal frames and the transverse bulkheads and web frames. Preliminary results with the onboard monitoring system demonstrated that acceleration precursors could provide timely warning of increased risk of excessive stress and slamming events. Together these early findings suggested that fracturing might be reduced with a practical program of structural modifications and avoidance of damaging seas, based on fatigue analysis and onboard tactical voyage planning. Developing and maturing the fatigue analysis and voyage-planning tools have occupied nearly 5 years. From the beginning, the fatigue analysis development work has steadily improved our ability to design successful structural modifications that eliminate fracturing aboard ship. As the monitoring program has evolved into a tactical guidance tool, bridge teams have become increasingly proficient at reducing the accumulation of fatigue damage and avoiding severe hull stress and slamming events. The following paragraphs provide an overview of the general characteristics and integration of the four key technologies that have made this dual approach possible.

Documentation of Fractures

The fractures found in classes A, B, C and D are recorded in that HFDB, a graphics-oriented relational database specifically developed in 1989 with Aerohydro, Inc. (Shook and Sucharski, 1991). HFDB provides a user-friendly tool for desktop computers that tracks the distribution, type of structure involved, severity, and frequency of fracturing. Accessed parametrically, these data are the basis for both an experiential approach to failure anticipation and verifying the reliability of the spectral fatigue analysis process.

During inspections, data describing the fractures and repairs are entered on a fracture record sheet (FRS). An FRS is an 8.5 in. by 11 in. drawing that shows a particular part of the structure of the ship, and a complete FRS set for a ship includes views of all of the structure within the cargo block. Seven parameters are used to identify and characterize each fracture, including ship name, ship class, three-dimensional location within the ship, severity, type of structure fractured, type of repair (or modification), and date. Each record is numbered, and additional comments may be appended to better describe the fracture or its repair. Thirteen types of structure are identified in the HFDB: shell plate; bulkhead plate; longitudinals; face plates on web frames, girders, stiffeners, and longitudinals; brackets; struts; web-frame web plate; horizontal-girder web plate; bulkhead stiffeners; web-panel stiffeners; flat-bar tie plates; longitudinal-girder web plates; and longitudinal-girder, web-panel stiffeners. Fracture severity may be recorded as less than 3 in., 3 in. to 6 in., 6 in. to 12 in., 1 ft to 2 ft, greater than 2 ft, and "through." Through describes a fracture that separates the structure into two pieces. Three types of repairs may be identified: repair, renew, and modify. Repair describes veeing and welding the fracture. Renew describes a repair in which some of the fractured structure is replaced with new material, but no change is made to the original design. Modify describes a repair that includes changes to the original design to achieve a more durable structure. When the inspection is complete, the data from the FRS set are entered into the HFDB computer database via a graphics tablet. Each FRS is placed on the graphics tablet, and the appropriate boxes on it are touched with the wand to select values for the various parameters. Data are recalled graphically and presented on longitudinal and transverse sections of the ship as a series of segmented bar charts that show the location and intensity of the fractures requested. The displays have a resolution of one bar for each transverse web frame in the longitudinal view and one bar for each longitudinal frame in the transverse view. The data in HFDB are managed by PARADOX, which allows selective retrieval of any group and range of parameter. It is possible to recall and display only fractures in modified brackets, for example, greater than 3 in. long, that have been found since 1988 for a specific ship. In this way, reoccurrence of fracturing in modified structure can be detected as a measure of the effectiveness of a modification. Selective data retrieval also allows the operator to request recently discovered fractures so that the current pattern can be compared with the historical record. In this way HFDB can provide an early identification of emerging fracturing trends as the ships age.

Predictive Analysis

An experiential basis like HFDB is essential, yet it is insufficient for successful design. By definition, experience is retrospective, and, while it may give valuable insight, it is not able to reliably give foresight. Successful design also requires predictive analysis, which forecasts the future performance of new designs and warns of both unsuspected departures from past practice and as yet unmanifested problems with existing structures. Spectral fatigue analysis is the key predictive technology for successful assessment of the future performance of new and existing structures. Beginning in 1989, MCA Engineers, Inc. (MCA), developed a practical spectral fatigue analysis tool (Sucharski and Chueng, 1993), which is now routinely used to achieve design modifications within critical structure with acceptable fatigue performance. This work was built on a foundation of many years of FEA experience for offshore structures and ships by MCA and a statistical characterization of the North Pacific sea environment by Ocean Systems, Inc. (OSI).

Figure 20 shows a functional block diagram of the process for deriving fatigue life from stress values calculated with a telescoping FEA. A telescoping approach is used because it is necessary to model at least the mid-length half–length of the ship to ensure realistic structural responses to the sea, but a very tight mesh FEA model is needed to capture the stress at key

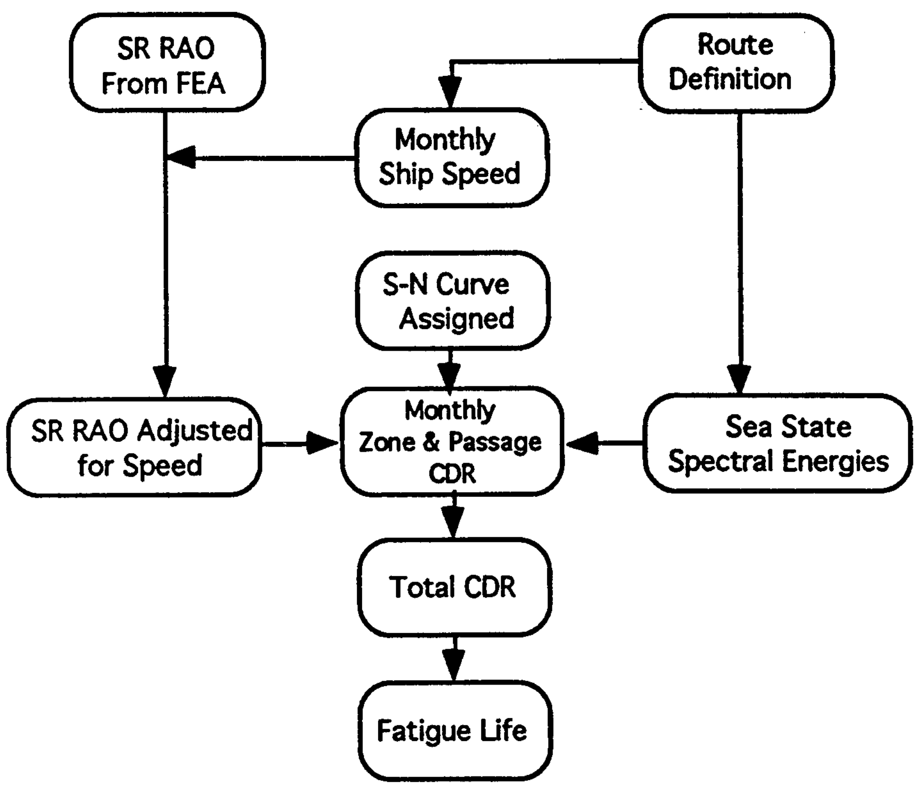


FIGURE 20 Spectral fatigue analysis outline.

details. The practical availability of computing power limits the number of elements that can be used, making it impossible to accomplish both of these models simultaneously. Typically the model is telescoped from a coarse-mesh global model, through an intermediate model, to a very fine-mesh local model to capture the stress values at the critical details. The distribution of stress is first determined for a global model, which includes about three cargo-tank lengths of the ship. The global model represents the shell, bulkheads, web frames, and girders as anisotropic plates that include the directional effects of stiffeners. The forces applied to this model include lightship weight, cargo and ballast weights, wave pressure, and the effects of the end moments and shears taken from the ship's hull girder. An intermediate model, which includes the final local area of interest, is then formed as a free-body segment taken from the global model. The intermediate model includes some structural details like the longitudinal frames and bulkhead stiffeners independent from the plating. Critical details, like the free edges of rat holes and the intersections of stiffeners and longitudinals, either are not modeled independently or modeled in too coarse of a mesh to provide the stress resolution needed. Along the boundaries between the intermediate and global models, there is a corresponding node in the intermediate model for every node in the global model. The internal forces at these global nodes become the external forces for the intermediate model. These external forces are in equilibrium with the internal pressure of the cargo and ballast and the external pressure of the waves. This process is repeated to create a local model that includes the critical detail of interest from the intermediate model. The local model is characterized by minimum mesh size in the areas of interest, with elements typically as small as one plate thickness.

The set of stress responses needed for the fatigue analysis is obtained by applying to the telescoping model a set of sinusoidal waves falling within the range of periods associated with the sea spectra for the ship's operational area. The cyclical stress range at local points of interest is the difference between the hogging and sagging principal stresses for a given wave period (length). Typically the cyclical stress range values are calculated for a matrix of 180 cases consisting of full load, winter (heavy) ballast, and summer (light) ballast conditions; head, following, and quartering seas; and 10 wave heights. The resulting cyclical stress ranges are divided by the wave height to obtain the normalized fatigue stress range, response-amplitude operators (SR RAO). The maximum SR RAO values are associated with wave lengths between 50 and 150 percent of the ship's length. The SR RAO values are corrected for the frequency of encounter, based on the ship speed. Peak SR RAO values are typically associated with wave-period encounters of about 10 s to 15 s at normal tanker sea speeds, depending of the ship's length and heading to the sea. SR RAO values may be as much as six times greater in head seas than in following seas. At normal tanker speeds and typical wave conditions, the effect of speed changes on the period of encounter and the SR RAO value in head seas depends upon the wave period and is increasingly influential with shorter period seas.

For TAPS trade tankers, the sea spectra needed to complete the calculation of fatigue damage were derived by OSI from the U.S. Navy's Global Ocean Wave Model (GSOWM) tapes for the areas of the North Pacific transited by the ships. The probabilistic sea is defined for each discrete zone along the ship's passage route for each month. For an existing ship, the log books are carefully analyzed to determine the time spent in each zone during each month for each passage versus the ship's heading, speed, and loading condition. For a new ship, an estimated lifetime operating service profile and sea spectra could be developed as the basis for the analysis.

The estimated service profile might consist of an envelope of plausible operating scenarios and monthly sea spectra for an envelope of different trading patterns.

The fatigue response spectra (cyclical stress range and frequency) are calculated by combining the SR RAOs with the monthly sea spectra and monthly zone operating profiles. The response spectra are compared with the U.K. Department of Energy S–N curves to derive a yearly cumulative damage ratio (CDR) according to Miner's method. With the telescoping FEA method, only the C (elements within a plate field and not adjacent to a weld) and the D (elements at a plate edge or adjacent to a weld) S–N curves are used. Using local models with a very fine mesh eliminates the need for other stress concentration considerations and the need for projecting the intersection nodal stress, since the FEA nodal or element center stress is known within about one plate thickness of the point of interest. The CDR values are integrated across each service month in each transit zone to determine the fatigue life prediction. In practice, a unit CDR (CDR/unit time) is calculated as a function of transit zone, month, ships heading, and loading condition. The unit CDRs are adjusted for each zone and month by the probabilities of encountering a given wave-height group and wave heading to derive a zone CDR. Knowing the ship's speed for each zone and month from the log books, the transit time through each zone can be determined. The passage CDR is calculated by summing the zone CDRs as a function of time spent in each of the zones for each passage. The total CDR is the sum of the individual-passage CDRs for the service period of interest. The fatigue life is calculated by dividing the service period by the total CDR corresponding to that period. CDR values greater than unity for a structural element mean that its fatigue life is less than the projected service life.

Spectral fatigue analysis is a robust technology that has proven to be a reliable predictor of fatigue performance in numerous applications for the eight ships discussed in this paper. For example, blind testing the method against actual fractures typically results in a predicted life within one shipyard period (2 to 3 years) of the actual life. On the other hand, this success, while encouraging, may only mean that for the present we are serendipitously making the correct number of compensating errors. Given the rigor of the spectral analysis approach, it is unlikely that a statistically significant comparison will be made anytime soon between predicted and actual fatigue performance. Even though comparisons of predicted and actual fatigue life are generally successful, designers should continue to ensure that the fatigue life of critical details is substantially in excess of the expected service life. Apparent success with spectral fatigue analysis has allowed the process to move forward with fracture-growth–rate prediction. The goal of this technology is to assess the risk of fracture growth to the critical length or to the point of shell penetration during the interval between shipyard inspections. Predicting fracture-growth rates may also demonstrate that, in at least some cases, fractures may grow from an initial stress concentration into a region of much lower stress and essentially stop. This technology is a logical extension of fatigue analysis and is expected to rationalize the inspection process and provide useful design criteria.

Although it has become a practical tool for routine use, spectral fatigue analysis remains a rigorous process that typically requires 3 to 6 months to complete for about five local structural details. Derivation of the sea spectra and operating profile can be developed simultaneously with the construction of the FEA models, but this work must be complete before the actual fatigue predictions can begin. Typically, developing the sea spectra and operating profile each require about 1 or 2 man months of effort, depending upon the age of the ship, quality of the log data and the number of transit zones. Once the FEA models and data are developed, reanalysis with modified models, or perhaps a new local model, can often contribute to the design process within 1 or 2 months. In the final analysis, the work's success depends upon sophisticated and experienced analysts, a representative operating profile and sea spectra, and an accurate model of the structure.

The spectral analysis process has taught us a great deal about fatigue and stress distribution in these tankers and is also an important tool for assessing the impact of modifying operating procedures on the accumulation of fatigue damage. For example, the effect of reducing the ship's speed on fatigue has been considered. Reducing ships speed increases the period of encounter for head seas of a given wave period and decreases the number of cycles experienced in a given time interval. Figure 21 shows how the plot of the SR RAO spectra versus wave period shifts to the right (increasing wave period) with increasing speed in head seas for Class B. At zero speed, the period of encounter and the wave period are the same. As the ship's speed increases into a head sea, the waves have to be longer (i.e., greater period) to keep the period of encounter constant. Changes in speed have the same practical effect as changing the wave length or the length of the ship and move the ship into or away from the critical wave-length range for maximum stress response.

It is difficult to achieve a net improvement in fatigue by slowing the ship because, although the cycles are accumulating at a slower rate, the longer passage times associated with slower speed increase the total time at sea and thus the total number of cycles. Although clearly beneficial and appropriate in storm conditions, routinely operating the ships at reduced speed is apparently not always an effective means of improving fatigue performance by itself. For

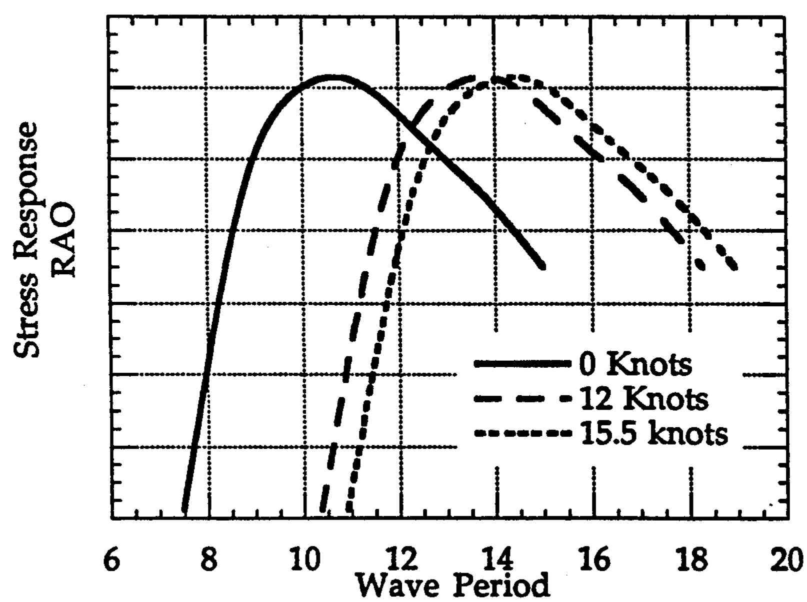


FIGURE 21 Class B stress response versus ship speed and head sea wave period.

example, slowing Class B from 15.5 knots to 12 knots in a uniform 15 s, sea-head sea increases time at sea to about 129 percent (assuming the same port time), decreases SR RAO to about 90 percent, and decreases the frequency encounter of the cyclic stress to about 94 percent for a cumulative effect of about a 9 percent increase in fatigue damage (cycles x stress). Even this large speed reduction is apparently unable to generate enough reduction in the either the SR RAO or frequency to achieve a net improvement. For wave periods less than about 14 s, the cumulative effect would be even worse since the SR RAO would increase with decreasing speed.

Fatigue damage can also be reduced by avoiding head seas. Like speed reductions, the effect of changing heading is generally smaller than the resulting increase in sea time, and the net effect is an increase in fatigue damage. Similarly, the still-water bending stress, which is the mean stress about which the cyclical stress oscillates, appears to have little effect on fatigue life (Sucharski and Cheung, 1993). It may be desirable for other reasons to minimize still-water bending and shear stresses or to adjust the ballast pattern. For example, heavier ballasting has been adopted for Class B to ensure a draft forward of about 60 percent of full load draft to minimize the risk of slamming. These findings for Class B regarding the effect of speed reduction, heading, and mean stress suggest that the principal operational means of improving fatigue performance is minimizing the wave heights encountered with an onboard voyage planning system.

Voyage Planning

Voyage planning is an interactive, computer-based decision support system that enables the ship's bridge team to acquire and use sea condition forecasts with predictive analysis of hull response to minimize structural damage (Lacey and Chen, 1993). Effectiveness requires accurate forecasts of sea conditions, reliable prediction of ship dynamic and stress response, and a user interface that communicates performance and allows for testing and optimization of voyage route options. Management and communication of information are the core technologies that make voyage planning possible. This system became practical only recently with the routine availability of sufficient computer power and satellite communication. Wave height and direction maps of the ocean area of interest are developed with data from the National Oceanic and Atmospheric Administration's Cray supercomputer and are downloaded to the ships via satellite at 12-hour intervals. Aboard ship, a desktop computer receives and stores the wave condition forecast, which provides the bridge team with daily enroute sea conditions for a period of up to 6 days into the future. Using a shipboard computer with the sea condition forecasts, the bridge team can predict the probability of slamming and exceeding preselected stress and motion responses and can project transit times for different route and speed options. The ship's computer also continuously monitors onboard instrumentation and alerts the bridge team if it detects response precursors that indicate that slamming or stress and/or motion responses exceeding preset maximums are imminent. The bridge team can then use the computer to test the effect of alternate heading and speed combinations. The development of this system occurred over a period of 5 years and included feasibility testing in 1990 and 1991; development testing in 1992 and 1993; a prototype development installation in 1993 and 1994; and installation of simplified, upgraded systems in

1994 and 1995. The following paragraphs describe the development path for the voyage planning system and give an example of its effectiveness in combating fatigue damage.

For the ships discussed in this paper, the development of voyage planning began in the winter of 1990 (Lacey and Edwards, 1991) with experiments aboard an ARCO 190 MDWT tanker to determine the frequency and intensity of slamming. This work was motivated by bottom damage in the forward part of the hull and initially was intended to determine if slamming was the cause. The 1990–1991 experiment was concurrent with the growing awareness of the importance of fatigue, and it was evident that hull monitoring might be developed into a broader tool for minimizing structural damage. Subsequent work focused on development of a voyage-planning tool that combined strategic and tactical features. The strategic features would make use of long-range wave forecasting to enable the bridge team to plan routes that minimized exposure to severe seas. The tactical features would make use of ship motion precursors, which signal the onset of damaging events, and stresses and/or motions that exceed preset maximums. Strategic use depends upon confidence in the reliability of the wave forecasts. Twice daily downloading of the wave height and direction forecast maps from OSI was initiated on an experimental basis and was compared with onboard observations and enroute wave-rider buoys. The standard deviation of the predicted versus measured wave heights was found to remain within 1 m during the first 72 hours of the forecast (Lacey and Chen, 1993). Tactical use depends on identifying precursors for the ship's response, which could be used to predict the onset of damaging events.

The precursor concept anticipates using easily monitored, empirical, intrinsic behavior to determine that conditions conducive to slamming are developing. The situation is analogous to riding in a closed elevator. If you release your briefcase and it doesn't fall to the floor of the car, you may reasonably conclude that a sudden slam is imminent, despite your ignorance about any external information. Similarly aboard ship, it might not be necessary to know very much to anticipate that slamming or exceeding limiting values of hull girder stress and/or motions is imminent. During the 1990–1991 program, triaxial accelerations, roll and pitch, and ships heading were the principal parameters recorded in an effort to determine whether slamming was occurring and to test the feasibility of using precursors. The 1990–1991 test program established that slamming was occurring and that ship-response precursors could be used to warn of impending slams and other exceptional responses (Lacey and Edwards, 1991).

In 1992 and 1993, additional shipboard tests were made to assess the feasibility of establishing a relationship between hull girder stress and accelerations (Lacey and Chen, 1993). Conceptually the ship is a large wave-rider buoy. Measuring the ship's acceleration response, and knowing the ship's characteristics, it is plausible that the associated hull bending moment can be determined. Acceleration is the preferred precursor because it is easily measured with comparatively reliable, simple, mature, and inexpensive instruments and can be measured at a convenient location near the computer. The issue is reliability. Use of acceleration eliminates the need for maintaining comparatively complex instrumentation suites, such as deck strain gages and forebody pressure sensors. Without this simplification, voyage planning risks degenerating into an instrument maintenance program. The 1992–1993 program was an ambitious effort to acquire sufficient field data to better understand the slamming process and to determine whether hull girder stress and slamming could be reliably correlated with accelerations. For this reason, a deck strain gage and forebody bottom and side-flare pressure sensors were included. During the winter of 1993–1994, this system evolved into a prototype voyage-planning tool. Clear correlations were established from the field data for vertical acceleration and midship strain and for vertical acceleration, pitch, and slamming in both the full-load and ballasted conditions. The system design was simplified during 1994, and subsequent installations in the 190 MDWT and 90 MDWT tankers during the winter of 1994–1995 relied only on inputs from the ship's acceleration, pitch, and roll sensors to determine hull girder stress and warn of imminent slamming (Lovdahl et al., 1995).

An experience during a severe North Pacific storm in mid-December 1993 provides a demonstration of the value of voyage planning for reducing fatigue damage. Two 190 MDWT tankers coincidentally met headed north to Valdez in ballast near the entrance to San Francisco Bay. Ship A had left Long Beach and sailed north at a modest speed, which placed it about one-half day steaming south of Ship B as it left San Francisco Bay. Ship B had the prototype voyage planning system, and the master had planned his route and speed to allow him to pass between two severe wave environments that would move across the Gulf of Alaska during the next 5 days. Voyage planning was then unavailable to Ship A because of a hardware problem. During the critical first 24 hours, Ship B increased speed and pulled away from Ship A to avoid the band of increasingly severe seas moving across the middle latitudes of the Gulf of Alaska. This was the only opportunity to position the ships to pass ahead of the approaching severe sea zone. From the wave forecast maps, the master of Ship B recognized the importance of getting the ship to the north side of the band of worsening seas developing across the middle latitudes of the Gulf of Alaska. The master of Ship A, without the benefit of the forecasts, followed conventional practice and slowed in the face of a rising head sea. During the next 24 hours, Ship B moved through decreasing seas that were shifting to the beam and quarter. Ship A continued to slow, and for approximately the next 36 hours struggled against increasing head sea and remained in severe seas for about 12 additional hours.

The stress component of fatigue damage is proportional to wave height, and fatigue damage is proportional to the produce of stress and exposure time. By comparing the wave heights encountered and length of exposure for the two ships, the increase in the cumulative fatigue damage was about 20 percent more for Ship A than for Ship B during this passage. The effect was probably greater than 20 percent since Ship A also spent more time in head seas than Ship B. At the same time, although receiving less damage, Ship B increased its lead over Ship A by about 21 hours, with the resulting improvement in efficiency. This incident is not unique. The record shows other comparative cases for sister ships and routine avoidance of severe seas by bridge teams using voyage planning.

CONCLUSIONS

Most fractures in tankers are small and do not propagate at a rate likely to cause a problem during the normal 2- to 3-year shipyard inspection cycle. A few are apparently more serious, but the risk is poorly understood and speculative. Considerable resources are currently focused on finding and documenting 1- in. fractures in tanker structures. This effort has largely served its purpose by significantly raising the collective consciousness about fracturing and causing the development of better documentation and predictive and operating tools, as described in this paper. We have learned a lot through this process. For example, we know with a sound experiential basis that fracturing isn't random—it occurs at the intersection of longitudinal and transverse structure—and that it is subtle, highly localized, and usually associated with production details. The importance of fatigue is now recognized, and we have learned that design and operating practices can prevent and mitigate its effects. The U.S. Coast Guard was the catalyst for this process, and it is yet another example of their distinguished history of contributions to the safety of ships.

Today it would be useful to move toward a better understanding of the risk implications of fracturing by differentiating the significant from the insignificant. This is difficult to do while the emphasis remains on frequent detailed inspections and the methodical and bureaucratic documentation of trivia. Recently a fracture risk-implication process was started for the eight ships discussed in this paper. This work is intended to provide a predictive basis for identifying the characteristics of fractures that may propagate through the oil–water interface and fractures that may propagate to lengths that compromise structural integrity during the interval between shipyard inspections. This capability will help rationalize the inspection process, provide a better basis for interim repairs, and develop an analytic predictive tool for designing ship structure to meet performance expectation. Differentiating between meaningful and trivial fractures is also needed to ensure that new fracture trends are not lost in a sea of data as they emerge. It is important that we nurture the experiential basis for design and not compromise the integrity of the data by emphasizing quantity over quality.

The fracturing described in this paper could have been prevented with a better design process. The essence of design is anticipating and preventing failure, not predicting that stresses are within allowable limits. Anticipating failure requires a strong experiential foundation. Preventing failure requires robust predictive analytical methods. Contributing to the experiential basis for design by sharing actual failure descriptions is an important goal of this paper. The other key step in anticipating failure is identifying the load path through the structure, especially as it crosses the boundary between components. The design should begin by answering the question, "Where and how will it fail?," based on knowledge of past failures and the load-path concentration points. A "plug and chug" application of classification rules or a broad-brush analysis that focuses on scantlings without designing the critical joints is not enough. The experience described in this paper shows that fracturing initiates in the details needed for production. It is essential that those details are included in the fatigue-life and fracture-growth–rate predictive analyses. It appears that, in the past, too often the structure was designed against yield and buckling and then forwarded to an independent process that added the production details. The nature of these details often determines the success of the structure. Great effort is applied to the development of the details to ensure producibility. A comparable level of effort is needed to ensure that the necessities of producibility do not compromise the integrity of the design. Improving this situation requires better availability and interpretation of failure data and the development of performance standards for predictive analysis and structural reliability. The experiences described in this paper show that in some cases the process permitted a favoring of the exigencies of production at the expense of the necessities of structural integrity. These separate design and production lives need to become integrated if we are to satisfy both efficiency and long-term reliability.

Experience already shows that voyage planning can significantly reduce the accumulation of fatigue damage in ships. These prototype programs are breaking down the boundary between the design/construction and operating lives of a ship and adding to our knowledge of the actual loads that ships experience. Much work is needed to improve the user interface, test the feasibility of surveillance of the near ship environment, and establish better correlation between ship response and the accumulation of fatigue damage. The user interface associated with the system described in this paper is characterized by a video game appearance and the ability to access historical, real time, and predicted ship behavior. These features provide the basis for improving design and operating practices, ongoing monitoring of the ship's performance, and the ability to test optional speed and heading combinations to mitigate damaging situations.

Within storm environments, areas of distinctly more severe waves may occur, analogous to the way that thunderstorm clouds develop and persist within the matrix of general storms in the atmosphere. If such conditions exist, they are outside of the envelope of resolution and persistence likely to be identified by wave forecasting anytime soon. In that case the ability to recognize and avoid these locals by prudent maneuvering must depend upon shipboard sensors. Preliminary experiments suggest that signal processing of the radar back scatter may offer a solution to this problem. The ability to detect the predominant representative wave direction has been demonstrated, and the pattern of crests is also discernible from the radar scatter. More rigorous processing of these data might also identify persistent areas of greater wave heights within the general sea matrix.

In the final analysis failure experiences, predictive analysis, and voyage planning are tools that will not improve the reliability of ship structure without capable people using them. More thought should be given to educating and developing designers, not just analysts. It is important that the design process as well as analysis receive emphasis and that appropriate attention is given to developing the cognitive and holistic cerebral mechanisms needed to create an integrated design. Similarly, maritime schools need to stay abreast of the evolution of voyage-planning technology and ensure that sailors can realize the promise of these systems and maximize their contribution to the successful lifetime performance of their ships.

REFERENCES

Lacey, P., and H. Chen. 1993. Improved passage planning using weather forecasting, maneuvering guidance, and instrument feedback. Presented at the meeting of the Los Angeles Section of the Society of Naval Architects and Marine Engineers held November 11, 1993 in Los Angeles, California. Jersey City, New Jersey: Society of Naval Architects and Marine Engineers.

Lacey, P., and R. Edwards. 1991. ARCO tanker slamming study. Presented at the West Coast Tanker Operations Symposium of the Society of Naval Architects and Marine Engineers held October 10–12, 1991 in La Jolla, California. Jersey City, New Jersey: Society of Naval Architects and Marine Engineers.

Lovdahl, R., H. Chen, and P. Lacey. 1995. Onboard computer assisted strategic and tactical voyage planning. Presented at the Society of Naval Architects and Marine Engineers California Joint Sections Meeting (CyberNautics 95) held April 21, 1995 in San Diego, California. Jersey City, New Jersey: Society of Naval Architects and Marine Engineers.

Murray, A.J. 1916. The Strength of Ships. New York: Longmans Green and Company.

Shook M., and D. Sucharski. 1991. Hull fracture database. Presented at the Marine Structural Inspection, Maintenance, and Monitoring Symposium held March 18–19, 1991 in Arlington, Virginia. Washington, D.C.: Ship Structure Committee and Society of Naval Architects and Marine Engineers.

Sucharski, D., and M. Cheung. 1993. Application of spectral fatigue analysis to the ARCO 190 MDWT tankers. Ship Structures Symposium '93 held November 16–17, 1993 in Arlington, Virginia. Washington D.C.: Ship Structure Committee and Society of Naval Architects and Marine Engineers.

Witmer, D.J., and J.W. Lewis. 1994. Operational and scientific hull structural monitoring on TAPS trade tankers. Society of Naval Architects and Marine Engineers Transactions 102:501–533.

Local Cracking in Ships—Causes, Consequences, and Control

Donald Liu and Anil Thayamballi

ABSTRACT

The larger trends related to cracking found on typical oceangoing vessels (primarily tankers and bulk carriers) are first reviewed on the basis of available damage data. The typical interrelated causes of such cracking are high local stresses, extensive use of higher-strength steels, inadequate treatment of dynamic loads, adverse operational factors (harsh weather, vessel handling), and preventable structural degradation (corrosion, wear, stevedore damage). Three consequences of cracking are illustrated: structural failure, pollution, and maintenance costs. The first two, while rare, are potentially of high consequence including loss of life. The types of solutions that can be employed to address cracking in ships are then presented. For existing vessels, these solutions range from repairs based on structural analysis or service experience, control of corrosion, and enhanced surveys. For new vessels, the use of advanced design procedures that specifically address dynamic loads and fatigue cracking is necessary. The efforts of the International Association of Classification Societies (IACS) in studying and addressing cracking-related effects in ships are also discussed. As the preferred solution to the problem of cracking in ships, this paper advocates prevention by explicit design consideration of relevant loads and failure modes, together with increased awareness and accounting of operation- and maintenance-related factors affecting structural durability.

INTRODUCTION

Oceangoing merchant ship structures are complex in geometry, detailing, fabrication, loading, operation, and inspection. To design such structures so as not to sustain cracking damage is, in principle, possible, but it is economically unfeasible. Ships are designed and operated to safety levels that society implicitly finds acceptable. To an engineer, these acceptable safety levels are a compromise between safety of life, property, and the environment and efficient use of resources; a tenet being that a certain level of risk is unavoidable in any human endeavor.

There is no one safety level that applies regardless of the type of vessel, and what society finds acceptable can change over time. Recently, for example, the safety levels for tanker structures have, in effect, been raised upward because of environmental concerns. Those for aging bulk carriers are in a similar state of change because of an increasing perception of unacceptability with the present state of affairs.

Ship structural design procedures and criteria over time have traditionally been experience driven. Experience, however, is relatively specific and tends to accumulate slowly. The use of advanced technology, calibrated by experience, is a better alternative.

It is against the above backdrop of structural complexity, changing societal demands, empiricism, and experience that the problem of cracking in ships should be seen.

BASIC CONCEPTS AND DEFINITIONS

This section defines the following:

- the "bath-tub" curve for expected failures over time
- safety-related failure modes for ships, viz., yielding, buckling, and fracture
- cracking, fracture, durability, and damage tolerance
- surveys and inspections

A ship, like any other structure subject to dynamic loads, should experience three distinct phases in terms of failure history: the teething phase, when failures from latent construction defects manifest themselves and during which the number of failures per year reduces with time; the stable phase, during which failures randomly occur, with the number of failures per year constant; and the aging phase, related to structural degradation (corrosion, fatigue, wear). During the aging phase, the number of failures per year increases with vessel age. The plot of number of failures per year versus age has the classic "bath-tub" shape. Needless to say, in a well-designed structure, failures typical of aging should occur past its economic life.

A ship must have adequate initial strength, durability, and damage tolerance. In addition, ships must be designed considering their impact on the environment. Strength-related failure modes in ships are three:

- yielding
- buckling
- fracture

Yielding implies material flow and, in some cases, separation of material. Buckling is due to instability of members under compression or shear and may occur together with yielding or without. In ship structures, some yielding is typically present with buckling since the members are not so slender. Whether yielding or buckling is significant depends on the extent and importance of the structural members affected.

Fracture is characterized by unstable tearing of material. In ships, fatigue cracks are the primary potential initiation points for fracture. Experience indicates fracture in ships to be relatively rare. This is primarily because of the good ductility of ship steels over typical loading rate and temperature ranges. Also, any tearing that occurs typically tends to arrest because of the presence of stiffeners, the presence of plating of relatively higher toughness in fracture critical areas of the vessel, or spatial changes in load (stress) levels.

Durability refers to the ability of a structure to remain in a form suitable for its intended function over its service life. Durability is a function of the structure as designed; the reserve strength present (over and above required minima); and systems and procedures designed to maintain structural condition in a satisfactory state, in particular those for corrosion protection, maintenance, and surveys. Fatigue and corrosion are the primary factors affecting durability. Any out-of-tolerance deviations in construction and changes in the structure because of accidental loads (e.g., dents) can also adversely affect durability. Durability characteristics are a function of time.

There is no crack-free, land or marine welded steel structure. Fatigue cracking refers to crack initiation due to damage accumulated from cyclic loading. This results in local cracks, which are primarily a maintenance concern. Service data seem to suggest that during the lifetime of a ship, an order of magnitude estimate for structural details expected to fail by fatigue is 1 in 100 (Jordan and Cochran, 1978).

Damage tolerance is the ability to sustain a certain level of damage, including cracking, without strength-related (catastrophic) failure. Damage tolerance can be characterized by the residual strength of the structure given a premised damage. In the presence of cracks, the demonstration of damage tolerance typically requires fatigue-crack growth and related residual-strength calculations.

In this paper, the term "surveys" refers to classification surveys. Inspection refers to inspections undertaken by others; in particular owners, operators, and flag administrations. This paper deals with causes, consequences, and control of local cracking in ships. Such cracking is primarily fatigue related and occurs during the aging phase, but not exclusively so. Cracks may also form, and material may tear, due to increased local stresses, alone, and such occurrences may be facilitated by scantling degradation due to corrosion.

CRACKING TRENDS IN SHIPS

What are the larger trends regarding cracking in tankers and bulk carriers? An early study by Newport News indicated that in terms of numbers, many cracks tend to occur at bracket connections, a fact still true today (Jordan and Cochran, 1978). The Newport News study, which covered several vessel types, indicated that nearly 33 percent of the over 6,800 damages it identified were in beam bracket connections, while 23 percent involved tripping brackets. These results are not surprising because the stress concentration factors associated with typical bracket connections are high. For similar reasons, another common location for cracking is a cut-out detail.

A more recent study of cracks in tankers was undertaken at Berkeley (Schulte-Strathaus, 1991). This particular study considered 10 ships (2 double hull, 2 double bottom, and 6 single hull; 4 of which were sister vessels). The database consisted of 3,600 cracks, of which about 2,000 were in the 4 sister ships. The Berkeley study indicated the following:

- Forty percent occurred in connections of side-shell longitudinals to transverse bulkheads or web frames.
- Ten percent were on bottom longitudinal-end connections.
- Ten percent were on horizontal stringers.
- Ten percent were on longitudinal bulkhead longitudinal-end connections.
- The rest occurred in various members, including 1 percent on deck longitudinal-end connections.

Figure 1 shows the crack distribution along the vessel length for all vessels in the Berkeley study. There is a tendency for more cracks to occur in the mid-body region; relatively less cracking occurs forward; apparently, even less occurs aft. Figure 1 also shows the crack distribution in the side shell, longitudinal bulkheads, and transverse members. Most side-shell cracks tend to occur in the middle third of the shell depth, while in transverse members, they tend to occur in the lower third.

The American Board of Shipping's (ABSs) experience indicates that tanker side-shell cracks tend to occur more at the connections at transverse bulkheads than at web frames and between the laden waterline and 8 m to 10 m below it.

In bulk carriers, ABS's experience indicates that cracking typically occurs in the areas indicated in Figure 2. The two most structurally significant cases of cracking are as follows:

- at bracket toe regions of hold frame connections to the upper- and lower-wing ballast tanks
- at transverse corrugated bulkhead intersections with topside tank structures

The resulting cracks can propagate fore and aft or athwartship, as the case may be. Other typical types of cracking in bulk carriers occur at:

- the intersection of inner bottom and hopper plating
- hatch corners and coamings

Bulk-carrier cracking can be exacerbated by grab-related wear, carriage of heavy cargoes, fatigue damage due to cyclical loading, and corrosion. Corrosion is important particularly with sulfur bearing coals. Corrosion, fatigue, and wear increase with vessel age, as may be expected. Cracking damage varies in significance, and we have listed some of the more significant cases above. In the case of tankers, the side-shell and bulkhead-bracket cracking noted are significant because of the potential for cargo leakage or mixing. In the case of bulk carriers, hold-frame end cracking is significant because it can result in the detachment of side shell from the internal framing—this too in an area of high shear stresses. Also, cracking at the transverse bulkhead top is significant because it can cause that bulkhead end to lose end support. The hatch corner and coaming cracks are significant in ore carriers because of the potential for water ingress. The significance of any given type of cracking can be determined by an analysis to determine the sequence of events that follow, for example, by an event tree methodology.

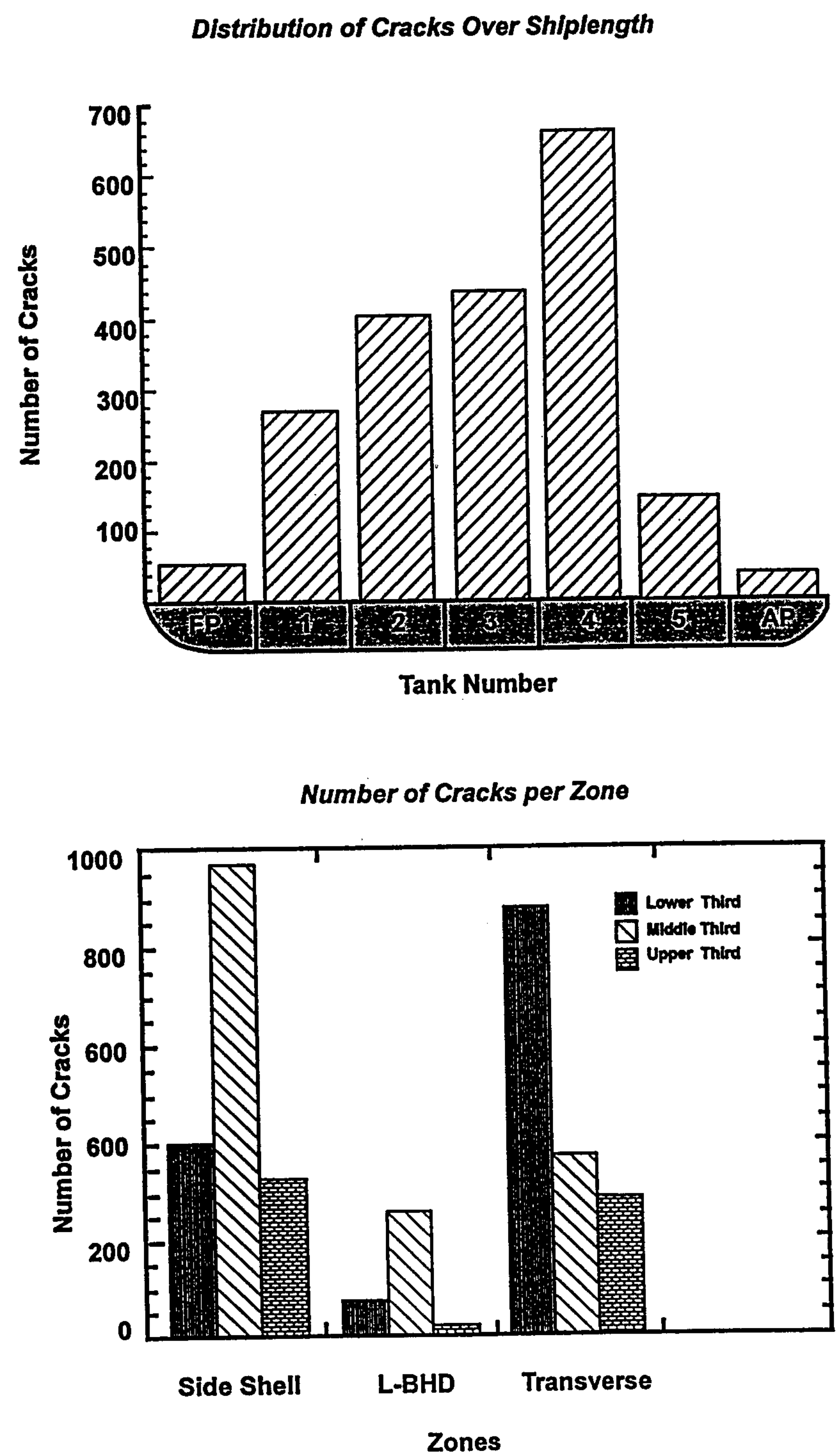
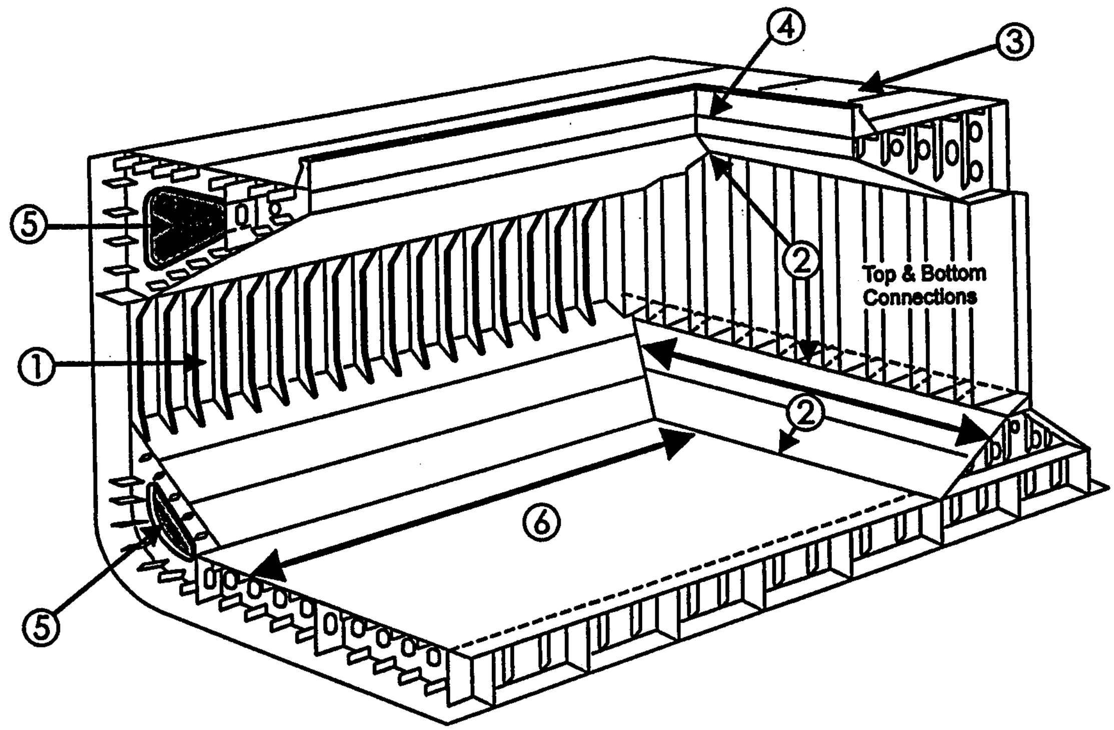


FIGURE 1 Crack distribution in tankers.
Source: Berkeley Study.



Problems Areas:

1. Hold Frame-Connection to upper & lower wing tanks and side shell.
2. Boundaries of transverse bulkheads and bulkhead stools.
3. Cross deck structure.
4. Hatch corners/hatch coaming brackets.
5. Localized cracking and buckling of web frames and breakdown of coatings in water ballast tanks.
6. Inner bottom plating/hopper plating intersection.

FIGURE 2 Bulk carrier problem areas—IACS.
Source: International Association of Classification Societies (IACS).

THE DOMINANT CAUSES OF LOCAL CRACKING

This section summarizes the lessons learned from several interesting analyses of cracking in tankers and bulk carriers, with a view to understanding why these damages occur. These studies did in some cases also include correlation with predictions of high stresses and low fatigue lives. The more important causes of cracking in oceangoing vessels are as follows:

- high local stresses
- extensive use of high tensile steels (HTS)
- uncertainties related to dynamic loads
- design sensitivity
- adverse operational factors (e.g., harsh weather, service routes, heavy cargoes, and vessel handling)
- structural degradation (e.g., corrosion, wear, and stevedore damage)

It will be appreciated that these causes are intertwined. In many cases, design features that by themselves were not a problem in the past may become marginal, given a combination of adverse factors and/or with age-related degradation.

High Local Stresses

Increased local stresses result in increased chances of material tearing and fatigue damage. The time to crack initiation in a welded structural detail is inversely proportional to approximately the cube of the stress range it is subject to. So the fatigue life of a structural detail subject to local stress fluctuations that are 10 percent higher than a basis detail is roughly 75 percent of the basis. Local stress-range increases above allowables can occur from any number of interrelated factors, including increased global stresses, barely acceptable structural detailing, increased stress-concentration factors, over-tolerance construction defects, and structural degradation due to corrosion and wear.

Extensive Use of High Tensile Steels

In the last 20 years, HTS has increasingly been used in ships. Such use provides a saving in initial costs because it leads to thinner material. The resulting reduction in steel weight can be significant, perhaps up to 15 percent. Unfortunately, the fatigue life of a welded structural detail of higher-strength steel is little different from that of a similar mild-steel structural detail. In a structure of H32 steel (Q = 0.78), the operating stresses are roughly 30 percent higher than mild steel for the same load, while the same figure is 40 percent for an H36 (Q = 0.72) structure. If the mild-steel structure is the comparison base, the fatigue life of an H32 structural detail is 50 percent of the mild-steel structure, and that of a H36 steel is 40 percent of the mild-steel structure (assuming that fatigue damage is proportional to the cube of stress). Clearly, such shortcomings can be designed out, provided fatigue is explicitly considered in ship structural design.

ABS is already on record with concerns about the somewhat related problem of light scantling short-life vessels (ABS, 1994c). Apart from the matter of reduced fatigue life, cracking due to overstressing is also more likely in HTS construction; simply because the stress levels are higher, and reserve margins (e.g., against overloads and corrosion) are lower. Also, higher-strength steels can be more difficult to weld and less tolerant of deviations from optimally established welding parameters.

Uncertainties Related to Dynamic Loads

Several studies of side-shell cracking in tankers conducted by ABS have indicated that on the regions of the side shell where cracks had occurred, the major contributor was fluctuating lateral pressure. Predicted fatigue damage was typically the largest, about 5 m below the laden waterline, consistent with actual damage experience.

On the side-shell regions of interest, one expects that there is a non-negligible hull girder horizontal-bending moment. However, the local stress accounting for stress concentration in bending under pressure is significantly larger than the local stress accounting for stress concentration under axial loading, and the fatigue damage is nearly entirely due to external pressure.

The types of loads involved (external hydrodynamic pressure and seaway-influenced or -driven internal cargo loads) had traditionally not been directly considered in design, and, until quite recently, the state of technology was such that it was not possible to consider such loads explicitly. Design changes, including the growth in vessel sizes and increasing use of HTS have evolved due to market economics and may not be supported by the state of knowledge at the time they occur. Further, there was nothing to indicate, until recently, that a closer look at loads was needed. This situation was due to two factors: the large reserve margins present in the older vessels; and the relatively young age of the newer vessels with more efficient structure. With time and age, experience with the newer vessels accumulated, (e.g., local cracking in areas where the fluctuating load to the total load ratio is high), and the recognition to more directly address dynamic loads in design grew.

In bulk carriers also, there is now recognition for the need to better address dynamic load components, both internal (ballast, heavy cargoes) and external, particularly in relation to side-shell and transverse-bulkhead structures. There is also now the growing recognition that structure in cargo holds should be designed for dynamic loads not only in the intact condition but for accidental flooding of the compartment.

Design Sensitivity

Certain design and construction factors that are normally adequate can, in some situations:

- contribute to cracking
- make the consequences of cracking worse

A classic example is the use of lapped brackets, which may be perfectly acceptable in some situations, but not others. Another example is the connection of the deck and the transverse corrugated bulkhead in a bulk carrier. Of particular interest is a design where a transverse box structure is not fitted, and the corrugated bulkhead is welded directly to the underside of the cross-deck plating. With heavy iron ore low in the hold, the cross-deck structure tends to deflect upward and away from the bulkhead, and wastage or other defect at the connection bulkhead upper connection detail can promote detachment of the deck from the bulkhead, particularly with thinner scantlings that make them more sensitive to corrosion and wear-related effects.

Certain potentially detrimental effects increase with lighter scantlings. Our studies have indicated, for example, that on the side shell of tanker structures, the choice of an unsymmetrical stiffener section leads to higher flange stresses because the stiffener tends to twist more under lateral pressure. This stress increase can be significant (in some cases, 50 percent more than the symmetrical flange case), and its effects on cracking are relatively more important in the thinner HTS construction, which tends to deflect more. For the same reasons, the relative deflections (and associated stresses) at the connection of longitudinals with transverse bulkheads is greater in HTS construction. These effects were undoubtedly always present, even in mild-steel construction, but to a less significant degree.

Harsh Service Routes/Cargo

In studies of the relative importance of various factors affecting cracking in ships, two factors that often arise are as follows:

- harsh trade routes
- certain types of cargo

Regarding harsh trade routes, comparative fatigue damage calculations indicate that certain vessel routes are in a relative sense more severe than an average North Atlantic wave environment that is the agreed basis for design for unrestricted worldwide service (Thayamballi, 1990). Hence vessels that operate exclusively in such areas may experience somewhat higher rates of fatigue-related cracking than those on other routes unless other corrective measures are taken.

As an illustration of the effects of a harsh trade route, we might note the study of U.S. flag tank vessels trading in the Trans-Alaska Pipeline Service (TAPS), carried out by the U.S. Coast Guard (USCG), which highlighted the above average incidence of cracking on that route (USCG, 1990). A subsequent ABS study indicated that not all such vessels were equally susceptible, which means that trade route alone is not the sole factor as may be expected (ABS, 1991). Another study also indicated that extreme hull girder loads to be expected on the TAPS trade were about the same as the design North Atlantic wave environment, even accounting for preferred vessel headings, but the fatigue damage was evidently different (ABS, 1989a).

In the case of bulk carriers, there is some evidence that a combination of heavy seas, harsh trade routes, the type of cargo carried, and other factors (such as corrosion and wear) may in some cases lead to an above average incidence of damage. Figure 3 shows known locations of bulk-ship failures worldwide from 1989 to 1992. This figure, fashioned after an Australian study, indicates the vessels to have suffered failure in areas of known harsh weather (BTCE, 1994). The same study also showed that bulk ships carrying heavy cargo, such as iron ore, are associated with a larger proportion of failures than would be expected based on their at-risk voyage exposure.

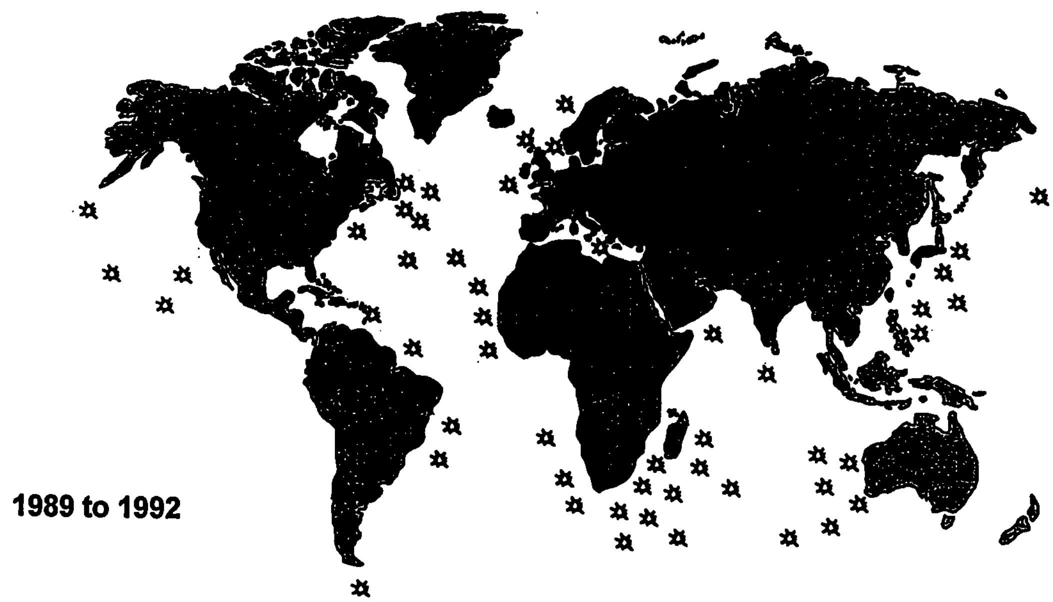


FIGURE 3 Locations of bulk carrier losses.

Structural Degradation

A common theme running through the majority of damages to bulk and combination carriers has been the wastage of members. This is predominantly in the upper and lower areas of the side shell and adjacent transverse structures within cargo holds of conventional bulk carriers and in ballast tanks of both combination and conventional bulk carriers. The wastage is due to two causes: corrosion and wear from routine vessel operations. The wastage can facilitate cracking and member separation, including by fatigue. Figure 4 shows such a member separation.

Corrosion is always a concern, but with the ballasting operations and particular types of cargoes, it is even more of a concern in bulk carriers. The frequent ballasting/deballasting cycles and the resulting tank humidity can accelerate the corrosion in topside tanks. The worst affected areas are the tank bottoms, which contain the important connections between the longitudinal framing of the ballast tanks and the transverse framing of the cargo hold. The level of corrosion may be masked by the mud and debris present. In the cargo holds, certain bulk cargoes, such as high-sulfur coal, can contribute to increased rates of corrosion. High-temperature cargoes, such as pelleted iron ore, can also accelerate corrosion by promoting condensation.

The structural wear due to routine vessel operations has recently been of interest. The effects of today's very high speeds of loading were essentially unanticipated in vessel design. In particular, grabs today may weigh as much as 30 tons, and their impact on comparatively light,

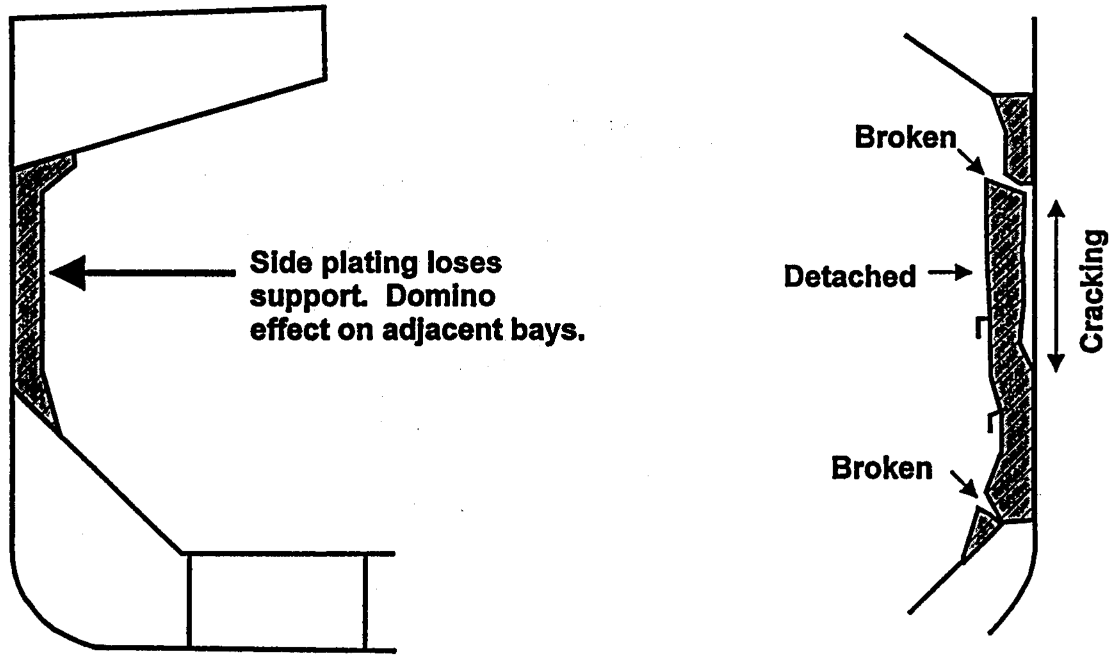


FIGURE 4 Bulk carrier side frame damage.

and possibly corroded, structure may be considerable. Wear can also be promoted by bulldozer blades and pneumatic hammers; the latter used to pound cargoes loose from the vessel. Also, in the relatively narrower forward aft holds, it is possible for the side structures to be damaged because they are within reach of cargo-handling equipment lowered through deck hatches.

In tankers, corrosion in ballast tanks was recognized to be a problem earlier on, which owners, operators, and classification societies actively addressed through efforts such as those of the Tanker Structure Cooperative Forum (TSCF) (TSCF, 1986). Today, segregated ballast tanks are required to be coated, thus reducing levels of general corrosion therein. What is of more concern in tankers today is cargo-tank pitting corrosion, particularly in the bottom (and other horizontal surfaces). Also worth noting are occasional instances of grooving (e.g., in way of side-shell longitudinals), corrosion in areas of residual stagnant water, localized corrosion in areas of high flow rate (such as bell mouths), and the "area ratio effect" in coated situations. In the last mentioned, when coatings start to break down, small exposed areas of steel become anodic to the rest of the coated steel and can be subject to accelerated corrosion. These effects are, again, relatively local.

CONSEQUENCES OF CRACKING

The potential larger consequences of cracking are as follows:

- structural failure, loss of vessel and cargo, and loss of lives
- pollution
- increased maintenance costs and downtime

These consequences vary in degree. For example, the events indicated in the first item above are of high consequence. Cases of significant pollution can also be similarly characterized. From a safety point of view, cracks that are primarily a maintenance concern are of relatively lower consequence.

Structural Failure

Although rare, cracks can lead to structural failure and vessel loss. Such nonaccidental structural failures are thought to be a factor in the recent spate of bulk carrier casualties. Spontaneous and substantial unstable fracture of the hull girder rarely takes place. The reasons for this are as follows:

- the relatively low incidence of substantial cracks in the hull envelope
- the good ductility (notch toughness) of ship steels
- the relatively low loading rates prevalent in the sea environment, even under slamming conditions
- the inherent crack-arrest properties of the hull

Where such substantial unstable fractures occur, these are often the result of plain human error (e.g., by misloading). A major newsworthy event on the TAPS trade, namely the grounding of the *Exxon Valdez*, was unrelated to structural causes.

What usually happens is that a sequence of progressive failure events leads to overall structural failure and vessel loss. In bulk carriers, for example, the following is thought to be one such possible sequence:

- Water ingress, because of a propagating hold-frame end crack and subsequent side-shell cracking; or water entry, because of hatchway damage.
- Collapse of transverse watertight bulkheads at the ends of a flooded cargo hold, and, hence progressive flooding and loss of buoyancy.
- Hull-girder breakup due to the resulting, unfavorable loads. Such breakup may be facilitated by structural degradation. Cargo liquefaction from seawater ingress, and resulting sloshing forces could also be a complicating factor.

The above events can occur with little sign of external damage, making them hard to detect early enough to prevent vessel loss. This is particularly the case if the forward hold is affected.

Casualty statistics published by the International Maritime Organization, Lloyds casualty reports, the Marine Index Bureau, the U.S. Coast Guard, and the Institution of London Underwriters all concur in the finding that over the past 20 years, there has been a declining rate of serious casualties of oceangoing vessels. However, the one exception to this general trend is aging bulk and combination carriers. A specific review of recent bulk carrier casualties may be found in an ABS paper (Grove, 1992). While it is unclear whether this is a general trend or a short-term anomaly, it nevertheless is a cause for concern.

Pollution

Oil spills are an emotionally charged societal issue. Data on such spills and the costs related to them are subject to interpretation and, in some cases, speculative.

It has been estimated that, worldwide, tanker accidents contribute less than 10 percent of the nearly 2,400,000 tons of oil that enter the world's seas every year from all sources, including natural leakage, land-based releases, and offshore activities. The world maritime operational and accidental losses of oil have actually been on the decrease for some time, as noted in a National Research Council (NRC) study (NRC, 1991), because of better vessel design (e.g., limitation in cargo tank size and protective location of ballast tanks) and improved operational procedures, including clean ballasting. At the same time, society's reluctance to tolerate pollution incidents has increased.

The amount of oil spilled from 50 major spills of tankers from 1960 to 1990 was studied in the NRC study (NRC, 1991). The volume varied from year to year and was quite unpredictable as to cause and occurrence. Based on the same study, Figure 5 shows (for tankers of size greater than 10,000 DWT and spills greater than 30 tons) the alleged causes of oil loss worldwide. While apparently roughly equal volumes of oil are lost from groundings, collisions, and fire and explosions, that lost from "structural/other" causes is significantly lower. It is also worth emphasizing that the figure pertains to a subset of tanker casualties, in fact a very small percent of the total.

Instances of significant pollution, when they occur, can result in considerable near-term harm to the environment, depending on where and when they occurred. Costs related to the damage fall under many categories: cleanup expenses, restoration costs, and lost use and non-use values. The third category includes intrinsic values, such as the depletion of sea life. Cleanup costs are typically large; the highest to date being the *Exxon Valdez* (reportedly in excess of $2 billion).

Legal claim payments are one way to judge the total cost of a spill and are said to be typically about $30,000 per ton of oil spilled but can be as large as $100,000 per ton (NRC, 1991). In any event, the costs of pollution incidents are significant.

U.K. Protection and Indemnity Club data also confirm that pollution is one of the more expensive incidents involving claims (U.K. Club, 1993). Of their major pollution claims, over 100 involved a total of $132 million (i.e., an average claim amount of $1 million each). About 12 percent each of the total number of pollution claims was caused by bunkering, collision, valve, and shell-plate failure. Tankers accounted for about one-half the number and two-thirds of the total value of claims.

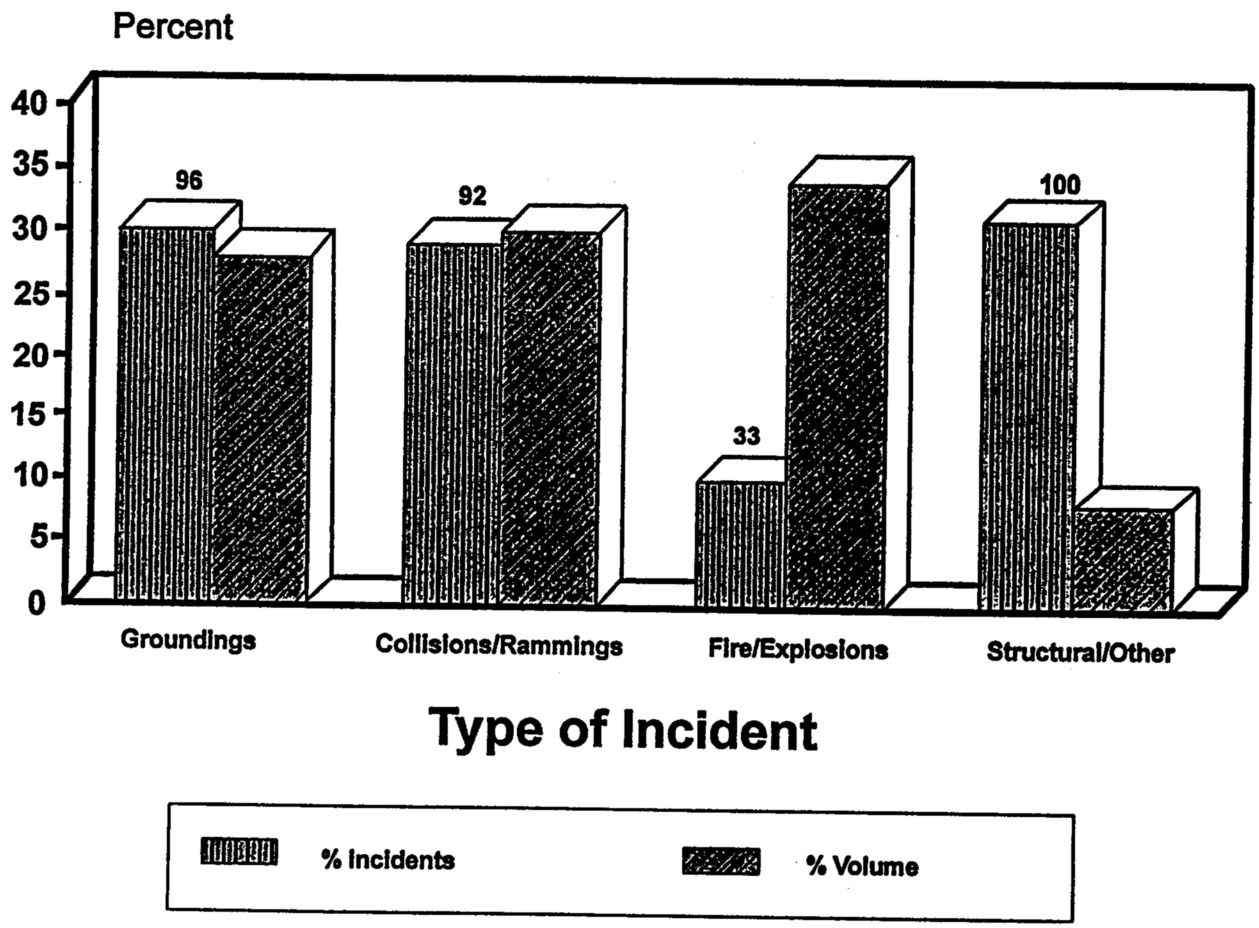


FIGURE 5 Causes of significant spills worldwide.
Source: NRC, 1991.

Increased Maintenance Costs and Downtime

Most steel renewals occur because of corrosion, and any cracks are typically addressed by local repairs. Data on maintenance costs and downtime are difficult to provide and depend on the circumstances. We will illustrate some of these costs, with the disclaimer that what follows is an incomplete list. In particular, we have omitted the cost of downtime, which is substantial.

The most obvious question is how much steel is typically renewed on the average? There are data related to this in the work of the TSCF (TSCF, 1986). The relevant numbers (tons/ship/20-year life) are approximately as follows: cargo tanks—50 tonnes uncoated and 25 tonnes partially coated; ballast tanks—350 tonnes uncoated, 210 tonnes coated, and 130 tonnes partially coated (epoxy and anodes). As expected, the cargo tank renewals are much lower than ballast tank renewals. The type of corrosion protection system employed obviously makes a difference. In coated situations, renewals in the first 5 to 10 years are low. No similar data exist for bulk carriers.

What does it cost to renew steel? There are many possible components to that cost, as illustrated below:

- Dry Dock Charges—For vessels above 150,000 GRT, the minimum charge for the first 2 days is about \$0.5 per GRT. The charge for each subsequent day is about \$0.2 per GRT.
- Tank Cleaning—Ranges from \$2 to \$12 per metric ton capacity, depending on type and location of tank, gas freeing and ventilation excluded.
- Steel Renewal—For mild steel, about \$4,000 to \$5,000 per tonne of steel renewed.
- Staging—About \$5 per m^3 of volume covered.

These illustrative costs are from a yard in the Far East, and costs do vary between yards.

Effect of Vessel Age

Recent insurance claims experience from a protection and indemnity (P&I) club is reviewed in this section to illustrate the effect of vessel age. The data come from the U.K. P&I Club, which insures roughly one-fourth of the world's oceangoing tonnage and publishes extensive analyses of their claims experience (U.K. Club, 1993). The 8,000 ships to which the club extends coverage are said to be representative of the world fleet in general, both in terms of tonnage and types of ships.

Because of the increasing concern with bulk carriers, the U.K. P&I Club report cited analyses-related claim experience in detail. Bulk carriers accounted for about 16 percent of the club profile and 30 percent of the structural claims; while in the case of tankers and cargo ships, the claims experience was more consistent with the vessel types' proportion of the total. Space limitations preclude a detailed review, but vessel age was an important factor in all types of claims. This is illustrated in Figure 6 relating to structural claims, with ships 15 to 20 years old contributing the most number of claims. The club data analysis also showed a correlation between human-error–related claims and vessel age, between cargo claims caused by hatch-cover failure and the age of the ship, and between vessel age and pollution claims.

Such experience is consistent with ABS's own analysis (Grove et al., 1992). Age is arguably not the direct cause—day-to-day incidents occur in ships of all ages. But apparently, the probability that a minor situation turns into a major one (of larger consequence) is greater for the older vessels.

Aging vessels are a reality. In fact, about 60 percent of the world tanker fleet and 40 percent of the bulk carrier tonnage are over 15 years old. This underscores the importance of proper vessel maintenance, operation, and inspection and also the specific consideration of durability-related factors in vessel design. These are primarily the responsibility of the owner, supported by the professionalism and innovation of the classification societies involved.

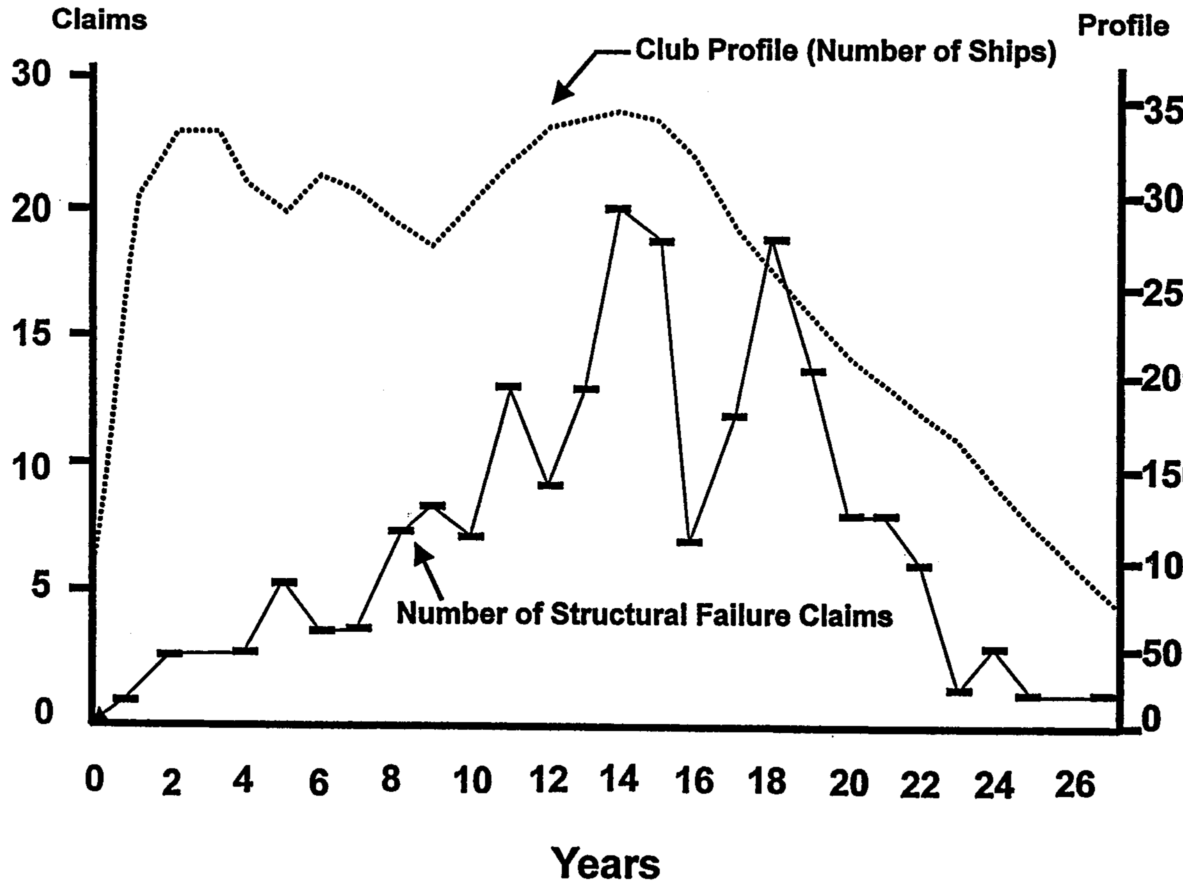


FIGURE 6 Age and structural claims.
Source: U.K. Protection and Indemnity Club.

Role of Human Error

Direct human error predominates in 60 percent of the major claims analyzed by the U.K. P&I Club (U.K. Club, 1993). Structural and shell-plate failures and claims due to sinking are also significant, with structural failures figuring in about 12 percent of the major claims. Some of these again, presumably indirectly, involve the human element (e.g., through lack of maintenance or error of judgment on the part of an otherwise qualified individual).

Human error can broadly be defined as a departure from acceptable or desirable practice, on the part of individuals on their own or as part of a team, that results in an unacceptable or undesirable event (Bea, 1994). Such errors occur due to many reasons: cognitive mistakes; ignorance and unawareness; intentional violations; accidental lapses; lack of planning, preparation, and procedures; lack of training; limitations and impairment, such as fatigue; increased stress, discomfort; and temperamental factors, such as boredom, anger, overconfidence, recklessness, and pride. The U.K. P&I analysis gives some interesting examples: pride in not asking for help in a complex situation leading to both property damage and personal injury; potential for misunderstandings and miscommunication with mixed nationality crews; and fatigue, with smaller crew sizes and short turnaround times in harbor. The control of human error is important to reduce the causes and contain the consequences of marine incidents including those arising from cracking.

TRADITIONAL CONTROL OF CRACKING

Fatigue crack initiation and fracture are facilitated by a combination of adverse factors: presence of a significant surface or imbedded flaw; high tensile stresses; low temperature; high loading rates; and over-tolerance construction defects, including misalignments. In this section we discuss the control of fatigue and fracture within the framework of traditional classification rules.

The following traditional design, construction, operation and maintenance, and fracture-control considerations are briefly outlined:

- explicit control of fatigue—allowable stresses, types of loads, and loading manuals
- improved design of structural details
- limitations on construction tolerances
- weld acceptance criteria
- control of corrosion by design margins, inspections, and surveys
- selecting steels of adequate ductility (fracture control)
- selective deployment of higher-ductility steel (fracture control)

It is worth noting that the following considerations, which we do not discuss, are either implicit or not within scope of classification:

- The vessel has adequate reserve and residual strength.
- The vessel condition is adequately maintained by its operator.
- The vessel is correctly loaded and operated by trained personnel under proper supervision and monitoring.

Implicit Control of Fatigue

In traditional classification rules, members are sized for local pressure heads, and the hull girder is required to have a section modulus adequate to withstand still water and wave-induced bending moments and shear forces. This is essentially an approach wherein stress limitations are used to control yielding, buckling, and fatigue implicitly. Loading manuals were required in some cases. On balance, this experience-based approach has worked reasonably well.

In the last 20 years, the design of tankers has moved toward more explicit consideration of failure modes. Buckling assessments became common. Fatigue calculations were still not the norm but were made in certain cases. Global and local stresses were increasingly calculated using finite element analysis. These changes in tanker structural design came about because of a recognition that vessel sizes far exceeded the rule experience base until then. Also, operating stress levels continued to increase because of efficiencies gained through advances in knowledge and the use of steels of higher-yield strength to save on initial costs. Bulk carriers, however, continued to be designed the traditional way, simply because the rule experience base (apparently) covered them adequately.

Improved Design of Structural Details

The design of structural details to reduce stress concentrations, provide smooth transitions, and reduce local bending, for example, by providing adequate secondary (e.g., back) support, has always been a primary defense against fatigue cracking. Special attention is typically paid to the termination of structural members.

As examples it may be recalled that shell fractures in tankers have tended to occur at the ends of bracket toes and sniped ends, particularly where the member stops too far from the adjacent members or where the ending is not soft (i.e., snipe angle is large).

Guidance on detail design may be found in documents from the TSCF and IACS (TSCF, 1986; IACS, 1994). The guidelines in these documents, useful in both design and repair contexts, are experience based.

Construction Tolerances and Weld Acceptance Criteria

Tolerances attainable in the construction of a vessel can depend on the country and the yard where the vessel is built. Major yards, in fact, have their own specific fabrication standards, including construction tolerances (Basar, 1978). These standards are sometimes not viewed strictly as limits pertaining to safety, but there is an implicit assumption in ship design procedures that the vessel is built so that deviations from nominal values are within tolerances. If not, additional local stresses can result, with particular relevance to fatigue cracking.

Except for plate rolling tolerances, classification societies do not always mandate specific construction tolerances for ships, but their surveyors do use certain guidelines for judging structural imperfections in new construction. These relate to the following (ABS, 1992):

- dimensions of built-up sections
- misalignments in butt- and fillet-welded connections
- weld undercuts
- plate and stiffener unfairness and out of straightness

Undersized, fractured, or missing welding is normally unacceptable. Construction defects are potential sources of cracking; hence, surveys and inspections during construction are important. Such surveys include those at the shipyard and at steel mills and manufacturers. Surveys during construction are not an alternative to the shipyard's quality assurance. The rules do specify attendance of the surveyor at certain manufacturing and construction activities, as well as at tank testing. However, the overall responsibility for a well-built vessel belongs to the builder.

Control of Corrosion

Corrosion control is important to the ability of the structure to maintain adequate strength throughout its economic life. Traditionally, a margin for corrosion had always been built into the classification rules for scantlings. More recently, a "net ship" approach has been used for member sizing, with an explicit corrosion margin added to obtain the "as designed" structure (ABS, 1994). This makes possible the direct treatment of sensitivity to corrosion.

In addition to the design corrosion margin, other important aspects of corrosion control in ships are as follows:

- protection systems (coatings, anodes, and impressed-current cathodic protection are the techniques commonly employed)
- surveys and inspections

For certain types of vessels where corrosion was a significant problem, classification requirements have tended to emphasize coatings as the solution. For example, ballast tanks in oceangoing vessels are now required to be hard coated.

Classification surveys after construction are of three types: annual, intermediate, and special surveys. At annual surveys, the hull is examined for general condition. In special surveys, which included requirement for dry docking and essentially occur once every five years, parts of the hull in which the classification society has an interest are examined in more detail, using close-up surveys and gaugings, and a recommendation is made as to the necessary work to enable the vessel to operate for another five years in apparent good order. Intermediate surveys are in addition to annual surveys and occur between special surveys to look at the general condition of ballast tanks and cargo holds.

Other than classification surveys, owners and operators should (and do), as part of their preventive maintenance programs, undertake their own rigorous inspections to detect and correct corrosion and fatigue effects on an ongoing basis.

Material Selection and Deployment

The ferritic types of steels used in ships undergo a change in behavior from ductile to brittle depending on the loading rate and temperature. Metallurgical processing cannot change such behavior but can change the loading rate or temperature at which such ductile to brittle transition occurs.

The toughness of a steel is a measure of its relative brittleness. ABS was the first classification society to recognize the importance of toughness, which was introduced into the rules in the 1940s. Prescriptive steel chemistry and certain metallurgical processing can help obtain a specific level of toughness. Charpy toughness testing can be used to evaluate the same.

Steel grades used in ships are A, B, D, E, DS, and CS for mild steels; and AH, DH, and EH for the higher-strength steels. All grades except A, CS, and DS require Charpy testing in some form. Grade A steel costs roughly $500 per ton. The approximate relative costs of these various steel grades are as follows: A (0.80), B (0.80), D (0.85), DN (1.05), E (1.05), CS (1.00), AH (0.90), DH (1.05), EH (1.10). These costs are normalized to grade CS, which is a heat-treated, high-toughness steel that does not require production Charpy testing. Costs include those for any heat treatment and production testing required by the rules (ABS, 1992).

With regard to toughness, steel is generally selected depending on the expected service temperatures; the more ductile steels being selected for the lower-temperature applications. In addition, some parts of the vessel structure (e.g., the sheer strake) employ a relatively tougher grade of steel. These are sometimes called "crack arrester strakes."

CONTROL OF CRACKING OF VESSELS IN SERVICE

In this section, we review approaches that may be taken to assess/reduce the incidence of cracking. These are as follows:

- structural analysis (stress, fatigue)
- fitness-for-service assessments
- repair/redesign of local details
- reduction and monitoring of loading
- control of corrosion and wear
- fatigue life improvement techniques
- enhanced inspections and surveys
- emergency management systems
- awareness of human factors

These are now briefly described.

Structural Analysis

Stress and fatigue analyses are typically undertaken to identify the possible causes and solutions to cracking in ships. The analysis procedures are fairly standard and are similar to those undertaken for new designs; the largest unknown for existing ships being the loading (past and future).

Stress analyses attempt to predict stresses in the structural areas of interest. Apart from the obvious application of determining stresses and deflections, the analyses are also useful to optimize local detail design, decide appropriate placement or relocation of large openings, and indicate the adequacy of local structural reinforcement.

Fatigue analysis calculates the damage accumulated within a given period of time. Fatigue analyses may be simplified or more direct, the latter being the spectral fatigue approach that is part of the ABS dynamic load approach (DLA) system (ABS, 1989). This involves an environmental description by a wave scatter diagram; determining the local stress range transfer functions, using ship motion and spectral fatigue analyses; determining the stress range histogram; and calculating the fatigue damage on the basis of S–N data. In a calibrated, simplified approach, such as that of ABS SafeHull (ABS, 1994), the stress range histogram is built up by determining the extreme stress range possible in the time period under consideration and using this together with predefined information on the possible shape of that histogram.

One important aspect of finite-element–based analysis is development of a structural model or models of a large portion of the structure. The modeling time is substantial and could conceivably be a factor in the scheduling of repairs. The following are typical options when that is likely to happen:

- Dispense with the analysis. Use experience-based repairs.
- Use a simpler analysis procedure (e.g., one that uses a beam model for stresses instead of a finite element model).
- Have the finite element models prepared in advance. Other time consuming aspects (e.g., load computations) could also be so handled.
- Use automated modeling tools, such as those in ABS SafeHull.

Fitness-for–Service Assessments

A fitness-for–service assessment uses stress analysis (e.g., the finite-element method), fatigue-crack–growth modeling, and fracture-mechanics calculations. The typical aim of such analyses is to show that there is no danger of crack extension or unstable fracture during a certain period of time; (for example, a voyage duration, time to reach a repair yard, or time until next scheduled dry docking. The decision to repair or not is, however, a function of other considerations, including what regulatory authorities require, possible threat to the environment, or potential liability.

A similar approach could also be applied to assessing the need for rectification of out-of-tolerance defects in newbuilds, reassessment of nondestructive testing acceptance criteria in specific cases, judging the effectiveness of specific repairs, and developing inspection plans.

The fitness-for-service analysis for a ship with a fracture typically consists of the following:

- Determining the Loads and Stresses—The accurate prediction of stresses is a necessity but often needs to be conservatively made because of the uncertainties involved.
- Estimating the Material Toughness—Typically, because no specific testing is possible, this is conservatively done (e.g., using assumptions of plane strain, and a lower-bound Charpy, energy-based, fracture-toughness–correlation procedure).
- Determining the critical flaw size.
- Estimating Sub-critical Flaw Growth—Again, because of lack of specific data, upper bound crack-growth–rate curves extrapolated to the threshold may be used for this purpose.

Established conservative procedures such as BSI PD 6493, Level 1, are typically used. The necessary stress-intensity factor calculations, used conservatively, resolve idealized crack situations (BSI, 1991).

It is fair to say that fitness-for-service assessments are not fully accepted in the marine sector and that even where an assessment is conservatively made authorities and owners alike would yet require that fractures be repaired—even for short passages.

Repair or Redesign of Local Details

The repair or redesign decision is based on input from:

- insights gained from inspection of the fractures
- structural and fatigue analyses (if performed)
- past experience (the predominant approach)
- repair management systems

The first and second items are self-explanatory. Hence, this subsection will deal primarily with the third and the fourth.

The Experience-Based Approach

Cracks, when found, typically are repaired. Small, single, or relatively straight cracks in plating may frequently be repaired satisfactorily by welding, but after locating the end of crack and veeing out as required. No attempt should be made to weld up multiple-branching or star-shaped fractures indicative of fatigue. Such defective areas should be renewed, strengthened, or modified as appropriate.

With regard to structural details, where cracks are typically experienced, and possible repairs for them, reference is made to the TSCF guidance document for tankers and a similar IACS document for bulk carriers (TSCF, 1986; IACS, 1994). The TSCF and IACS references contain the following types of information:

- sketches of the failure and proposed repair
- factors contributing to damage
- repair alternatives and unsuccessful repairs

An example relating to the repair of cracking at the connection of longitudinals to transverse webs in tankers is shown in Figure 7. Finite element studies of similar situations have indicated an interesting fact—brackets backing the flat-bar stiffeners on web frames tend to help because they reduce their out-of-plane deformation (and related stresses). We might also state the obvious, that softening of the bracket toes helps. Repeated occurrences of cracking in the same area may indicate a need for structural strengthening or modifications. Options for such modifications include provision of soft toe brackets, providing backing brackets on the other side

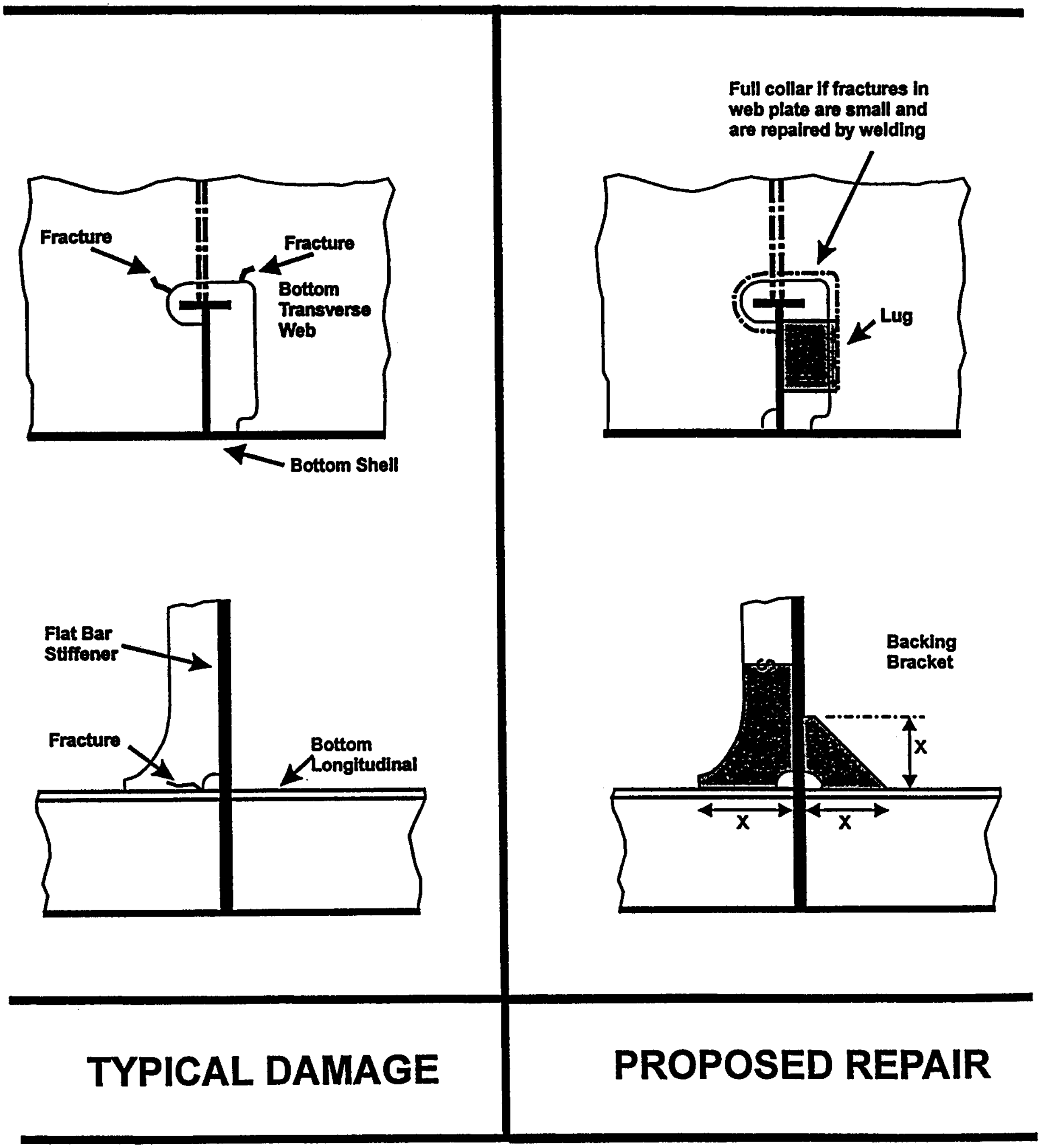


FIGURE 7 Tanker damage and repair example.
Source: Tanker Structure Cooperative Forum (TSCF)

of plating, and increasing local member thicknesses with a view to reducing stress levels. One can also increase local fatigue strength by measures such as weld-toe grinding. There is typically no one-effective–repair strategy to suit all situations, the best approach being to specifically address fatigue and local stress effects at the design stage.

Repair Management Systems

The main component of a repair management system is a purpose-written computer program that facilitates the identification of the best repair option in a short period of time. A repair management system may contain the following types of capabilities for comparing and evaluating repair options rapidly:

- graphical representations of possible repair options
- finite-element models of possible repair options
- simplified fatigue-life calculations
- economic trade-off analysis
- a repair-record database

A first generation repair management system for "critical structural details" in ships has been developed by Ma and Bea (1994). An example of the types of repair options for which graphical representations and finite element models exist in that system is shown in Figure 8. Purpose-written repair management systems can facilitate rapid comparison of alternatives, especially where finite element models are prepared and stored in advance. These systems are typically designed to provide comparative rather than absolute measures of stresses or fatigue lives.

Control of Corrosion and Wear

Corrosion and wear are significant compounding factors in the incidence of cracking. Corrosion-control measures, such as the use of coatings, thus help indirectly to reduce the occurrence of cracks or other material separation.

As previously discussed, corrosion and wear are factors in bulk-carrier holdframe end cracking. Corrosion can also be a problem with the corrugated bulkheads in bulk-carrier cargo holds, where its uniform nature often makes it hard to detect visually. Because of such experience, the holdframes and corrugated transverse bulkheads of newbuild bulk carriers are now coated.

Coatings are expensive and can account for 10 percent to 20 percent of the cost of a vessel in some cases (e.g., double-hull tankers).

Reduction and Monitoring of Loading

The reduction of wave loads experienced by the vessel structure will typically improve fatigue performance since fatigue lives are proportional to the cube of the stress range (at least).

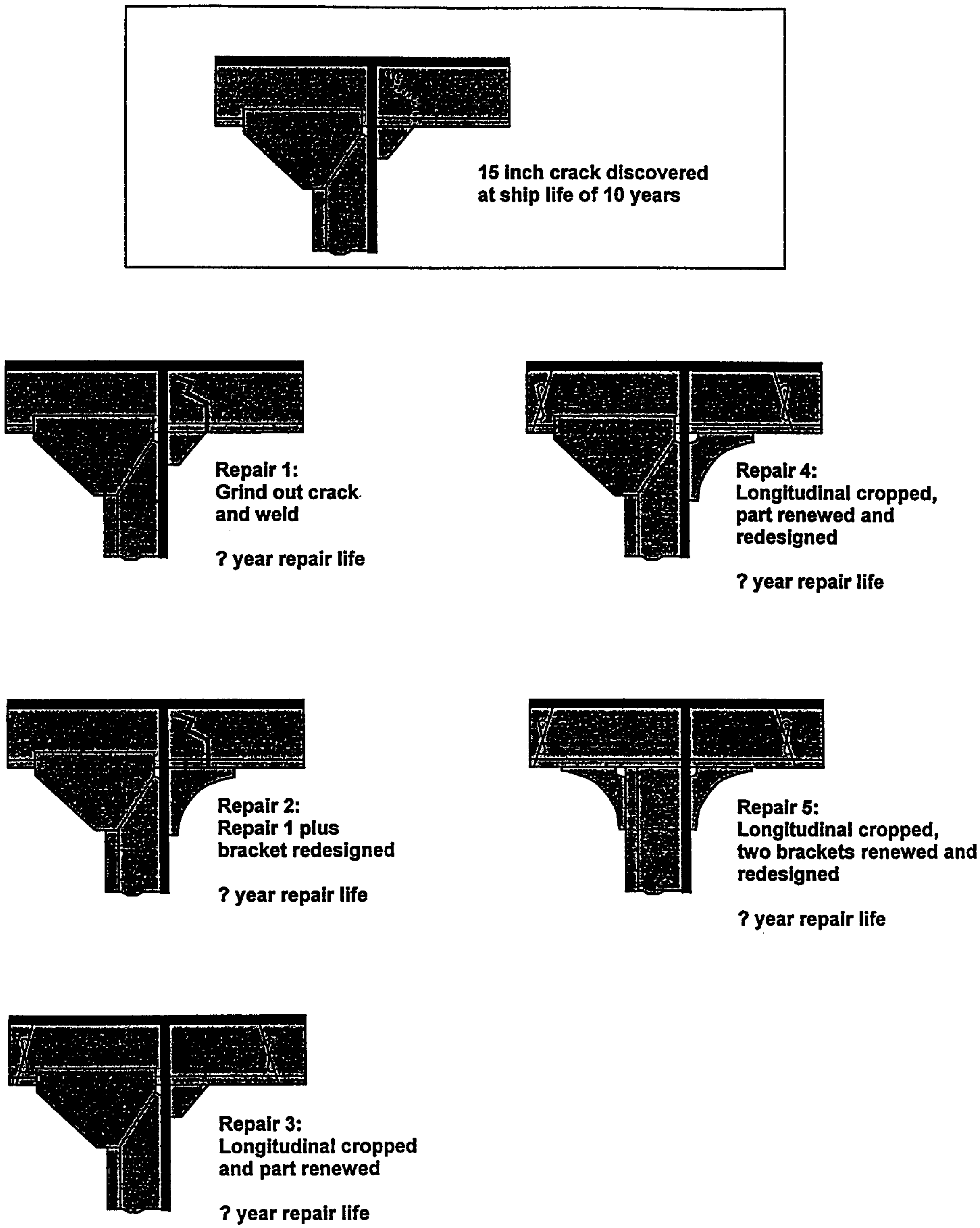


FIGURE 8 Repair management system.
Source: Ma and Bea, 1994.

Such load reduction may be possible in specific cases by voyage planning, changes in vessel route, speed and heading limitations, changes in tank fill heights, or (the rare) reduction of vessel length.

The reduction of still-water loads producing tensile mean stresses can help in containing the possible consequences of cracking and reducing the chances of unstable crack propagation. Loading manuals and loading computers help in this regard.

Hull and hold load-and-stress monitoring systems can provide valuable real-time checks on the load effects being experienced by a vessel and support proper vessel loading and handling, whether at a terminal or at sea.

Fatigue Life Improvement Techniques

These fall in two categories:

- those that alter local geometry of the weld
- those that alter the residual stress field at the weld

In the first category are the mechanical and argon arc treatments of the weld. The second category includes methods for relieving residual stresses or introducing favorable (compressive) residual stresses. Here, we briefly discuss these two classes of methods, but only the first class of methods is a reliable option in ships, and even then only in local areas in specific cases.

Alterations to Local Weld Geometry

The aim is to create a smoother transition between the weld and parent metal. Weld toe grinding using abrasive disks or mill cutters is an example. The best results are achievable for butt welds, where the overfill can be completely ground out or the transition zone given a large enough radius (e.g., 5 mm) and smoothened depth. The fatigue limit is then essentially the same as that of the parent metal. In other joint types, the fatigue limit after treatment may be increased anywhere from 5 percent to 90 percent—depending on the type of joint and treatment—which is a large range.

In the argon arc method, the transition between weld and parent metal is smoothed by fusion of the weld boundaries by a nonconsumable electrode. Good results can be obtained in many cases, depending on the type of electrode and its diameter, but it is also possible that tensile residual stresses are increased in other cases.

Alterations to the Residual Stress Field

These treatments aim to either reduce the level of tensile residual stresses present after welding or induce the more beneficial compressive residual stresses. In welded details, tensile residual stresses of high magnitude in effect make both tensile and compressive parts of the load cycle damaging in the context of fatigue. It is mainly for this reason that S–N data for welded construction typically neglect the effect of mean stresses.

Thermal treatment by tempering is one effective way of relieving tensile residual stresses, but the treatment is usually possible only for small components. Other ways are: static overloading, vibration, spot and local heating, surface cold working (e.g., by shot blasting), explosion treatment, and local peening. For a discussion of these, see Gurney (1979), for example. Ultrasonic impact peening is being evaluated for use in ship structures (Trufyakov et al., 1993).

Enhanced Surveys and Inspections

In accordance with International Maritime Organization Resolution A. 713, enhanced survey requirements for tankers and bulk carriers were implemented in the rules in 1993. These emphasized and expanded the then-existing survey requirements for annual, intermediate, and special surveys as a function of vessel age.

Now, the condition of coatings is graded and recorded. The survey program also calls for identification of so-called suspect areas that show substantial corrosion or rapid wastage. If not remedied, such areas are subject to special attention during the subsequent survey. Certain documentation, such as summary of hull condition, may be retained on board the vessel.

The enhanced survey program has a provision for a survey planning document, prepared by the owner, well in advance of a special survey. The purpose of the document is to identify critical structural areas and to stipulate the minimum extent, locations, means, and access arrangements for close-up survey and gaugings of sections and internal structures, as well as to nominate suspect areas consistent with rule requirements. The stated bases for nomination include (Horn and Johnson, 1994):

- design features, such as extent of HTS and local details item former history with respect to cracking, buckling, indents, and repairs for the vessel or similar vessels
- information on cargoes carried, tank usage for ballast, protection of tanks, and condition of coatings, as applicable

Survey planning, via a planning document, can help focus inspections and surveys based on general and particular experience. In a sense, it is similar to the critical area inspection plans required by the U.S. Coast Guard in certain cases, but broader in scope (USCG, 1991).

Emergency Management Systems

These provide manpower and technical assistance to understand and contain the consequences of an accident, whether fracture related or not. An example is the ABS rapid response damage assessment (RRDA) program, which accommodates around-the-clock multiple responses for enrolled vessels, using naval architectural and salvage analysis software (ABS, 1994b). This particular program offers hull girder stress and bending moment calculations and residual stability assessments in response to an emergency situation as standard. Other types of required information could also be provided (e.g., local stresses using previously developed and stored structural models).

Two other emergency-related programs, emphasizing procedures and training, are oil-spill management exercises and the requirement of the International Maritime Organization convention on marine pollution (MARPOL) for a shipboard oil pollution emergency plan (SOPEP), both offered by ABS Marine Services.

Awareness of Human Factors

Awareness of human factors is important in limiting the undesirable consequences of cracking previously discussed. Human error is inherent to any system and cannot be eliminated. It has to be recognized and controlled by management of potential causes and consequences. The three main ways of accomplishing this in practice are as follows:

- personnel selection and training
- implementation of monitoring procedures
- design of durable and damage tolerant systems, with adequate reserve and residual strength

The International Management Code for the Safe Operation of Ships and for Pollution Prevention, commonly referred to as the International Safety Management (ISM) Code, is a management system standard designed to ensure the safety of life, property, and the environment (see ABS, 1994a). The code essentially interprets and applies ISO 9002 quality system requirements to particular cases of marine management and vessel operation. The ISM Code addresses personnel responsibility; management system structure; ship and cargo status identification; procedures for deck, cargo, and engine room operations; procedures for navigation, safety, emergency preparedness, maintenance, communications, pollution prevention; and the line. The scope does not include design or design control.

An amendment to Chapter IX of the Safety of Life at Sea (SOLAS) Convention now mandates the ISM Code by 1998 for all passenger ships, tankers, chemical carriers, gas carriers, and bulk carriers over 500 GRT. The requirements are to extend to cargo ships over 500 GRT by the year 2002. Certification is required to demonstrate that the owner or operator of the ship is in compliance with the ISM Code. Such compliance, demonstrated by the company's adherence to a safety management system, is evidenced by a document of compliance for the company and a safety management certificate for each ship.

LIFE CYCLE CONTROL THROUGH ADVANCED TECHNOLOGY

An integrated approach to design, construction, maintenance, operation and inspection of the vessel over its life cycle is the goal. In such an approach (Jan, 1993):

- The structure itself should be designed using advanced technology, considering various failure modes including fatigue. There should be an appropriate accounting of dynamic loads.
- Design safety margins should duly reflect the consequences of failure and, in some cases, may need to increase.
- Durability should be explicitly addressed in design. There needs to be awareness and appropriate accounting of construction, maintenance, and operational factors that affect structural adequacy, including residual strength.
- Maintenance of structural condition would be effectively monitored by ongoing in-service inspections and planned surveys.
- Necessary information databases will be established.

Service experience has repeatedly indicated that incidents of cracking in service are sometimes eliminated and sometimes not (i.e., they can recur). Structural defects may exist, but the incidence of cracking can be reduced and monitored and consequences can be contained. To this end, we now discuss:

- advanced design and condition assessment systems
- damage tolerant design
- use of probabilistic methods
- structural management systems

Design and Condition Assessment Systems

We now review the following two types of computer-assisted approaches (procedures, criteria, software) for design and condition assessment:

- direct analysis (including spectral fatigue analysis)
- design-oriented first principles procedures

The consideration of dynamic loads, durability, and reserve strength by such systems is discussed. Investigations during the development of the ABS SafeHull design system have revealed that, in order to obtain a comparable degree of durability and safety as with tankers, certain structural areas and conditions specific to bulk carriers warranted explicit consideration and possibly increased safety margins. These are also briefly reviewed.

The Direct Analysis Approach

In this approach, structural and spectral fatigue analyses are performed using directly calculated dynamic and static loads unique to the specific vessel. For vessels for unrestricted service, the North Atlantic wave environment is commonly used. The structure can be sized initially by standard rule procedures. The seaway induced loads are calculated using ship motion analysis, including internal structural and cargo inertial loads and external hydrodynamic pressures. Special loads, such as those due to tank sloshing, are also obtained by direct calculation. Stresses in the "as built" structure are then determined by finite element analysis (using a global model of the entire vessel, followed by local models) and checked against allowables for yielding, buckling, and fatigue.

The Dynamic Load Approach (DLA), introduced in 1991, is a good example of an integrated software system for direct analysis. DLA is essentially a formalization and enhancement of in-house direct calculation methods long used by ABS (Liu et al., 1992). An emphasis in DLA is on the direct calculation of dynamic loads, reflecting their importance and relatively large uncertainty. Vessels checked using DLA can have scantlings greater than rule requirements, but not less. Thus the procedure provides an extra margin of safety, potentially contributing to a stronger, robust, and more durable vessel.

Design Oriented First Principles Procedures

Specific direct analysis using first principles is suitable for the assessment of ship structures after the design is developed. However, such procedures need simplification and streamlining and a very high level of software support before they can be used as true structural design systems. Such simplification has its benefits and drawbacks. For example, a direct analysis approach can be utilized to evaluate any type of ship of any structural configuration, including novel ones that fall outside today's experience base. A design oriented first principles engineering system, on the other hand, is purpose-built for typical structural configurations of a given type of vessel for which the analysis requirements, including finite-element modeling, have been reduced and streamlined.

Some other differences between the two approaches may also be highlighted. For example, in the direct analysis approach, dynamic loads are specifically calculated by advanced procedures, while in a design-oriented first principles system, simplified formulae based on previously performed parametric calculations are used (including for loads such as those due to bottom and bow flare slamming and to liquid sloshing). Additionally, a design-oriented system typically considers a lesser number of (standardized) load cases. It treats fatigue design by a calibrated simplified approach, whereas direct analysis uses the more involved spectral fatigue approach.

A first principles design procedure should ideally be based on a "net ship" (i.e., excluding corrosion margin) limit-state approach. This facilitates explicit consideration of durability issues and corrosion sensitivity since the net ship scantlings are those required by strength alone. Another important safety enhancing design feature that may be made available by such systems is consideration of residual strength of the hull girder after damage. The provision of minimum hull-girder strength to resist premised damage situations can reduce the risk of post-accident collapse or disintegration of a damaged hull during tow or rescue operations.

First principles advanced technology is also useful for the assessment of existing ship structures, for example, for condition assessment. The aim is to give owners, operators, charterers, underwriters, and other interested parties the means for objectively judging the condition of their ships through a condition assessment survey and subsequent structural evaluation. The evaluation typically compares current and projected conditions of the hull structure with the net ship, so that

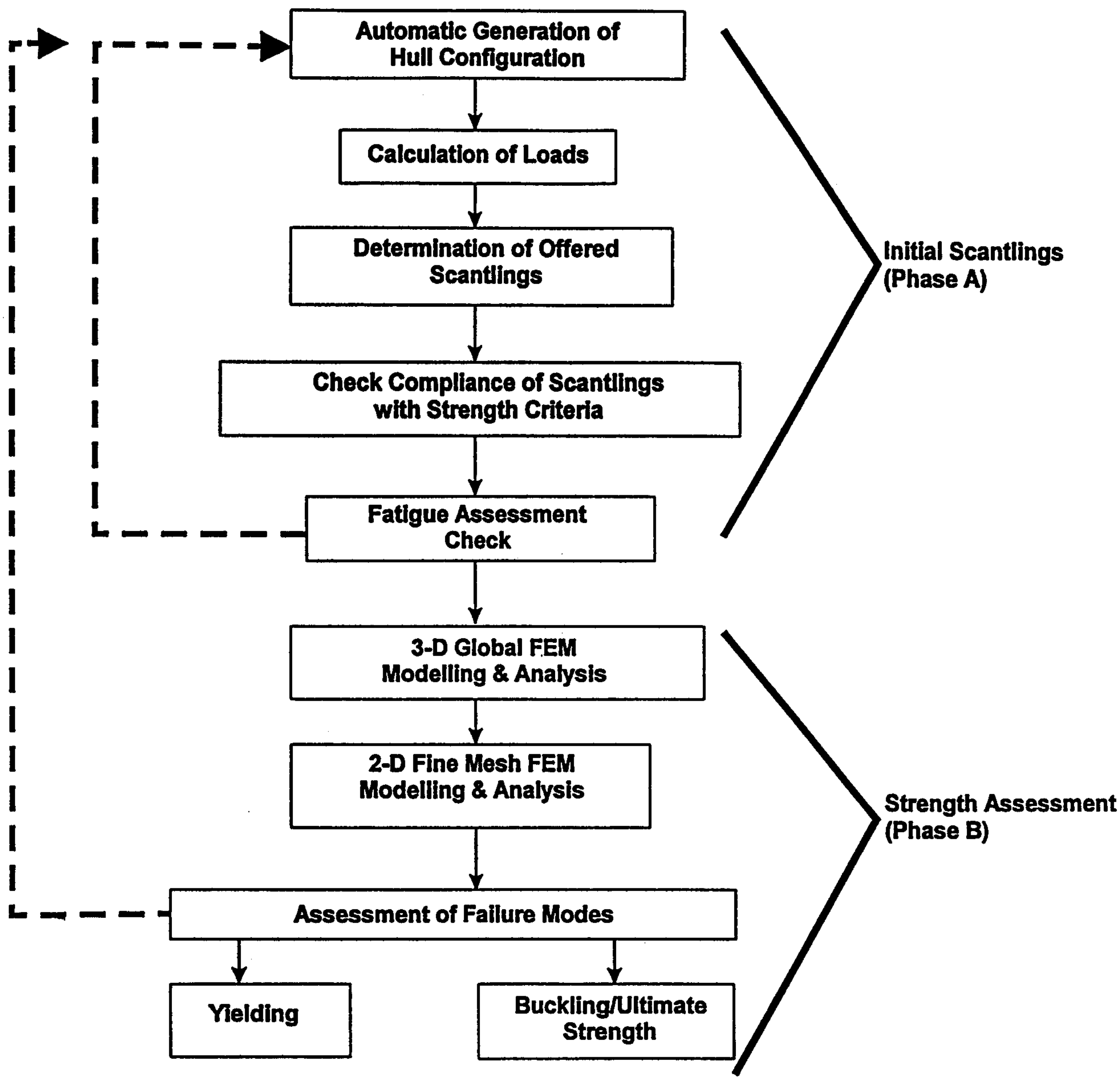


FIGURE 9 A computer-assisted design system.
Source: ABS, 1994.

maintenance requirements, including steel replacement, may be objectively assessed/projected.

The ABS SafeHull system, introduced in 1993 is a true first principles design system with easy-to–use software that meets all of the above requirements and objectives (Chen et al., 1993; ABS, 1994). A flow chart for the SafeHull design system is shown in Figure 9. SafeHull procedures exist for tankers and bulk carriers, with those for container ships and liquefied-gas carriers to follow. It is the intention that SafeHull will eventually replace traditional rules, as it has for tankers today. A specific version of Safehull is available for condition and maintenance needs assessments of existing vessels.

Bulk Carrier Structural Studies

Studies of existing bulk carriers undertaken by ABS using the SafeHull system have indicated that increased design margins may be warranted in some cases. The investigations revealed that, in order to obtain a comparable degree of durability and safety as with tankers built to the SafeHull criteria, five structural areas and conditions specific to bulk carriers need specific consideration. These are transverse corrugated bulkheads in dry cargo holds, vertical-hold frames, cross-deck structure, fore-body structure, and effects of possible cargo overloading.

In summary, the findings were as follows:

- Dry Cargo Hold Transverse Corrugated Bulkheads—Accidental flooding may in some cases cause collapse of the bulkhead based on current requirements. The necessary scantling increases to avoid this may be 20 percent to 30 percent higher than at present.
- Vertical Hold Frames—These are generally the weakest link in a bulk carrier structure. Their continued effectiveness can be increased by stricter corrosion margins and explicit fatigue assessment.
- Cross Deck Structure—Higher loads and stresses can occur in these areas than previously thought. These stresses result from torsional effects imposed by twisting and compressive loads imposed by heavy cargoes in adjacent holds. Consequently, such loads need to be explicitly considered.
- Fore Body—Dynamic loads resulting from bow slamming, bow flare impact, and green water loads fore deck can in some cases be substantial. Therefore, both the local and global effects of these loads on the structure should be taken into account.
- Cargo Loading—Shear forces in the structure during certain loading sequences of ore carriers can be larger than the permissible shear values. As a result, extra margins in the side-shell structure are necessary to account for the possibility of overloads.

Such insights through comparative analyses of different types of vessels (in this case, bulk carriers in comparison with tankers) are one of the many benefits of advanced technology. In this case, the analysis findings closely mirror service experience regarding known bulk-carrier problem areas.

Damage Tolerant Design

The damage tolerance of a vessel is a function of initial design margins, and system effects (Paliou et al., 1987). Increased initial design margins (above minimum requirements) will increase the reserve capacity and, thus, the load at which damage can initiate. Once damage initiates, the residual strength of the vessel, which is a function of the structural system rather than merely its components, decides vessel damage tolerance. The residual strength is a function of the strength of components, their arrangement (and thus load paths), and material ductility. Conventional design procedures do not typically provide for any consideration of residual strength of damaged vessels; an exception being the SafeHull system.

Use of Probabilistic Methods

In this section, we discuss the use of probabilistic methods, in particular, reliability techniques and risk assessment. The former is useful for design and maintenance decisions related to the effects of cracking and for model updating based on service data. The latter is useful to obtain numerical measures for structural, environmental, personnel, and property risk associated with fracture-related undesirable events. Both disciplines account for variability in parameters affecting loads and strength.

ABS, in particular, has invested a considerable amount of resources in the development and application of probabilistic methods for ship structural design and analysis. In fact, it was as early as 1970 that ABS sponsored the early pioneering work of Mansour (1970). Quite recently, during the SafeHull project, a separate independent study was devoted to a reliability-based assessment of structural safety levels implied by the SafeHull procedures and criteria for tankers. Another notable series of projects are those being undertaken by the Ship Structure Committee as part of its reliability thrust area (Mansour, 1989; Nikolaidis and Kaplan, 1991; Hughes et al., 1994) and ongoing projects SR-1344, Assessment of Reliability of Existing Ship Structures and SR-1345, Probability Based Ship Design: Implementation of Design Guidelines for Ships.

Reliability Methods

Reliability methods, which directly account for variability in loads and strength, are better suited for the following than their deterministic counterparts:

- Estimating the structural safety levels (probabilities of malfunction or failure) implicit in existing structural design practice. These can be used to develop target values for new, reliability-based, limit-state design procedures and criteria. Safety levels can be made uniform across failure modes if desired.
- Incorporating new information that becomes available after the design process (e.g., from actual service experience related to cracking) in contrast to what would be expected from design presumptions. Such information, which can be useful in updating design parameters (e.g., load limits), may also be available from the construction stage (e.g., actual material data and fit-up information), proof-testing, and inspections (corrosion and fatigue experience) and from survival of high-demand events, such as storms. Updating is generally difficult with deterministic methods, unless done in an overly pessimistic way.
- Inspection planning for fatigue effects (Shinozuka and Deodatis, 1989), with direct accounting of variability in loads, strength, and effectiveness of inspection procedures

(by use of so-called probability of detection curves). Inspections can be scheduled so that structural reliability is improved for a given inspection budget, or inspection costs are reduced while maintaining a required level of reliability.

- As an aid to decision making considering sensitivity to changes in parameters (e.g., labor and material cost variations). In addition to the probability of failure information, reliability analyses also provide sensitivity indices as a byproduct.

Risk Analysis and Evaluation

Risk is a composite measure of the likelihood and the severity of an adverse accidental event. A risk assessment (analysis and evaluation) involves the following steps:

- identification of major hazards (e.g., those arising from cracking)
- evaluation of the likelihood that these hazards are realized
- estimation of risk to persons, property, or the environment from the potential consequences of the hazards
- evaluation of preventive, protective, or containment measures to reduce the risks as low as is reasonably practicable

In the process of risk analysis, methods, such as failure mode and effect analysis (FMEA) and fault and event trees, may be used to define and study the relationships between affected components and subsystems, initiating events, and outcomes.

Recently, risk assessment techniques have been advocated for the purpose of making a personnel safety case for certain types of ships (see UK DOE, 1990; UK HL, 1992).

Structural Management Systems

The aim of a structural management system is to monitor and manage structural safety of the vessel by an integrated approach to analysis and assessment of information from design, construction, operation, maintenance, inspection, and surveys (Bea, 1991). While there is no one, unique structural management system, two features are common to many:

- an information database
- a collection of analysis and assessment tools working off that database

Structural management systems may include or use hull stress monitoring systems; repair management systems; tools for reliability analysis and risk assessment; tools for fitness for service assessments; tools for stress and fatigue analysis, including prefabricated finite element models; inspection planning software; and emergency response systems.

The types of information used by structural monitoring systems vary but can include a design-stage stress and fatigue life map of the vessel; knowledge of actual construction materials and fit-up defects; ship tank usage and trading history; input from loading computers and hull and hold stress monitoring, inspection, and survey data (e.g., for corrosion effects and cracking); and data related to structural modifications and repairs.

A structural management system is a desirable life-cycle–control tool for vessel safety. More frequent monitoring and assessment of structural condition and response to changes in it are made possible by use of such a system than was traditionally possible. A schematic of a structural management system is shown in Figure 10.

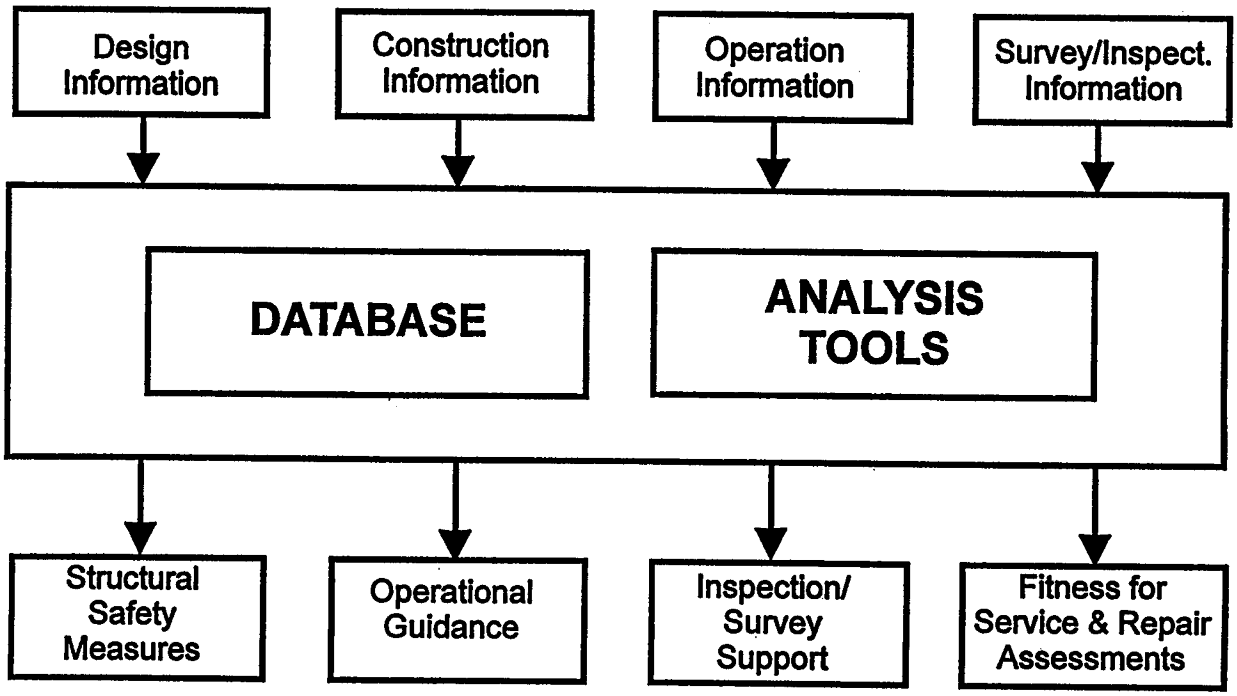


FIGURE 10 Structural management system.

CLASSIFICATION SOCIETY RESPONSE TO CRACKING IN SHIPS

Classification societies have traditionally been very responsive to addressing problem areas in vessel structures. This process consists of early identification of problem areas, using service experience and analyses, and subsequent modification of rules and survey procedures. With regard to cracking in ships, and the related factor of corrosion, we now summarize some of the actions taken by classification societies in general, and the IACS, in particular.

The actions we now describe are of the following types:

- fundamental improvements in design procedures, criteria, and safety margins
- survey improvements
- awareness of operational demands

- effective use of feedback from service experience
- ongoing internal and external research and development
- IACS efforts

Changes in Design Procedures and Criteria

Although most cracking-related problems appear to be primarily maintenance and operation related, ABS feels that changes in initial design criteria and procedures should also be considered to mitigate them. The most far reaching change that ABS has initiated in this regard is its SafeHull system, which explicitly considers relevant failure modes, including fatigue, can also rationally address durability issues and maintenance, and is also suitable for the condition assessment of existing vessels.

SafeHull procedures and criteria, which have been previously discussed, are in a basic engineering format that facilitates direct quantification of load and stress effects. The ABS DLA, with its more sophisticated but more involved approach, complements the design-oriented SafeHull. These procedures make possible a more specific and safer design, considering structural, material, and loading pattern differences.

We previously indicated that, in some cases (e.g., bulk carriers), ABS feels that safety margins actually need to increase from present levels, and indications are that this can be accomplished for the most part by a redistribution of material, that is, without any significant added weight.

Survey Improvements

Corrosion of vessel structures has been the subject of study for many years, including by the TSCF, which consists of owners, operators, and classification societies. Recently, corrosion and damage experience in bulk carriers has been intensively considered by IACS.

For both tankers and bulk carriers, survey requirements have been tightened over time. Consider the example of bulk carriers. When many of these, which have experienced problems, were delivered, they were built without the coatings now required for ballast tanks. Previously, uncoated ballast tanks were not very frequently surveyed. Now, they are surveyed not only at special (5-year) and intermediate surveys but also at each annual survey.

For cargo holds in bulk carriers, the previous requirement was examination at special surveys and on an annual basis at a late age. From 1992, ABS rules have required general examinations of bulk carrier cargo holds on an annual basis, commencing after the second special survey. Additionally, bulk carrier intermediate-survey requirements are also now more stringent and specify close-up examination of 25 percent of cargo hold framing.

Survey requirements are subject to continuous review. It is imperative that appropriate action be taken by all parties concerned—ship owners, operators, and classification societies—to increase the frequency and expand the scope of structural surveys of vessels experiencing frequent cracking-related problems and to detect unacceptable changes in structural condition as early as possible.

Awareness of Operational Demands

The responsibility to assure the structural integrity of a vessel is with its owner, who can use the classification society effectively, for example, by facilitating regular surveys and structural access.

Although classification societies are not normally aware of the terms of charters, it has become apparent that some terms and conditions may, in some cases, make it difficult for owners or operators to meet the demands of time and preparation required for proper survey and maintenance. Hence, charterers must consider the needs of structural integrity in drafting charter agreements.

Terminal operators may place additional demands on the vessel structure. Recent concerns about potential damage due to high rates of loading and other particular features of cargo operations in bulk carriers are also worth considering.

Effective Use of Feedback From Service Experience

Currently, using feedback from service experience, classification societies provide circulars of instruction to surveyors, informing them of information on the critical areas of the hull structure to be surveyed. Owners are also contacted to make them aware of particular concerns as they arise or are evident from feedback of service data, such as, the need for their close-up inspection of the welded attachment of the side-shell frames in forward cargo holds of bulk carriers.

A classification society's computerized database of hull service experience allows systematic collection, storage, and retrieval of a vessel's hull condition and local failure data, its ready processing and trend analysis; and generation of related reports, including those to surveyors and for the IACS early warning system.

Such data are beneficial in identifying structural problem areas, whether vessel specific or general, in an efficient way. They also support research and development and adjustment of procedures and criteria consistent with experience. Additionally, they provide surveyors and owners and operators with a vessel's history of hull damages and repairs, facilitating survey planning and making surveys and inspections more efficient and effective.

Ongoing Internal and External Research and Development

The major classification societies are actively engaged in internal and external research and development related to various aspects of vessel experience and structural adequacy. In the case of cracking of ships, two efforts are the work of the TSCF on repairs of cracks and the Berkeley joint industry project on structural maintenance of tankers.

IACS Efforts

Apart from ABS, Lloyds, and Det Norske Veritas, the members of the IACS are Nippon Kaiji Kyokai, Bureau Veritas, China Classification Society, Germanischer Lloyd, Korean Register, Polish Register, Registro Italiano Navale, and the Russian Register of Shipping. There are two associate members, the Croatian and Indian Registers. IACS is a forum for the improvement of merchant ship safety and prevention of marine pollution and also liases with IMO and industry bodies such as the TSCF. IACS also has a mandate for unifying classification requirements; some related examples being unified requirements on materials and minimum longitudinal strength. In addition, IACS provides unified interpretations of international conventions to its members.

IACS membership implies certain capabilities, including research and development and audited compliance with a set of formal quality requirements. The 11 IACS members together class 90 percent of the world merchant fleet by tonnage and 50 percent by number—more than 40,000 ships and 400 million gross tons. At least one P&I club currently makes a classification certificate from an IACS society mandatory, but this is an exception rather than the rule.

The technical work of IACS is carried out by permanent working parties and ad hoc work groups. The working parties and groups whose efforts have some relevance to our subject of cracking in ships are the IACS working parties on hull damages (WP/HD), marine pollution, materials and welding, strength of ships subdivision, stability and load line, survey reporting and certification, and fatigue.

We now present a review of relevant IACS efforts under the following headings:

- enhanced surveys and survey planning
- corrosion protection
- the IACS early warning scheme
- study of bulk carrier hull damages
- fatigue design procedures

Enhanced Surveys and Survey Planning

IACS has developed unified requirements for the enhanced surveys of tankers and bulk carriers (UR Z10.1, 10.2) and survey planning. In fact, the previously described ABS procedures for enhanced surveys reflect the IACS unified requirements. Also, IACS, which participates in the International Maritime Organization (IMO) deliberations as an observer, contributed to the IMO developments on enhanced surveys.

Corrosion Protection

IACS members have long recognized the importance of corrosion protection of ballast spaces and cargo holds and have developed the following unified requirements addressing it:

- UR Z8—Corrosion Protection Coatings for Salt Water Ballast Spaces.

- UR Z9—Corrosion Protection Coating for Cargo Hold Spaces on Bulk Carriers.

In addition, the unified requirements for enhanced hull surveys of oil tankers (Z10.1) and hull surveys of bulk carriers (Z10.2) previously referred to also address corrosion-related matters, including the types and extent of close-up surveys and gaugings as a function of vessel age.

The IACS Early Warning Scheme

IACS has recently (1993) developed procedures for collecting and disseminating information related to hull damages among its members. The information collected pertains to significant hull damages that will or could develop to jeopardize structural performance and thus endanger the ship, its cargo, or the environment. Essentially, members of the IACS WP/HD are to submit confidential data sheets reporting relevant hull damages. This information is studied by the WP/HD, which makes recommendations regarding further action, including reflecting the same in guidelines or booklets.

Study of Bulk Carrier Hull Damages

Because of concern with the continuing bulk carrier casualties and related loss of life, IACS has been investigating bulk-carrier hull damages as a matter of priority. This work continues to look at factors contributing to structural failures, sequences of events leading to casualties and losses, and measures to reduce structural failure. From this study, two publications have so far resulted, IACS, 1992; AND IACS, 1994. The former is an advisory document for ship owners and operators; the latter provides detailed guidelines for surveys, assessment, and repair of hull structure of bulk carriers.

Fatigue Design Procedures

An IACS ad hoc working group on fatigue is developing procedures for the fatigue design of ship structures. The selected approach is to use a simplified procedure wherein the long-term distribution of stresses is approximated by a Weibull curve, similar to that in ABS SafeHull, but undoubtedly with differences. A unified method for fatigue design is expected to result from this work. A possible future task is the potential unification of pressure load calculation methods for fatigue and local bending loads. We might also mention that there is another IACS working group that is considering standardization of wave data for use in the direct calculation of a ship's behavior at sea.

CONCLUDING REMARKS

In this paper, we have discussed a number of matters related to the subject of local cracking in ships, its causes, consequences, and control. We also addressed how cracking is controlled in the traditional classification rule approach and how it can be even better controlled throughout the vessel life cycle using advanced technology and structural management. In addition, we discussed classification society and IACS efforts related to crack occurrences in vessels.

The following concluding remarks are appropriate:

- Cracking is of concern.
- Crack-free vessels are unusual.
- Cracking typically does not occur due to one isolated factor.
- Cracks, when found, have to be effectively repaired.
- Repeated incidence of cracking, particularly early in vessel life, requires in-depth analysis.
- Designers need to be aware of construction and operation and account for them as necessary.
- Better design and analysis using advanced technology should, to a large measure, reduce cracking in ships and should also provide for planned control of its consequences.

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All information described this paper has by necessity been simplified and abbreviated, and, hence, is not definitive.

REFERENCES

ABS (American Bureau of Shipping). 1989. Fatigue Analysis of Tankers. R&D Division Technical Report RD-89020F. New York: ABS.

ABS. Assessment and Comparison of Wave Data in the North Atlantic Ocean, North Pacific Ocean, and Along the TAPS Route. New York: ABS.

ABS. 1991. Service History of Trans-Alaskan Pipeline Service Vessels. New York: ABS.

ABS. 1992. Rules for Building and Classing Steel Vessels. Part 1: Classification, Testing and Surveys: Appendix 1/B: Hull Surveys of Oil Tankers, January 1993; Appendix 1/C: Hull Surveys of Bulk Carriers, July 1993. New York: ABS.

ABS. 1994. Technical Information—The SafeHull System. New York: ABS.

ABS. 1994a. Marine Management and Ship Operation Guideline—ISO 9002/ISM Code, Rev. 03. New York: American Bureau of Shipping.

ABS. 1994b. ABS rapid response damage assessment (RRDA) program. ABS Quest 2(4):8.

ABS. 1994c. Pp. 4–7 in Proceedings Bulk Carrier Safety Seminar, ABS Europe held December 1994 in London, England. New York: ABS.

Basar, N.S., and R.F. Stanley. 1978. Survey of Structural Tolerances in the United States Commercial Shipbuilding Industry. Report SSC-273. Washington, D.C.: Ship Structure Committee.

Bea, R.G. 1991. Marine Structural Integrity Programs. Report SSC-365. Washington, D.C.: Ship Structure Committee.

Bea, R.G. 1994. The Role of Human Error in the Design, Construction, and Reliability of Marine Structures. Report SSC-378. Washington, D.C.: Ship Structure Committee.

BSI (British Standards Institution). 1991. Guidance on Methods for Assessing the Acceptability of Flaws in Fusion Welded Structures. PD 6493:1991. London, England: Her Majesty's Stationery Office.

Bureau of Transport and Communication Economics. 1994. Structural Failure of Large Bulk Ships. Report 85. Canberra, Australia: Australian Government Publishing Service.

Chen, H.H., H.Y. Jan, J.F. Conlon, and D. Liu. 1993. New approach for the design and evaluation of double-hull tanker structures. Pp. 215–246 in SNAME 1993 Annual Transactions. Jersey City, New Jersey: Society of Naval Architects and Marine Engineers.

Grove, T.W., G. Rynn, and G.M. Ashe. 1992. Bulk Carriers, A Cause for Concern. ABS Paper Series, 1992. New York: American Bureau of Shipping.

Gurney, T.R. 1979. Fatigue of Welded Structures. London, England: Cambridge University Press.

Horne, G.E., and M.D. Johnson. 1994. Using the ABS SafeHull Approach to Optimize Repairs. New York: American Bureau of Shipping.

Hughes, O., E. Nikolaidis, B. Ayyub, G. White, and P. Hess. 1994. Uncertainty in Strength Models for Marine Structures. Report SSC-375. Washington, D.C.: Ship Structure Committee.

IACS (International Association of Classification Societies). 1992. Bulk Carriers—Guidance and Information to Ship Owners and Operators. London, England: IACS.

IACS. 1994. Bulk Carriers—Guidelines for Surveys, Assessment and Repair of Hull Structure. London, England: IACS.

Jan, H.Y. 1993. The Net Ship Approach and a Systematic Maintenance Program. SafeHull Internal Memo dated July 13. New York: American Bureau of Shipping.

Jordan, C.R., and C.S. Cochran. 1978. In-Service Performance of Structural Details. Report SSC-272. Washington, D.C.: Ship Structure Committee.

Liu, D., J. Spencer, T. Itoh, S. Kawachi, and K. Shigematsu. 1992. Dynamic Load Approach in Tanker Design. Pp. 143–172 in SNAME 1992 Annual Transactions. Jersey City, New Jersey: Society of Naval Architects and Marine Engineers.

Ma, K.T., and R.G. Bea. 1994. Repair Management System for Fatigue Cracks in Ship Critical Structural Details. Report No. SMP III-1-1 of the Structural Maintenance for New and Existing Ships Project. Department of Naval Architecture. Berkeley: University of California.

Mansour, A.E. 1970. Methods of Computing the Probability of Failure under Extreme Values of Bending Moment. MIT Report No. 70-15. Cambridge, Massachusetts: Massachusetts Institute of Technology. (See also Journal of Ship Research, vol. 16, no. 2, June 1972. Jersey City, New Jersey: Society of Naval Architects and Marine Engineers.)

Mansour, A.E. 1989. An Introduction to Structural Reliability Theory. Report SSC-351. Washington, D.C.: Ship Structure Committee.

Nikolaidis, E., and P. Kaplan. 1991. Uncertainties in Stress Analyses on Marine Structures. Report SSC-363. Washington, D.C.: Ship Structure Committee.

NRC (National Research Council). 1991. Tanker Spills, Prevention by Design. Marine Board. Washington, D.C.: National Academy Press. (See also the Tanker Structure Cooperative Forum Work Group Project Report on Examination of the National Academy of Sciences Report dated June 1991. London, England: Witherby).

Paliou, C., M. Shinozuka, and Y.N. Chen. 1987. Reliability and durability analysis of marine structures. Pp. 143–158 in Proceedings SSC-SNAME Marine Structural Reliability Symposium held October 5–6, 1987 in Arlington, Virginia. Jersey City, New Jersey: Society of Naval Architects and Marine Engineers.

Schulte-Strathhaus, R. 1991. Fatigue Database Development and Analysis. Report No. SMP-1-1. Department of Naval Architecture. Berkeley: University of California.

Shinozuka, M., and G. Deodatis. 1989. Reliability of Marine Structures Under Bayesian Inspection. New York: American Bureau of Shipping.

Thayamballi, A. 1990. Fatigue Screening of Tankers. ABS Research Report No. 90005. New York: American Bureau of Shipping.

Trufyakov, V.I., P.P. Mikheev, Y.F. Kudryavtsev, and D.N. Reznik. 1993. Fatigue endurance of welded joints—Residual stresses and fatigue improvement treatment. Pp N-1–N-14 in Proceedings of Ship Structure Symposium held November 16–17, 1993 in Arlington, Virginia. Jersey City, New Jersey: Society of Naval Architects and Marine Engineers.

TSCF (Tanker Structure Cooperative Forum). 1986. Guidance Manual for Inspection and Condition Assessment of Tanker Structures. London, England: Witherby. (Also see TSCF, 1992. Condition Evaluation and Maintenance of Tanker Structures. London, England: Witherby).

U.K. Club (U.K. Protection and Indemnity Club). 1993. Analysis of Major Claims. The United Kingdom Mutual Steam Ship Assurance Association (Bermuda) Limited. London, England: U.K. Protection and Indemnity Club.

U.K. DOE (U.K. Department of Energy). 1990. The Public Inquiry into the Piper Alpha Disaster. London, England: Her Majesty's Stationery Office.

U.K. H.L. (U.K. House of Lords). 1992. Safety Aspects of Ship Design and Technology. London, England: Her Majesty's Stationery Office.

USCG (U.S. Coast Guard). 1990. Report on the Trans-Alaska Pipeline Service Tanker Structural Failure Study. Washington, D.C.: USCG.

USCG. 1991. Critical Area Inspection Plans. USCG Navigation and Inspection Circular 15-91. Washington, D.C.: USCG.

Ship Fabrication—From Shipbuilder's Viewpoint

Hidetoshi Sueoka

ABSTRACT

There are increased efforts today by many governments and regulatory bodies to increase the reliability and safety of ships and provide for more rigorous protection of the marine environment. However, the depressed market for ships has led shipbuilders to offer reduced prices for ships. The demand placed on shipbuilders is to ensure greater quality ships at competitive costs. These problems are being addressed by developing standards of quality and increasing shipyard productivity. While these changes in ship reliability are important, there are other changes occurring in ship design. Tankers are now required to have double hulls, and higher-strength steel is being used in construction. Improved structural analysis methods are being used in design, and shipyards are changing the details of the structure to improve reliability and are increasing the control of construction tolerances.

TODAY'S DEMANDS FOR SHIP FABRICATION

Reflecting the recent movement for more reliable safety of ships and more rigorous protection of marine environment, various regulations have been and will be proposed and/or put into force by the relevant international or domestic organizations, governments, and so on.

On the other hand, regardless of such movement, the marine market has been depressed, unfortunately, so that shipbuilding industries have been requested to offer competitive prices of ship in tankers, bulk carriers, and other ship types' fields.

Under these circumstances, today's demands for ship fabrication seem to be those listed below:

- how to ensure some quality of ship fabrication, including during the design and construction stage
- how to achieve competitive costs in reply to present markets under some quality level

- how to design and construct new-generation ship structures, such as double hull (DH) or mid-deck (MD) tankers or equivalent, as required by the International Maritime Organization's (IMO) convention on marine pollution (MARPOL)

This chapter outlines such subjects very briefly, focusing on the ship structures, especially for the case of DH tankers, together with the introduction of our shipyard's activities.

Standards of Quality

One familiar standard of quality regarding ship fabrication is the Japan Shipbuilding Quality Standard (JSQS), which has been revised several times in accordance with changes in ship size and type, progress of materials and production facilities, and other technical innovations since the first issue of the standard in 1966 in Japan (JSQS, 1991).

This standard, JSQS, has been accepted not only in Japan but also in other countries and has contributed to ensuring some quality level on accuracy of products. The contents cover the following stages: material, marking, gas cutting, fabrication, subassembly, accuracy of hull form, welding, alignment and finishing, deformation, and miscellaneous.

Unlike the standard on production accuracy, such as JSQS, few standards on structural details and other practices could be found. The reason is that such practices may be strongly dependent on each shipyard's technical situation through the experiences of design and construction.

Some shipyards have their own good standards on structural details and other practices. Others have few or poor ones. In that sense, it may be difficult to make up such standards as applied worldwide. However, regarding the maintenance of existing ships, some guidelines have been issued, as follows:

- Bulk Carriers—Guidelines for Surveys, Assessment and Repair of Hull Structure (IACS, 1994)
- Guidance Manual for the Inspection and Condition Assessment of Tanker Structures (TSCF, 1986)

These guidelines show the typical structural damages and recommended repair plans for the important primary structural members, in addition to the practice for inspection and condition assessment. Such technical information is so effective for shipowners/operators, as well as shipyards, as to improve the quality for structural details from the fracture and fatigue strength viewpoints.

Shipyards will expect IACS or other relevant parties to carry out the technical works continuously for establishing such guidelines through many experiences of damage or non-damage of structures for existing and/or new-built ships.

Another recent movement is the application of ISO 9001, which is the international standard to examine and approve the procedure in the written format of the management system of shipyards for quality assurance. Many shipyards have just received the certificate of ISO 9001, which will satisfy shipowner and ship operators from the viewpoint of the adoption of the qualified shipyards.

As mentioned here, standards of quality for ship fabrication will be increasingly improved and introduced in reply to one of today's demands: how to ensure some quality level of ship fabrication.

High Productivity

In order to achieve competitive costs, some shipyards have made an effort to challenge the high productivity of ship fabrication by means of the development and the application of advanced computer-aided design/computer-aided manufacturing (CAD/CAM), robots and computer integrated manufacturing for shipbuilding (CIMS) (Funaoka, 1992).

In the design stage, the advanced CAD system, which is commercially developed by a software house or developed by shipyards, has been applied. In the case of Mitsubishi Heavy Industries (MHI), the advanced CAD system consist of two main parts, one of which is named MARINE (Mitsubishi Advanced Realtime Initial Design and Engineering System), for initial design uses and the other is named MATES (Mitsubishi Advanced Total Engineering System) for basic and detailed design use of hull structure, machinery, and outfitting and electric system.

MARINE has the following functions as a powerful tool for design on inquiry or short-list bidding:

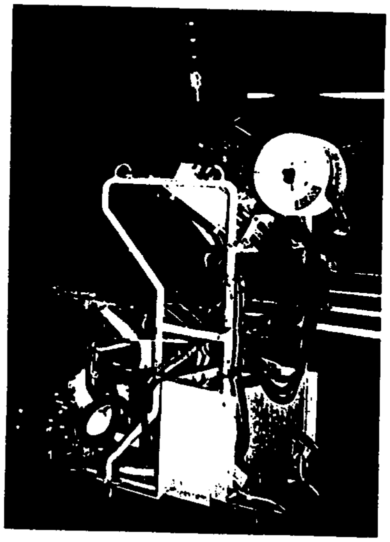
- design data support of similar ships for subject ship
- calculation of performance and drawing of arrangement under the selected principal dimensions and other conditions such as ship speed, draft, and so on
- cost estimation and preparation of full specifications

The functions of MATES on the hull structure design system are as follows:

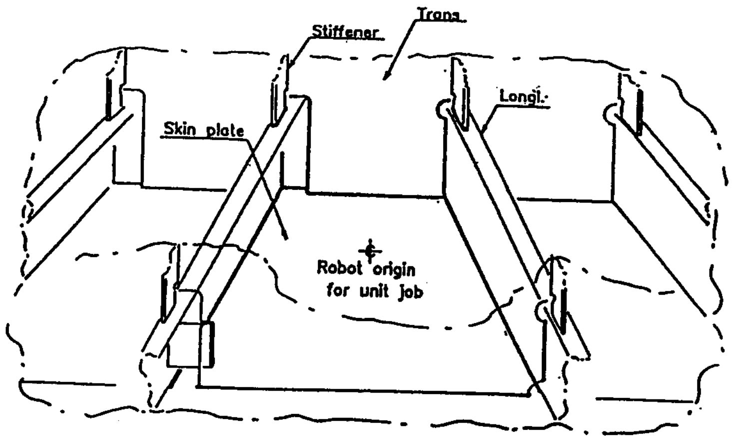
- supporting lines fairing, basic and detailed structural design work
- storing of generated data into the hull database
- drawing of key and detail plans
- generating small parts data, such as stiffeners, brackets, and similar parts, and numerical control (NC) data for fabrication stage
- summing up of materials for purchasing and fabrication planning use

In addition to the existing automatic equipment for cutting, bending, welding, and fabricating plates, newly developed robots have been introduced into the assembly stage and are now working very effectively. For example, some robots weld many stiffeners on web plate simultaneously, and others weld a couple of the complicated connection part of longitudinals and transverse web at once, as shown in Figure 1 (Miyazaki, 1990). Such robots will be powerful and useful, especially, to fabricate DH structures.

The applications of robots will take an advantage not only on cost saving by man-hour reduction but also on keeping good quality by stable machine work. These developments of



(a) Appearance of robot-origin transfer unit



(b) An example of applied structure

FIGURE 1 Example of welding robot.
Source: Miyazaki, 1990.

automation technology have reduced the man-hours in production to about half those of 20 years ago, as shown in Figure 2 (Miyazaki, 1994), and changed the fabrication method at the indoor stage, as shown in Figure 3 (Miyazaki, 1990).

The research and development (R&D) of CIMS in Japan started about six years ago as a project of the Ship and Ocean Foundation, with the participation of the related members from some major shipyards.

The purpose of CIMS is to integrate the information of procedures of ship fabrication from design to completion of construction and to simulate the production methods and processes, including the interface between design and construction planning, and consequently to give the optimum solution for control of job and material flow in the production process.

To develop the advanced CAD/CAM, robots and CIMS for high productivity is one of the recent activities of some major shipyards in reply to one of today's demands: how to achieve competitive costs under some quality level.

New-Generation Ship Structure—Tanker Case

One of the greatest interests on new generation ship structure is DH or MD tankers as measures to prevent outflow of cargo oil from cargo tanks at ship's damage by collision or grounding.

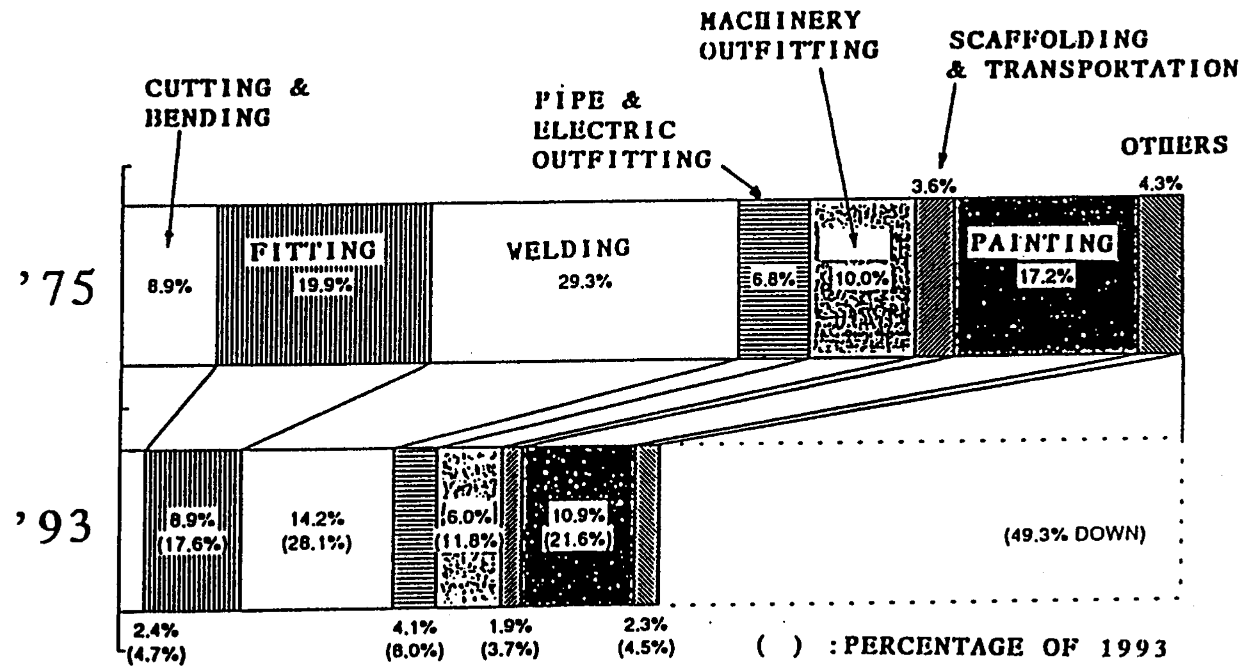


FIGURE 2 Change of the man-hour ratio (in the case of single-hull very large crude carrier production).
Source: Miyazaki, 1994.

In the early stage, while the efficiency of the DH tanker consisted of double-sided hull and double bottom was discussed mainly on necessary depth of DH, our shipyard-proposed MD tanker consisted of double-sided hull and two-tier cargo oil tanks divided by mid-deck, the performance and efficiency of which were researched and confirmed through model tests and simulation by analysis especially in grounding cases (Hirai, 1992). The DH and MD tanker were agreed to at IMO, but the MD tanker has not yet been accepted by the U.S. Coast Guard.

DH tankers have already been constructed and delivered as new generation ships that satisfy the MARPOL regulation.

In MHI case, one DH very large crude carrier (VLCC) was delivered about two years ago, and five DH VLCCs as sister ships are now under construction. Both types are for foreign shipowner.

Before the construction of such DH VLCCs, the detailed investigation and study for hull structure planning were carried out in the design stage. The reliability of structural strength, the suitability for production facilities, material and labor cost impacts, and other factors for the initial design of the DH VLCC have also been investigated. The experiences of DH VLCCs as new-generation ships are just beginning in the fields of design, construction, and operation.

Regarding oil-pollution prevention methods, some R&D tasks have been carried out continuously. For example, in Japan, the Association for Structural Improvement of Shipbuilding Industry (ASIS) has carried out a seven-year research program, named "Protection of Oil Spill

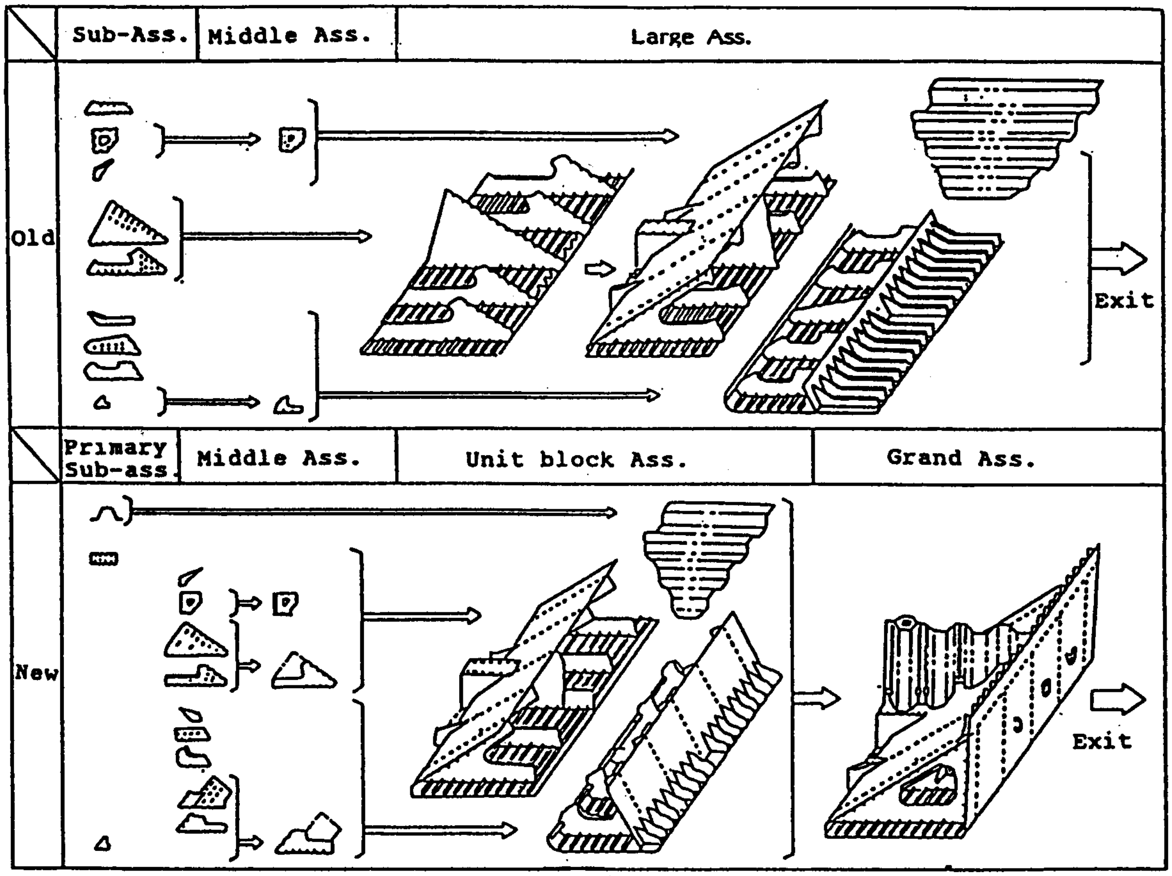


FIGURE 3 Change of fabrication method (at in-door stage).
Source: Miyazaki, 1990.

from Crude Oil Tankers," since 1991; some parts of which have been done by MHI (Kuroiwa, 1994).

The program consists of the following items:

- to establish methodology to predict the extent of structural damage due to collision and grounding
- to establish methodology to predict the expected quantity of oil outflow from damaged tanks
- to develop structural design methodology to prevent oil spill from tankers by collision and grounding

The studies have been done by various experimental works and numerical simulations. Recent big experiments are large-scale grounding tests using the double-bottom structure models of one-fourth scale of DH VLCC against an artificial rock set up in the waterway. In accordance with the today's demands for marine pollution prevention, many efforts will continuously be made to improve the protection methods by the relevant organizations, governments, and shipping and shipbuilding industries. In the structural technology field, the present focus is how to design and construct new-generation ships such as DH Tankers.

CHANGES IN DESIGN AND CONSTRUCTION STAGE

Changes of ship fabrication may be brought about by internal and/or external motivations. While the external ones are derived from the social and economic demands on safety of ships and life, protection of marine environment, and marine market, as stated before, the internal ones largely depend on the progress of technology and the spread of advanced engineering methods and products, such as materials, computers, automatic equipment for fabrication use and so on.

One of the big topics during the last decade is the use of higher-tensile (HT) steel in ship hull. The development of a thermo-mechanical controlled process (TMCP) for HT steel triggered the extensive use of HT steel for main hull structures because of the easy and good weldability of its materials.

Extensive use of HT steel in ships (HT ships) has brought the economic benefit of fuel-oil saving or cargo weight increase by reduction of hull steel weights compared to the conventional mild steel ships, but it has also caused anxiety about the margin for corrosion and possibilities of fatigue cracks from the ship maintenance viewpoints.

The evaluation of HT ships shall be studied continuously through the experiences of design, construction, operation, inspection, and maintenance for many delivered HT ships, and then more reasonable design would be expected.

This chapter introduces the changes of ship fabrication in the design and construction stage for material items, as mentioned here, and other key items of analysis methods, design considerations, and so on.

Discussions on HT Ships at the Initial Stage

At the initial stage of inquiry or short-list bidding, some shipowner and ship operators have asked shipyards to explain the considerations for HT ship because the extent of use of HT steel is not uniform. Mainly it is dependent on ship type and size and also the design considerations of each shipyard.

Figure 4 shows the trend of the application of HT steel. In the late 1970s, TMCP was developed in Japan as an advanced production process of HT steel. The higher strength and good toughness of TMCP HT steel are given by the very fine crystalline structure under the lower carbon content of its chemical composition compared with the conventional HT steel. Figure 5 shows the comparison of TMCP HT steel and conventional HT steel.

While the conventional HT steel needs special care during fabrication work, such as preheating before welding, TMCP HT steel is free from such extra work mainly due to its lower carbon content (Yasuda, 1983). This advantage encouraged shipbuilders to start using HT steel extensively.

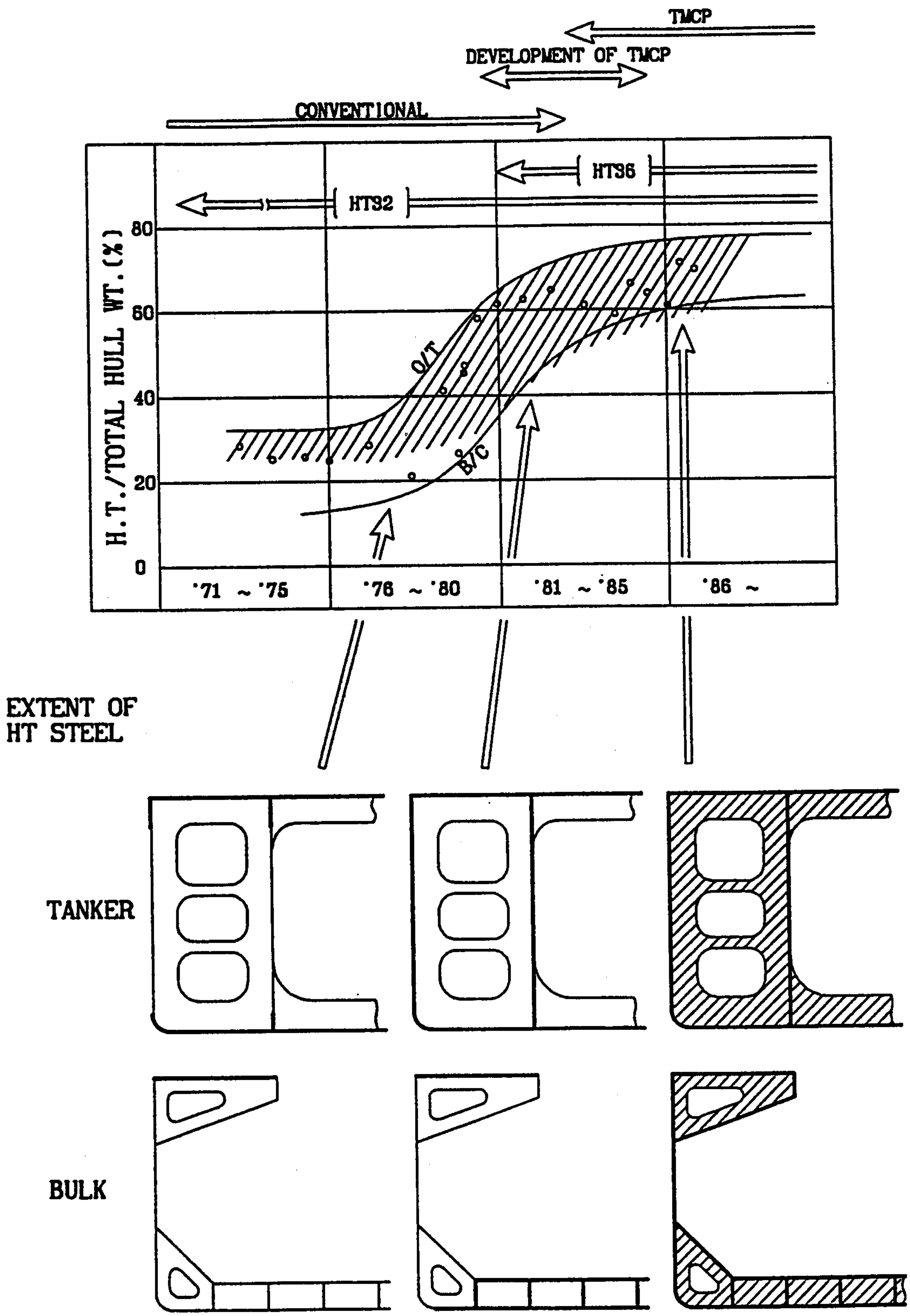
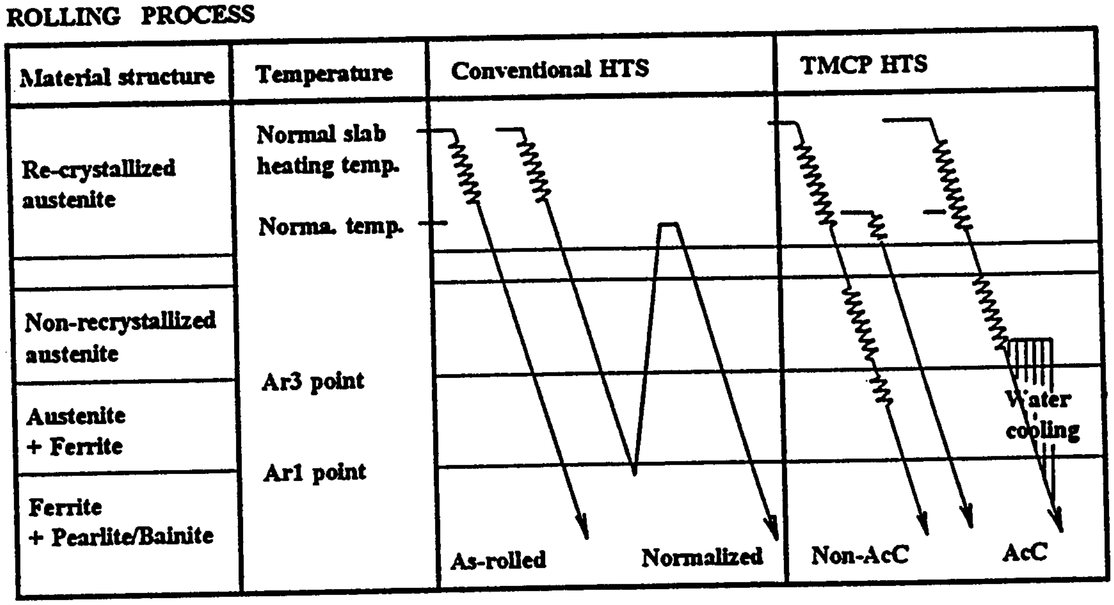


FIGURE 4 Application of HT steels in hull structures.

ROLLING PROCESS



STANDARD FOR WELDING OPERATIONS

Item		Conventional HTS	TMCP HTS
Short bead	tack weld	> 50 mm	> 10 mm
	repair weld	> 50 mm	> 30 mm
Pre-heating	temperature need pre-heat	+ 5 deg.C and below	0 deg. and below
	pre-heating temperature	> 50 deg.C	> 20 deg.C
Line heating (Max steel temp)	water cooling after heating	< 650 deg.C	< 1000 deg.C
	air cooling after heating	< 900 deg.C	< 1000 deg.C

(Guidance for the survey and construction of steel ships
Part M 1994 NIPPON KAIJI KYOKAI)

FIGURE 5 Comparison of conventional and TMCP HT steels.
Source: Yasuda, 1983.

At present, the standard hull structure is designed using HT steel of the appropriate strength level for longitudinal members and part of the transverse members, including transverse bulkhead in way of whole tank parts.

The special design care for application of HT steel is paid from the fatigue strength viewpoints. For example, the applications for the critical areas are under the adoption of soft toe or soft heel structure to the connection of longitudinals and transverses, the maintained use of mild steel, and the control of alignment in the construction stage.

The structural strength for yield, buckling, and fatigue has been evaluated by three-dimensional finite element method (FEM) analysis for typical tank parts and other necessary calculations, such as zoom-up FEM analysis for critical parts on fatigue.

The design for HT ships has been confirmed and improved not only by analysis but also by some experiences for hull structural conditions of HT ships in services. In the case of single-hull (SH) VLCCs with extensive use of HT steel delivered to the domestic shipowner since the mid-1980s, no buckling damages have been found, but a few fatigue fractures in side longitudinals of HT steel were reported, as shown in Figure 6. The studies on this damage were carried out so that countermeasures for these cracks were immediately taken, and the design for detail structures in side frames of HT steel has been improved together with the improvement, of the simplified fatigue design method (Watanabe, 1995).

In the studies, the cyclic loads with different phases due to wave pressure, hull girder bending, and so on, are investigated, and also the stresses of side longitudinal caused by these multiple loads are evaluated by FEM analysis in Figure 7. The characteristics of load components affecting the fatigue strength of side longitudinals, one of which is the significant amplitude of wave pressure as shown in Figure 8, and the stress concentration factor corresponding to the shape of the detailed structure of side longitudinals are obtained, and consequently, the simple formula for fatigue design are proposed. Such works contribute to the improvement of design for detail structure.

This is one example of shipyards' activity for design consideration on HT steel use. Through discussions on HT ships at the initial stage between shipowner and ship operators and shipyards, some shipowner agree on the extensive use of HT steel, and some shipowner still prefer the maintained use of mild steel, judging from their experience. More experience and more R&D on HT ships would lead to building reasonable HT ships under both parties' understanding.

Introduction of Analysis Technology for Design

Some shipyards have developed, applied, and improved the structural analysis system for a long time, which has often been a powerful tool for structural design—especially when ship types changed and ship sizes enlarged—and also was useful for structural damage analysis to research causes and effective countermeasures.

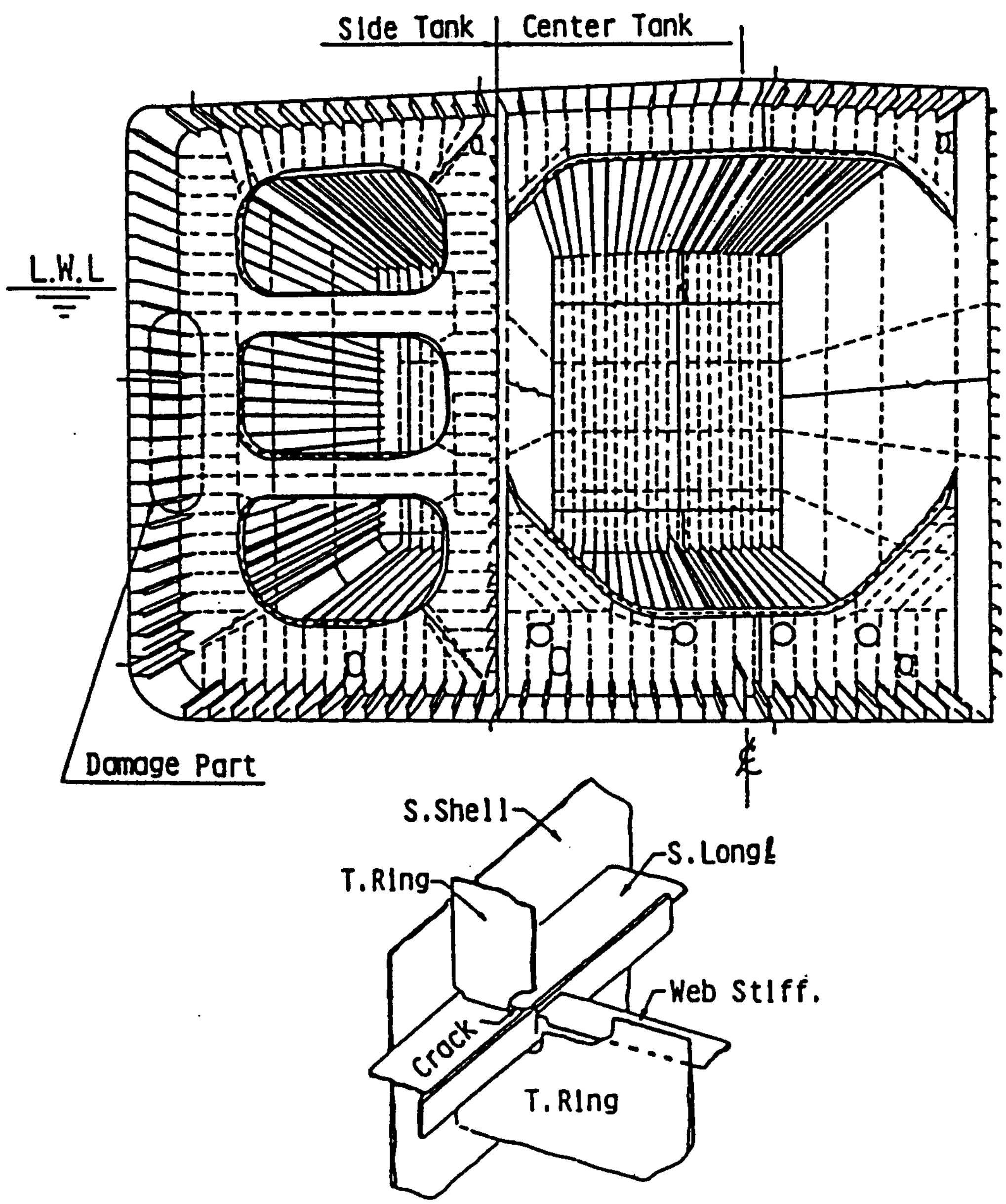


FIGURE 6 Example of fatigue crack of side longitudinals.
Source: Watanabe et al., 1995.

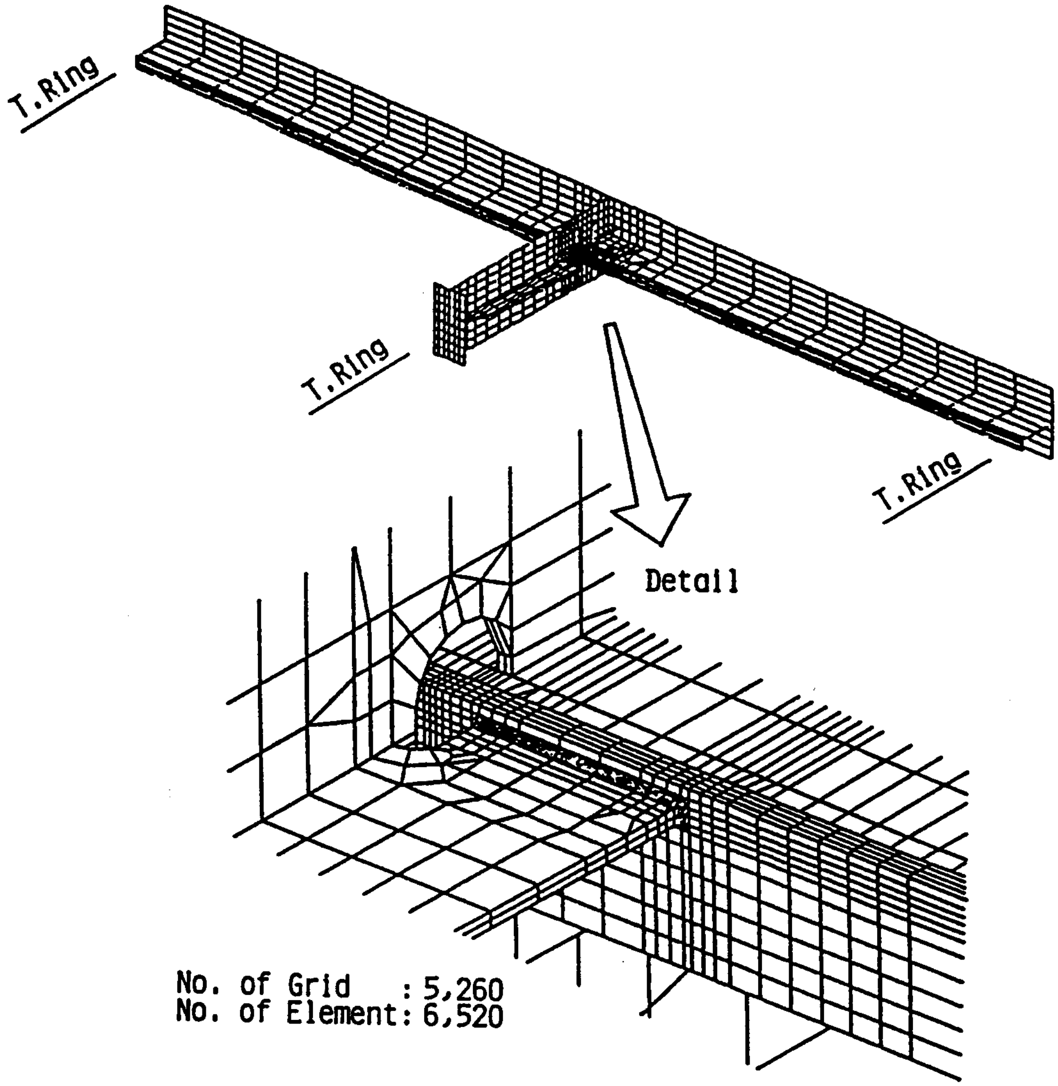


FIGURE 7 Solid FEA modeling of side longitudinal.
Source: Watanabe, 1995.

The technology for the development of a structural analysis system is composed of the following items:

- the estimation method for ship motions and applied loads by waves and ship accelerations in sea conditions
- the stress analysis method for three-dimensional hull structures under given load conditions

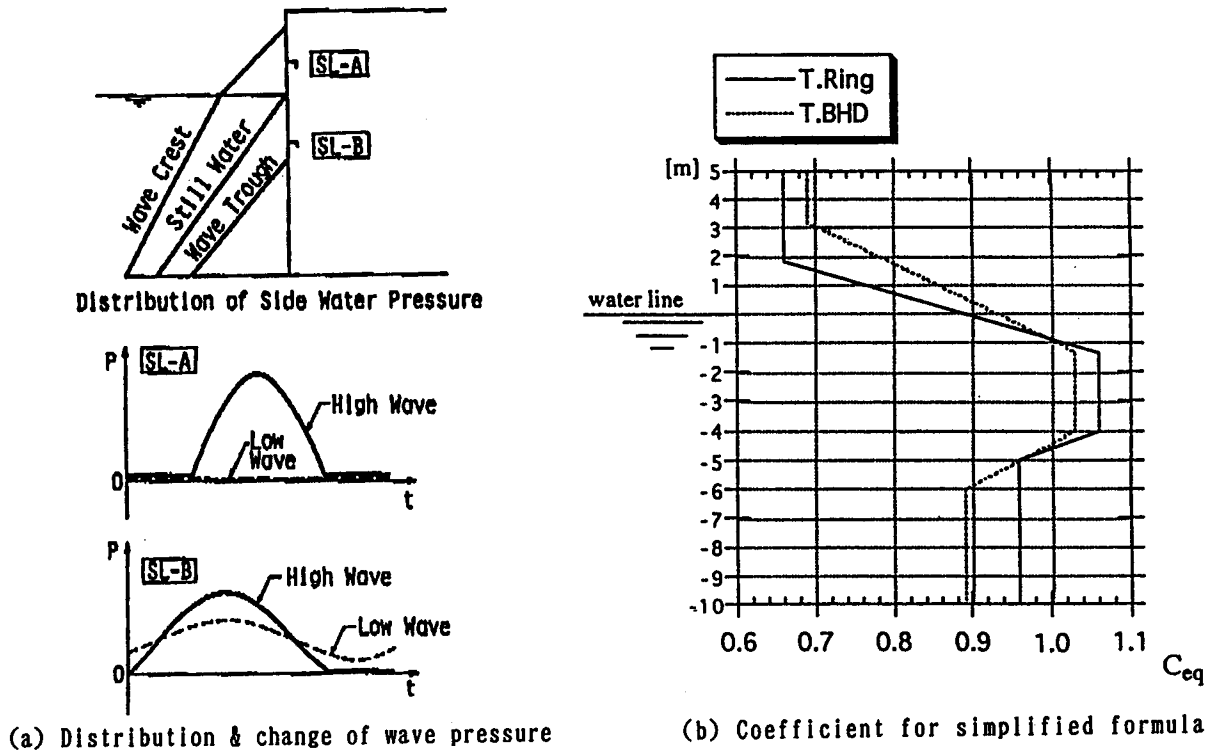


FIGURE 8 Significant amplitude of wave pressure on side longitudinals.
Source: Watanabe, 1995.

- the evaluation method of structural strength for buckling, fatigue, and ultimate strength under given structure and working stress conditions

A brief review on the developments of a structural analysis system (mainly in Japan), including MHI, is outlined here. Figure 9 shows the change of the methods for structural analysis. The R&D in each period are described as below.

1960s and 1970s

Under the demands of enlargement of ship size, especially in the tanker field, the research on ship motions, wave loads, and structural responses was carried out to obtain reasonable methods for structural design of large-size ships. For the evaluation of ship motions and wave loads, the strip method was established as the most practical and useful approach, and computer programs for calculation were developed using the host computers. Structural analysis methods

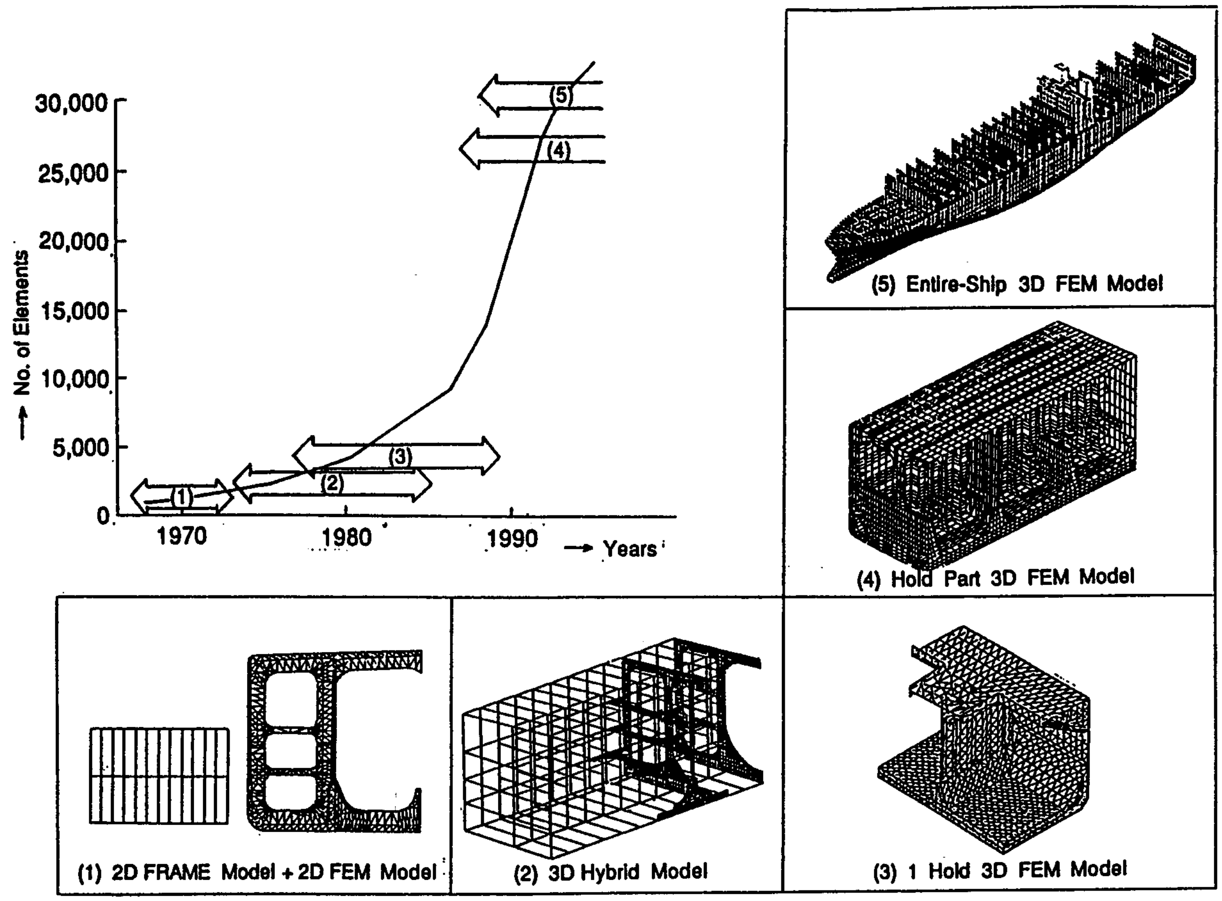


FIGURE 9 Change in the methods for structural analysis.

for calculating the structural responses of its deformations, member forces, and stresses under the given loads were also developed step by step from framework analysis to FEM analysis, as shown in Figure 9. For example, MHI developed the Strength Analysis System of Midship Part of Tanker (SASMIT) composed of the longitudinal-strength calculation of the hull girder, whole-tank–part structural-response calculation by three-dimensional framework analysis, and stress calculation of the transverse ring by two-dimensional FEM analysis, which was used in the initial design of a 400,000 DWT oil tanker.

At the latter half of the 1970s, to predict extreme values of wave bending moments, stresses of primary members, and other items in sea conditions, the so-called total system of analysis on longitudinal strength and/or transverse strength, which includes statistical analysis to get short-term and long-term predictions, was developed on the basis of the integration of both analysis methods for ship motions and structural responses. The results of the investigations by "total system" gave useful criteria for structural design. At that time, a lot of energy in time, money, and technical personal was spent for development of programs and work of analysis due to the computers' circumstances.

1980s and 1990s

With the rapid expansion of computers in hardware and software, analysis has been increasingly used in the design stage, and methods for analysis have been also improved in the field of the evaluation of structural strength, especially on buckling and fatigue.

Recently it has been popular to carry out three-dimensional FEM analysis, where the solver and pre- and post-programs in public use have often been applied, for the typical hold part structures. In order to assess the structural reliability, these analyses have been introduced into the design stage not only for new generation ships but also for conventional ships. Figure 10 shows the typical flow of structural design and strength assessment with direct analysis methods. Moreover detailed FEM analysis with the solid elements, as well as shell/bar/rod elements, is applied to evaluate the fatigue strength in the critical structural part where necessary.

In parallel with the activities of the expanded application of analysis, the basic research has been carried out continuously in order to throw light on the difficulties of complicated ship behavior of structure in actual sea wave conditions.

Today's topics are how to superpose the fluctuating stresses with different phases at moments affected by direct wave loads and other factors encountered in wave conditions and how to estimate the initiation and propagation life of fatigue crack of various detailed structure under the actual corrosive conditions of sea water, cargo oil, and so on.

As for the former, discrete analysis method (DISAM) for hull structure response in waves has been developed by MHI and applied to the design of new-type ship structures, such as DH VLCCs, and also to damage analysis and design improvement, as was previously introduced (Watanabe, 1995).

Regarding the fatigue problems on hull structures, the R&D for the estimation of crack-initiation life and crack-propagation life has been carried out continuously due to its complicated phenomena (Kawano, 1993).

In the flow of structural strength assessment, the crack-initiation life for design use is generally estimated by Miner's Rule, as shown in Figure 11, and the crack-growth life for the prediction of the remaining structural life after crack initiation is generally estimated by the Paris Law, as shown in Figure 12, where the reference stress for estimation is calculated by detailed FEM analysis.

This is the outline of the introduction of analysis for design on hull structures at the fields of shipyards mainly in Japan.

Improvement of Structural System and Details

In reply to today's demands, as mentioned before, the improvement of structural system and details has been done on the basis of the shipyard's technology of design and construction including R&D activities. The example of the improvement on DH VLCCs is

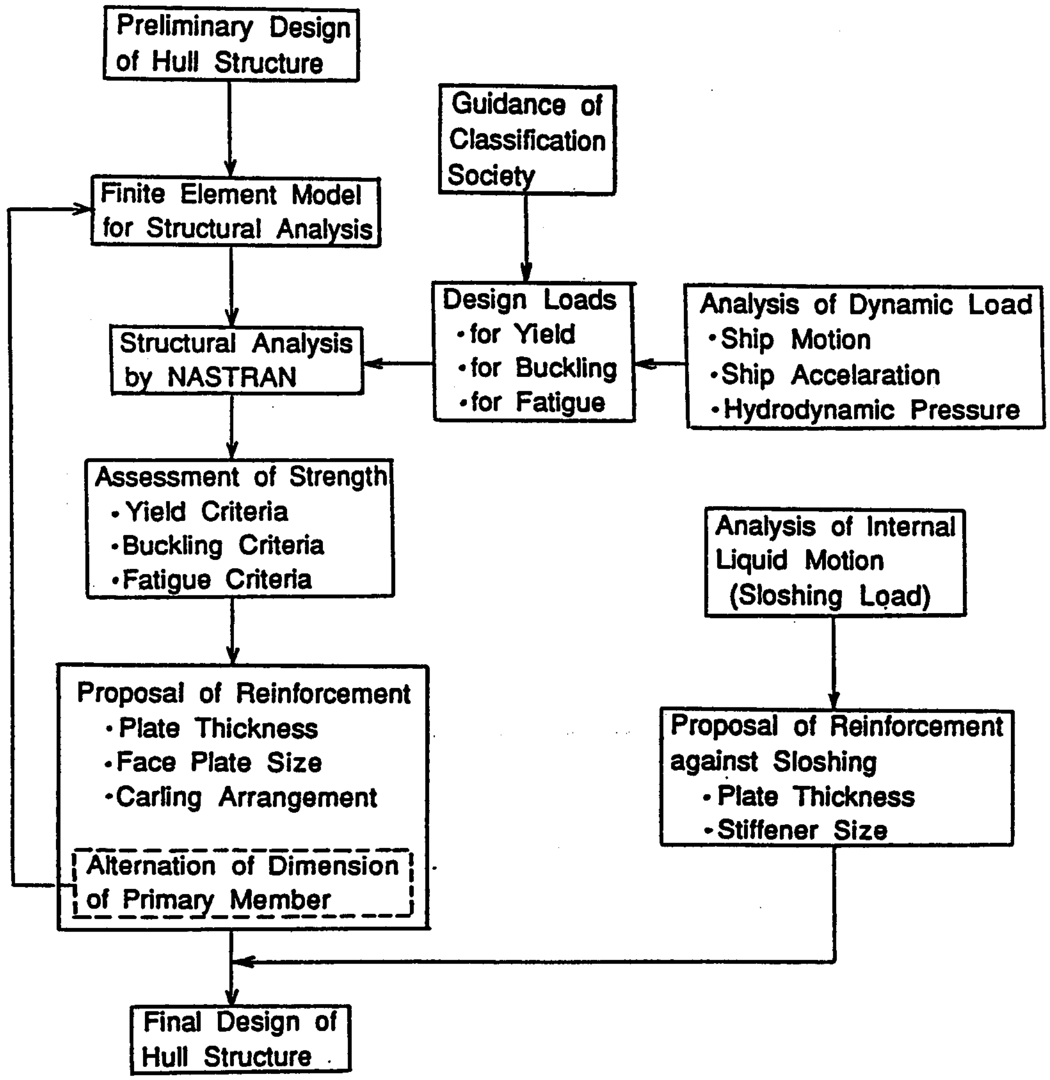


FIGURE 10 Typical flow of structural design with direct analysis.

discussed here.

One important point for design is how to arrange the DH, which is used as water ballast tanks, in size and shape. Necessary ballast capacity will give deeper depth of the DH than the minimum value required by IMO's MARPOL. DHs will generally have a hopper section at the corner of the double-sided hull and double bottom, mainly from the structural strength viewpoints.

As the depth of the DH will not be determined uniformly, some DH VLCCs will have a wider double-sided hull compared with double bottom, and others will have opposite ones. Shipyards have studied the optimization of the size and shape of DH, including the structural

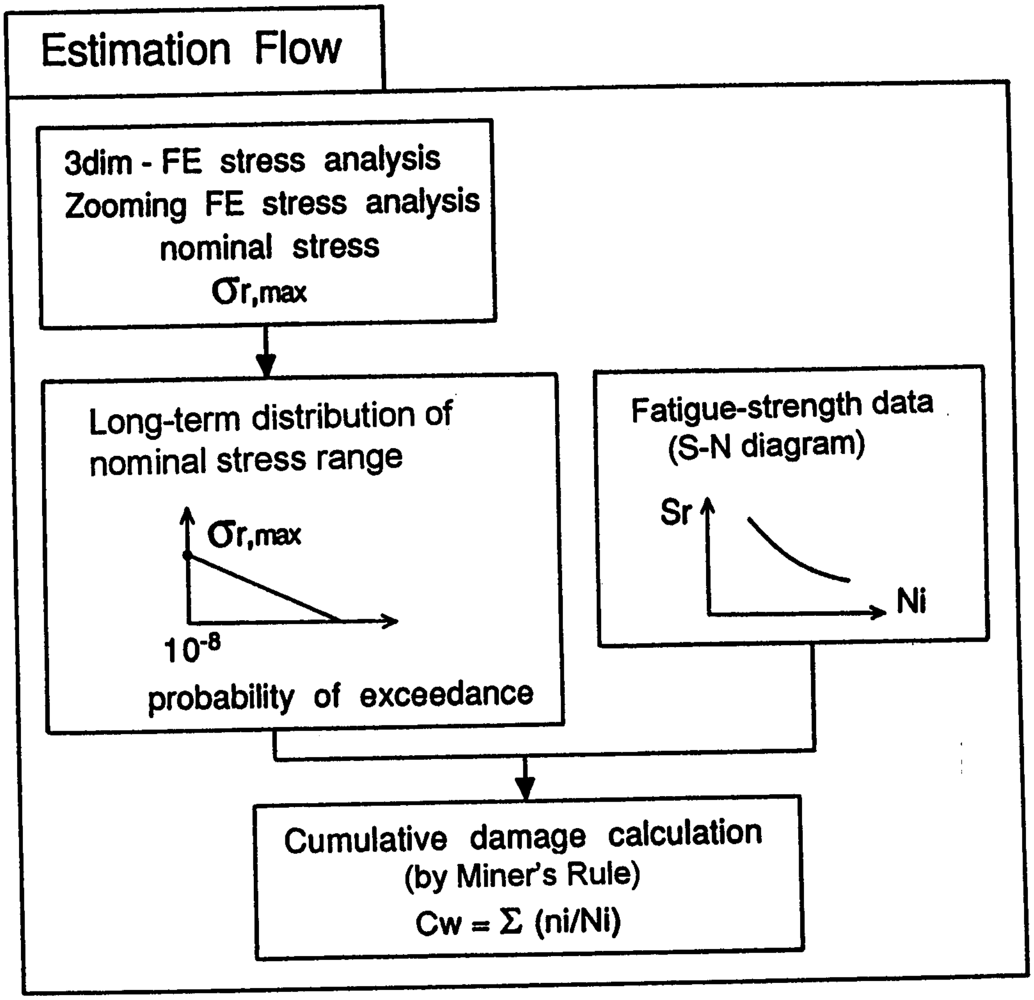


FIGURE 11 Estimation flow of crack initiation life.

system of transverse ring in center tank or side tank, from the structural-strength and fabrication viewpoints according to their design and construction considerations. Ship owners and ship operators may consider them from the inspection and maintenance viewpoints. Effective improvement on this point will be brought by both parties' cooperation.

Another important point for design and construction is how to specify the critical structural part on the fatigue fracture and how to progress the practical design of the related detail structures. For example, the inner corner of the hopper tank is one of the important structural areas from the fatigue-strength viewpoint, as shown in Figure 13.

Some papers discuss and investigate the feature of structural system and the evaluation of structural strength for yielding, buckling, and fatigue at typical and critical structural parts of DH VLCCs (Watanabe, 1994; Niho, 1994).

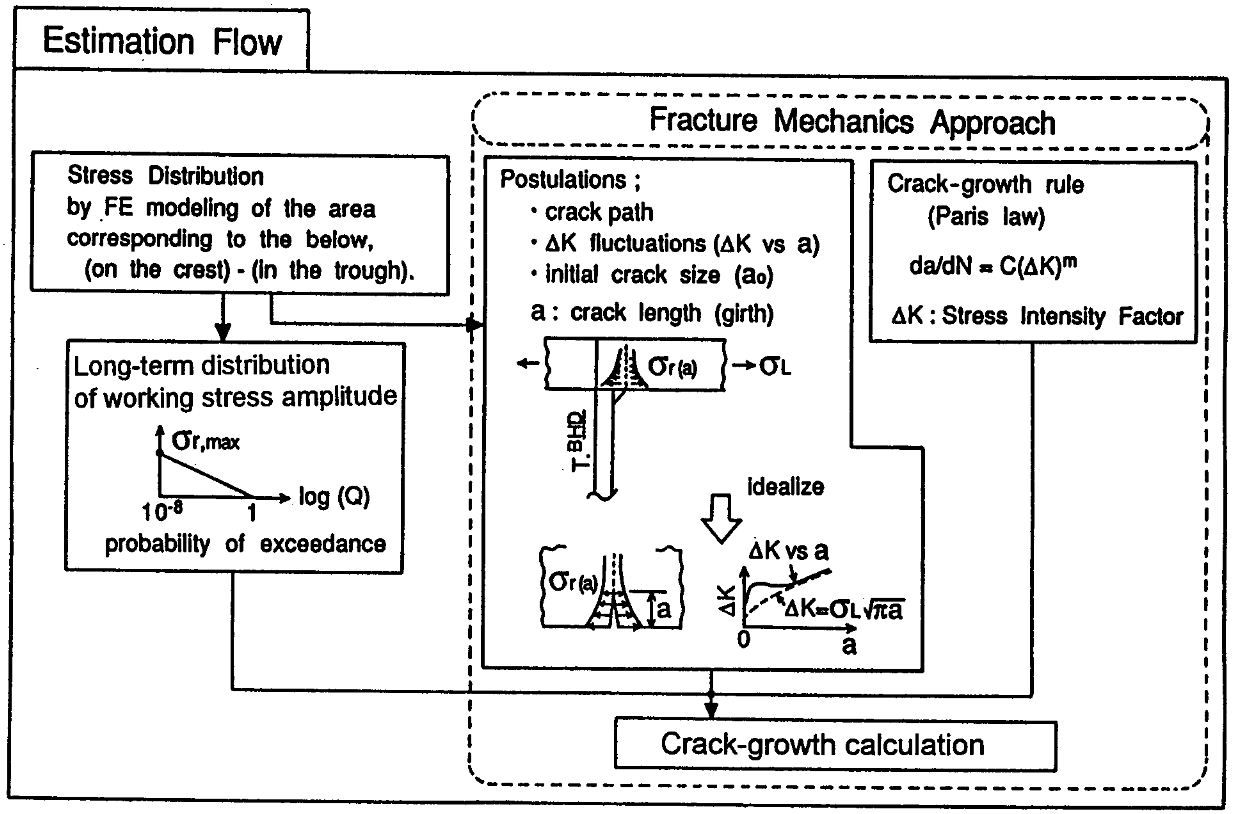


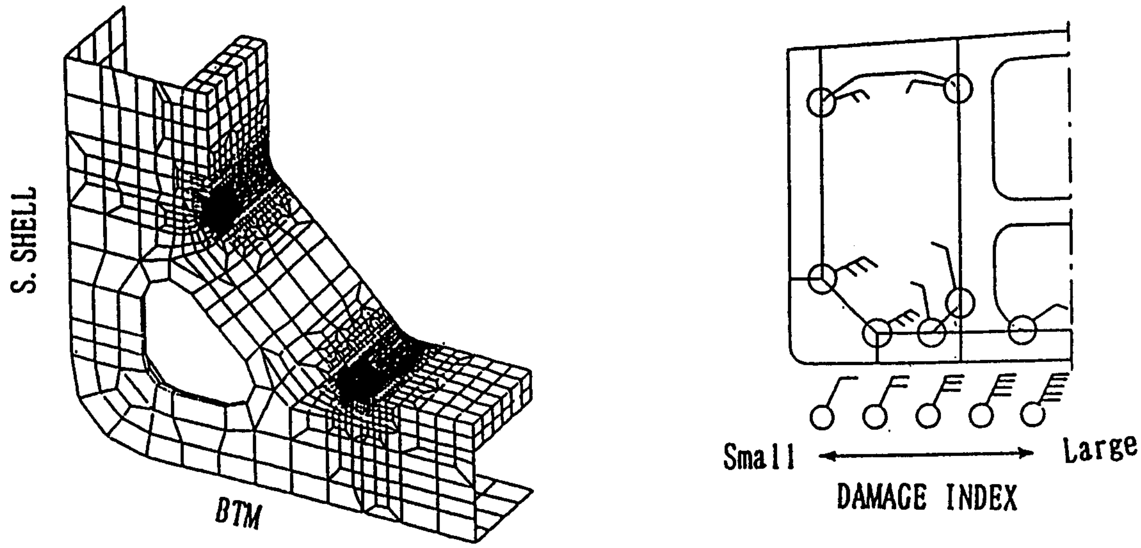
FIGURE 12 Estimation flow of crack growth life.

Regarding the specifications of critical parts in the hull structural system of DH VLCCs, including other smaller size tankers, draft general guidelines are being prepared by the TSCF, "Guidelines for the Inspection and Maintenance of DH Tankers."

Referring to the guidelines and the results of shipyards' own investigations, shipyards are making an effort for the improvement of structural details; however, there may be some difficulties in assessing the reliability of applied structural details because the occurrences of fatigue fracture in expected ship's life are caused by many factors of cumulative working stress histogram, corroded steel surface conditions, and so on, in each ship's history.

The information that shipyards have on the damage or non-damage data of various ships has been and/or will be so limited that the investigations and analysis based on the reports by operations, inspections and maintenance will be expected in order to clarify the reasonable design criteria for fatigue strength.

In the construction stage, it is very important to construct the hull structures with the appropriate control of production tolerances, especially, for the alignment of cruciform welding joints from the fatigue-strength viewpoints. For example, the recent research in Japan says that the fatigue life of a cruciform welding joint with the difference of its plate thickness



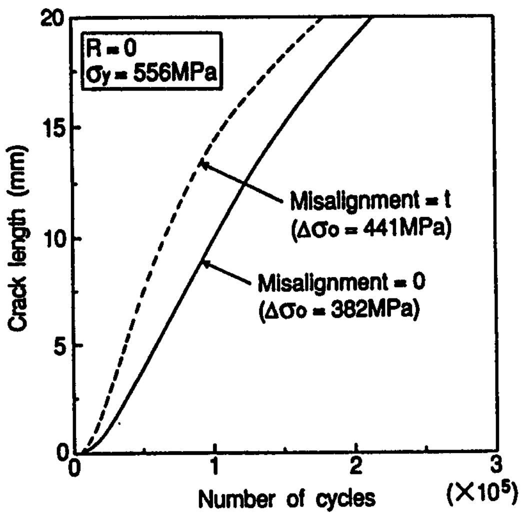
(a) Detailed FE modelling for hopper structure   (b) Example of result of fatigue investigation

FIGURE 13 Fatigue strength investigation for DH VLCC.
Source: Watanabe, 1994.

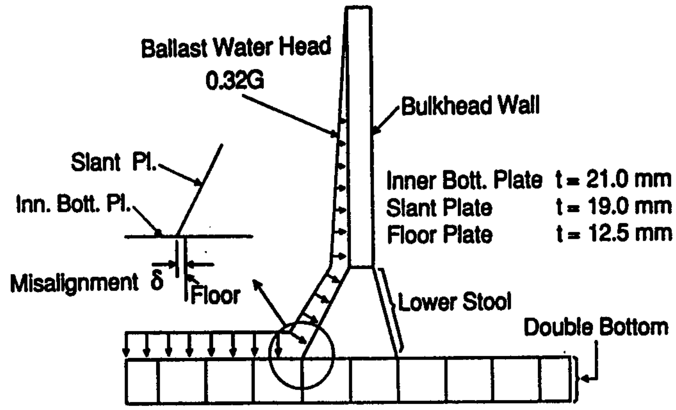
in alignment is about 60 percent of that without difference, as shown in Figure 14, through the studies carried out for the structures at the connection parts of double bottom and bulkhead stool of bulk carriers (JSRA 397). In DH VLCCS, the typical critical cruciform welding joints are located as shown in Figure 15. Corresponding to the critical level, special care is taken for alignment control.

In order to improve the structural system and details for achieving structural reliability under the practical ways, the current changes in the design and construction stage are summarized as follows:

- to increase the discussion on the extensive use of HT steel and the configuration of DH structures in the initial design stage
- to increase the design by analysis using three-dimensional FEM analysis and other necessary direct calculations
- to improve the design through shipyard's experiences of analysis on fatigue fractures of HT ships
- to rearrange the construction tolerances in accordance with the new-generation ship such as DH tankers
- to introduce some robots in addition to existing automatic equipment in the construction stage



(a) Estimated fatigue life with/without misalignment



(b) Analyzed structure (Connecting part of lower stool and double bottom of bulk carrier)

FIGURE 14 Influence of misalignment in production on fatigue strength.
Source: JSRA 397.

FUTURE DEVELOPMENTS IN THE SHIPBUILDING FIELD

Reviewing today's demands for ship fabrication and changes in the design and construction stages, some considerations for future development on hull structures in the shipbuilding fields will arise.

The first is the necessity of lifetime assessment for ships, especially from the fatigue-strength viewpoints. At present, the necessary data for assessing fatigue phenomena on actual hull structures of ships are not always arranged. Shipyards have design data but little information on the hull conditions of ships in service. Shipowner and ship operators have inspection data through maintenance but little information on the structural strength assessment. Data that both shipyards and shipowner and ship operators have are to be integrated and analyzed to improve the technology in the field of fatigue fracture.

If the lifetime assessment, composed of statistical analysis for many ships and follow-up surveys of each ship, will be introduced and established in the future, the design criteria and methodology in the shipbuilding fields will be changed to more reliable ones, and the maintenance program for each ship in the ship owner and ship operating fields will be much more optimized for expected ship life.

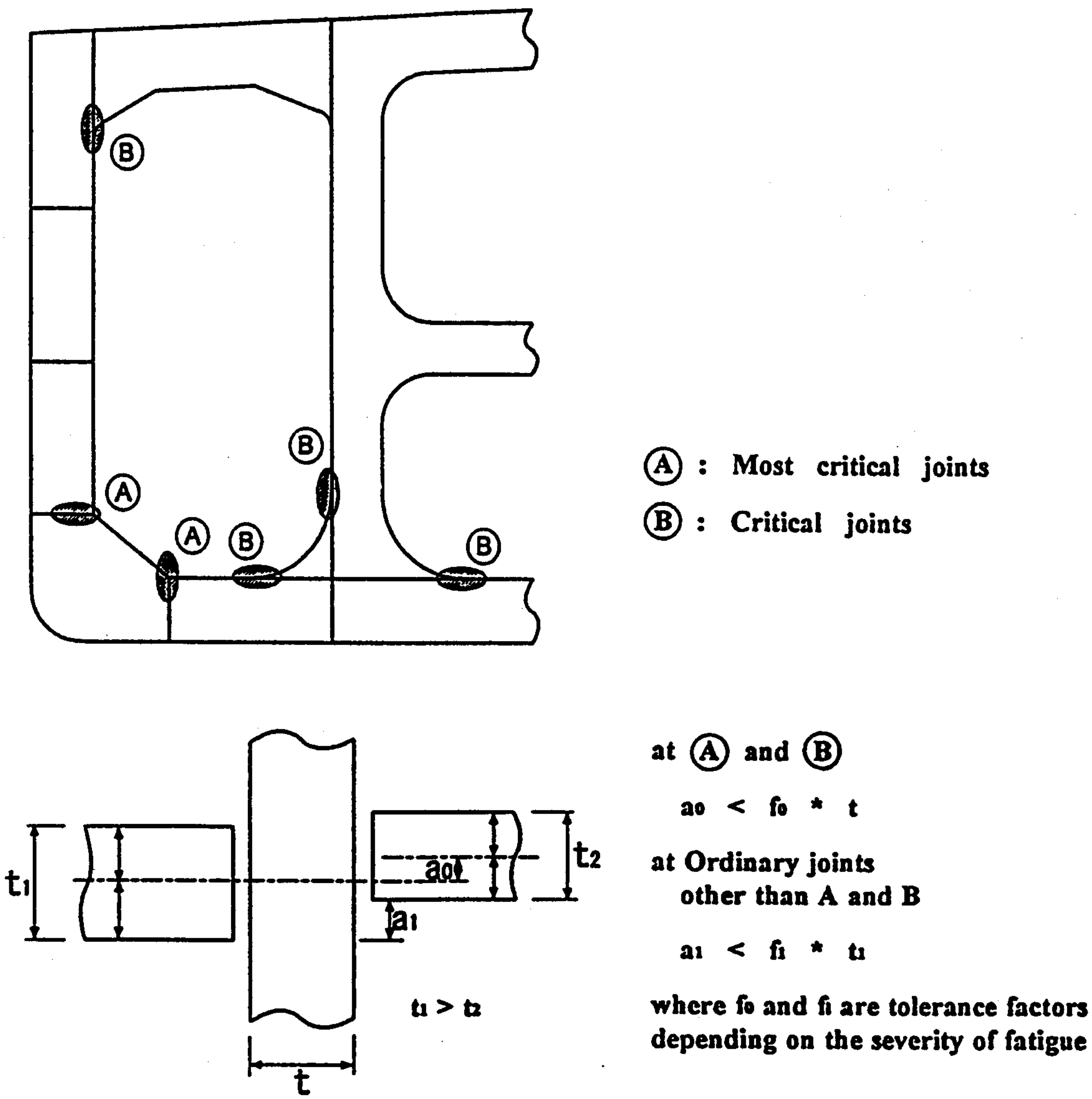


FIGURE 15 Critical cruciform joints and tolerances in alignment.

The second is the application of a hull-monitoring system and development of a diagnosis system for hull structural conditions, which will support the basic technology in the lifetime assessment. For bulk carriers, a hull-monitoring system has been already discussed by IMO's SOLAS and by classification societies because of many losses of existing bulk carriers. The function of this hull monitoring system at present is to indicate the longitudinal stresses and the ship accelerations at several typical positions in navigation or in harbor loading and unloading conditions. If the function will be expanded to save the obtained data and analyze them statistically, together with the encountered wave conditions during the period of navigation, such a system will be more effective to offer ship's history.

Using the data of cumulative stresses, by monitoring systems, and the data of aged plate thickness or structural scantlings, by inspection at periodic survey, the diagnosis for hull conditions will become feasible by means of the structural analysis.

In order to improve the structural reliability during ship's life, it would be necessary to progress these systems, where the role of classification societies will become more important.

The final consideration is the construction of a new procedure for ship fabrication. As stated in today's demands, the development of advanced CAD/CAM, robots, and CIMS are in progress in order to achieve high productivity.

In order to achieve more reliable ship structures, it would be necessary to improve the design criteria and design methodology on hull structures under the future conditions of lifetime assessment with monitoring and diagnosis system. The key is how to introduce more reasonable design criteria into design methodology.

The newly developed design methodology will have to be integrated into the advanced systems, such as CIMS, and then contribute to the construction of a new procedure in ship fabrication as satisfied with high productivity and high reliability.

REFERENCES

Funaoka, K. 1992. CIM trends and CIM concepts for shipbuilding. Welding Technique, Japan. 40(1):96–100.

Hirai, A., and H. Isoda. 1992. The tanker design for prevention of oil pollution—mid-deck tanker. Presented at the 5th International Symposium on the Practical Design of Ships and Mobile Units held May 17 1992 in Newcastle Upon Tyne, United Kingdom. London: Elsevier Publications, Ltd.

JSQS (Japan Shipbuilding Quality Standard). 1991. Japan Shipbuilding Quality Standard (Hull Part). Tokyo: The Society of Naval Architects of Japan.

JSRA 397. Research on Ship Structural Strength Assessment and Quantification of its Control Parameters. Research Report No. 397 (in Japanese). Tokyo: The Shipbuilding Research Association of Japan.

Kawano, H., and K. Inoue. 1992. A Local Approach for the Fatigue Strength Evaluation of Ship Structures. Fatigue Design '92. ESSIS 16. London: Mechanical Engineering Publications.

Kuroiwa, T., N. Nakamura, Y. Kawamoto, M. Kondo, and S. Kusuba. 1994. Study on structural damage of ships due to collision and grounding. Pp IV-416–420 in ISOPE '94 held April 10–15, 1994 in Osaka, Japan. Golden, Colorado: International Society for Offshore and Polar Engineering.

Miyazaki, T. 1990. Welding Robots for Shipbuilding. Shipbuilding Technology International: Maritime Technical Information Facility.

Miyazaki, T. 1994. Automation technologies for shipyard CIMS implementation. Presented at SINGAPORT'94 held in 1994 in Singapore.

Niho, O., S. Yanagibayashi, and T. Akashi. 1994. Structural design of double hull tankers. Pp IV-534–541 in ISOPE '94 held April 10–15, 1994 in Osaka, Japan. Golden, Colorado: International Society for Offshore and Polar Engineering.

Watanabe, E., K. Hashimoto, Y. Kuramoto, H. Kawano, and S. Tozawa. 1994. Strength evaluation of hull structure of oil tanker for prevention of oil pollution. New S-tec. Conference '94.

Watanabe, E., S. Inoue, K. Hashimoto, K. Sato, H. Sueoka. 1995. Proposal of simplified fatigue design method for side longitudinals. Journal of the Society of Naval Architects of Japan 177:391–398.

Yasuda, K. 1983. A Synopsis of High Tensile Hull Structural Steels through the Thermo-mechanical Control Process (TMCP) in Japan. Tokyo: Nippon Kaiji.

Cost-Effective Analysis for Tanker Structural Repairs

Maxwell C. Cheung

ABSTRACT

Structural failures cost operators dearly in terms of down time and repair expense. Although some fractures can be traced to material or construction defects, many can be designated as "class" or design problems. Many modifications resulting from repair analysis should be incorporated into new ship design. This paper addresses the field of repair analysis and the potential benefits to both operators and new construction design.

This paper uses an actual case study of a tanker double-bottom structure to describe the various degrees of analytical complexity associated with ship structural repairs. The results at each level of analysis are compared to indicate the accuracy at each level. The comparison is for reference only because the solutions may be sensitive to the particular geometry and loading. The comparison does demonstrate the choices available to repair decision makers when responding to a failure.

INTRODUCTION

There is ample evidence in modern tanker design that basic structural scantlings are well within classification society (i.e., American Bureau of Shipping, Lloyd's, et al.) rule requirements. Construction details usually initiate ship structural problems. If these small problems are not corrected promptly and effectively, they may soon become unmanageable.

Under the pressure of an operating schedule, many structural problems detected during unscheduled inspections receive "vee-and-weld" repairs or simple, localized reinforcement. Although these procedures are acceptable as temporary remedies, they should not be considered permanent solutions. Weld repairs do not renew the fatigue life of the parent material, and local reinforcement (such as doublers) may move the problem to the surrounding area. Prudent owners will take the precaution of analyzing the cause of class (design) failures and developing appropriate repair plans synchronized with scheduled repair availability.

With recent advances in personal computer technology, ship designers can now apply powerful finite element analytical tools to ship structural analysis (Stiehl et al., 1991; Witmer et al., 1991). Advances have occurred in both finite element model size and analytical depth. A decade ago, structural analysts would respond to a class problem with a single "worst case" analysis using a model of only a few thousand elements. Today, spectral fatigue analysis procedures are used routinely to predict the life expectancy of structural details with finite element analysis (FEA) models ten times larger than before (Liu et al., 1992; Sucharski et al., 1993; Payer, 1994).

Although comprehensive spectral fatigue analysis is still time consuming and costly in comparison to the typical repair "window," a well-planned fatigue analysis would be beneficial in the long term, as it would minimize repair costs and down time. It is, therefore, vital for the ship repair industry to use a properly simplified analysis for rapid problem diagnosis and short-term repair, eventually augmented by some form of extended benchmark or fatigue analysis to achieve a permanent solution. This paper discusses three optional levels of analysis, demonstrated through an actual sample case.

DESCRIPTION OF CASE PROBLEM

Figure 1 illustrates a typical double-bottom construction. The area of interest to this paper is boxed in phantom line and enlarged as Figure 2. The bottom longitudinals (BL) are continuous, while the frame web plating has cutouts to fit over the BLs. The BLs are connected to the frame plating by an 8 in. by 1/2 in. vertical flat bar on top and either one or two collar plates at the side to fill in the cutout. The flat bar and collars transfer the pressure-induced vertical loads from the BL to the frame plating. The class problem with this design is a horizontal crack starting at the heel of the 8-in. flat bar just above the fillet weld. A large number of these cracks were found during a recent inspection, most between 2 in. and 3 in. in length. Heavy rust was observed only around the cracked portion, indicating that the cracks grow slowly. Obvious questions included:

- Why do the cracks start at the heel?
- Why is crack propagation so stable?
- Because of the number of cracks, what is the optimal solution to prevent recurrence during the ship's economic life?

The flat-bar fractures at the heel suggest that load transfer between the bottom long to the flat bar is more tensile than bending. Since the edge at the heel is much stiffer in tension than the edge at the toe, the heel side would be the first to fail.

On-site inspection revealed that all but a few smaller cracks occurred at locations where only one collar was installed. This finding substantiated that the collars shed a significant portion of the vertical loads from the BL to the web frame. As the flat bar progressively fails and weakens, an increasing portion of the vertical load is taken by the collar(s), resulting in a reduction of flat-bar crack stress. This explains why crack growth is stable.

The on-site inspection was also key to developing an analytical "plan of attack." By observing the sharing relationship between the vertical flat bar and collar plates, it was determined that the primary fracture-inducing load was hydrostatic differential pressure across the shell boundary rather than hull-girder bending load. This determination allowed the rapid development and analysis of a simplified local stress model with idealized loading and boundary conditions.

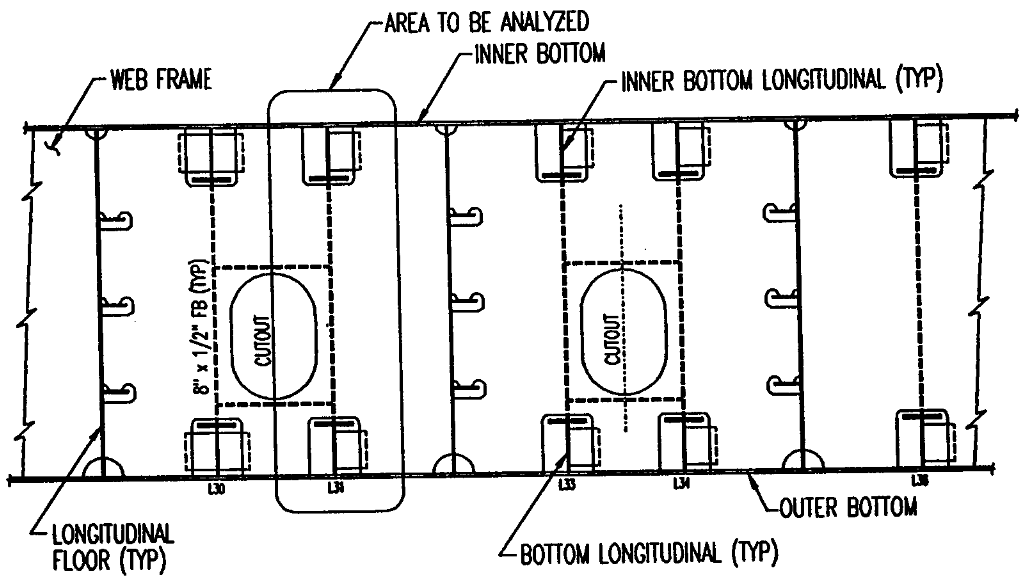


FIGURE 1 Double-bottom construction at the web frame (typical).

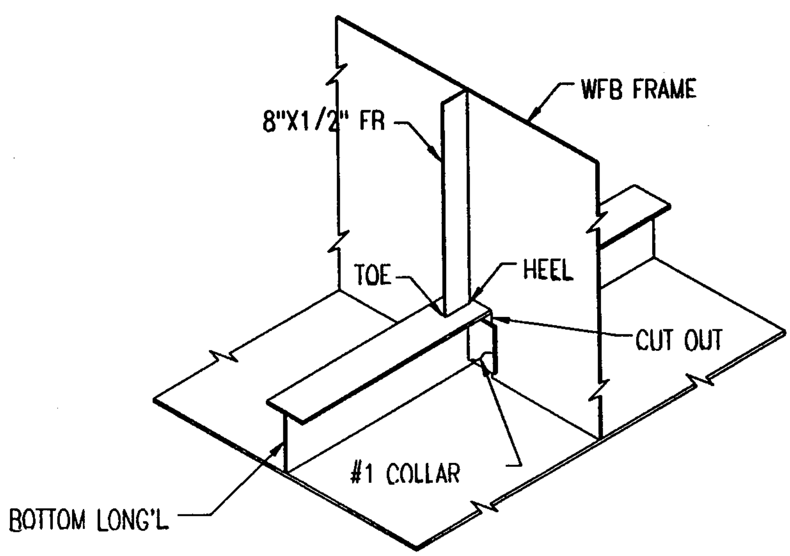


FIGURE 2 Bottom longitudinal crossing frame (typical).

Given the observed fracture pattern and probable loading mechanism, it was possible to initiate a multilevel structural analysis intended to provide both a short-term solution and a long-term, optimized solution.

ANALYTICAL APPROACH

Three general levels of finite element analysis (FEA) are frequently used by structural engineers to resolve class (design) repair problems:

- local stress analysis
- extended stress analysis
- spectral fatigue analysis

The following paragraphs describe each analysis, and their results from the case study are compared.

Local Stress Analysis

As mentioned before, the structural problems of a tanker are initiated in the small-detail region. Analyzing a given hull detail often requires as much information about the surrounding portions of the hull structure. However, a localized structure can sometimes be isolated using a one-step, detailed finite element model if the interactive information (loads and boundary conditions) can be idealized. This process permits quick diagnosis and parametric study of design alternatives.

As the quickest form of analysis, this first level involves the development of a relatively simple local model of the affected structure, analyzed with simple assumptions for boundary conditions and applied loads. If drawings or shipcheck information is available, this level of analysis can be concluded within several days. Because the local FEA model involves significant assumptions concerning boundary constraints and loads, the accuracy of such assumptions dictates the accuracy of the results.

In general, a good candidate for this type of analysis should have simple or repetitive geometry. In addition, the weaker the coupling of the global and local load, the better the results.

When developing the local FEA model mesh, some basic guidelines to follow include:

- Keep quadrilateral element shape in plane and as equilateral as possible. Avoid warping, extreme skew, and high-aspect ratios.
- Minimize mixing triangular with quadrilateral elements in the region of interest for improved stress representation.
- Minimum shell (plate) element size should be comparable to plate thickness.
- Increase mesh size gradually when moving away from the region of interest.

A section of the double-bottom, as outlined in phantom in Figure 1, was modeled using FEA, as shown in Figure 3. Centered around a typical web frame, the model extends half a frame spacing forward and aft and half a longitudinal spacing port and starboard. Since the structure is repetitive, the surrounding structure is replaced with symmetric boundary conditions imposed on the edges of the model. The six "benchmark" load cases listed in Table 1 were analyzed.

TABLE 1 Local Stress Analysis Load Cases

Ship Loading Condition	Wave Height	Wave Phase	Wave Angle
Ballast	Stillwater	--	—
	LBP/20	Hog	Head-on
	LPB/20	Sag	Head-on
Cargo	Stillwater	—	—
	LBP/20	Hog	Head-on
	LBP/20	Sag	Head-on

The stress resultants are listed in Table 2 in the "As-Built Configuration/Local FEA" column. All load cases resulted in highest stresses at the vertical bar heel, where the fractures actually started. The cargo-hog wave load case induced the highest stress level of 50.0 ksi (illustrated in Figure 4). The maximum cyclic hog-sag stress ranges for cargo and ballast conditions were 26.3 ksi and 24.3 ksi, respectively. The stress range reported here is the vertical component, S_{yy}, which is normal to the direction of crack propagation.

TABLE 2 Comparison of Local and Extended FEA Results (Von Mises, KSI)

Load Case		As-Built Configuration		Three-Collar Configuration	
				Local FEA	Extended
Ballast	Stillwater	36.4	37.6	24.8	19.5
	L/20 Hog	25.1	27.2	17.0	23.2
	L/20 Sag	49.0	58.3	33.6	32.1
Cargo	Stillwater	36.8	36.1	25.2	18.9
	L/20 Hog	50.0	56.9	34.0	31.9
	L/20 Sag	23.7	27.3	16.1	27.2

NOTE: All tabulated stresses are unaveraged element Von Mises stresses. Plotted stresses are averaged nodal Von Mises stresses.

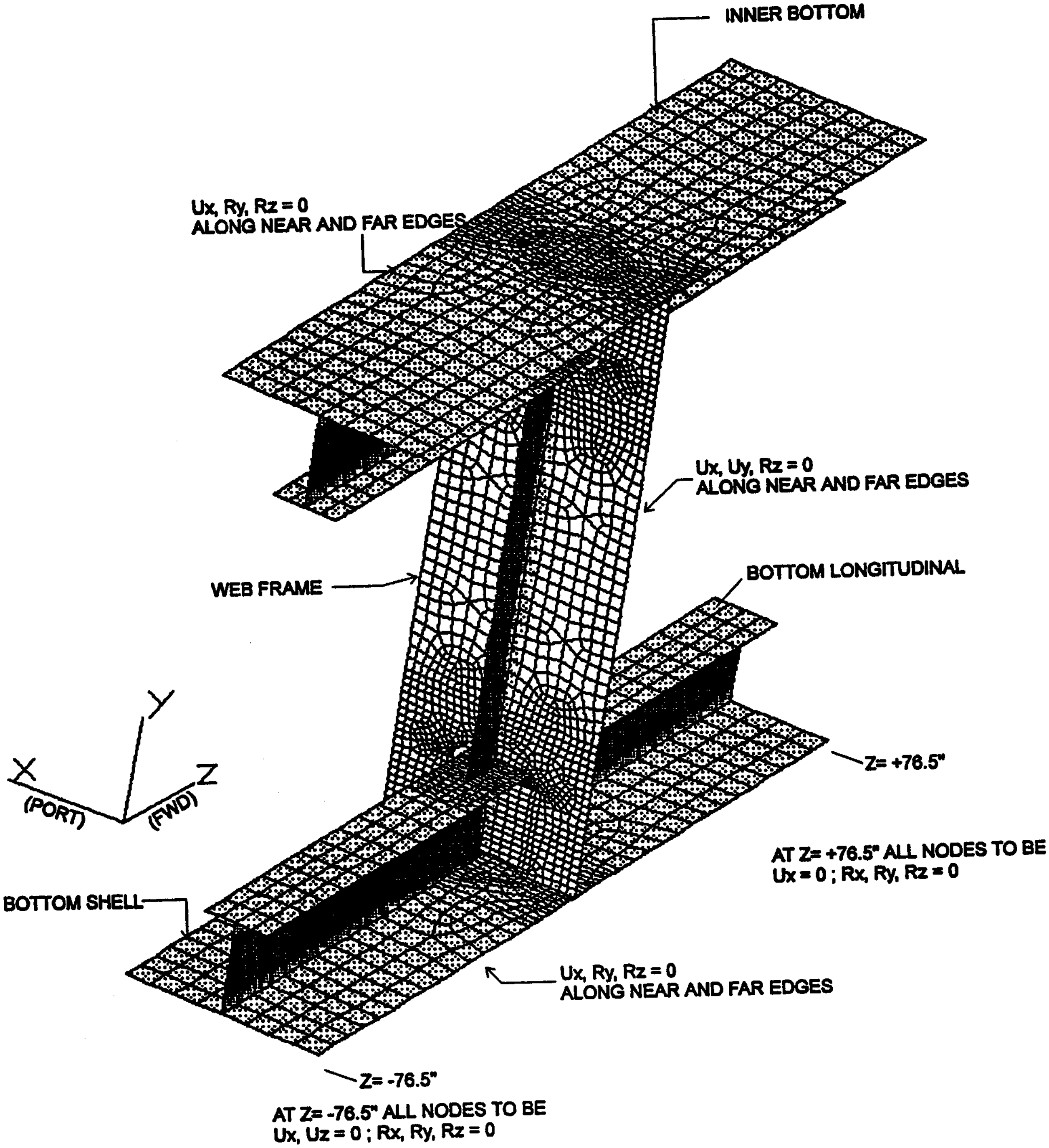


FIGURE 3 Boundary conditions for the local model.

After studying several alternative designs, one configuration emerges as the best overall solution. This is to add a third collar on top of the BL (Figure 5 and the top part of Figure 13). Although maximum stresses still occurred at the heel/longitudinal intersection, the magnitude was reduced to 34.0 ksi. Cargo and ballast stress ranges were reduced to 12.0 ksi and 11.9 ksi,

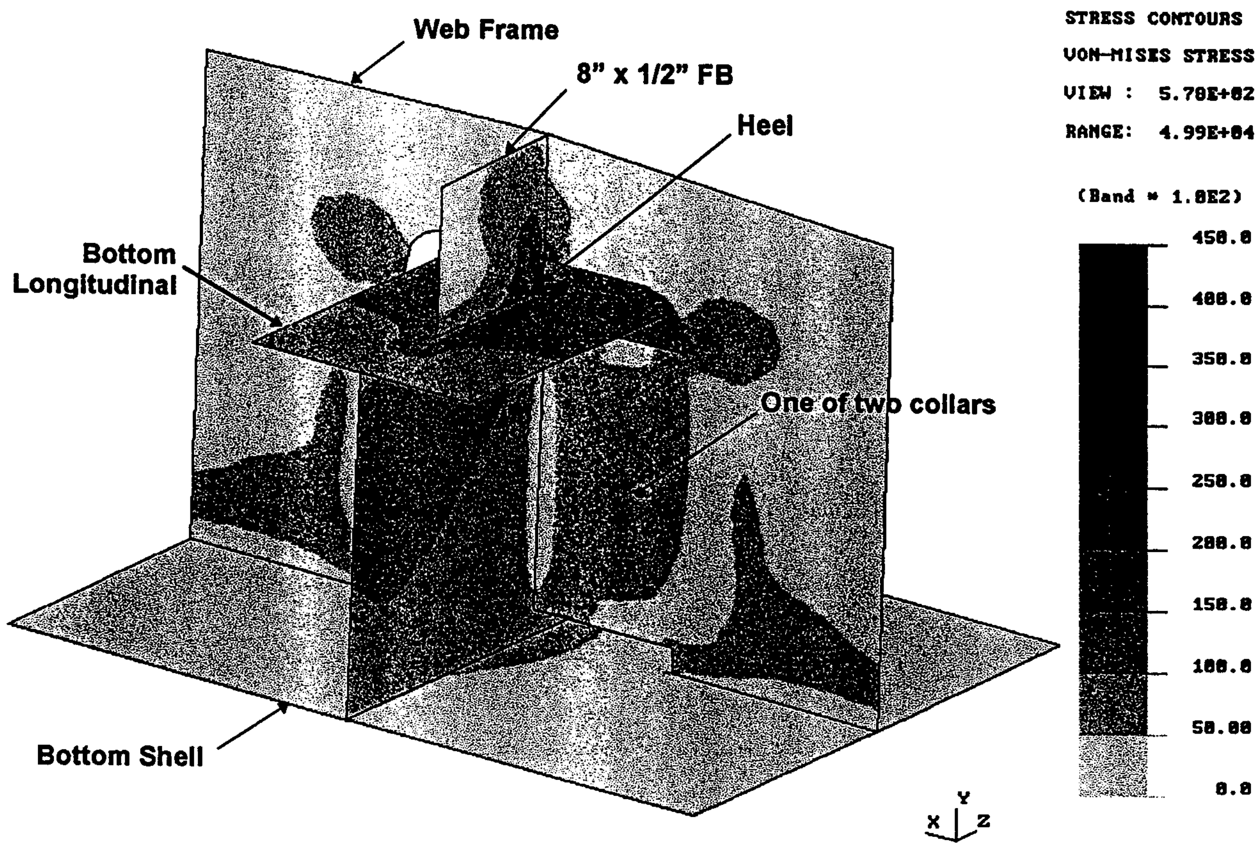


FIGURE 4 Stress contour of as-built.

respectively. Compared to the original design, the cyclic stress range reduction of 50 percent improves fatigue life by a factor of eight (based on the cubic rule of the S–N curve). Other areas now become slightly more critical in fatigue.

Extended Stress Analysis

Highly indeterminate three-dimensional structures with complex loading patterns cannot usually be analyzed effectively using local stress analysis techniques. Extended stress analysis becomes necessary to transfer the overall hull-girder bending stress distribution into the detail under review. This procedure employs a "telescoping" technique of global, intermediate, and local finite element models to transfer hull girder loads and reactions into an accurate set of loads for a local structural detail model. Although this process usually requires over a month to build the models and process several "benchmark" load cases, lead time can be greatly reduced by building the models in advance as part of a "Ready Response" program.

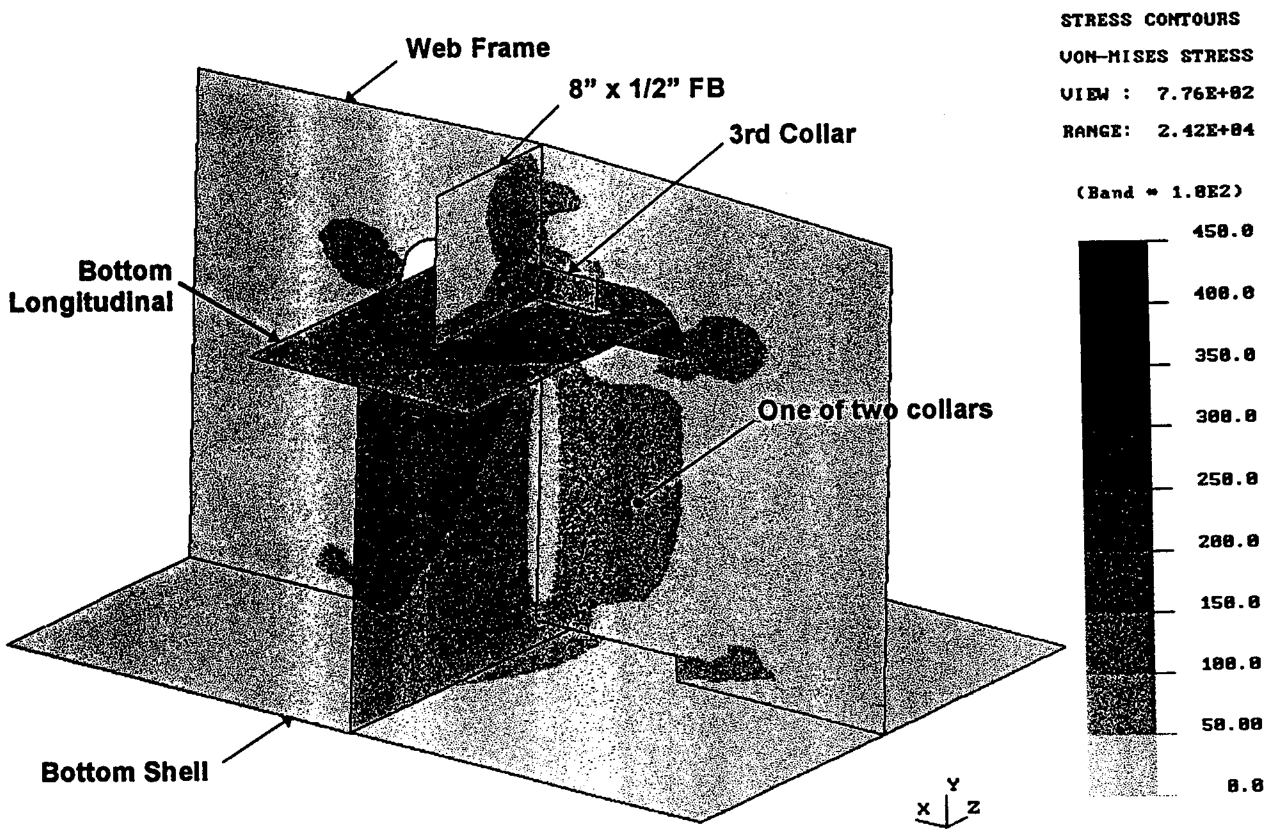


FIGURE 5 Stress contour of modified with third collar.

Because of the number of fractures, the double-bottom case was easily placed in the "class" or "design" problem category. The expense of repairing that many details justifies the cost of the extended analysis and, ultimately, fatigue analysis. Although many problems can be resolved by extended stress analysis of possible design alternatives, it is also an important first step in a fatigue analysis, since comparative results are often indicative of fatigue performance.

The extended stress analysis procedure starts with a finite element model representing a substantial portion of the hull. Although a full-hull model (Figure 6) is not mandatory, it is preferred because fewer boundary condition assumptions are required. All dynamic and static loads must be in equilibrium to achieve reliable results.

A complete stem-to-stern model requires a large number of elements. The smallest, practical element mesh is usually about one-frame spacing (over 10 ft per side). Although this mesh is sufficient to capture overall hull-girder bending stress distributions, it is obviously not sufficient to capture the intricate stress contours at the problem detail level. An intermediate model (Figure 7) with medium mesh density was created to transition the load from the global model to the final local model (Figure 8). The critical area of interest is enlarged in Figure 9.

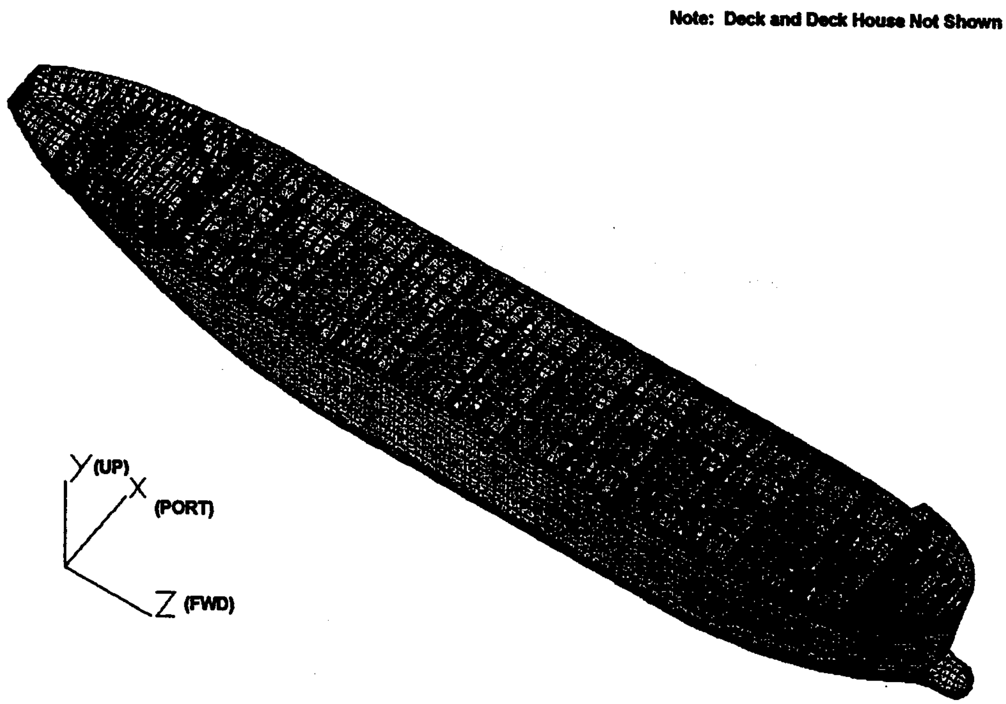


FIGURE 6 Global FEA model for the extended analysis.

The six benchmark load cases analyzed for the Local Stress Analysis were repeated for all three stages (i.e., global, intermediate, and local) of the extended stress analysis. Boundary element reactions are transferred from the larger model to the smaller model. Table 2 compares maximum stresses for the two levels of analysis, for both as-built and alternative configurations. Results for the two levels of analysis were sufficiently close to reach the same design conclusions.

Spectral Fatigue Analysis

The extended stress analysis is often used for preliminary selection of the "best" repair configuration by comparing the stress results from the extreme "benchmark" loading conditions. Calculated "stress ranges" can be used as rough fatigue-life estimates when comparing new designs against a known (damaged) as-built configuration. However, some details are sensitive to fatigue failure, where "moderate" stress levels (relative to yield) can induce fatigue, depending on weather and the route characteristics. To obtain more accurate fatigue life estimates, particularly where damage is generated mostly by moderate waves, a spectral fatigue analysis similar to the one described in Sucharski (1993) is necessary. As described in Sucharski (1993), and illustrated in Figure 10, key fatigue analysis components include structural modeling, weather, and material properties.

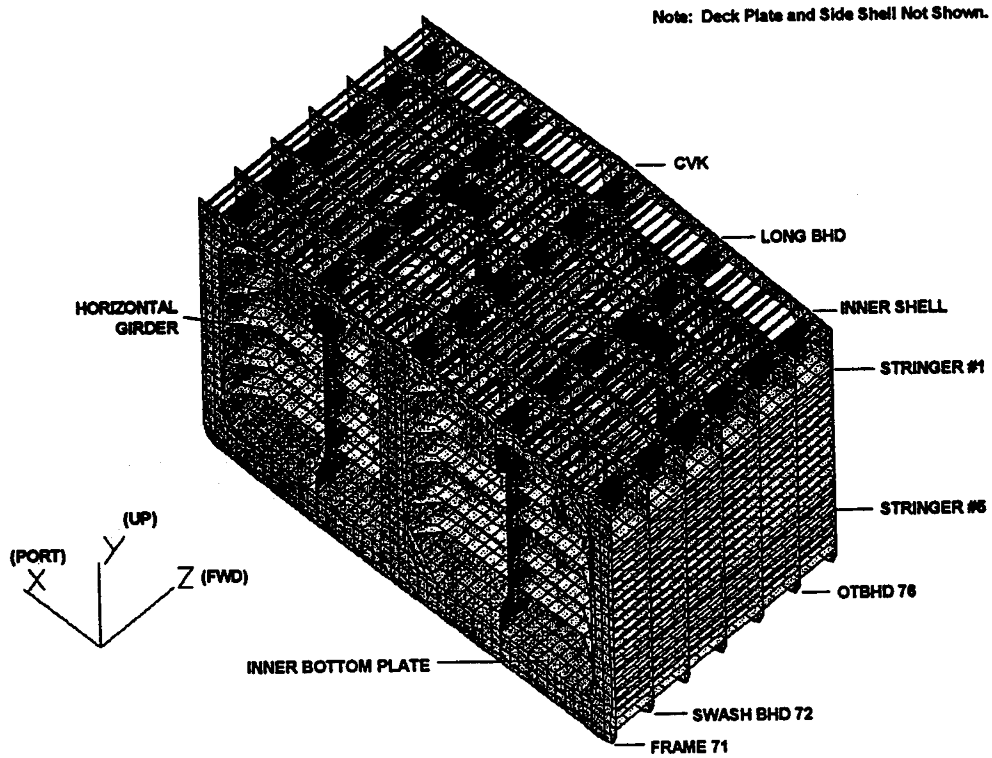


FIGURE 7 Intermediate FEA model for the extended analysis.

Spectral fatigue analysis extends the quality of the telescoping "extended stress analysis" by analyzing the telescoping models for a spectra of wave lengths, phases (hog/sag), encounter angle, and ship operating condition. This type of analysis is instrumental in identifying fatigue failures due to short-wave loading and higher-frequency, moderate-stress levels. Because of the hundreds of load cases and additional work to define sea state and voyage statistics, this process usually requires several months. Again, lead time can be greatly reduced by developing the models in advance.

Structural Modeling

The telescoping global, intermediate, and local "detailed" finite element models developed during the extended stress analysis phase are analyzed for a matrix of loading conditions and waves. Matrix dimensions include:

- Loading condition—The two or three loading arrangements typical of the majority of ship operating time must be included.
- Wave length—Ten or more wave frequencies or lengths are analyzed to accurately represent the given weather spectra.

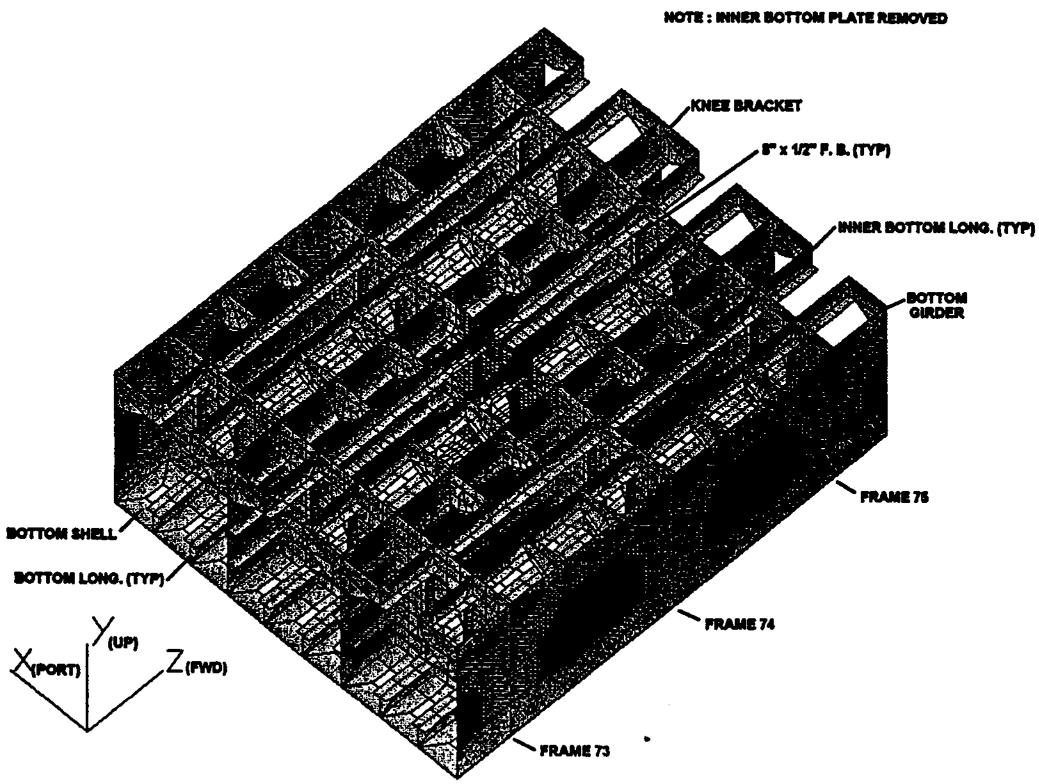


FIGURE 8 Local FEA model for the extended analysis.

- Wave incidence angle—Generally four or five wave incidence angles are necessary to describe the typical operating conditions.
- Wave phase—Both hog and sag wave positions are analyzed to allow computation of the cyclic stress range.

The stress range for each finite element for each load case is then normalized by wave height and stored as response amplitude operators (RAOs) for further processing.

Weather

A typical Trans-Alaska Pipeline Service (TAPS) trade route is shown in Figure 11. The route can be divided into zones. Monthly weather statistics based on the U.S. Navy's Spectral Ocean Wave Model (SOWM) hindcast project can then be compiled for each zone. Figure 12 shows a typical wave energy spectrum in the month of January in Zone 1363. By combining structural RAOs with route wave spectra, the cumulative damage ratios (CDRs) can be calculated.

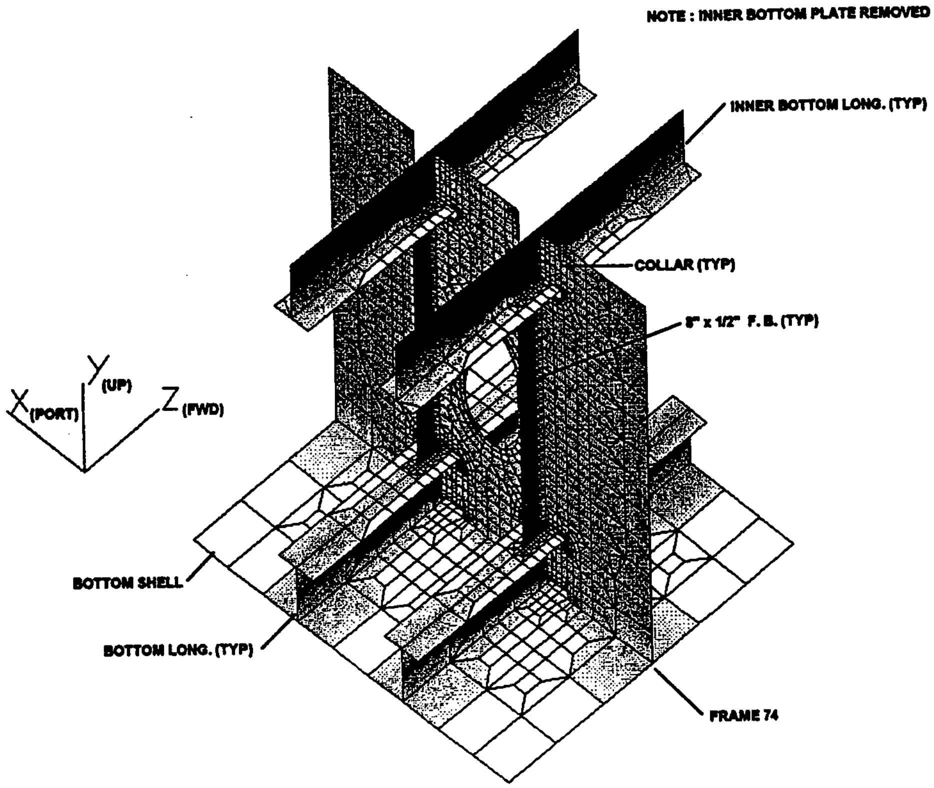


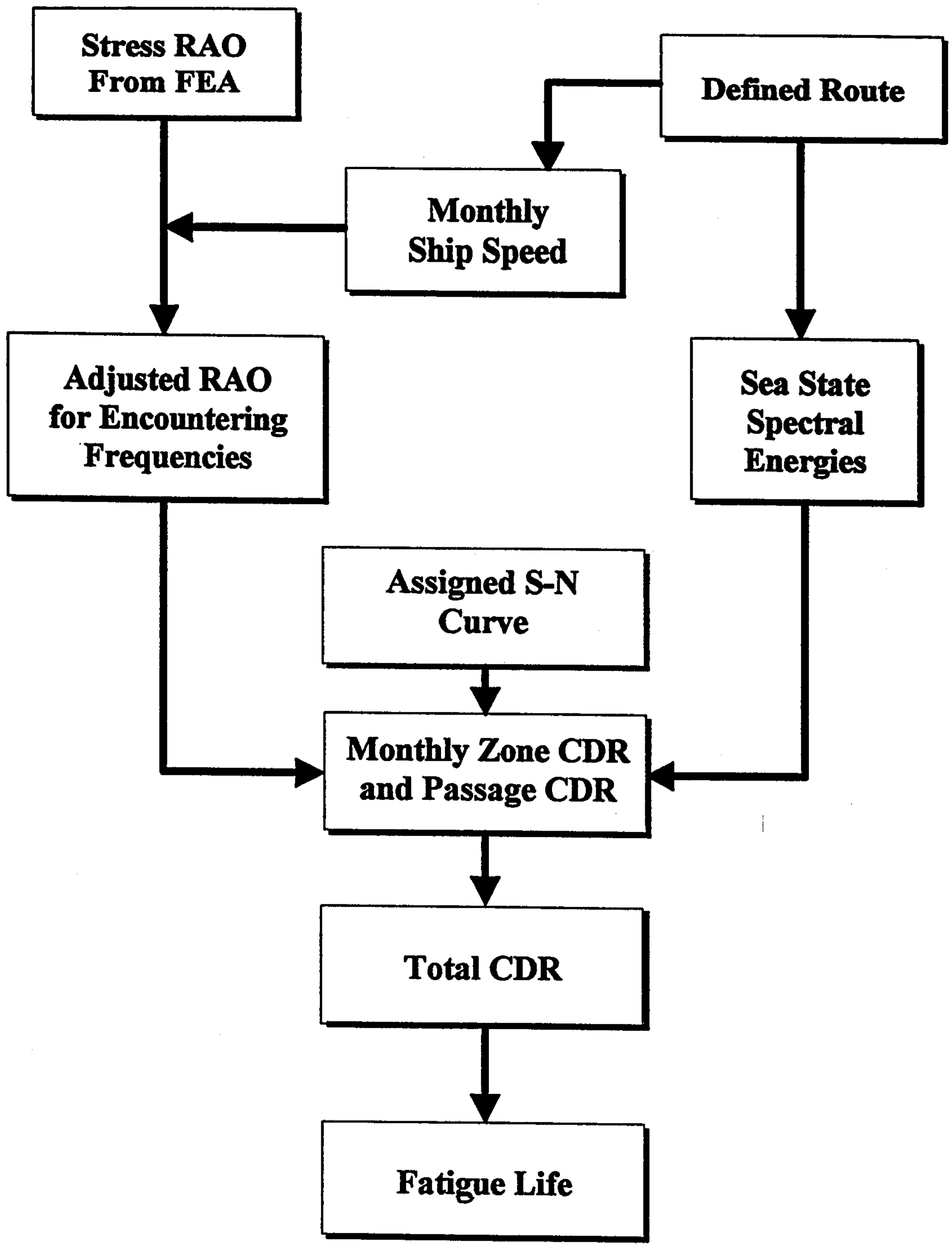
FIGURE 9 Enlargement of the critical area.

Material

The fatigue characteristics of most steels, from mild steel to high-strength steel, are quite similar. S–N curves published by the U.K. Department of Energy were used in the case study. The S–N curves define the number of cycles required to cause fatigue failure at a constant stress-range level. Most fatigue analyses use either the Miner/Palmgren technique of summing cumulative damage ratios to predict crack initiation or the Paris linear fracture mechanics equations to predict crack propagation. The case study used the Miner approach, using the U.K. "C" and "D" curves (depending on whether a weld abuts the element) since those curves are most appropriate for finite-element "hot-spot" stress calculation.

Fatigue Results

Table 3 summarizes fatigue results for the critical flat-bar "heel" element. Several conclusions can be drawn from the table. First, the cumulative damage sustained for the four years of service is 0.47. If service conditions and weather patterns are extrapolated into future years, the total life expectancy for the as-built detail is about eight years. Second, the damage



RAO: response amplitude operator
CDR: cumulative damage ratio

FIGURE 10 Fatigue analysis flow chart.

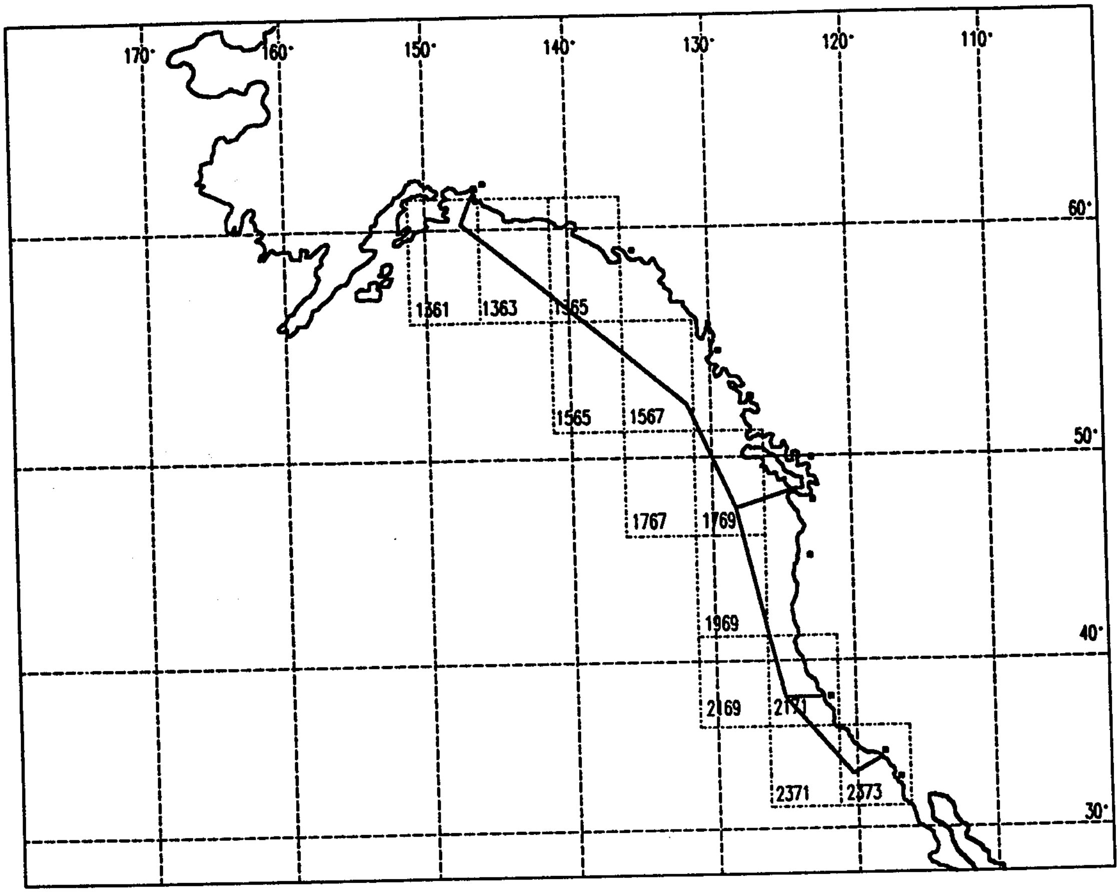


FIGURE 11 Map of trade routes with geographic zones.

sustained during the winter months is much greater than during summer. Finally, the damage sustained during the ballast legs is somewhat higher than for the cargo legs. The alternate design has about half the cyclic stress range, and should, therefore, have a significantly longer fatigue life (eight times, based on a cubic S–N curve) than the original design. In this instance, fatigue results closely parallel results from the local stress analysis and the extended stress analysis. Note that this will not always be the case.

CONCLUSIONS

As demonstrated in the case study, simplified local FEA can provide a reasonably accurate answer for short-term ship repairs if used properly. However, more complex structures and loading patterns cannot be analyzed reliably with this technique.

Extended stress analysis and fatigue analysis are far more accurate, but generally require more time than is available during an unscheduled repair period. The general procedure in this scenario is to "vee and weld," but extended analysis should be performed to identify a proper

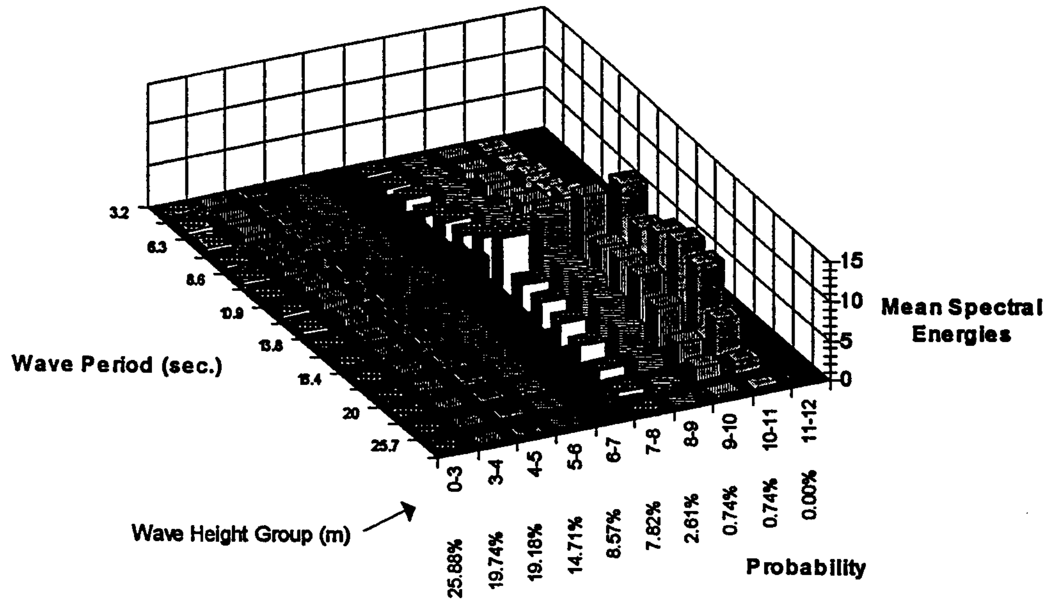


FIGURE 12 New design versus repair option.

permanent fix. Although a full, comprehensive fatigue analysis can be costly and time consuming, it could be quite economical compared to repair by trial and error.

The recommended third-collar repair developed during the case study described herein can easily be adapted to new construction using a design similar to that shown in the lower part of Figure 13. This is a typical example of how repair "lessons learned" can be fed back into new ship design to reduce overall life cycle costs and improve performance.

The schedule problems associated with extended and fatigue analyses can be overcome through a proactive approach of analyzing the structures before failures occur and by having a repair plan ready for the time when the fractures do occur. The cost of material in a well-planned repair exercise is usually a small portion of the total yard cost, which includes staging and downtime.

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TABLE 3 Fatigue Damage Summary

MODEL: BTM STR: As Built
ELEMENT: 6188

DATE ######
S–N CURVE: D

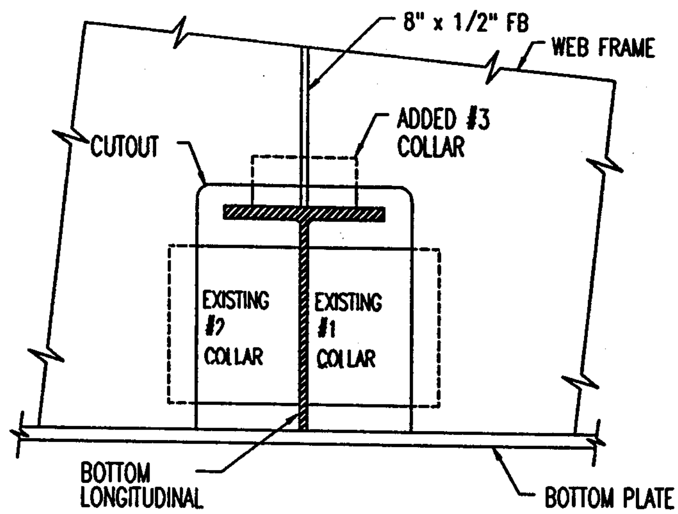
STRESS: Principal Stress
TABLE FOR MONTHLY CDR CALCULATION

MONTH	San Francisco to Valdez			Valdez to San Francisco			Seattle to Valdez			Valdez to Seattle			San Francisco to Seattle			Seattle to San Francisco			Los Angeles to San Francisco			San Francisco to Los Angeles		
	Trips	PCDR	MCDR	Trips	PCDR	MCDR	Trips	PCDR	MCDR	Trips	PCDR	MCDR	Trips	PCDR	MCDR	Trips	PCDR	MCDR	Trips	PCDR	MCDR	Trips	PCDR	MCDR
Jan	4	.0064	.026	4	.0043	.017	3	.0047	.014	5	.0031	.016	1	.0029	.003	3	.0018	.005	3	.0004	.001	4	.0002	.001
Feb	5	.0048	.024	0	.0028	.000	4	.0033	.013	7	.0020	.014	0	.0021	.000	3	.0011	.003	3	.0007	.002	1	.0003	.000
Mar	6	.0044	.026	3	.0025	.008	3	.0030	.009	6	.0018	.011	0	.0021	.000	4	.0010	.004	2	.0007	.001	4	.0003	.001
Apr	7	.0034	.024	0	.0018	.000	0	.0029	.000	8	.0016	.013	0	.0011	.000	6	.0006	.004	3	.0003	.001	2	.0001	.000
May	3	.0021	.006	1	.0010	.001	4	.0014	.006	6	.0007	.004	0	.0011	.000	3	.0005	.002	1	.0004	.000	1	.0001	.000
Jun	6	.0020	.012	1	.0009	.001	2	.0014	.003	7	.0008	.006	1	.0009	.001	4	.0003	.001	2	.0003	.001	1	.0001	.000
Jul	4	.0009	.004	3	.0003	.001	2	.0005	.001	2	.0002	.000	1	.0005	.001	3	.0002	.001	4	.0003	.001	5	.0001	.001
Aug	3	.0007	.002	3	.0003	.001	3	.0005	.002	3	.0002	.001	0	.0004	.000	0	.0001	.000	3	.0002	.001	3	.0001	.000
Sep	4	.0016	.006	1	.0007	.001	2	.0012	.002	2	.0006	.001	1	.0006	.001	2	.0002	.000	3	.0002	.001	1	.0001	.000
Oct	1	.0023	.002	2	.0014	.003	4	.0019	.008	6	.0012	.007	0	.0008	.000	2	.0003	.001	0	.0002	.000	2	.0001	.000
Nov	7	.0056	.039	3	.0031	.009	1	.0044	.004	4	.0023	.009	1	.0024	.002	3	.0013	.004	6	.0005	.003	5	.0002	.001
Dec	5	.0056	.028	3	.0032	.010	3	.0040	.012	4	.0024	.010	1	.0026	.003	3	.0013	.004	3	.0007	.002	3	.0002	.001
	55		.199	24		.052	31		.074	60		.092	6		.011	36		.029	33		.014	32		.005

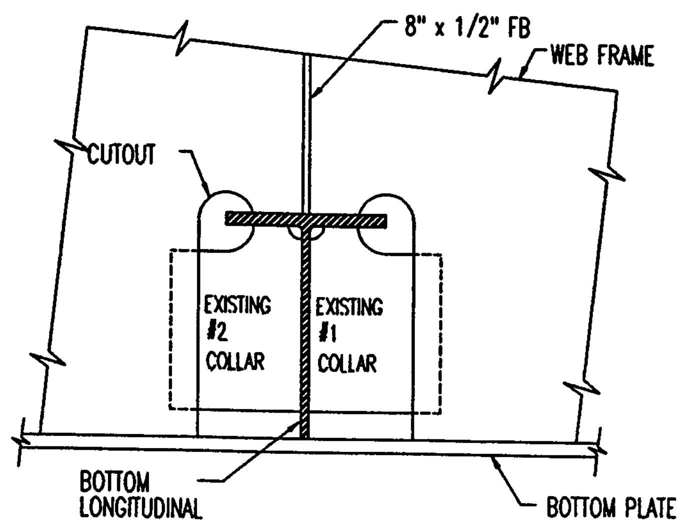
TRIPS: Total number of trips during four years
PCDR: Passage cumulative damage ratio
MCDR: Monthly cumulative damage ratio

Total CDR to Date = 0.476

Fatigue Life = 8.40 Years



Alternate Repair Plan



New Design

FIGURE 13 Typical wave spectral energy for Zone 1363 (January).

REFERENCES

Liu, D., J.S. Spencer, T. Itoh, S. Kawachi, and K. Shigematsu. 1992. Dynamic load approach in tanker design. Society of Naval Architects and Marine Engineers Transactions 100:143–172.

Payer, H.G., and W. Fricke. 1994. Rational dimensioning and analysis of complex ship structures. Society of Naval Architects and Marine Engineers Transactions 102:395–418.

Stiehl, G.L., G.G. Kojayan, and M.C. Cheung. 1991. The structural renovation of a liquefied gas carrier. Presented at the Society of Naval Architects and Marine Engineers West Coast Tanker Operations Symposium held October 10–12, 1991 in La Jolla, California.

Sucharski, D., and M. Cheung. 1993. Application of spectral fatigue analysis to the ARCO 190 MDWT tankers. Presented at Ship Structures Symposium '93 Ship Structure Committee and Society of Naval Architects and Marine Engineers held November 16–17, 1993 in Arlington, Virginia.

Witmer, D.J., and M.C. Cheung. 1991. The use of finite element modeling techniques in the investigation of structural failures in ships. Presented at the Society of Naval Architects and Marine Engineers West Coast Tanker Operations Symposium held October 10–12, 1991 in La Jolla, California.

Fatigue and Fracture Control in the Aerospace and Power Generation Industries

D. O. Harris

ABSTRACT

Approaches to fatigue and fracture control in the aerospace and electric power generation industries are reviewed, with emphasis on the use of fracture mechanics. Fracture mechanics considers the behavior of a dominant crack that has either initiated in service or was preexisting. The applicability of fracture mechanics to a wide variety of industries is emphasized. Fracture control in the aerospace industry is based on the fracture mechanics analysis the growth of assumed preexisting cracks of a size related to inspection detection capabilities. The fracture mechanics bases and the material properties involved are reviewed. Applications of fracture mechanics in the electric power generation industry, such as nuclear pressure vessels, steam turbine rotors, and the like, are reviewed. Probabilistic approaches, based on fracture mechanics with some of the inputs being considered as random variables, are mentioned, and an example of an application of probabilistic fracture mechanics to ships is reviewed.

INTRODUCTION

The purpose of this paper is to review the approaches taken for the control of fatigue and fracture in industries other than the marine industry. The emphasis here is on the use of fracture mechanics, which concentrates on the growth of a dominant crack. This area is also reviewed elsewhere in this symposium (see Reemsnyder), and detailed background information is not provided. Emphasis is, however, made regarding the applicability of fracture mechanics to a wide variety of industrial problems. An alternative approach to the problem is provided by the more conventional fatigue approach, which does not consider an explicit crack but concentrates on material behavior described in terms of cycles to "failure" for a given cyclic stress (or strain). This approach will not be discussed herein, but the fact that fatigue and fracture problems in marine and other structures occur mostly at welds and welds are notorious for containing crack-like defects (see Maddox), suggests a strong commonality between these two approaches when applied to welded structures. The fracture mechanics approach requires detailed stresses at the weld, but can, with proper information, account for the effects of geometry on crack growth and final instability. Approaches to fracture control in the power generation and aerospace industries are reviewed. Such approaches are most often deterministic, but probabilistic approaches are gaining in use, and examples of probabilistic analyses are provided—including one to marine structures. Reliability approaches are reviewed in this symposium by Wirsching.

REVIEW OF FRACTURE MECHANICS

Fracture mechanics considers the growth and stability of a dominant crack in a structure. The basic procedures are widely applicable, being suitable to a wide variety of industries and materials. Fracture mechanics was initially developed for linear elastic materials, which is the situation of most interest in marine structures. The technology has been extended to other material conditions, including plasticity and creep, and has been applied to a wide variety of materials, including metals, plastics, ceramics, concrete, and rock. Figure 1 summarizes the basic components of a deterministic fracture mechanics analysis of lifetime. The material behavior (e.g., linear elastic, plastic, creep, etc.) determines the relevant crack driving force, which is a measure of the stresses near the crack tip and the strain-energy release rate associated with crack growth. The relevant crack driving force for linear elastic materials is the stress intensity factor, K. The value of K is dependent on the crack size, stress level, and geometry of the cracked body. The "K-solution" can be obtained from handbooks, such as Tada (1985) and Murakarni (1987), and the basis of linear elastic fracture mechanics is the dependence of the crack behavior on the value of the applied K. Figure 2 provides an example of a K-solution for a cracked configuration of interest in the fracture mechanics analysis of welded ship structures. The value of K for the semi-elliptical crack shown at the top of Figure 2 varies with position along the crack front. The plots at the lower part of Figure 2 provide the values of F_T and F_B, which describe the value of K at the deepest point of the crack. A K-solution such as this is used in conjunction with material properties to analyze how a crack will grow under service loadings. The material properties needed include the subcritical crack growth characteristics, such as fatigue or stress corrosion crack-growth rate, and the toughness associated with crack instability. Numerous books have been written on fracture mechanics, and the literature is voluminous. The following are selected books on the subject: Kanninen and Poplar (1985), Broek (1978), Barsom and Rolfe (1987), and Anderson (1991).

In many instances, the inputs needed for a fracture mechanics analysis are not known with great certainty or may be random in nature or exhibit considerable scatter. In such situations, a conservative estimate of lifetime or failure load can be made using worst case conditions. This can lead to overly pessimistic results because conservatisms are stacked on one another. A different approach is to characterize the randomness and scatter of the inputs and perform a probabilistic analysis of lifetime (or failure load). The loads (and stresses) on marine structures are random in nature and naturally call for such an approach. Probabilistic fracture mechanics can be used in such situations and is based on deterministic fracture mechanics, with some of the inputs considered as random variables. Provan (1987) and Harris (1985; 1995) provide reviews of this area. Probabilistic fracture mechanics are naturally applied to marine structures because of the statistical nature of the loadings. The influence of inspection procedures and times of inspection on the reliability can be naturally treated by the use of probabilistic fracture mechanics,

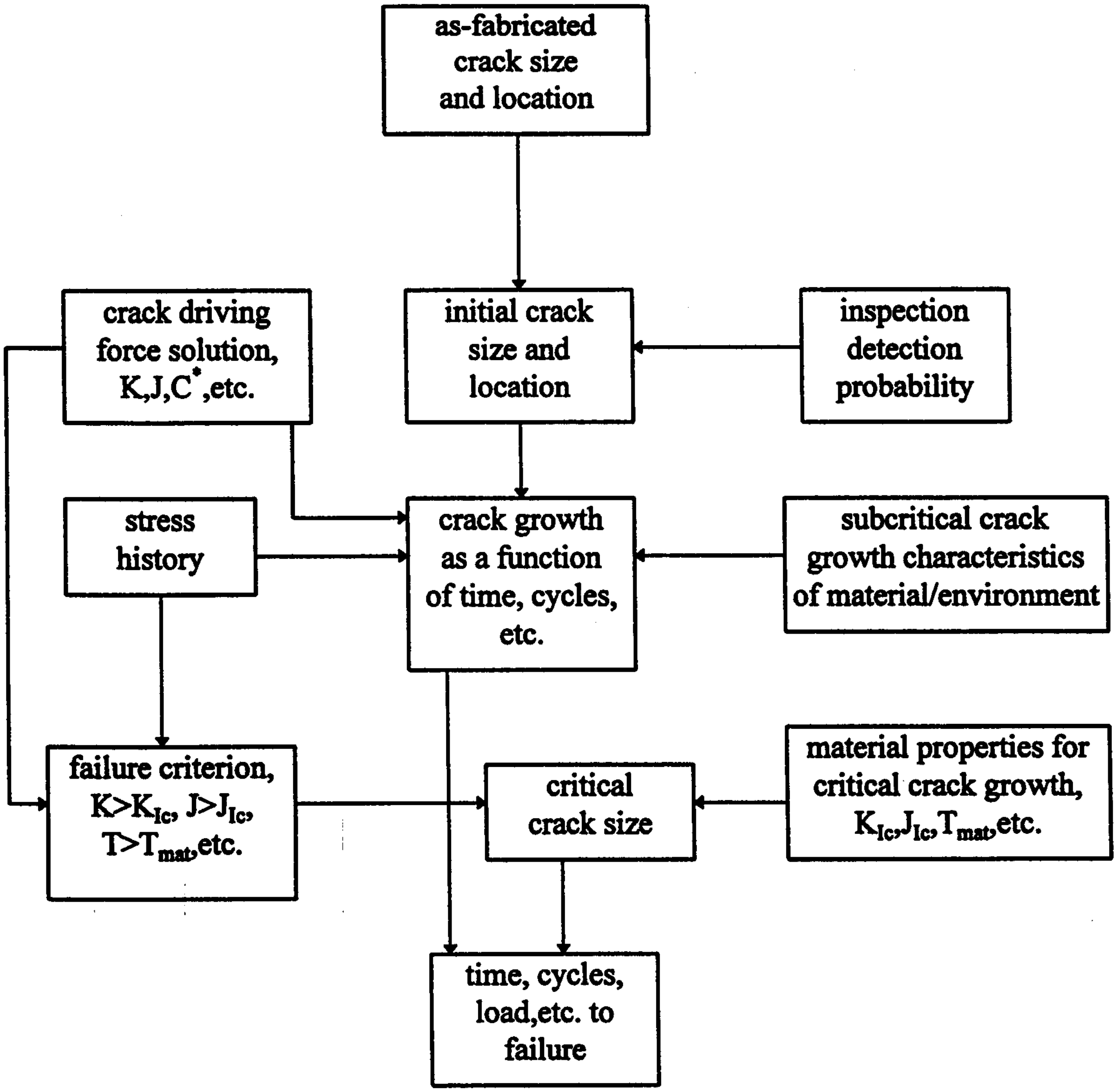


FIGURE 1 Basic components of a deterministic fracture mechanics model for prediction of crack growth and instability.

through the probability of detecting (and correctly dispositioning) a crack as a function of its size. This is an additional advantage of the probabilistic approach.

EXAMPLES OF FRACTURE CONTROL

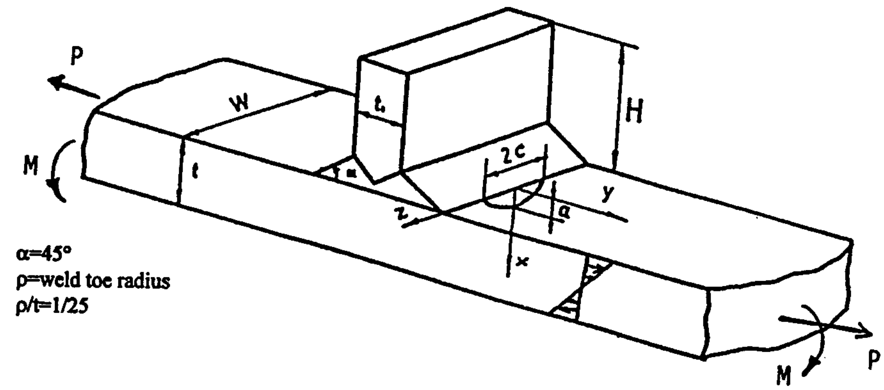
Various approaches have been taken to fracture control, with the following discussion providing selected examples.

Space Structures

Fracture control procedures for safety critical structures in space applications are specified by the National Aeronautics and Space Administrations (NASA) (NASA, 1988), with details for pressurized systems covered in MIL 84. The NASA requirements are applied to all pay loads in the space shuttle, as well as life/mission-critical items in space applications, such as the space station. These procedures call for a fracture mechanics analysis of the component using as an initial flaw size the size of a crack that would be found with a 90 percent probability (at the 95 percent confidence level). This is referred to as the nondestructive examination (NDE) size. A component passes the fracture control requirements if a flaw of this size can be demonstrated to survive four lifetimes. Median material properties are used in the crack growth calculations. Tables of the NDE size for various inspections and flaw configurations are provided (NASA, 1988; Forman et al., 1993), and smaller sizes can be considered if a better detection capability can be demonstrated for the particular inspection to be used on the component. Fracture mechanics crack-growth software is provided by NASA and is available from the Cosmic Code Center in Athens, Georgia (Forman et al., 1993). The FLAGRO code includes the NDE sizes for various inspection procedures, along with default median fatigue-crack–growth and toughness properties for approximately 400 aerospace materials. A very wide selection of crack/body geometries and loadings are included in the FLAGRO code, so that it is straightforward to perform the necessary fracture control analysis for many applications. Table 1 provides a summary of the NDE flaw sizes, and Figure 3 includes an example of the fatigue-crack–growth data and curve fits included in the FLAGRO software. FLAGRO has been available since 1986, and the supporting material database is under continuing development. The software currently runs on a personal computer with DOS, and a WINDOWS version is planned (Forman, 1994).

Aircraft Structures

Aircraft structures were one of the earliest applications of fracture mechanics and continue to be one of the major areas of application. This is because of the high required reliability and severe weight penalties for overly conservative design. Probabilistic analyses are occasionally used, because of the randomness of initial flaws and loading spectra. Various chapters in Provan (1987) discuss damage tolerance analysis of military aircraft structures, with strong emphasis on probabilistic aspects. The military approaches taken are known as "damage tolerance" and "fail safe." They are discussed in Vlieger (1988), with the Air Force Aeronautical Systems Division (AFASD) (AFASD, 1974), Federal Aviation Administration (FAA) (FAA, 1978a; 1978b), and Wood and Engle (1979) providing details. Palmberg et al. (1987) include additional discussion, with probabilistic considerations.



$$K = \left[\frac{P}{Wt} F_T + \frac{M}{Wt^2} F_B \right] \frac{2}{\pi} \sqrt{\pi a}$$

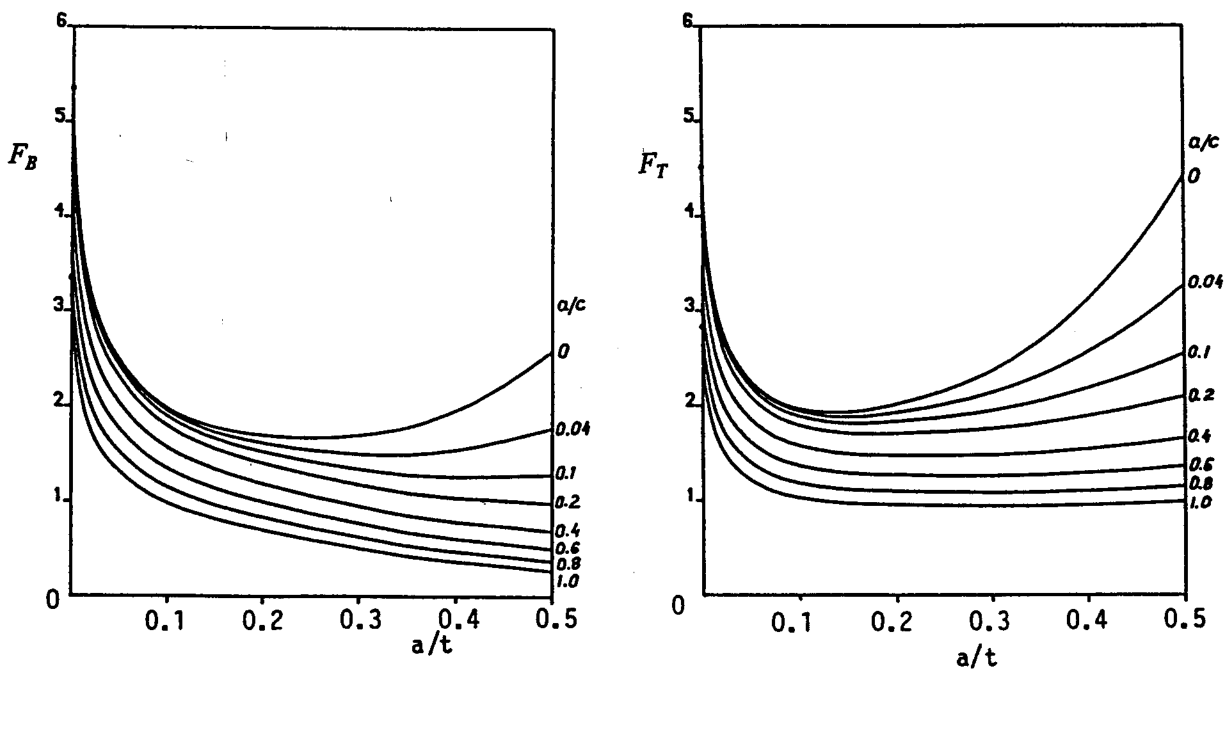
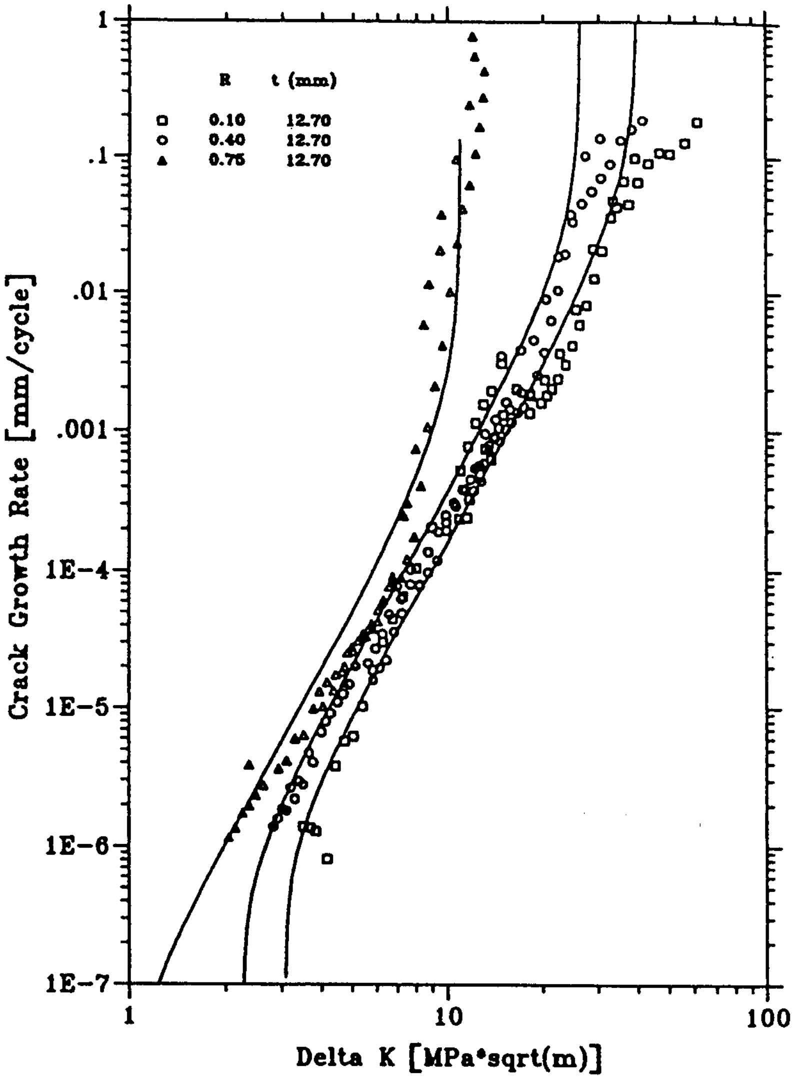


FIGURE 2 Stress intensity solution for a semi-elliptical crack at a weld toe.
Source: Murakami, 1987.



crack growth in mm/cycle, K in MPa-m^{½}

C_F=1.6x10^{-7} α'=1.90 S=0.3
p=0.5 q=0.75 n_f=3.635
ΔK_o=3.3 MPa-m^{½} K_c=25.2 MPa-m^{½}

1 ksi-in^{½}=1.0988 Mpa-m^{½}

FIGURE 3 An example of fatigue crack growth data and curve fit in FLAGRO materials compilation, 6063-T5 aluminum alloy.

TABLE 1 Summary of NDE Sizes Used as Assumed Initial Crack Size for NASA Fracture Analysis (Sheet 1)

Crack Case	NDE Inspection Technique or Flaw Size Criterion	Thickness Range (inches)	Crack Size (inches)[a]	
			a	c
TC01, TC06, TC07, TC08 (open surface)	EC	t ≤ 0.050	--	0.050
	P	t ≤ 0.050	--	0.100
	P	0.050 < t ≤ 0.050	--	0.15-t
	MP	t ≤ 0.075	--	0.125
TC02 (edge)	EC	t ≤ 0.075	--	0.100
	P	t ≤ 0.100	--	0.100
	MP	t ≤ 0.075	--	0.250
TC03, TC04, TC05, TC09 (hole)	EC	t ≤ 0.075	--	0.100
	P	t ≤ 0.100	--	0.100
	MP	t ≤ 0.075	--	0.250
	HPD-driven rivet	any thickness	--	0.005
	HPD-other holes	t ≤ 0.050	--	0.050
EC01	R	0.025 ≤ t ≤ 0.107	0.35t	0.075
	R	t > 0.107	0.35t	0.7t
	U	t ≥ 0.300	0.35t	0.065
CC01 (edge)	EC	t > 0.075	0.075	0.075
	P	t > 0.100	0.100	0.100
	MP	t > 0.075	0.075	0.075
	U	t > 0.100	0.100	0.100
CC02, CC03 (hole)	EC	t > 0.075	0.075	0.075
	P	t > 0.100	0.100	0.100
	MP	t > 0.075	0.075	0.075
	U	t > 0.100	0.100	0.100
	HPD-not driven rivet	t > 0.050	0.050	0.050
SC01, SC02, SC03 (open surface)	EC	t > 0.050	0.020	0.100[b]
			0.050	0.050[c]
	P	t > 0.075	0.025	0.125[b]
			0.075	0.075[c]
	MP	t > 0.075	0.038	0.188[b]
			0.075	0.125[c]
	R	0.025 ≤ t ≤ 0.107	0.7t	0.075
		t > 0.107	0.7t	0.7t
	U	t ≥ 0.100	0.030	0.150[b]
			0.06	0.065[c]

NOTES:

EC = eddy current
P = dye penetrant
[a] 1 in. = 25.4 mm
R = radiographic
U = ultrasonic
[b] minimum crack depth
MP = magnetic particle
HPD = hole penetration defect (max)
[c] maximum crack depth

Source: Forman (1993).

TABLE 1 Summary of NDE Sizes Used as Assumed Initial Crack Size for NASA Fracture Analysis (Sheet 2)

Crack Case	NDE Inspection Technique or Flaw Size Criterion	Thickness Range (inches)	Crack Size (inches)[a] a	c
SC04, SC05	EC (ext & int)	t > 0.050	0.020	0.100[b]
			0.050	0.050[c]
	P (ext)	t > 0.075	0.025	0.125[b]
			0.075	0.075[c]
	MP (ext)	t > 0.075	0.038	0.188[b]
			0.075	0.125[c]
	R (ext & int)	0.025 ≤ t ≤ 0.107	0.7t	0.075
		t > 0.107	0.7t	0.7t
	U (ext & int)	t ≥ 0.100	0.030	0.150[b]
			0.06	0.065[c]
SC06	EC (ext & int)	t > 0.050	0.020	--
	P (ext)	t > 0.075	0.025	--
	MP (ext)	t > 0.075	0.038	--
	R (ext & int)	0.025 ≤ t ≤ 0.107	0.7t	--
	U (ext & int)	t ≥ 0.100	0.030	--
SC07	Ec	--	Eq 17,18	0.050
	P	--	Eq 17,18	0.075
	MP	--	Eq 17,18	0.125
SC08 (rolled threads)	P	--	Eq 17,18	0.075
SC09, SC10 (machined threads)	max. machining defect size	--	thd. depth +0.005	--

NOTES:

EC = eddy current	R = radiographic	MP = magnetic particle
P = dye penetrant	U = ultrasonic	HPD = hole penetration defect (max)
[a] 1 in. = 25.4 mm	[b] minimum crack depth	[c] maximum crack depth

Source: Forman (1993).

The major purpose of a damage tolerance analysis is to ensure structural safety throughout the life of a structure (AFASD, 1974). This is achieved by the following steps:

- thorough identification of all critical areas in the structure
- analysis of the fatigue-crack-growth under realistic loadings in critical areas under the assumption that flaws exist in the structure at the beginning of the service life
- nondestructive periodic inspections of areas found to be fatigue critical in the fatigue-crack-growth analysis
- investigating the effects of accidental damage that might occur during the service life and verifying that the structure can withstand this damage during a period of time long enough to pursue operational service until the next inspection or, in the case of obvious damage, until the current mission is completed

The calculated fatigue life is strongly dependent on the assumed initial flaw size. The initial flaw size to be used is mandated to be a small imperfection equivalent to a 5-mil (0.005-in. or 0.127-mm) radius corner flaw that exists at each hole of each element in the structure. The imperfection is taken to be in the most unfavorable orientation relative to loadings and material properties. Figure 4 depicts flaw sizes and locations to be assumed at other types of structural details. The material properties to be used in the fatigue analysis, such as fracture toughness and fatigue-crack–growth rates, are summarized in the *Damage Tolerance Handbook* (Wood and Engle, 1979). Inspection procedures and frequencies are based on the results of the fatigue analysis. For instance, it is specified that a fatigue crack may not grow to failure in two times the inspection interval.

Fatigue-crack–growth analyses require the use of computer software. There is no code in place for this application that is equivalent to FLAGRO for NASA applications. Vlieger (1988) discusses some of the software in use at his organization, which is the National Aerospace Laboratory in The Netherlands.

Electric Power Generation

The most extensive effort in fracture control in the electric power generation area is in the commercial nuclear power industry, where fracture mechanics has been used extensively used in the analysis of the nuclear reactor pressure vessel and piping. The requirement for extreme reliability and the prohibitive cost of full-scale testing (as used in the aircraft industry) necessitates use of analytical procedures for prediction of defect behavior in components. The American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code, which dates back many decades, describes procedures for the design of pressure boundary components in general, with Section Ill pertaining to nuclear power applications. Allowable stresses are defined for various materials, which are based on tensile and fatigue properties. The code concentrates on the design procedure, with little consideration of problems that may arise during subsequent operation. Due to the need to maintain a very high level of reliability of the pressure boundary in commercial reactors, Section XI of the code was developed to cover in-service nondestructive inspection intended to detect cracks before they grow to produce a failure. The locations to be inspected and procedures to be used are defined. If a crack is found, procedures for analyzing its future behavior are described in Appendix A of Section XI (ASME, 1989).

The necessary material properties are provided. Procedures for defining the crack size for the fracture mechanics analysis in terms of the defect indications are provided, and the stress intensity factor solutions are included. Very extensive materials testing programs over several decades provided the material information required for definition of default values that are provided in the code. The testing and analysis development efforts are ongoing. Figure 5 provides the default values of the fracture toughness for crack-growth initiation (K_{Ic}) and arrest of a running crack (K_{Ia}) included in the code. These are conservative bounds. The fatigue-crack–growth properties are also provided and are also being steadily updated. Figure 6 includes the default fatigue-crack–growth properties to be used in the analysis. Once again, these results are based on very extensive testing performed by many investigators over many years. In order to reduce the need for a fracture mechanics analysis of every detected crack in piping and pressure

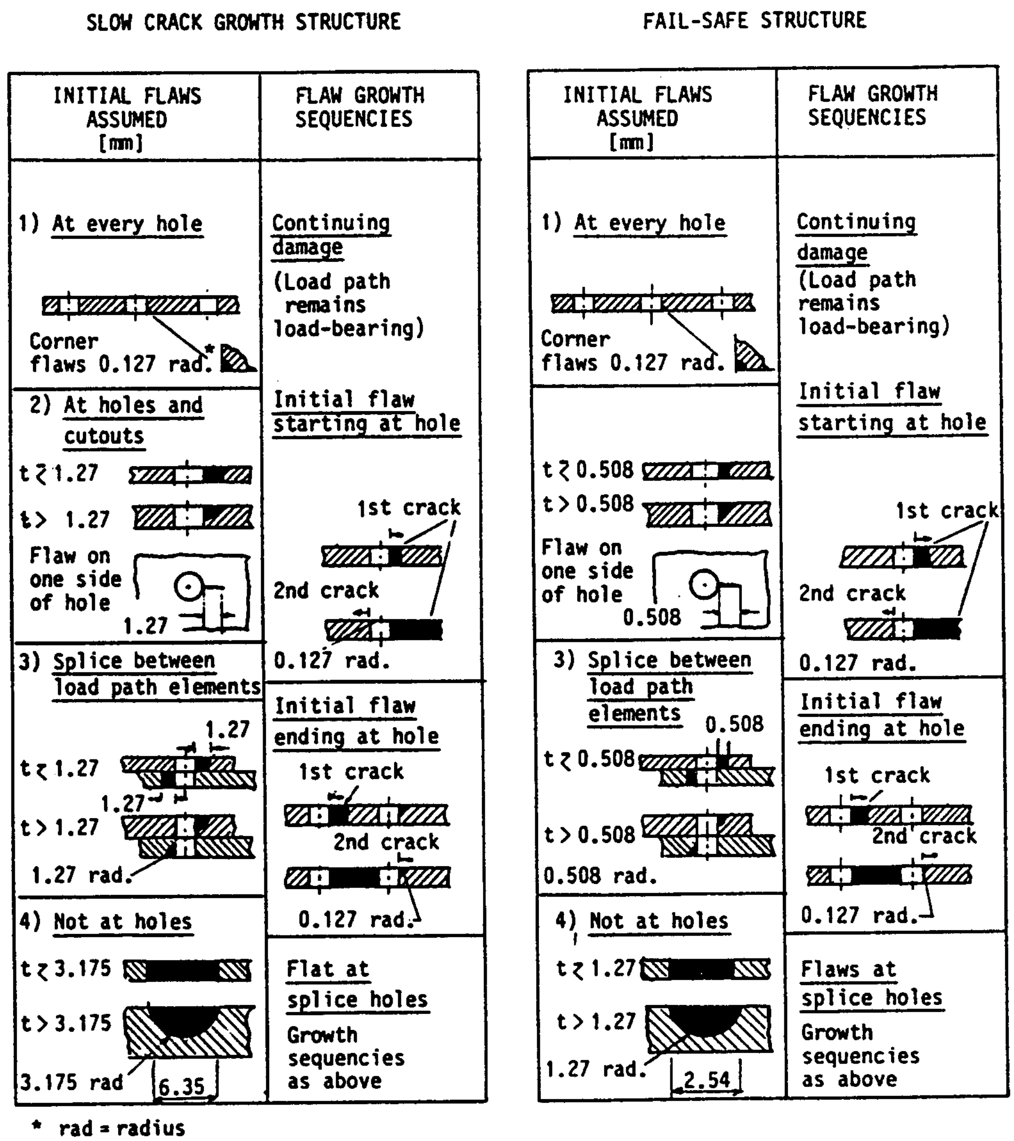


FIGURE 4 Initial flaw size assumptions for damage tolerance analysis of aircraft structures. Source: Palmgren, 1987.

vessels, the code provides tables of crack sizes that need not be further analyzed if the detected size is smaller. These tables are based on bounding calculations, using the default material properties along with stresses that are at the code allowable limits. Cracks larger than these tabulated values can still be left in service based on a more detailed analysis that shows them not to grow beyond a specified fraction of the critical crack size in the remaining desired lifetime.

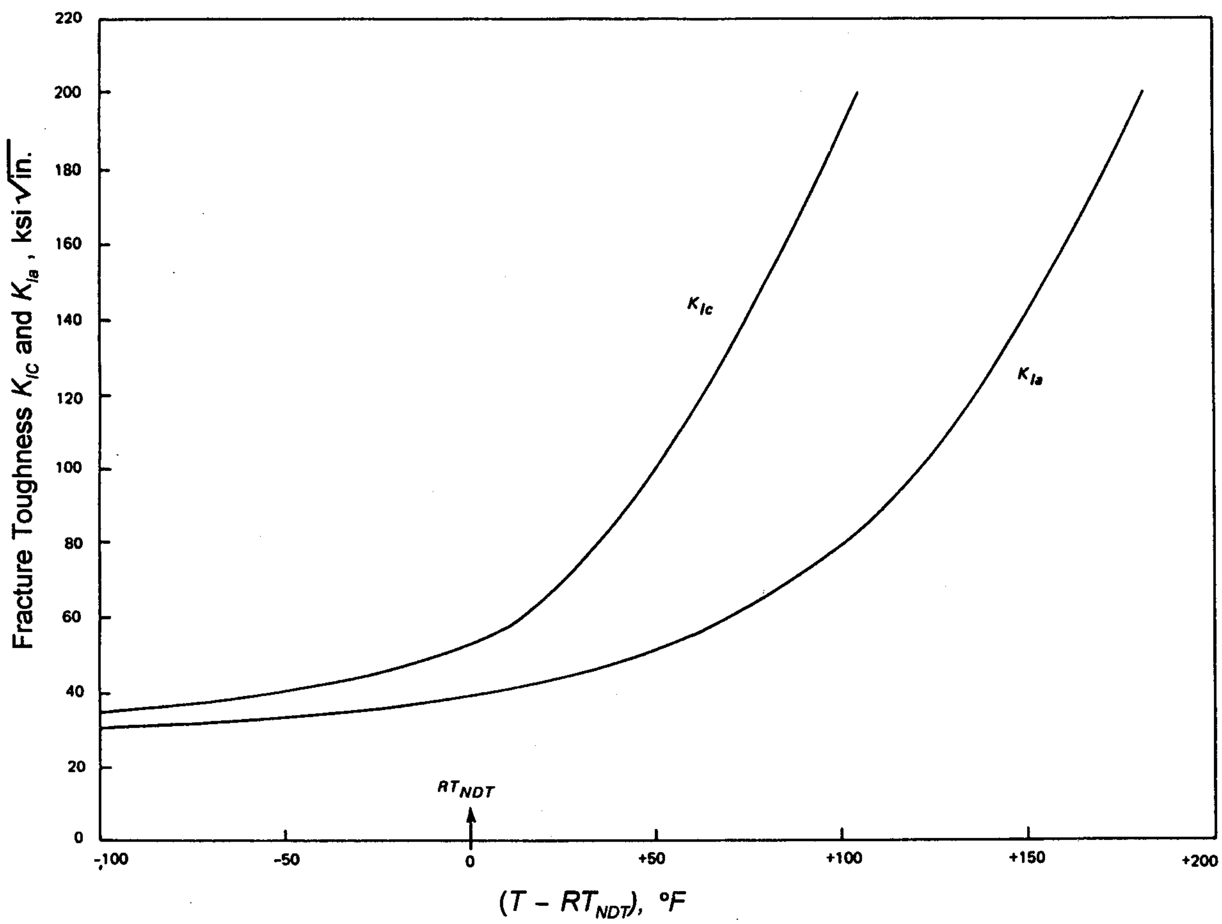


FIGURE 5 Default lower bound values of initiation and arrest toughness for pressure vessel steels from the ASME Boiler and Pressure Vessel Code.

These same fracture mechanics procedures have been used to set operating limits (i.e., combinations of pressure, temperature, and radiation embrittlement) for commercial power reactors that become embrittled in service. This may be the life-limiting degradation process for many reactors and is a major concern in the industry. Regulatory requirements have been written based on fracture mechanics procedures (NRC, 1982). Related probabilistic analyses have also been performed that do not require stacking conservatisms, with results being generated for the probability of crack instability for a given set of conditions. The VISA code was developed for such analyses (Simonen, 1986). Figure 7, which is drawn from NRC, 1982, provides the probability of failure for a given pressure, cooling rate (β), and degree of embrittlement (T_f - RT_{NDT}). Such results are useful in defining regulatory requirements for allowable combinations of conditions. Similar probabilistic calculations have been performed for reactor piping. Radiation embrittlement is not a problem, but cyclic stresses (including seismic) may be higher, and other degradation modes may be present (such as stress corrosion cracking). The PRAISE code was developed for probabilistic fracture mechanics analysis of reactor piping (Harris et al., 1981; 1992).

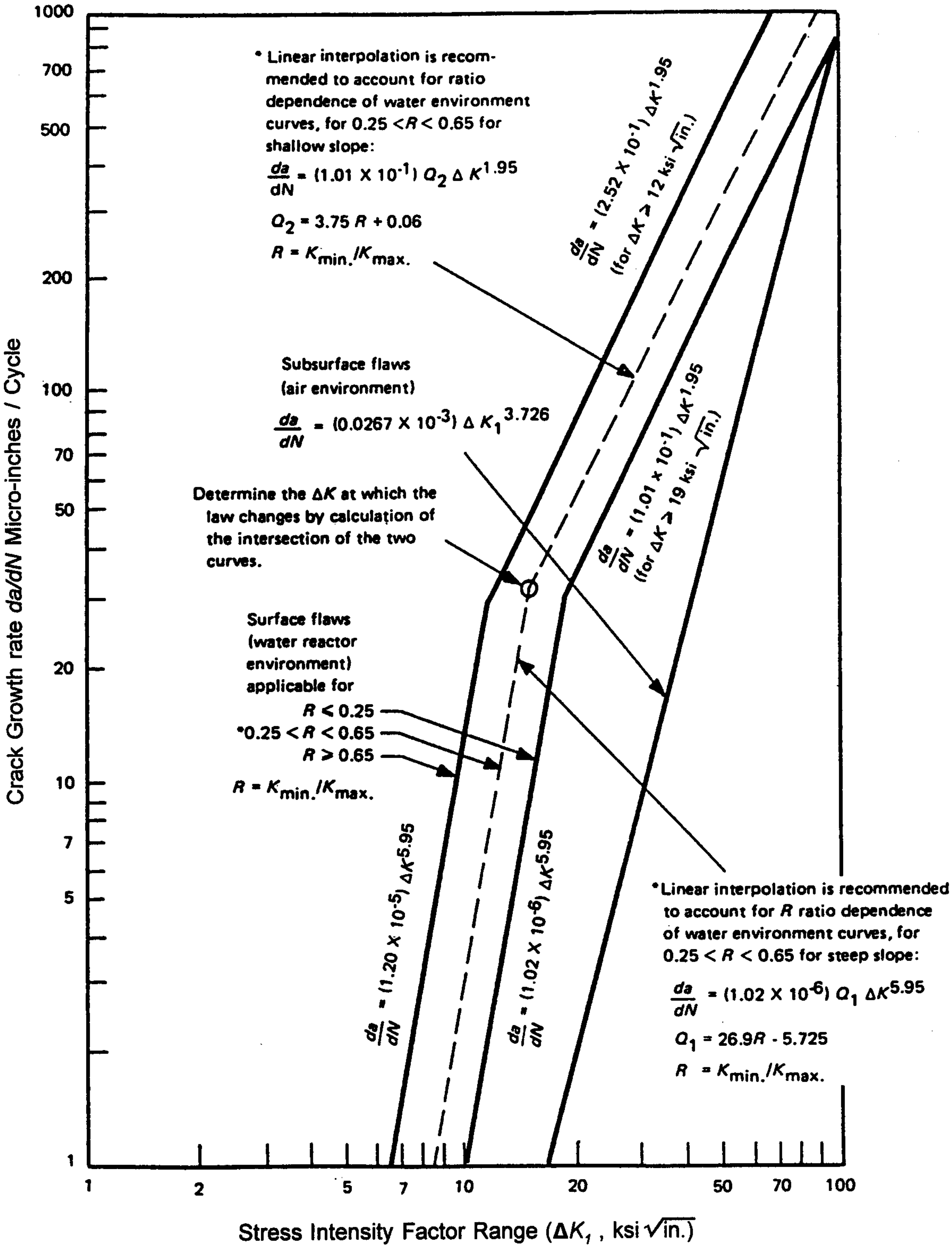


FIGURE 6 Default reference fatigue-crack–growth curves for carbon and low alloy ferritic steels from the ASME Boiler and Pressure Vessel Code.

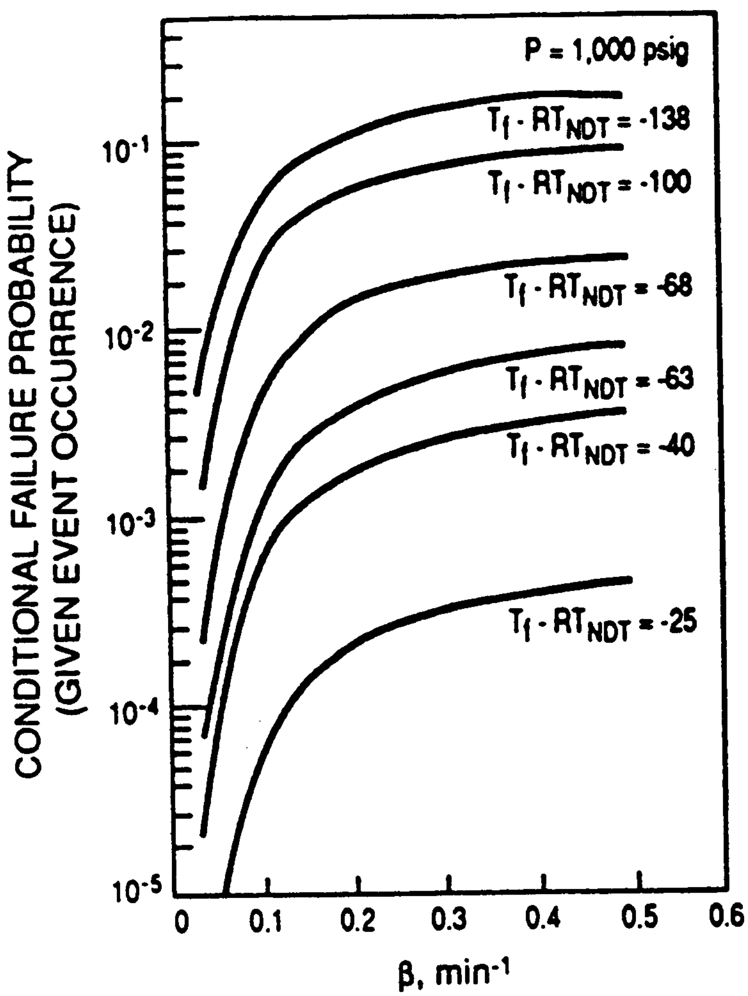


FIGURE 7 Conditional failure probability of a pressure vessel as a function of cool-down rate (β) and temperature.

Section XI of the ASME Boiler and Pressure Vessel Code defines the locations in the pressure boundary of a nuclear reactor to be inspected without explicit consideration of the probability and consequences of a failure. Efforts are underway to develop procedures for selecting locations based on the risk of failure at a location. The most risk-prone (risk = probability of failure – consequences of failure) locations are concentrated on in the inspections, which should provide a greater risk reduction for a given number of inspections or the same risk reduction for fewer inspections. General guidelines are provided in ASME, 1991. Other ASME volumes (1992, 1994) address nuclear and fossil-fired power plants. Probabilistic fracture mechanics and expert opinion have been used to identify the locations of highest failure probability.

Fracture control procedures based on fracture mechanics have also been widely applied to fossil-fired power plants. Examples include boiler components (such as headers), piping, and turbine/generator components. A series of turbine rotor failures in the 1970s sparked an effort to apply fracture mechanics and nondestructive testing to the assessment of such failures. Tools were developed for the stress analysis and fracture mechanics analysis of steam turbine rotors, as exemplified by the SAFER code, which was developed under support of the Electric Power Research Institute (EPRI, 1983). In conjunction with this, the NDE tools for rotor inspections were developed, and such inspections and analyses are now performed on a fairly routine basis. Probabilistic capabilities were incorporated into the SAFER software that include the influences of inspection sizing uncertainty and scatter in material fracture properties (Wells et al., 1986).

Failures in seam-welded reheat lines, which operate at about 1,000°F and are in the creep regime for the materials employed, are of current concern and are being addressed in the industry by NDE combined with fracture mechanics analysis of detected cracks (Foulds et al., 1994). Much of the current technology of fracture mechanics analysis of creep cracking was developed in conjunction with these efforts, which has also involved considerable effort in generation of the required material properties. Computer software has been developed to facilitate the fracture mechanics analysis, and capabilities for headers has been included. The software was developed under support of the Electric Power Research Institute and is known as the BLESS code (Grunloh et al., 1992; Harris et al., 1993).

ADDITIONAL AREAS

Fracture mechanics has been used in additional areas for fracture control purposes, including offshore structures and bridges. The fracture mechanics methodologies are based on the same principles discussed above. In these instances, the randomness of the loadings suggests a probabilistic approach. Yazdani and Albrecht (1990) provide an example applying probabilistic fracture mechanics to bridges, and the references that they provide give additional details on both the deterministic basis and probabilistic considerations. Applications to offshore structures is a large field in itself, and Skjong (1995) is suggested for information in this area. Fracture mechanics considerations also play a major role in the analysis and control of failures in the chemical and petroleum industries, where the term "fitness-for-service" is employed. Assessments concerning the safety implications of defects found in service are a major concern in these industries, and fracture mechanics is heavily relied upon in such assessments, although many degradation mechanisms not involving a dominant crack are encountered. Buchheim et al. (1992) provide a review.

Example of a Marine Application

Fracture mechanics applications to marine structures are fairly numerous and are discussed elsewhere in this symposium. An example of the use of probabilistic fracture mechanics to determine an optimum inspection and repair strategy for container ships that had been observed to crack in service is provided by Sire et al., 1992. In the case considered, deck-doubler plates had been welded to the deck of a fleet of container ships during the process of adding cargo bays. Soon after being placed in service, cracks were observed to be occurring in the deck-doubler plates at the welds. The cracks were found to have initiated from large internal weld defects, such as lack of fusion at the butt welds.

The simplest repair scheme would have been to remove the entire length of doublers, reweld the butt joints, and complete the associated fillet welds. However, the time and cost of such a procedure was prohibitive. An effective repair strategy was needed in which the cost of periodic inspection and repair could be balanced against the cost of service failures. The technical issues involved ultrasonic inspection procedures, repair schemes, welding procedures, and alternative joint geometries for the main deck doubler butt joints. The economic factors included inspection time and interval, lost service time required for repairs, the costs associated with future crack development, and the potential for catastrophic failure. Several weld repair alternatives were considered.

A probabilistic fracture mechanics approach is ideally suited to this type of life prediction/maintenance optimization problem, and such an approach was used to predict the expected life of doubler butt joints for any set of input variables, such as initial-flaw size distribution, flaw-detection criteria, flaw-rejection size, weld fracture toughness, joint residual stress, and inspection interval. A probabilistic model of deck-doubler butt weld lifetime based on a fracture mechanics analysis of fatigue growth of preexisting weld defects was constructed. The model was based on a deterministic fatigue-crack–propagation model. Initial crack size, cyclic stress level, and fracture toughness were taken to be random.

The initial flaw size distribution was estimated from the reported results of early ultrasonic inspections. This provided two points on the cumulative crack depth distribution, which allowed the parameters of the assumed lognormal distribution to be evaluated. The nondetection probability was assumed to be a step, with a value of 1 for a crack of size less than a_d (i.e., it would never be detected) and a value of 0.05 for a crack size greater than a_d (i.e., 1 in 20 cracks of size greater than a_d would be missed). The detectable size, a_d, was varied and was taken to also be the inspection rejection crack size. Any detected crack was assumed to be repaired. Mean stresses due to still-water bending and as-welded residual stresses were considered. The cyclic stresses due to wave-induced bending and slamming and whipping were estimated, taking into consideration the distribution of significant wave heights that the ships were expected encounter on their routes.

The probabilistic fracture mechanics model was exercised to determine the relative improvement in reliability for various combinations of rejection flaw size (a_d), inspection schedules, and residual stresses. Figure 8 provides an example of the results and shows the predicted number of deck doubler failures in the fleet in 20 years for various inspection flaw sizes and schedules. The result for no inspections is also shown. Figure 8 consistently shows that in-service inspection can greatly reduce the expected number of failures in the next 20 years. The smaller the inspection rejection size and the more frequent the inspection, the greater the reduction in the number of failures. These reductions need to be balanced against the increased cost of inspection and repair associated with the more frequent and/or sensitive inspections. Results such as shown in Figure 8 provide the failure probability inputs needed for the economic optimization calculations.

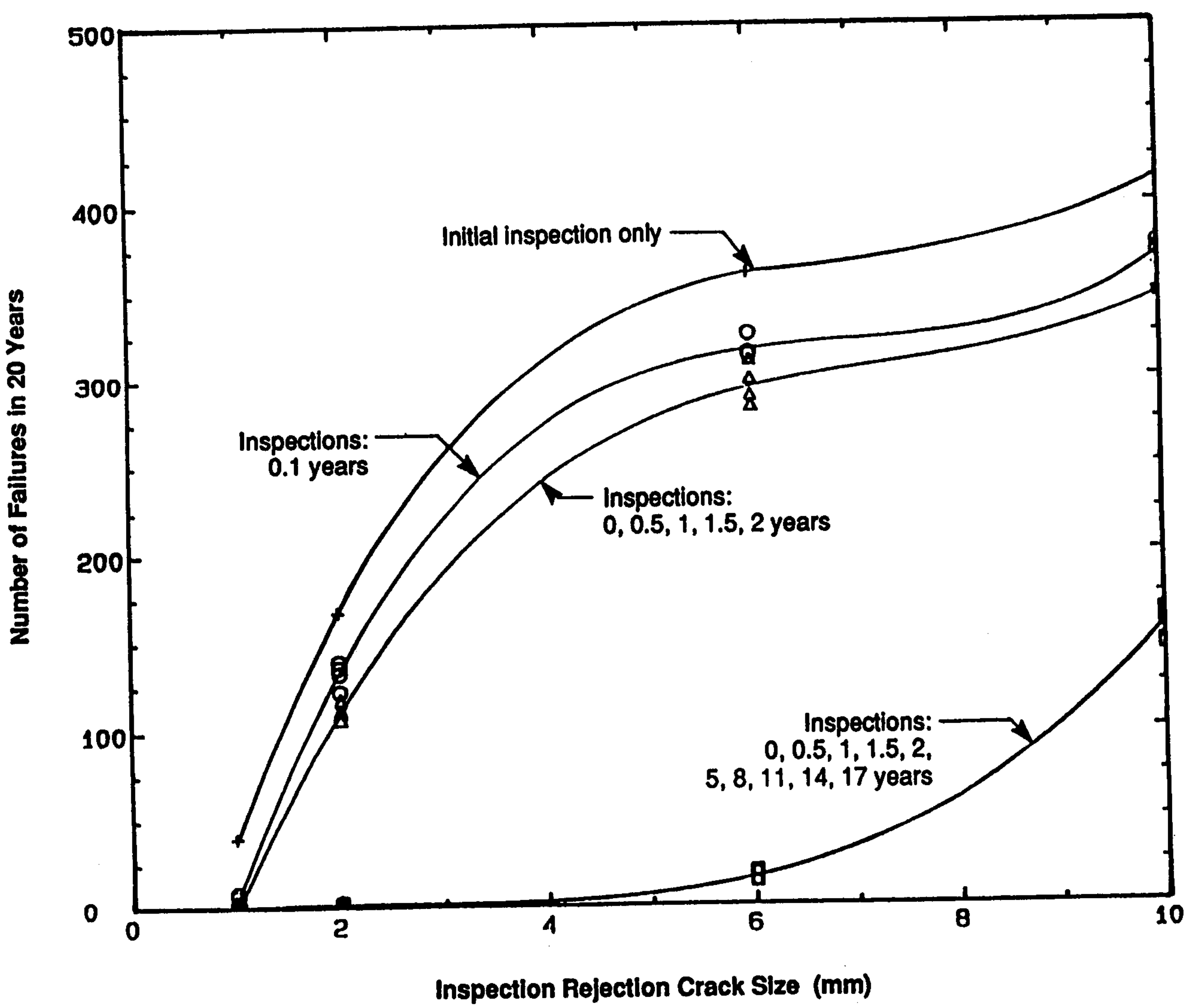


FIGURE 8 Effect of inspection frequency and inspection/rejection crack size on the expected number of failures in the fleet of deck doublers.
Source: Sire. 1992.

CONCLUDING REMARKS

A review of fatigue and fracture control procedures in the aerospace and power generation industries was provided, with emphasis on the use of fracture mechanics. The general applicability of fracture mechanics to many industries was emphasized, and the marine industry has used the procedures in certain areas, such as offshore structures. In the instances reviewed, the fracture mechanics procedures appear in codes and regulations, such as the ASME Boiler and Pressure Vessel Code and NASA fracture control requirements. The necessary material properties are provided, and software for ease and standardization of implementation is often available. Such software is tailored to the particular industry through the types of materials and the crack configurations included. In many instances the inputs to a fracture mechanics analysis are subject to scatter and/or are not well known, and a probabilistic approach is appropriate. Examples of probabilistic applications are given, including one in the marine industry.

The implementation of a fracture mechanics-based approach to fatigue and fracture control in the industries reviewed has involved the generation and characterization of the necessary material properties (such as fracture toughness and fatigue-crack–growth rates) along with compilation of the K-solutions for the crack geometries relevant to that industry. This has occurred through the development of computer software (such as NASA/FLAGRO) or codification (such as the ASME Boiler and Pressure Vessel Code). Much of the material property information for marine structures is already available, but additional data are undoubtedly needed, such as fatigue-crack–growth rates in sea water. Many of the relevant K-solutions already exist. Hence, much of the technology for a standardized approach to fatigue and fracture control already exists, but not in a single location or document. Outlines for development of a standardized approach for fatigue and fracture control for marine structures is a goal of this symposium and workshop. Obstacles to such developments appear to be surmountable. A good set of stress inputs is necessary. A probabilistic approach appears to be well suited, with either explicit consideration of reliability or implicit use in setting design criteria.

REFERENCES

AFASD (Air Force Aeronautical Systems Division). 1974. Airplane Damage Tolerance Requirements. MIL-83444. Wright-Patterson AFB, Ohio: AFASD.

Anderson, T.L. 1991. Fracture Mechanics: Fundamentals and Applications. Boca Raton, Florida: CRC Press.

ASME (American Society of Mechanical Engineers). 1989. ASME Boiler and Pressure Vessel Code. Section XI, Rules for In-service Inspection of Nuclear Plant Components. New York: ASME.

ASME. 1991. Risk-Based Inspection—Development of Guidelines, vol. 1. General Guidelines, CRTD—vol. 20-1. New York: ASME.

ASME. 1992. Risk-Based Inspection—Development of Guidelines, vol. 2, part 1. Light Water Reactor (LWM) Nuclear Power Plant Components, CRTD—vol. 20-2. New York: ASME.

ASME. 1994. Risk-Based Inspection—Development of Guidelines, vol. 3, Fossil Fuel-Fired Electric Power Generating Station Applications, CRTD-vol. 20-3. New York: ASME.

Barsom, J.M., and S.T. Rolfe. 1987. Fatigue and Fracture Control in Structures, 2d ed. Englewood Cliffs, New Jersey: Prentice-Hall.

Broek, D. 1978. Elementary Engineering Fracture Mechanics. Alphen aan den Rijn, Netherlands: Sitjhoff & Noordhoff.

Buchheim, G.M., D.A. Osage, W.R. Warke, and M. Prager. 1992. Update of fitness-for-service and inspection for the petrochemical industry, service experience and reliability improvement: nuclear, fossil, and petrochemical plants. ASME. PVP 288:253–260.

EPRI (Electric Power Research Institute). 1983. User's Guide for Safer Turbines. EPRI Nondestructive Examination Center. RP 1570-2. Palo Alto, California: EPRI.

FAA (Federal Aviation Administration). 1978a. Damage Tolerance and Fatigue Evaluation of Structure. FAA Regulations, part 25-571. Washington, D.C.: FAA.

FAA. 1978b. Damage Tolerance and Fatigue Evaluation of Structure. FAA Advisory Circular AC 25-571-1, Washington, D.C.: FAA.

Forman, R.G., V. Shivakumar, J.C. Newman, A.U. De Konig, T.K. Henriksen, J.M. Beek, J.A. Henkener, V.B. Lawrence, W.R. Mettu, S.M. Piotrowski, L.C. Williams, and F. Yeh. 1993. Fatigue Crack Growth Computer Program NASA/FLAGRO, ver. 2.0. Houston, Texas: National Aeronautics and Space Administration.

Forman, R.G. 1994. Development Plans for the NASGRO Computer Program. Presented at U.S. Air Force Aircraft Structural Integrity Program Conference held December 7, 1994 in San Antonio, Texas.

Foulds, J.R., R. Viswanathan, J.L. Landrum, and S.M. Walker. 1994. Guidelines for the Evaluation of Seam-Welded High Energy Piping. Draft Final Report on Research Projects RP8046-04 and 2819-24. Palo Alto, California: EPRI.

Grunloh, H.J., R.H. Ryder, A. Gattuso, J.M. Bloom, D.R. Lee, C.C. Schultz, D.D. Sutherland, D.O. Harris, and D. Dedhia. 1992. An Integrated Approach to Life Assessment of Boiler Pressure Parts, vol. 4. BLESS Code User's Manual and Life Assessment Guidelines. Draft Final Report on Research Project 2253–10. Palo Alto, California: EPRI.

Harris, D.O., E.Y. Lim, and D. Dedhia. 1981. Probability of Pipe Fracture in the Primary Coolant Loop of a PWR Plant, vol. 5. Probabilistic Fracture Mechanics Analysis. NUREG/CR-2189. Washington, D.C.: U.S. Nuclear Regulatory Commission.

Harris, D.O. 1985. Probabilistic Fracture Mechanics. Pressure Vessels and Piping Technology—1985—A Decade of Progress. New York: ASME.

Harris, D.O., D. Dedhia, and S.C. Lu. 1992. Theoretical and User's Manual for pc-PRAISE. NUREG/CR-5864. Washington, D.C.: U.S. Nuclear Regulatory Commission.

Harris, D.O., C.H. Wells, H.J. Grunloh, R.H. Ryder, J.M. Bloom, C.C. Schultz, and R. Viswanathan. 1993. BLESS: Boiler Life Evaluation and Simulation System, A Computer Code for Reliability Analysis of Headers and Piping. Reliability and Risk in Pressure Vessels and Piping, vol. 251. New York: ASME.

Harris, D.O. 1995. Probabilistic Fracture Mechanics. Pp. 106–145 in Probabilistic Structural Mechanics Handbook. R. Sundararajan, ed. New York: Chapman & Hall.

Kanninen, M.L., and C.H. Poplar. 1985. Advanced Fracture Mechanics. New York: Oxford University Press.

MIL-STD (Military Standard). 1984. Standard Requirements for Safe Design and Operation of Pressurized Missile and Space Systems. MIL-STD-1522A.

Murakami, Y. 1987. Stress Intensity Factors Handbook. New York: Pergamon Press.

NASA (National Aeronautics and Space Administration). 1988. Fracture Control Requirements for Payloads Using the National Space Transportation System (NSTS). NASA NHB 8071.1. Washington, D.C.: NASA.

NRC (Nuclear Regulatory Commission). 1982. U.S. Nuclear Regulatory Commission Policy Issue, Enclosure A. NRC Staff Evaluation of Pressurized Thermal Shock. Washington, D.C.: U.S. Nuclear Regulatory Commission.

Palmberg, B., A.F. Blom, and S. Eggwertz. 1987. Probabilistic Damage Tolerance of Aircraft Structures. Pp. 47-130 in Probabilistic Fracture Mechanics and Reliability, J.W. Provan, ed. Boston, Massachusetts: Martinus Nijhoff Publishers.

Provan, J.W. 1987. Probabilistic Fracture Mechanics and Reliability. Boston, Massachusetts: Martinus Nijhoff Publishers.

Simonen, F.A., K.I. Johnson, A.M. Liebetrau, D.W. Engel, and E.P. Simonen. 1986. VISA II—A Computer Code for Predicting the Probability of Reactor Pressure Vessel Failure. NUREG/CR-4486. Washington, D.C.: U.S. Nuclear Regulatory Commission.

Sire, R.A., J.E. Kokarakis, C.H. Wells, and R.K. Taylor. 1992. A probabilistic structures life prediction system for container ship repair and inspection. International Journal of Pressure Vessels and Piping 50:297–315.

Skjong, R. 1995. Applications in Offshore Structures. Pp. 608–639 in Probabilistic Structural Mechanics Handbook, R. Sundararajan, ed. New York: Chapman & Hall.

Tada, H., P.C. Paris, and G.R. Irwin, 1985. The Stress Analysis of Cracks Handbook. St. Louis, Missouri: Paris Productions, Inc.

Vlieger, H. 1988. Damage Tolerance of Stiffened-Skin Structures: Prediction and Experimental Verification. Pp. 169–219 in Fracture Mechanics: 19th Symposium. ASTM STP 969. Philadelphia, Pennsylvania: ASME.

Wells, C.H., D.O. Harris, and D.P. Johnson. 1986. An Overview of Life Assessment Technology for Turbogenerator Rotors. Pp. 1-3–1-11 in Life Assessment and Improvement of Turbo-Generator Rotors for Fossil Plants. R. Viswanathan, ed. New York: Pergamon Press.

Wood, H.A., and R.M. Engle, Jr. 1979. Damage Tolerance Design Handbook: Guidelines for the Analysis and Design of Damage Tolerant Aircraft. AFFDL-TR-79-3021. Wright-Patterson AFB, Ohio: Air Force Flight Dynamics Laboratory.

Yazdani, N., and P. Albrecht. 1990. Probabilistic fracture mechanics applications to highway bridges. Journal of Engineering Fracture Mechanics 37(5):969–985.

Fatigue and Fracture Design Procedures Used in Construction Industries

S.J. Maddox

ABSTRACT

Based on the premise that fatigue and fracture control of most structures will be governed by the performance of weldments, this paper highlights the fatigue design and assessment methods available and used in construction industries. Brief reference is also made to fracture-avoidance measures. The main message is that these procedures, which are used successfully in a wide range of industries, are equally applicable to ships.

INTRODUCTION

Fatigue and fracture avoidance measures are required for the whole range of engineering components and structures, as is evident from the continuous supply of cases of service failure. Around 90 percent of these are attributable to fatigue, and recent cases reported to the International Institute of Welding (IIW), which collects and publishes case histories, include offshore structures, pressure vessels, cranes, earth moving equipment, pipelines, ships, and vehicles (PA, 1967; 1979a). A major contributing factor is undoubtably the presence of welds, which, as a result of flaws, severe geometric stress concentrations, high-tensile residual stresses, and degraded material, can have relatively low resistance to fatigue and fracture (Gurney, 1979a; Maddox, 1991). Indeed, welded joints are invariably the most fracture-critical features in a structure and hence the most likely sites for failure. Consequently, measures directed at the avoidance of failure in welded joints are a good starting point for considering fracture prevention in ships. This paper provides an overview of the design procedures concerned with the avoidance of fatigue and fracture adopted by the various industries that should be suitable for application to ships.

DESIGN TO AVOID FATIGUE

Fatigue Design Methods

The fatigue behavior of welded joints is well characterized, and reasonably comprehensive design rules exist for many structures. Considerable international collaboration has taken place, leading to international rules that are in good agreement with many national codes. This has been possible because of the general acceptance of certain basic principles upon which fatigue design rules for welded joints should be based (Maddox, 1992a). Nevertheless, other codes are still in use that are not based on these principles, although in most cases they are being reviewed. Both approaches are based on the use of S–N curves, but the fatigue data from which they were derived differ, as described in the following subsections.

Type 1 Approach: S–N Curves for Weld Details

Most fatigue design rules present a series of S–N curves (e.g., Figure 1) based on data obtained from constant amplitude fatigue tests on particular weld details. The design S–N curve incorporates the notch effect of the weld and the effect of the stress concentration due to the detail. Consequently, it is expressed in terms of the nominal stress in the vicinity of the joint. In some cases, the fatigue lives of different details are approximately the same; therefore, the same design curve can be used for all of them. A classification system relates details of the welded joint and the appropriate design curve. In general, the design category depends on the joint type, geometry, and direction of loading and relates to a particular location of fatigue cracking (see Figure 2). At present, comprehensive guidance is provided only for arc welded joints.

Depending on the precision with which the applied stress can be estimated by the designer, there are two variants of the type 1 approach. First, if the stress directions, or even the precise location of the highest stress, are not known, the lowest category for the particular detail type would need to be adopted. Second, if highly detailed stress information is available, it may be possible to adopt the so-called hot-spot stress approach for the fatigue design of weldments. In this case, the calculated stress includes the stress concentration effect of the welded joint, so the design S–N curve only needs to allow for the notch effect of the weld itself. Thus, the S–N curve for transverse groove welds may be sufficient to cover all cases of fatigue failure from a weld toe. Interest in the hot-spot stress approach is increasing, but before it can be adopted more widely agreed procedures for deriving it are needed. Recommendations produced by the IIW, which refer to both measured (e.g., strain gauges) and calculated (e.g., finite element analysis) stress distributions, are expected to be suitable (Niemi, 1992).

Type 2 Approach: S–N Curves for Materials

An alternative approach incorporated in some codes, notably those used for designing pressure vessels, is to base the design S–N curve on fatigue test data obtained from polished specimens of the material of interest. These S–N curves are used in conjunction with fatigue-

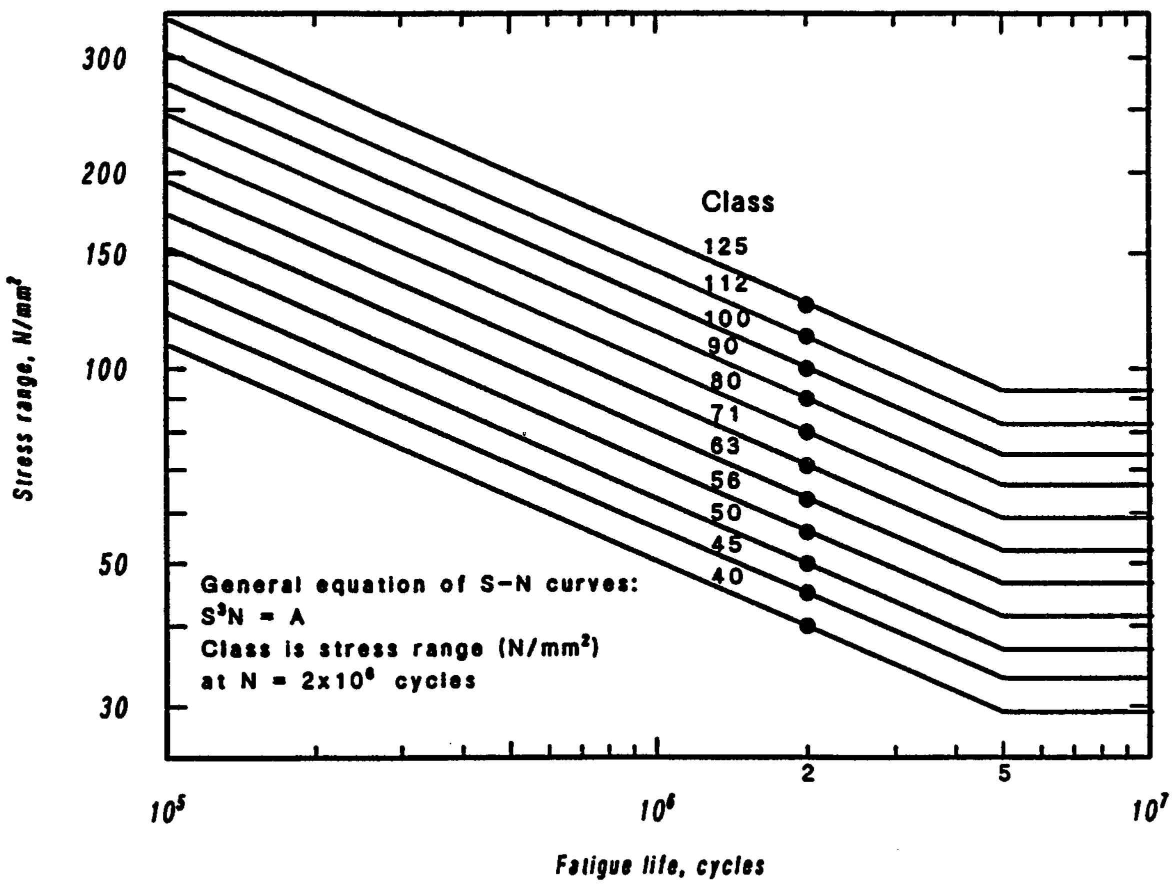


FIGURE 1 IIW fatigue design S–N curves for welded steel.

strength reduction factors, which take account of structural discontinuities in the structure, including welds.

The fatigue lives of machined specimens are dominated by fatigue-crack initiation. In contrast, because of the presence of welding flaws and severe geometric stress concentrations, the fatigue lives of welded joints are dominated by fatigue-crack growth. Since the processes of fatigue-crack initiation and propagation are so different and depend on different factors, the latter approach is now widely considered to be unsuitable, and even potentially unsafe, for application to severely notched components like welds (Maddox, 1992b).

Statistical Analysis of Fatigue Data

Fatigue design S–N curves usually represent some lower bound to experimental data. Traditionally, safety factors have been applied to the estimated mean S–N curve to establish

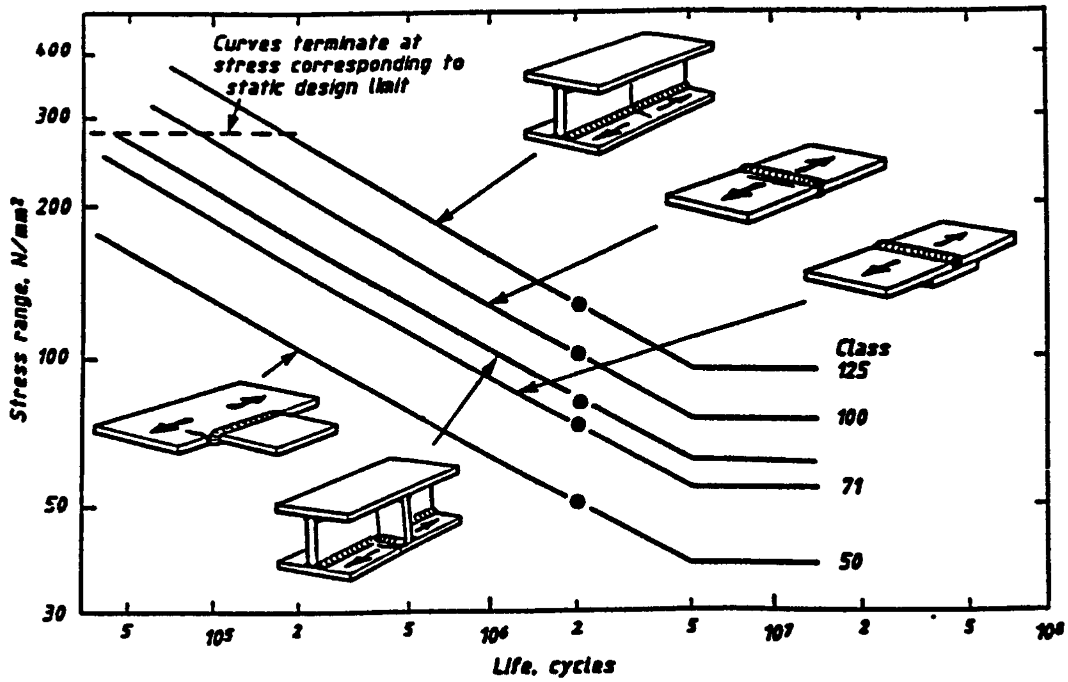


FIGURE 2 Typical IIW design S–N curves and corresponding weld details, showing site of fatigue cracking and loading direction considered.

design limits. However, most codes now rely on statistical analysis of the data (e.g., Gurney and Maddox, 1973) to allow the probability of failure to be estimated. In most cases, linear regression analysis of log S versus log N is used, and design is based on the curve representing approximately 2.3 percent probability of failure (i.e., the line approximately two standard deviations of log N below the mean).

Residual Stress Effects

Fatigue design is usually based on applied stress range, regardless of the applied mean stress, to take account of the influence of high tensile residual stresses in as-welded joints (Fisher, 1971; Gurney, 1979b). In some cases, as supported by experimental evidence (see Maddox, 1982a and Figure 3), it is assumed that this requirement applies even for fully compressive loading; whereas other codes relax the requirement for part-compressive loading, assuming that part of the compressive cycle will be non-damaging.

It is not usually assumed that stress relief is beneficial from the fatigue viewpoint. Although it is recognized that the reduction of tensile residual stresses will be beneficial, the practical view is that the effectiveness of stress relief, particularly of large welded structures,

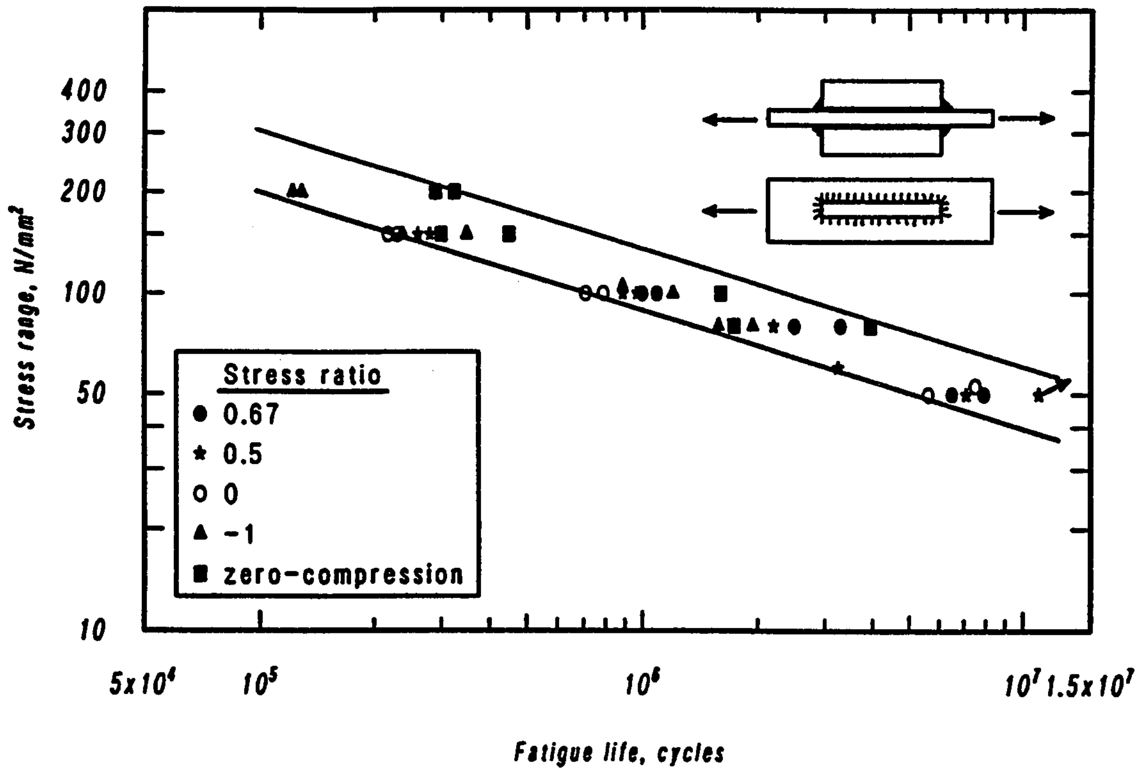


FIGURE 3 Fatigue data for steel fillet welded joints containing high tensile residual stresses showing lack of influence of applied stress ratio.

cannot be guaranteed. Also, long-range residual stresses will be produced during the assembly of a structure.

Cumulative Damage

Fatigue design S–N curves are based on data obtained under the constant amplitude loading, and, in order to estimate the fatigue life of a structure subjected to variable amplitude loading, use is normally made of Miner's rule ($\Sigma n/N = 1$ at failure). In some circumstances, for conservatism, a code may require a designer to assume $\Sigma n/N < 1$. Recent research has identified load spectra for which Miner's rule is unsafe (Gurney, 1983), and it may be necessary in the future to adopt $\Sigma n/N < 1$ more widely.

Some codes take account of the damaging effect of stress ranges that are below the constant amplitude fatigue limit but which, under variable amplitude loading, become damaging during the fatigue life of a weld. A method widely adopted is to extend the S–N curve beyond the constant amplitude fatigue limit at a reduced slope (see Figure 4).

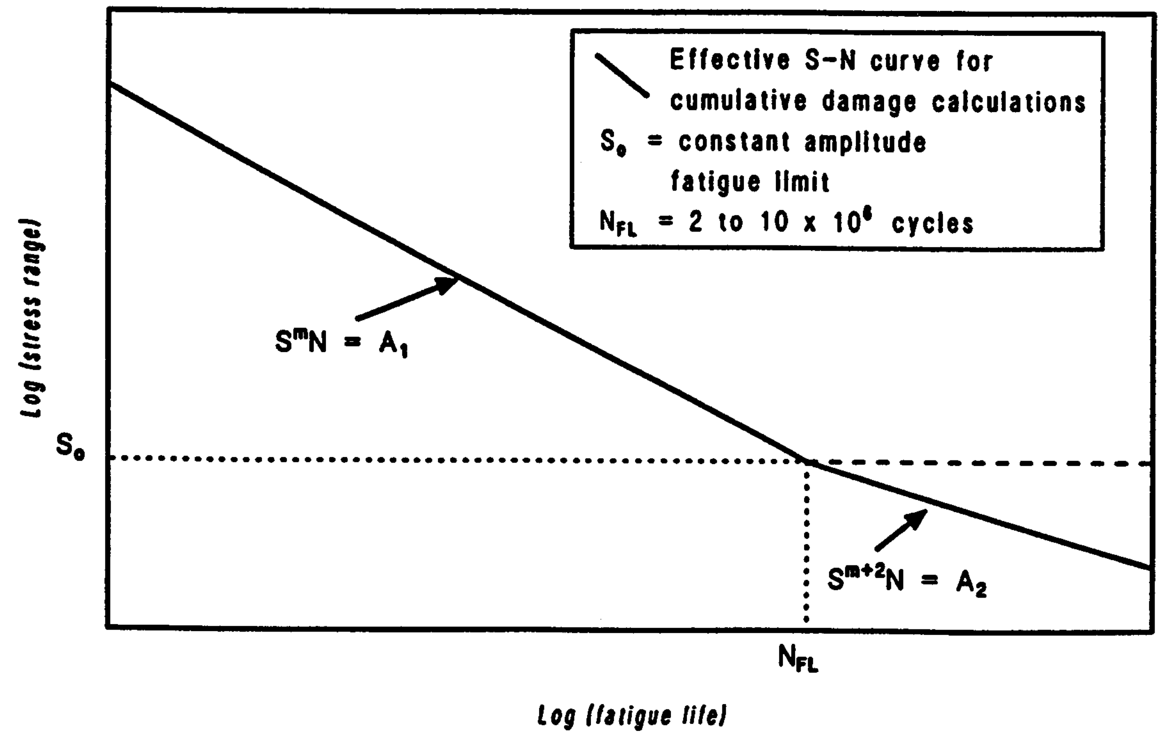


FIGURE 4 Effective S–N curve used in cumulative damage calculations.

Significance of Fatigue-Crack Growth

Since, in most cases, the fatigue process in a welded joint consists of the growth of a pre-existing flaw, the fatigue lives of welds can be calculated using fracture mechanics, by integrating the fatigue-crack–growth relationship. The resulting calculated S–N curve has the equation $S^n N$ = a constant, where the slope, n, is the same as the exponent, m, in the Paris crack growth relationship $da/dN = C\ (\Delta K)^m$. In view of this, most of the S–N curves in type 1 rules are parallel (see Figure 1), with a slope compatible with the fatigue-crack–growth relationship for the material ($m = n = 3$ is usually adopted for steels). In those cases where fatigue-crack initiation is significant, the S–N curves are shallower (i.e., $n > 3$). Clearly, fatigue data obtained from polished specimens result in shallower S–N curves than those relevant to welded joints (see Figure 5), which is another reason why type 2 rules are unsuitable for severely notched components.

Scale Effect

There are two important scale effects associated with welded joints that need to be taken into consideration in producing design data. The first concerns the effect of residual stresses. Real

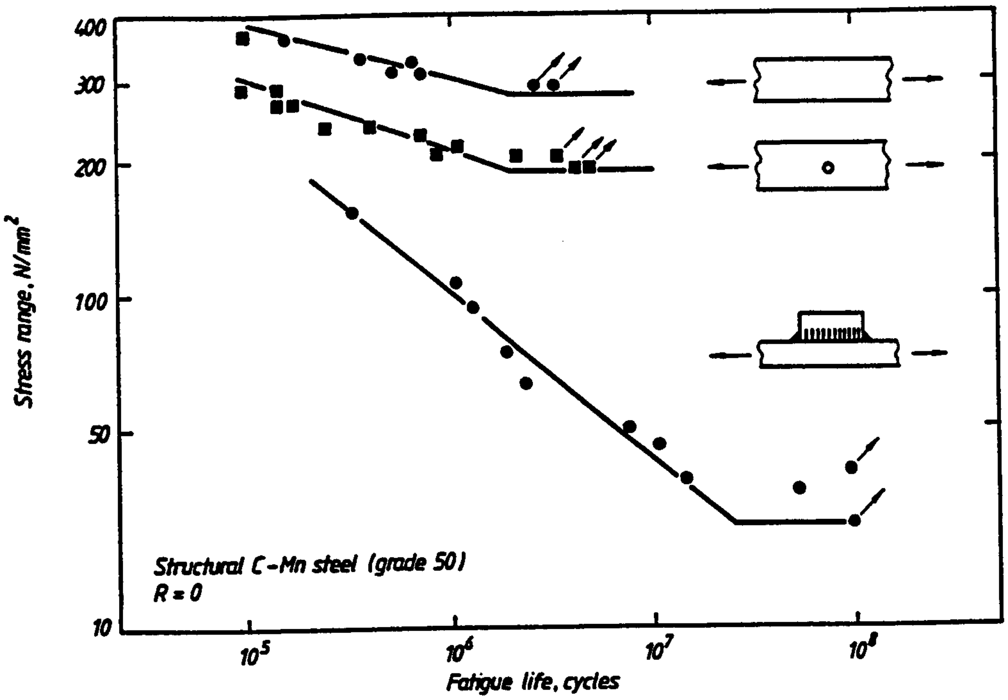


FIGURE 5 Comparison of S-N curves obtained from plain steel plate, notched plate, and plate with a fillet-welded attachment.

welded structures can be expected to contain very high tensile residual stresses (Fisher, 1971; Gurney, 1979a), which determine the effective mean stress (see Figure 6). However, typical laboratory specimens may be too small to contain such residual stresses. The solution is to rely on data obtained from large-scale specimens, such as beams, or to generate fatigue data under high-tensile, mean-stress conditions.

The other scale effect refers to the influence of plate thickness and overall joint geometry on the stress concentration at the weld toe. It is now well established that increasing plate thickness can lead to a reduction in fatigue strength (Gurney, 1979a), and some codes include a design penalty (see Figure 7). However, current research is investigating the influence of other dimensions and seeking ways of relaxing that design penalty. For example, it has been found that the thickness effect is strongest if all the relevant joint dimensions are scaled in proportion to thickness (Maddox, 1987), but it may disappear altogether if this is not the case (e.g., thick plates with small attachments).

Effect of Material Strength

All recent fatigue design rules recognize the fact that fatigue lives of welded joints are not influenced by the tensile strength of the parent material. This situation arises because of the dominance of the fatigue-crack–propagation process—the rate of fatigue-crack propagation being

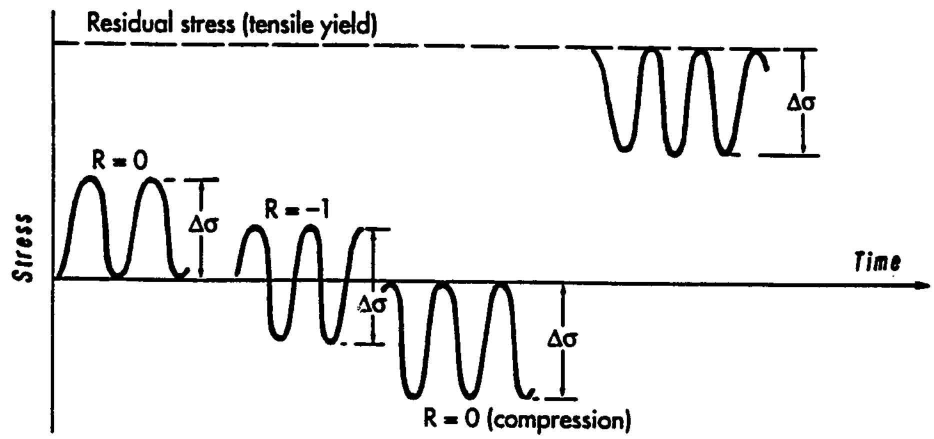


FIGURE 6 Effective stress range resulting from superposition of applied cyclic and yield magnitude residual tensile stresses.

relatively insensitive to material tensile strength. This again is highly relevant to the type 2 design approach based on fatigue data obtained from polished specimens. Such test data will indicate a beneficial effect of increasing tensile strength (see Figure 8), an effect that is not seen in the fatigue performance of severely notched components.

Fatigue Design Codes

Bridges

Most of the development that has resulted in modern fatigue design rules for welded joints originated from bridge standards. Consequently, several countries have comprehensive fatigue design rules for steel bridges (BSI, 1979; AASHTO, 1989). These are all type 1 rules; consequently, although there are differences in design stresses resulting from the use of the various codes, they are all based on essentially the same principles.

Offshore Structures

Fatigue is a major problem for structures situated in the North Sea and has been the cause of many failures. Consequently, when the boom in the oil- and gas-recovery industry started in

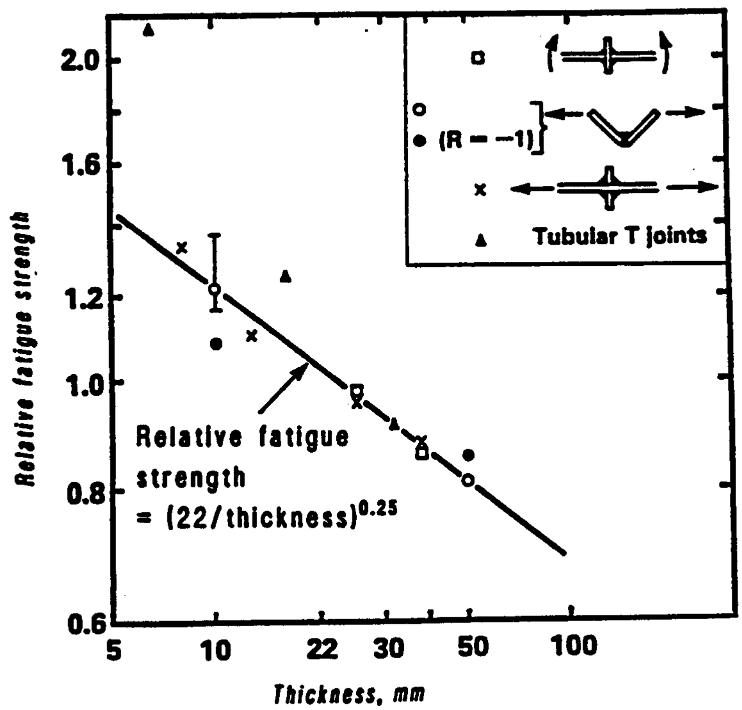


FIGURE 7 Fatigue data upon which the thickness-related design-penalty was based.

the late 1960s, several European countries embarked on major studies of the fatigue performance of tubular offshore structures. This led to the production of reasonably comprehensive fatigue design rules, notably those published by the U.K. Department of Energy, which drew very heavily on the new research findings (HMSO, 1990; HMSO, 1984). The basic principles of the fatigue rules are the same as those in the bridge rules and indeed, where appropriate, identical rules are presented. The most important connection in a steel offshore structure is, however, the intersection point of tubular members (node); and, in this case, the hot-spot stress approach is used. This can be estimated using parametric formulae or calculated directly using finite element analysis. The agreed definition of the hot-spot stress relates to the stress distribution as the weld is approached. The corresponding design S–N curve is that for transverse groove welds in plate.

One factor not covered in other design codes is an allowance for seawater corrosion fatigue. Currently, fatigue life is halved under free corrosion conditions but assumed to be restored to air behavior if cathodic protection is used. Also, there is no fatigue limit in free corrosion conditions. These rules are based on limited experimental evidence, and it is likely that account will be taken of more recent work that suggests a greater influence of seawater in future revisions of the European codes (Gurney and Sharp, 1991).

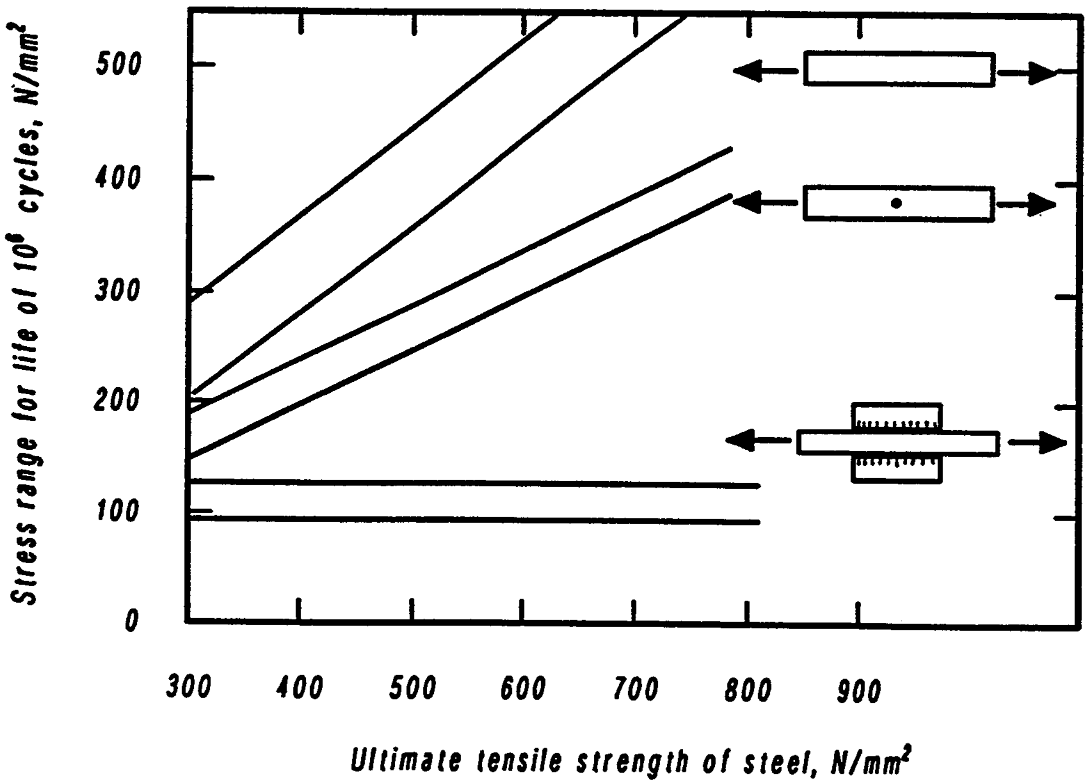


FIGURE 8 Effect of steel tensile strength on fatigue strengths.

Pressure Vessels

Fatigue rules for pressure vessels throughout the world are based on the type 2 approach described earlier, that is material S–N curves derived from fatigue test data obtained from polished specimens. The approach originates from the American Society of Mechanical Engineers' codes. Since the approach is now thought to be unsuitable for welded joints (Harrison and Maddox, 1980), alternative type 1 rules are being developed for new codes in Europe (Maddox, 1992b). However, it is understood that ASME, including the code for nuclear components, will continue to use the present type 2 approach.

Welded Aluminum

A great deal of activity has gone in to the production of fatigue design rules for welded aluminum in Europe in recent years, and two new codes have appeared (BSI, 1991a; ECCS, 1992). These are both type 1 rules and, as far as possible, they have been written to be similar to the rules of steel structures. Broadly speaking, it can be assumed that fatigue design stresses for welded aluminum are a third of those for the same geometry in steel (see Figure 9) (Maddox, 1982b), although that relationship becomes increasingly conservative as the severity of the stress

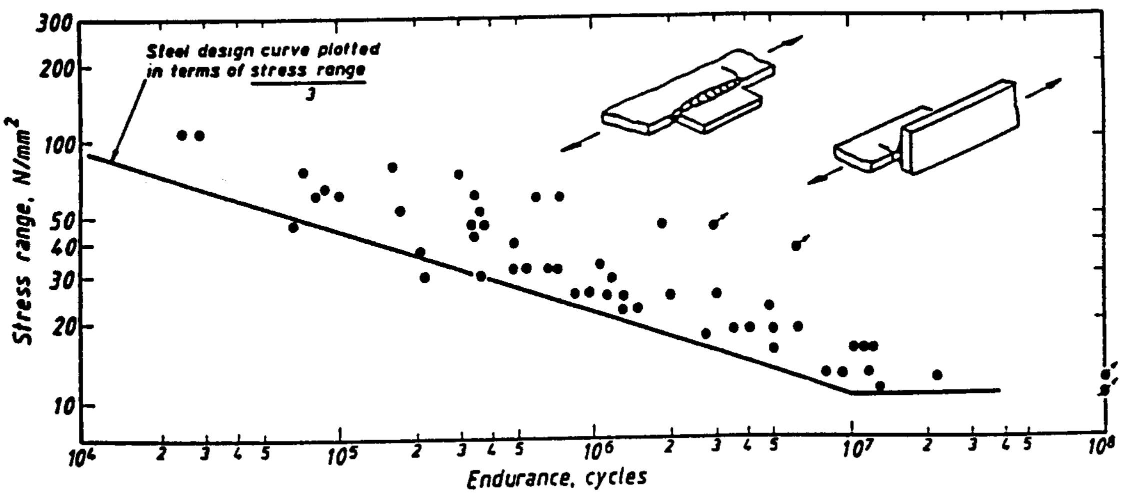


FIGURE 9 Fatigue data for an aluminum weldment for which fatigue strength is one-third of that for steel.

concentration due to the weld decreases.

General Code

Most available fatigue design rules are directed at particular industries, which are required to use them. However, industries that are not bound by any particular code or standard also find it convenient to adopt available rules if they are considered to be well founded, not unduly conservative, and relevant. This has proved to be the case with the type 1 approach, and general fatigue rules for any application are available. This reflects the fact that modern type 1 rules are closely related to fatigue test data obtained from actual details of the types that might be used in any fabrication. In the United Kingdom, the British Standards organization recently published a new code that presents general fatigue design data for the assessment of steel structures (BSI, 1993). At present, this simply embodies all the fatigue rules contained in current British standards for specific applications, but the intention in future is to introduce new information and to update the current rules. Similarly, rules produced by IIW and the European Union (Eurocode 3) can be used as a general basis for fatigue design of steel structures (IIW, 1982; European Union, 1992). It may be noted that the former is the basis of new rules for ships in Germany (Germanischer Lloyd, 1992).

Examples of industries that are known to be using type 1 fatigue rules on a routine basis are given in Table 1. A selection of weld details, some of which will be of particular relevance to only one industrial sector, and the corresponding design S–N curves from the IIW rules (IIW, 1982), for illustrative purposes, are shown in Table 2.

TABLE 1 Industries Using Type 1 Fatigue Design Rules for Welded Components and Structures

General	Applications
Structures	Bridges, cranes, fixed offshore tubular platforms, mining roof supports, fairground rides
Construction, mining and agricultural plant	Earthmoving equipment, tractors, fork trucks, agricultural equipment
Vehicles	Trucks, buses, chassis, railcars, railway bogeys, engine frames
Engineering equipment	Power generators, fans
Process plant	Pressure vessels, food and chemical processing equipment, storage tanks, pipework
Marine	Mobile drilling rigs, ships

Future Developments

The following topics, which are likely to be reviewed or introduced into fatigue design codes in future, are particularly relevant to ships.

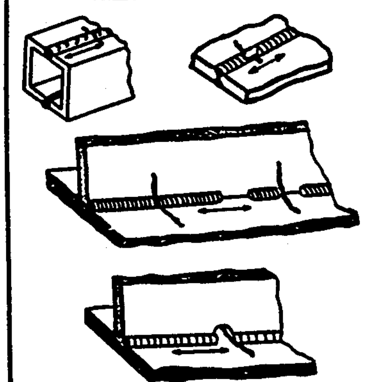
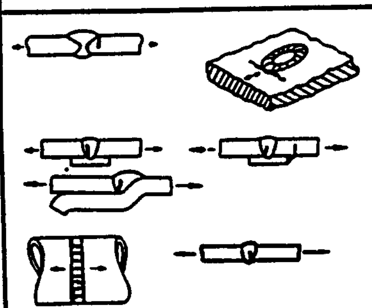
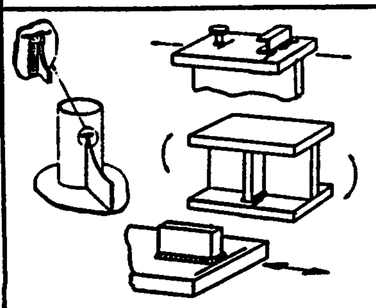
- inclusion of non-arc–welded (e.g., laser, electron beam, resistance) joints
- clearer guidance on the definition of nominal stress to be used with the S–N curves when detailed stress analysis information (e.g., from finite-element analysis) is available
- extension of the hot-spot stress concept to non-tubular joints
- introduction of recommendations on the use of fatigue life improvement techniques (e.g., weld toe grinding or peening)
- guidance on the influence of corrosive environments
- refinement of the thickness-effect design penalty to include overall joint geometry and to allow benefit from the use of thin sections

Inspection and Repair

Welding Flaws

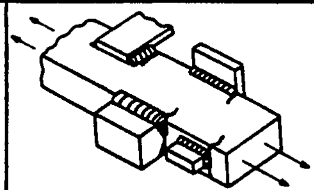
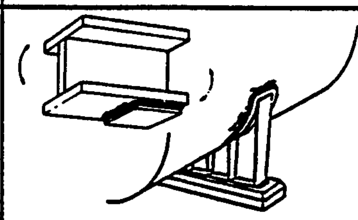
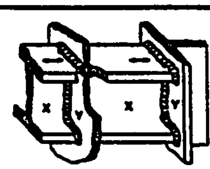
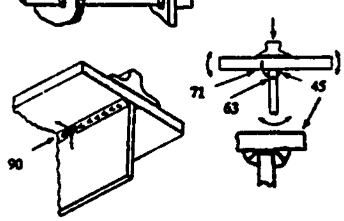
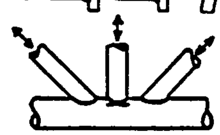
Welding flaws can limit the fatigue resistance of a welded structure; therefore, some form of nondestructive inspection is invariably performed. Most attention, for example, involving the use of radiography or, increasingly, ultrasonic testing, is focused on groove welds since these transmit the full load in a member. It is usually a condition of the fatigue design category that a groove weld is proved to be free from significant flaws. Fillet welds might also be inspected,

TABLE 2 Examples of Weld Detail Categories in Type 1 Fatigue Design Rules (Sheet 1)

Detail	Examples	Crack propagates from:	Fatigue strength* N/mm²	Conditions
Longitudinal groove and fillet welds		Weld ripples Stop/start Weld end in dis-continuous weld Weld end at rat-hole	125 90 80 71	Continuous welding Continuous welding
Transverse groove welds		Weld toe Weld toe Weld toe Weld toe	100 80 71 63	Stop weld made in flat position Positional and/or site welded Weld made on permanent backing strip Weld made from one side without backing – full penetration assured
Stiffeners and small attachments to surfaces of stressed members		Weld toe or end	71 63	Short attachments (≤ 150mm) Long attachments (> 150mm)

* Design stress range at $2x10^6$ cycles on IIW S–N curve, S^3N = constant

TABLE 2 (Continued) (Sheet 2)

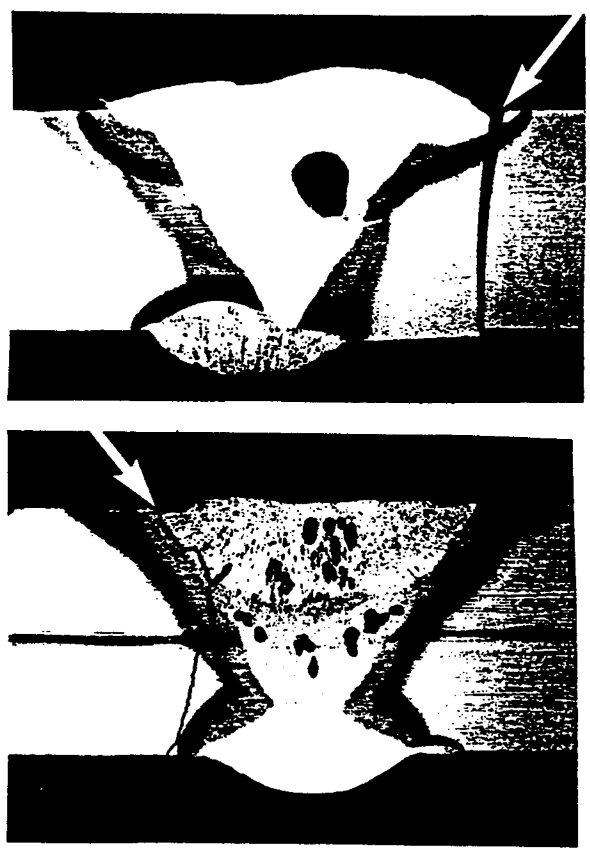
Detail	Examples	Crack propagates from:	Fatigue strength*, N/mm^2	Conditions
Attachments to edges of stressed members		Weld toe or end	50	
Cover plates		Weld toe or end	50	
Load-carrying T and cruciform joints		Toe of full penetration weld	71	
		Toe of partial penetration or fillet weld	63	
		Weld root across throat	45	Based on nominal stress on weld throat
		Weld toe at nozzle	71	Allow for SCF
		Weld toe in tubular joint	90	Based on hot spot stress

* Design stress range at 2×10^6 cycles on IIW S–N curve, S^3N = constant

but usually only for surface-breaking flaws; for example, using magnetic particle inspection or dye penetrant testing. Fatigue cracks can initiate at either ofthe two general types of welding flaw, that is planar (crack-like flaws, such as incomplete fusion or penetration) and non-planar (volumetric discontinuities, such as inclusions and porosity), although the severity of the weld toe makes welded joints surprisingly tolerant to embedded flaws (see Figure 10).

Acceptance Criteria

Fitness for purpose is now widely accepted as the most rational basis for the assessment of weld imperfections, such that an imperfection would not need to be repaired unless its presence was harmful to the integrity of the part concerned. The value of such a criterion may seem obvious, but, in fact, it is not the basis of acceptance criteria in virtually any code. Instead, they are either arbitrary or meant to reflect good workmanship. Consequently, weld imperfections are repaired even though they are not harmful. This situation is undesirable from two viewpoints.



(a)

(b)

FIGURE 10 Examples of insignificant welding flaws. Groove weld failed from toe (arrowed) regardless of presence of (a) slag inclusion and (b) porosity.

First, the repair welding must often be performed under more difficult conditions than the original weld, with the consequential risk of more defects. The effect of this is brought home very forcibly in a review by Swedish shipbuilders (Johansen et al., 1981), who found that service fatigue cracking in groove welds invariably originated from weld repairs. Second, repair welding is costly. For example, allowing for direct and consequential losses, it was estimated in 1980 that the extra cost in U.S. shipyards was $0.6M to $1M per ship (Sandor, 1981). Repair costs can escalate in cases where, as well as the need for expensive specialist equipment to perform repairs, the consequential costs include lost production, such as in the power-generation or, especially, oil- and gas-recovery industries. One case of potential repair in an oil rig, which was avoided on the basis of a fitness-for-purpose assessment, was estimated to have saved around $100M (Harrison, 1989).

Fitness-for-Purpose Applied to Fatigue

A number of fitness-for-purpose–based procedures exist (e.g., IIW, 1990; BSI, 1991b), largely directed at the pressure vessels and offshore industries but, in fact, generally applicable (Maddox, 1993). These allow both planar and non-planar flaws to be assessed. Planar flaws are assessed using fracture mechanics, in particular integration of the relationship between rate of crack growth and the stress-intensity–factor range mentioned earlier. Non-planar defects are assessed using S–N data obtained from welded specimens containing the actual flaws of interest. It is also possible to allow for the influence of shape imperfections, such as misalignment, by calculating the secondary bending stress due to misalignment and adding this to the applied stress when assessing the welded joint. The Welding Institute (TWI) software, Fatiguewise, is now available for performing fitness-for-purpose assessments strictly in accordance with the procedures laid down in British Standard PD 6493 (BSI, 1991b).

DESIGN TO AVOID FRACTURE

Fracture of Welded Structures

As in the case of fatigue, features such as welding flaws, geometric stress concentrations, and high-tensile residual stresses have a profound effect on fracture resistance. Indeed, severe stress concentrations are a requirement for unstable fracture, although these could, of course, be cracks formed in service (e.g., fatigue cracks). In spite of the obvious importance of fatigue cracking on the fracture resistance of a structure, the two design procedures are usually presented quite independently. Indeed, some codes do not even refer to the link, and fracture-avoidance measures tend to be directed only at the original weld. Since the fatigue life referred to in most design codes corresponds essentially to through-thickness fatigue cracking, ideally all structures in which there is a risk of fracture should be designed to tolerate through-thickness cracks.
Apart from the features noted above, and in contrast to fatigue, fracture resistance is also strongly influenced by the material degradation, notably embrittlement, that can be caused by welding.

Material Selection

The most widely used approach to fracture avoidance is based on the specification of particular material properties, notably impact properties, at specified temperatures. Although it is recognized that impact tests on notched specimens do not provide quantitative information on fracture resistance, correlation of such data with service experience and realistic large-scale fracture tests has enabled safe guidelines to be developed for a wide range of steels. In the case of welded structures, specified impact properties would be required for both the parent material and the weld zone. This is important because, in general, both the weld metal and the heat-affected zone offer lower resistance to fracture than the parent material.

Fracture Assessment

For particularly critical applications (e.g., nuclear), the assessment of thick structures and applications where it may be necessary to quantify fracture resistance (e.g., to assess the effect of welding flaws), alternative approaches are available. These are based on fracture-mechanics principles, involving the identification of a single parameter that can be used as a measure of fracture resistance. For brittle materials, linear elastic fracture mechanics provide the stress-intensity factor, K, and the critical value for fracture (fracture toughness), K_{Ic}. For the more widely used ductile materials, like weldable structural steels, parameters related to general yielding fracture mechanics (e.g., J_{Ic} and critical crack-tip–opening displacement) have proved to be more useful. The key element of the fracture toughness is that it relates the applied stress conditions, flaw size, and the material's fracture resistance. Like the impact test, standard fracture mechanics test methods are available for checking the fracture toughness of particular materials in procedural tests. These are now widely used in nuclear, pressure-vessel, and offshore industries as the basis for material and welding procedure acceptance.

In the case of welding flaws, unlike fatigue, unstable fracture is unlikely to initiate from a non-planar flaw, and, hence, only crack-like discontinuities are potentially significant. Where relevant, an assessment should also consider the effect of propagating cracks (e.g., fatigue) on fracture resistance. It should be noted, however, that tolerance to a large fatigue crack does not necessarily guarantee tolerance to smaller flaws because the latter may be situated in more brittle material and may experience higher residual stresses.

Codes

Design codes for bridges, pressure vessels, offshore structures, cranes, storage tanks, and pipelines all include material selection requirements based on impact properties. Fracture-mechanics–based design procedures are also provided in some pressure vessel codes, particularly those relevant to nuclear components, generating equipment, and offshore structures. However, undoubtably the most comprehensive guidance on the use of fracture mechanics for assessing fracture resistance is that contained in codes dealing with the acceptance of welding flaws on a fitness-for-purpose basis (e.g., Harrison et al., 1986; BSI, 1991b). Assessments of the likelihood of ductile fracture can be extremely complex, and it is normally necessary to resort to computer methods, such as the TWI software, Crackwise.

CONCLUDING REMARKS

Fatigue or fracture in ships in most likely to be associated with welded joints. There are well-established procedures already being used throughout the construction industry to design to avoid fatigue and to select materials and fabrication conditions to avoid fracture. Since these do not depend on the type of construction being considered, they are suitable for ships. In particular, type 1 fatigue rules, based directly on experimental data obtained from welded specimens, as used in the design of offshore structures, bridges, cranes, earthmoving equipment, and land-based vehicles, are immediately applicable. Current developments of the hot-spot stress approach will also be applicable and, indeed, particularly suitable for ship design based on detailed finite element analysis.

Similarly, fracture control procedures, for example, those used in bridge and offshore industries, based primarily on material selection via impact tests but backed up by more realistic fracture mechanics tests, are certainly applicable to ships.

Finally, weld quality assessments as well as weld-flaw acceptance criteria, that are based on fitness-for-purpose could, and should, be adopted more widely in the context of ship construction.

REFERENCES

AASHTO (American Association of State Highway Transportation Officials). 1989. The American Association of State Highway Transportation Officials Standard Specification for Highway Bridges, 14th ed. Washington, D.C.: AASHTO.

BSI (British Standards Institute). 1979. Steel, Concrete and Composite Bridges, part 10. Codes of Practice for Fatigue. British Standard Specification BS 5400. London: BSI.

BSI. 1991a. Structural Use of Aluminum. British Standard Specification BS 8118. London: BSI.

BSI. 1991b. Guidance on Methods for Assessing the Acceptability of Flaws in Fusion Welded Structures. British Standard Published Document PD 6493. London: BSI.

BSI. 1993. Code of Practice for Fatigue Design and Assessment of Steel Structures. British Standard BS 7608. London: BSI.

ECCS (European Convention for Structural Steelwork). 1992. European Recommendations for Aluminum Alloy Structures: Fatigue Design. ECCS Report No. 68. Brussels, Belgium: European Convention for Structural Steelwork.

European Union. 1992. Design of Steel Structures. Eurocode 3. Luxembourg: European Union.

Fisher, J.W. 1971. Fatigue strength of welded A514 steel beams. Pp. 135–148 in Proceedings International Conference Fatigue of Welded Structures held June 1970 in Brighton, England. Cambridge: The Welding Institute.

Germanischer Lloyd. 1992. Rules for Sea-Going Steel Ships. Section 20, Fatigue Strength. Hamburg, Germany: Germanischer Lloyd.

Gurney, T.R., and Maddox, S.J. 1973. A re-analysis of fatigue data for welded joints in steel. Welding Research International 3(4):1–54.

Gurney, T.R. 1979a. Fatigue of Welded Structures, 2d ed. Cambridge, United Kingdom: Cambridge University Press.

Gurney, T.R. 1979b. The influence of thickness on the fatigue strength of welded joints. Pp. 523–534 in Proceedings of the 2nd International Conference, Behavior of Offshore Structures (BOSS '79) held August 28–31, 1979 in London, United Kingdom.

Gurney, T.R., and Sharp, J.V. 1991. The effect of seawater on the fatigue lives of welded plates—A comparison of results from U.K. and other national programmes. Pp. IV-288–IV-296 in Proceedings of ISOPE-91 held August 11–16, 1991 in Edinburg, Scotland. Golden, Colorado: International Society for Offshore and Polar Engineering.

Harrison, J.D., and Maddox, S.J. 1980. A critical examination of rules for the design of pressure vessels subject to fatigue loading. Presented at special session of the 4th International Conference on Pressure Vessel Technology "Design Codes—Philosophies and Criteria" held May 1980 in London, England. Also available as document IIW-XIII-941-80. Cambridge, England: International Institute of Welding.

Harrison, J.D. 1989. Significance of weld defects with regard to fatigue behavior. Fatigue Aspects in Structural Design. Delft, The Netherlands: Delft University Press.

HMSO (Her Majesty's Stationery Office). 1984. Background to New Fatigue Design Guidance for Steel Welded Joints in Offshore Structures. London: Her Majesty's Stationery Office.

HMSO. 1990. Offshore Installations: Guidance on Design and Construction, 4th ed. London: Her Majesty's Stationery Office.

IIW (International Institute of Welding). 1982. IIW design recommendations for cyclic loaded welded steel structures. Welding in the World 20(7/8):153–165.

IIW. 1990. IIW Guidance on Assessment of the Fitness-for-Purpose of Welded Structures. IIW/IIS-SST-1157-90, ISBN 876-8877107-6. Cambridge, United Kingdom: International Institute of Welding.

Johansen, B.G., S.L. Kjellander, and N.G. Leide. 1981. Is it good enough? A study of tolerances for welded ship structures. Pp. 6-1–6-11 in Proceedings Conference on Fitness for Purpose Validation of Welded Constructions held November 17–19, 1981 in London, England. Cambridge, United Kingdom: The Welding Institute.

Maddox, S.J. 1982a. Influence of Tensile Residual Stresses on the Fatigue Behavior of Welded Joints in Steel. ASTM STP 776. Philadelphia, Pennsylvania: American Society for Testing and Materials.

Maddox, S.J. 1982b. Fatigue design of welded aluminum alloy structures. Pp. 4.1–4.17 in Proceedings of the 2nd International Conference on Aluminum Weldments held May 24–26,1982 in Munich, Germany. Dusseldorf, Germany: Aluminum-Verlag.

Maddox, S.J. 1987. The Effect of Plate Thickness on the Fatigue Strength of Fillet Welded Joints. Cambridge, United Kingdom: Abington Publishing.

Maddox, S.J. 1991. Fatigue Strength of Welded Structures, 2d ed. Cambridge, United Kingdom: Abington Publishing.

Maddox, S.J. 1992a. Fatigue design of welded structures. Pp. 31–56 in Proceedings International Institute of Welding Conference Engineering Design in Welded Constructions held September 7–8, 1992 in Madrid, Spain. Oxford, England: Pergamon Press.

Maddox, S.J. 1992b. Fatigue Analysis of Pressure Vessels in BS 5500. WRC Bulletin 374. New York: Welding Research Council.

Maddox, S.J. 1993. Recent advances in the fatigue assessment of weld imperfections. Welding Journal 72(7):42–51.

Niemi, E. 1992. Recommendations concerning stress calculation for fatigue analysis of welded components. IIW Document XIII-1458-92. Cambridge, United Kingdom: The Welding Institute.

PA (Publications de la Antogène). 1967. Fatigue Fractures in Welded Constructions, Vol. I. Paris, France: Publications de la Antogène.

PA. 1979. Fatigue Fractures in Welded Constructions, vol. 2. Paris, France: Publications de la Antogène.

Sandor, L.W. 1981. A perspective on weld discontinuities and their acceptance standards in us maritime industry. Pp. 16-1–16-7 in Proceedings Conference on Fitness for Purpose Validation of Welded Constructions held November 17–19, 1981 in London, England. Cambridge, United Kingdom: The Welding Institute.

Hull Structural Design for a 40,000 DWT Double-Hull Products Carrier

Mark Debbink

ABSTRACT

This paper is intended to provide a global to local overview of the design process and analytical methods used to evaluate the structural design of a 40,000 DWT products tanker for strength, producibility, and fatigue resistance. The global design process will discuss selection of key parameters, such as transverse web spacing, longitudinal stiffener spacing and access requirements. The local design will consider critical structure for reduction of stress concentration fatigue as well as producibility issues.

INTRODUCTION

With the decline of naval ship demand it is necessary for shipyards to offset military ship construction with commercial ship construction. Part of Newport News Shipbuilding's (NNS) strategic plan to reenter the commercial ship markets was to design and offer a product carrier for both domestic and international markets.

This paper summarizes the design process that is performed by NNS for the structural configuration of a 40,000 DWT products carrier. The design was developed as a standard design to be offered for international as well as domestic service, with generic owner/operator's design criteria. The design is based upon marketing research and a midship section optimized to suit construction facilities, regulatory double-hull requirements, and owner/operator concerns for inspection and maintenance. Additionally, the interaction between design engineering, production control, and industrial engineering will be described as they form a concurrent design team.

STRUCTURAL FRAMING

There are many critical factors that affect the initial design of a ship. Of these key design parameters, shown in Figure 1, the structural framing system will have a direct influence on strength, deadweight, cargo capacity, and access.

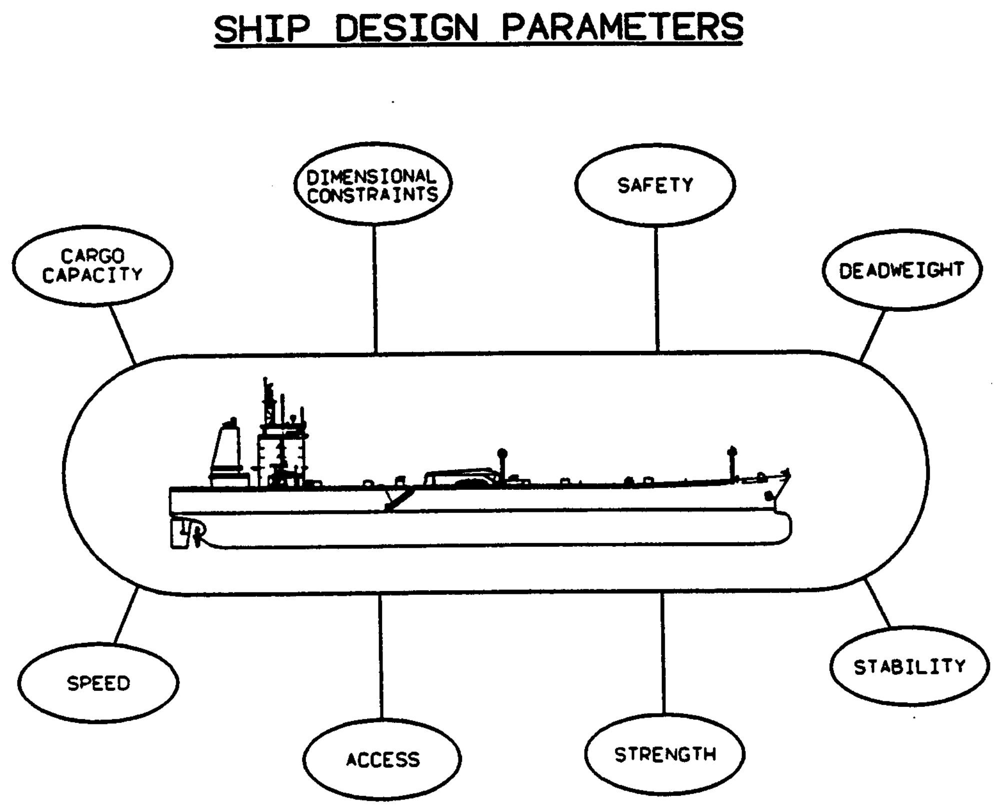


FIGURE 1 Ship design parameters.

The objective of arriving at a structural framing design is to determine the most economically producible arrangement to be constructed with existing facilities. As previously mentioned, the dimensional constraints and number of cargo tanks for this design resulted from a marketing study. With a length over all (LOA) set, subtracting the length of the forepeak tank and the engine room leaves the cargo box and slop tank length to be subdivided into eight port and starboard cargo tanks. A plan view of the ship's main deck, identifying the tank arrangement, is shown in Figure 2.

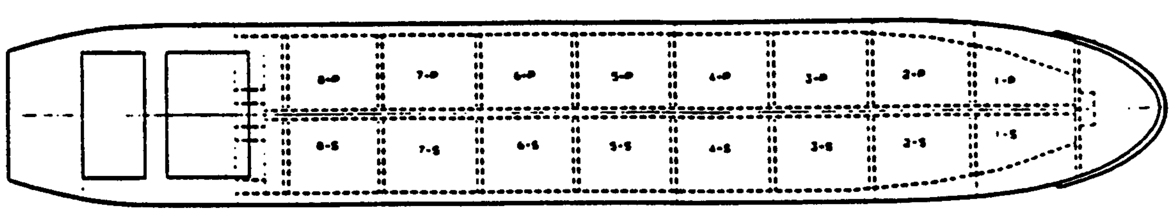


FIGURE 2 Plan view of tank arrangement.

Optimal scantlings were developed early in the design. This was accomplished by varying transverse web and longitudinal stiffener spacing then comparing the total cost of each alternative. Scantlings for each alternative were designed in accordance with American Bureau of Shipping (ABS) Rules for Building and Classing Steel Vessels. A matrix of three transverse web and three longitudinal stiffener spacings was designed for input to a cost model. To simplify the design and cost models, a one-tank–length innerbottom section from the centerline of the ship to the innerhull bilge knuckle was used as a representative section of structure for the ship. Table 1 identifies the web and longitudinal spacings used in the scantling selection. Figure 3 shows the portion of the ship's innerbottom used in the alternative spacing and cost models.

TABLE 1 Web and Longitudinal Spacing

Members	Spacing and Number		
Web Spacing	9 ft, 4 in.	11 ft, 3 in.	14 ft, 0 in.
Number of webs	5	4	3
Longitudinal Spacing	22 in.	33 in.	39 in.
Number of longitudinals	19	13	11

Structural Framing "Cost Model"

A cost modeling technique was developed by the industrial engineering department to compare the structural spacing models. The major cost drivers for this analysis were as follows:

- steel cost
- surface coating cost
- construction labor cost

The approach taken was to assign dollar values to these three items and then develop a total cost differential that could be compared as an indicator of producibility.

Steel costs were determined by multiplying categories by the appropriate price per pound. Raw material categories were flat plate, U.S.-manufactured shapes, and foreign-manufactured shapes. Coating costs were estimated by multiplying the total tank surface area by a dollar value that includes labor and material. Construction labor costs were determined by multiplying manhours of affected operations by an average dollar value per operation. Manhour estimating

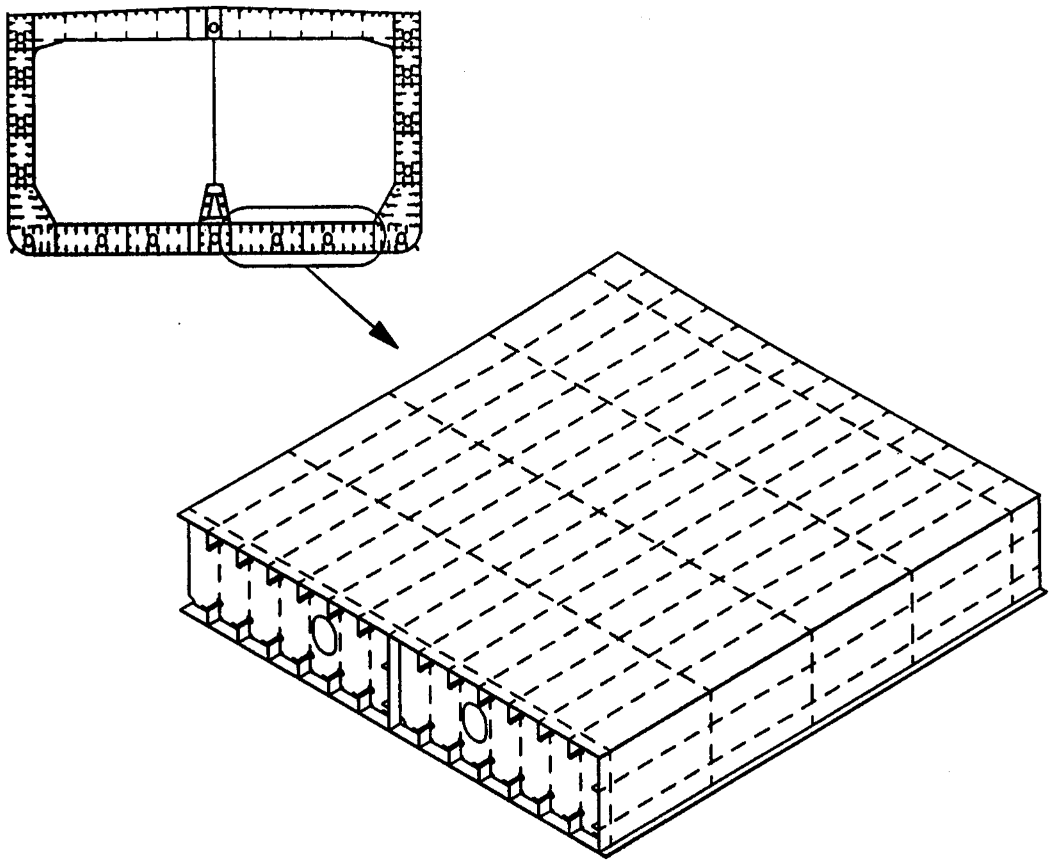


FIGURE 3 Inner hull cost model.

formulas were developed that grouped work steps into categories that could easily relate to structural construction steps. The cost formula and work-step descriptions are stated in Table 2.

Structural Framing "Results"

A chart was developed (see Figure 4) to graphically display the results of the cost model application. The values plotted represent the cost differential at various transverse and longitudinal spacings. Cost differentials were calculated by subtracting out the value of the lowest cost

TABLE 2 Cost Formula and Work Step Descriptions

Manhour estimate = Plate joining manhours + (No. of web frames x manhours per web frame) + (No. of stiffener pairs x manhours per stiffener pair) + (No. of stiffener pairs x No. of web frames x manhours per stiffener/web intersection) + (No. of longitudinal floors x No. of web frames x manhours per longitudinal floor/web intersection)

Work Step Descriptions

Work step	Description
Plate joining manhours	Considers fitting and welding of plate blankets and joining to adjacent assemblies. Varies according to plate thicknesses.
Manhours per web frame	Considers fabrication and installation of web plate.
Manhours per stiffener pair	Considers fabrication, installation on plate blankets joining to adjacent unit, and installation of oil tight collar.
Manhours per stiff/web intersection	Considers fabrication and installation of web panel breakers and two non-tight collars.

alternative from all other totals. The cost curves indicated that a transverse spacing of 11.5 ft and longitudinal spacing of 33 in. were the optimum solution. As a result of this analysis, we determined that the 11.5-ft transverse spacing would be adopted. Other analyses were needed, however, before a final decision on longitudinal spacing could be made.

One important factor in the selection of optimal spacing was the sensitivity of the optimum solution to labor and material cost variations. Actual labor costs could be affected by productivity better or worse than expected or by changes in the applicable dollar value per hour. Material cost could be affected by the raising or lowering of steel prices or a change in coating costs. To evaluate the affect of these variations, a new graph was developed, which is presented in Figure 5. The various plotted bands allow the design engineer to assess the impact of increasing or decreasing labor and material costs on the optimum decision. Each band's centerline

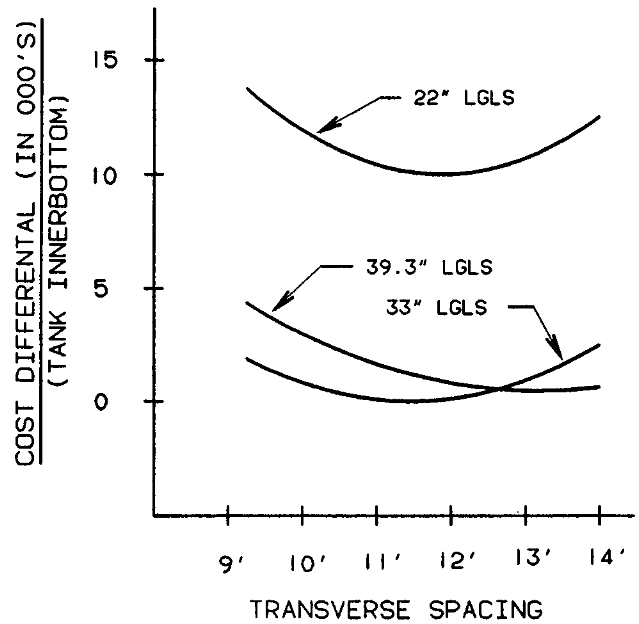


FIGURE 4 Structural spacing cost comparison.

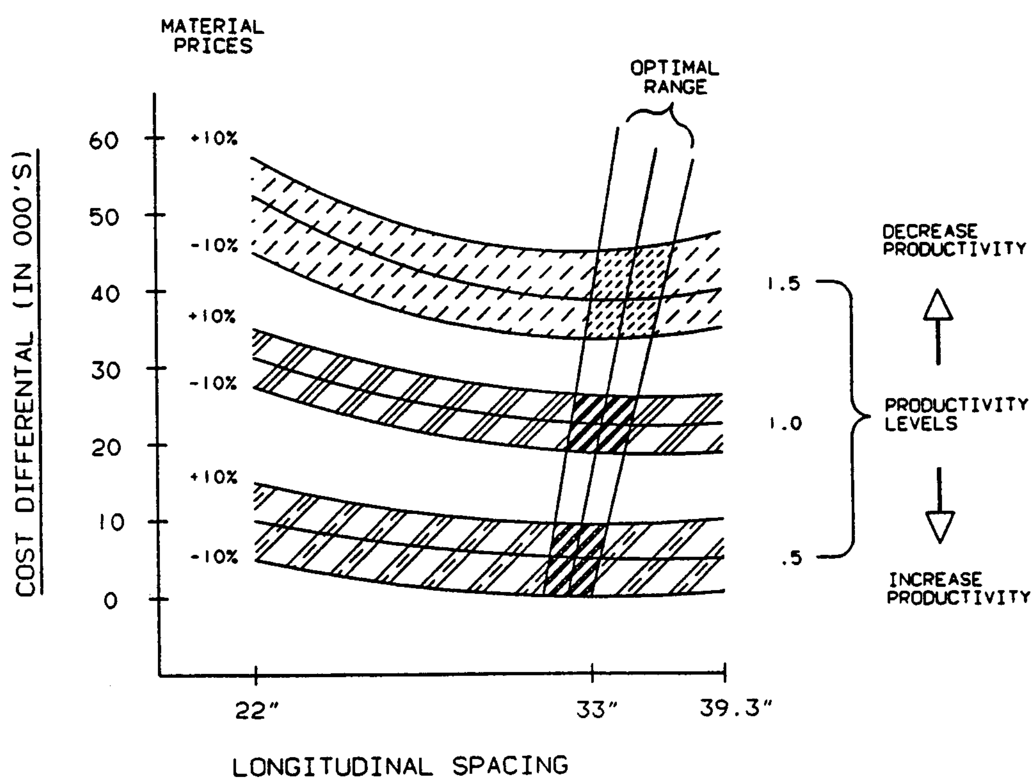


FIGURE 5 Sensitivity of optimum structural spacing to changes in labor and material cost.

represents a unique productivity level while its width reflects the effect of material price variations. The diagonal band highlights optimal longitudinal spacing at the various productivity levels. The slope of all productivity levels indicates that significant cost reductions occur as longitudinal spacing increases from 22 in. to 33 in. At this point, the slope of each curve remains relatively flat showing little cost variation between the 33-in. and 39.3-in. spacings. The direction of the diagonal zone's slope indicates how the optimal solution shifts to larger frame spacing (fewer pieces) as productivity decreases.

Another important factor in the design optimization process was weight. An increase in ship's weight will correspondingly reduce cargo deadweight. The effect of longitudinal spacing on ship's weight is illustrated in Figure 6. The line's slope indicates a relatively small weight increase as longitudinal spacing changes from 22 in. to 33 in. At this point, an increase in longitudinal spacing significantly affects the ship's weight.

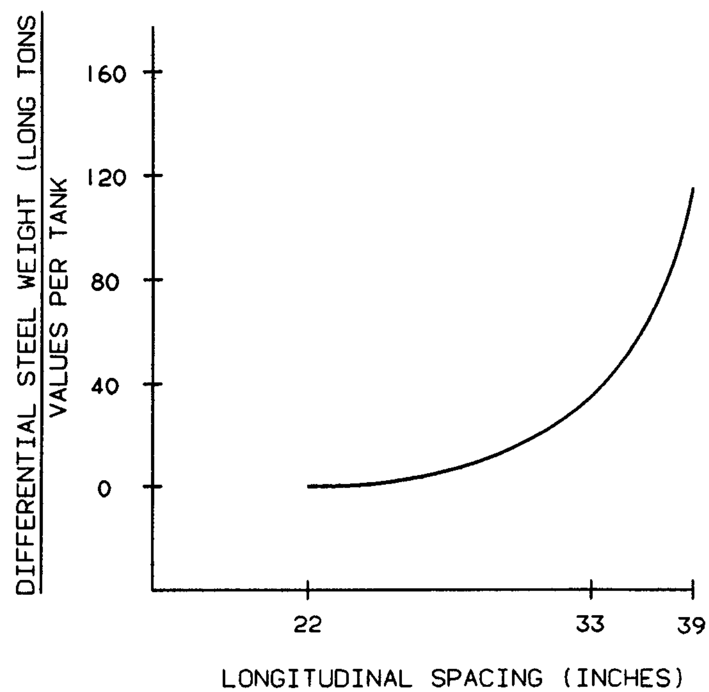


FIGURE 6 Relationship between longitudinal frame spacing and ship weight.

After a complete assessment of the information presented above, it was concluded that a 33-in. longitudinal spacing with the previously selected 11.5-ft transverse spacing was optimum. This decision balances ship construction costs with the potential owner's cargo deadweight requirements.

Structural Framing "Access"

Another key design parameter that must be addressed in the initial structural framing design of a double-hull tanker is ballast tank access. Our objective when designing access is to incorporate safe and effective access features for survey, maintenance, and construction.

Access requirements should:

- permit the surveyor to safely examine (close-up) the tank structure and coating system
- allow efficient maintenance and repair of coatings, structure, and piping systems
- allow the removal of injured personnel
- enhance outfitting during construction

To accomplish these objectives, each wing tank or double bottom tank should have two means of access from the open deck. Access on one end of the tank should be direct, vertical in-line access for equipment loading and unloading. The other access opening should have offset vertical openings in the longitudinal stringers for the safety of personnel entering and leaving the tanks. The minimum access size in longitudinal stringers and transverse webs should be 600 x 800 mm. Figure 7 shows the final access system for this design.

It should be noted that access holes in transverse webs are not located arbitrarily. Finite element analysis of a transverse web frame will show areas of high stress where openings should be avoided or must be compensated for.

Structural Framing "Construction Erection Plan"

With a midship section optimized by design and industrial engineering, production planning joins the concurrent design team to determine how the ship will be constructed. Review of the design for applicability of construction facilities will be performed to ensure the most efficient breakdown into superlifts and erection units.

Superlift and unit erection breaks identify the ship's and final assembly platen's structural products and, in doing so, determine the work that will be required in the drydock. In essence, it is the manifestation of the shipyard's construction strategy and an important cost determinant. Once the structural breakdown is completed, it becomes the basis for scheduling and for future manufacturing, construction, and outfitting planning. A good breakdown does not necessarily ensure efficient production, but it does establish the performance limits that will be achievable. Once the plan is developed, costs must be further minimized through the use of efficient production methods and by developing a practical component-installation plan.

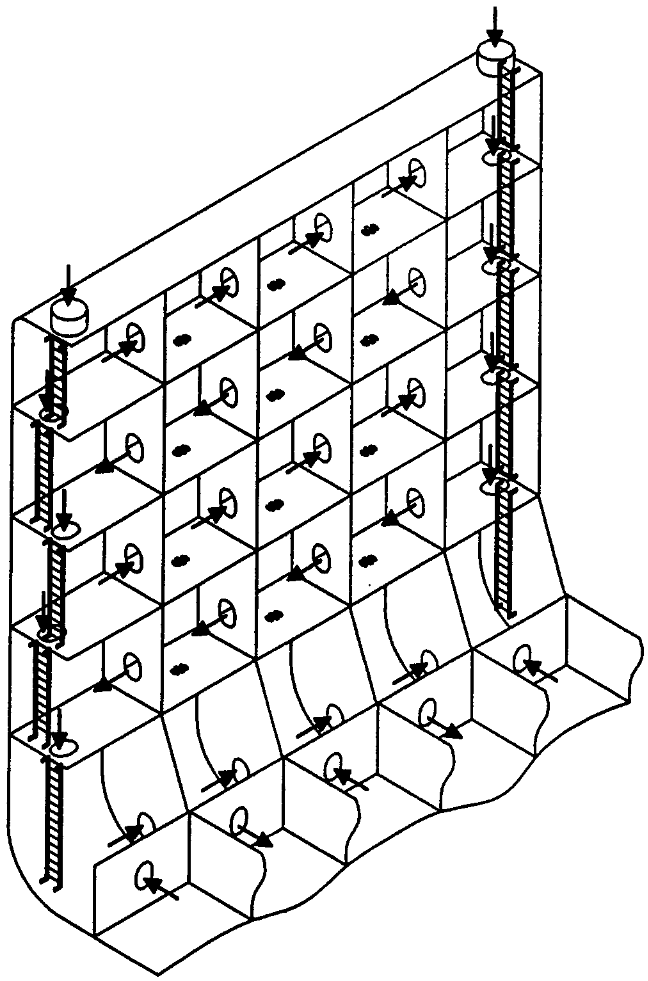


FIGURE 7 Wing ballast tank access.

Seven alternative construction erection scenarios were evaluated for the most efficient method. Based upon unit weights and sizes for this design, the decision was made to construct the vessel in dry dock number 12 under the 900 tonne gantry crane. A band concept was developed. Units similar to that shown in Figure 8, one cargo tank in length each, would be built for assembly in the dry dock.

STRUCTURAL ANALYSIS

With the arrangement of the side access girders complete and the transverse web spacing and longitudinal stiffener spacing set, the remainder of the midship scantlings are sized in accordance with regulatory body empirical equations.

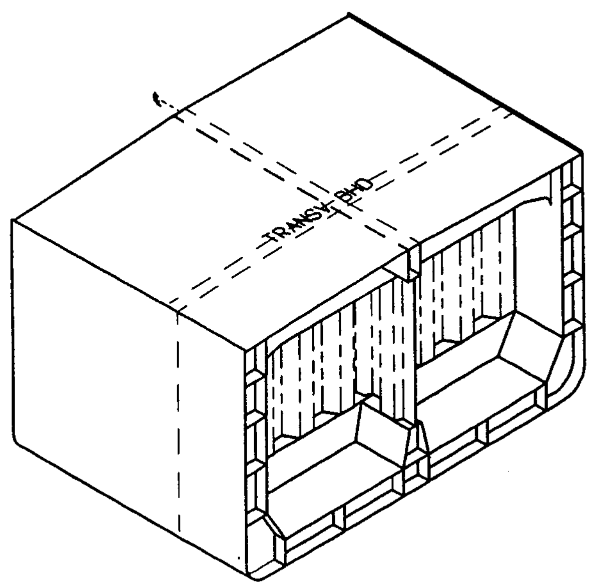


FIGURE 8 Construction erection scenario—single tank bank.

The longitudinal hull girder section modulus is calculated on an in-house personal computer program. Modifications are made to the midship design until the deck and bottom structure meet regulatory-body–minimum section-modulus requirements. This usually involves thickening the deck plating and longitudinals for a double-hull tanker. The bottom section modulus is next checked for possible reductions in plating thickness and longitudinal stiffener section modulus due to excess hull-girder section modulus. Global longitudinal hull-girder section modulus and local longitudinal-section modulus requirements for the design are now in compliance with regulatory requirements.

The next step to verify the structural design is to evaluate a transverse web. This is accomplished by creating a three-dimensional, finite element model of a complete transverse web including shell and longitudinal plating for one-half web spacing forward and aft of the transverse web. In this model longitudinal stiffeners are lumped with the plating, resulting in an increased plate thickness. Figure 9 is an example of this type of finite element model.

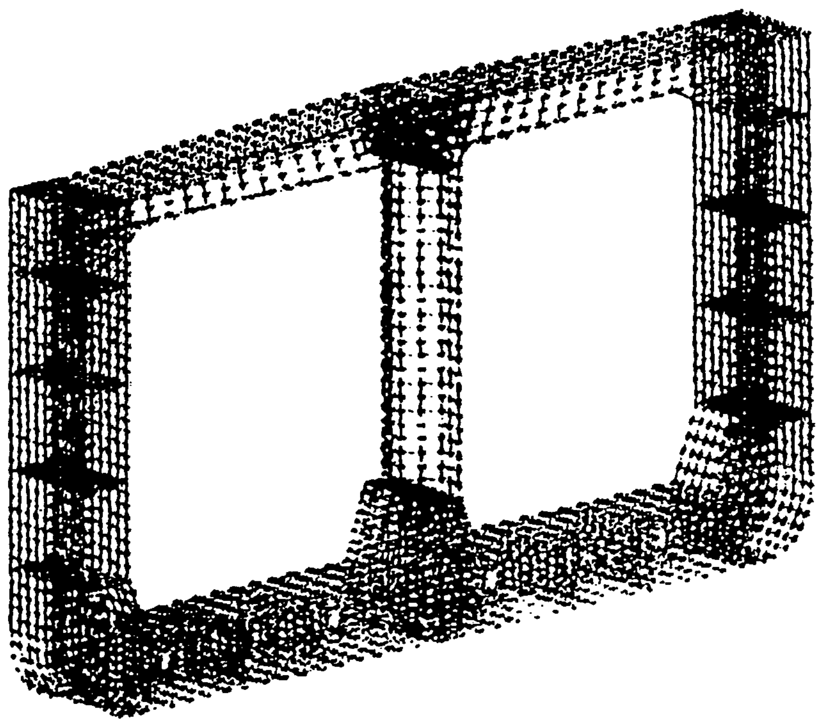


FIGURE 9 42,000 DWT transverse web model.

Loads for a worst case transverse loading condition are applied to the model. One of these load cases is a light draft, sag wave, transverse checkerboard (full-empty) loading pattern (see Figure 10).

The model was created using IDEAS for pre- and post-processing and using ABAQUS as the finite element solver. A model such as this would generally contain between 3,000 and 4,000 elements.

Post-processing review of the finite element model begins with deformed geometry plots. General deformed shapes should be consistent with those expected from the loading case with realistic deformation. After confirming that the model is performing properly, stress plots are examined. Von Mises stresses are compared to an allowable stress of 25,500 psi, or 75 percent of yield for mild steel, and shear stresses are compared to an allowable stress of 12,500 psi. Figure 11 shows a typical contour stress plot with areas of high stress identified with arrows.

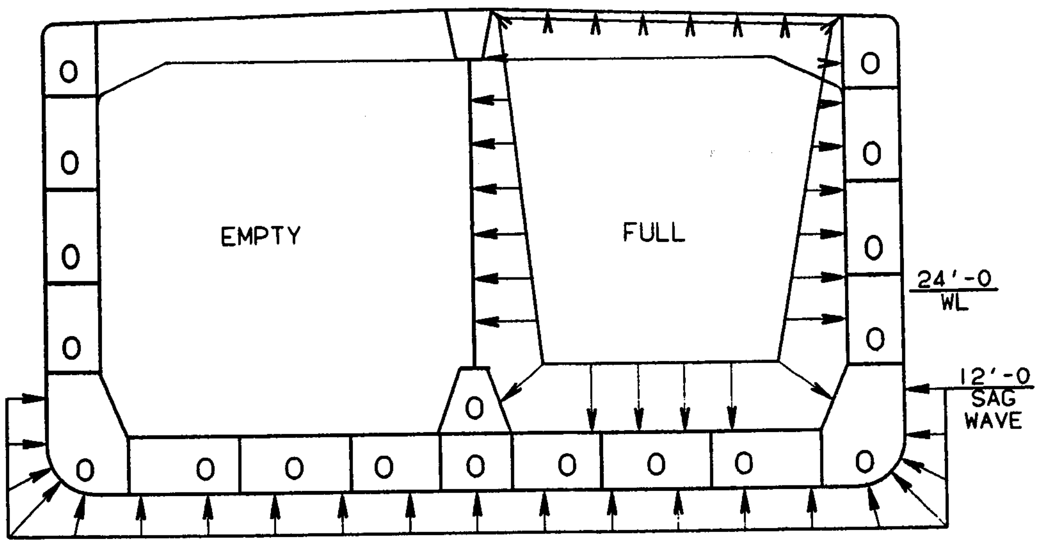


FIGURE 10 Finite element model—loading for transverse web.

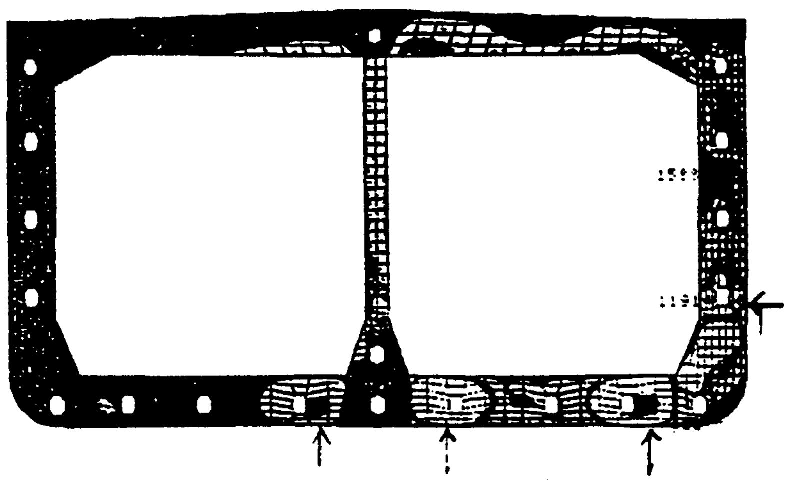


FIGURE 11 Finite element model—40,000 DWT Von Mises stresses.

The highest stress areas for this loading condition were located in the innerbottom on the loaded tank side to the right of the centerline stool girder, in the innerbottom to the left of the bilge girder, and in the side tank just above the upper bilge knuckle girder. It can be noted that the access openings attract stress concentrations and, therefore, must be located strategically or compensated for with thicker plating or coamings. The 12.5-mm plate in the innerbottom and the 14.0-mm plate in the side tank were adequate for this design, with stresses below allowable.

Preliminary weight estimates are calculated for the parallel midbody of the ship. Shape factors are used to estimate the weight of Tanks 1 and 2, and volume estimates are used to estimate the weight of the bow and stern. The structural steel weights are used in the calculation of the lightship weight. With a fairly accurate lightship weight, hydrostatics and stability calculations may be performed.

Stillwater bending moments for a scantling draft homogenous cargo loading at 98 percent full, with 10 percent and 90 percent consumables, along with ballast condition bending moments, with 10 percent and 90 percent consumables, are calculated. The highest of these sag and hog bending moments are then added to wave bending moments calculated from regulatory body requirements to create a combined maximum bending moment. This combined maximum bending is converted into an actual required hull-girder–section modulus by dividing by a maximum allowable stress of 11.33 long tons/in^2. The actual required section modulus is compared to the minimum allowable regulatory body hull-girder–section modulus.

The larger of the two numbers is used to determine if scantling reductions in the bottom shell and longitudinal stiffeners are possible. If the actual calculated section modulus from the midship design exceeds the minimum regulatory of the actual required, bottom reductions are taken. This amounted to a 1-mm, bottom-shell thickness reduction and a longitudinal stiffener reduction of 30 mm in depth.

At this point the preliminary midship design is considered complete.

LOCAL DESIGN

The structural design process involves review of detail connections to reduce stress concentration as well as to the improve productivity in manufacturing and minimize stress. One such example of this process was the redesign of the transverse web cuts for longitudinal stiffeners.

Review of the University of California Berkeley's hull crack database shows that approximately 36 percent of structural failures occur in panel stiffeners and clearance cutouts (see Figure 12). The number of problems in these details offered the ideal opportunity for design improvement.

A finite element analysis was performed with NASTRAN as the model solver and IDEAS for pre- and post-processing on three alternative web cut and panel stiffener configurations (see Figure 13). The locations for stress comparison are marked by "X."

Design 1 is a traditional web cut with small radiused corners and a sniped panel stiffener lapped on the longitudinal stiffener. Design 2 has a semicircular web cut and a sniped panel stiffener corner welded to the longitudinal stiffener. Design 3 has the same semi-circular web cut as Design 2 with an unsniped panel stiffener lap welded to the longitudinal stiffener. The idea

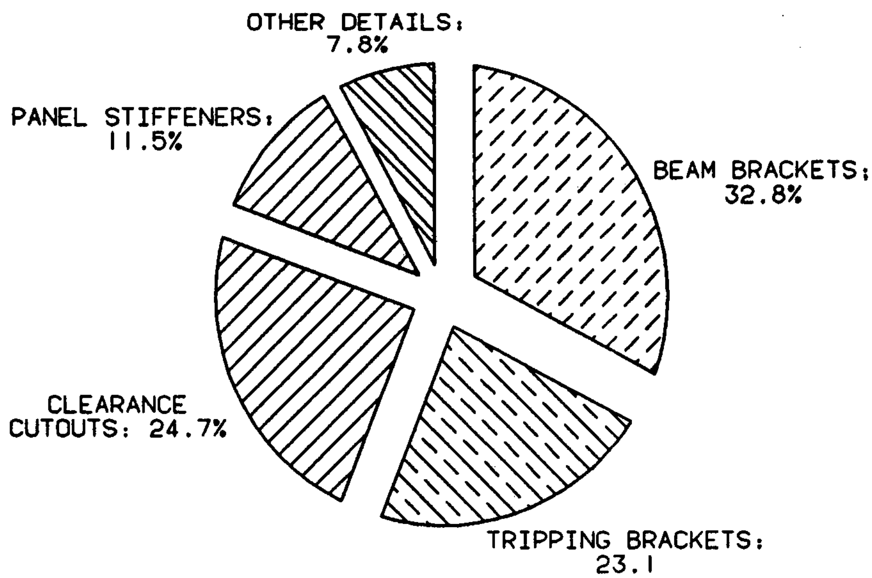


FIGURE 12 Failure percentages of structural details.

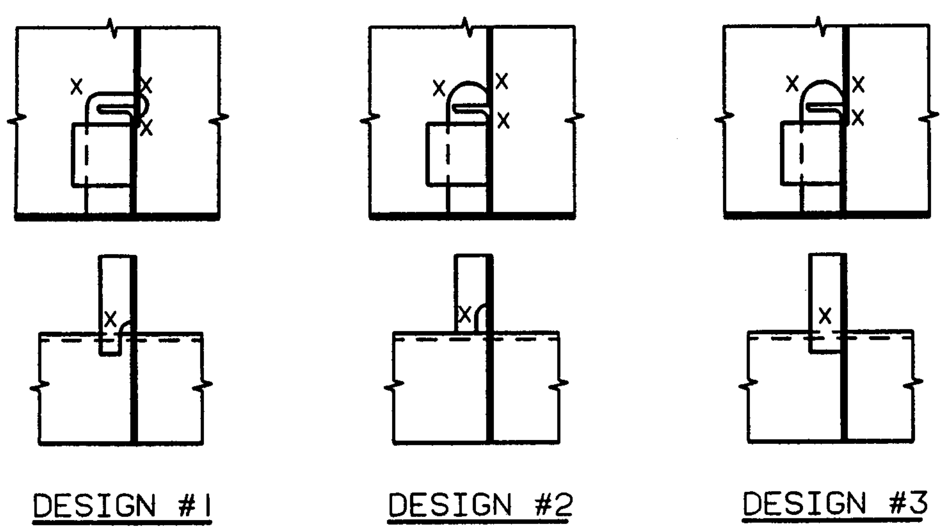


FIGURE 13 Web cut analysis.

behind the semicircular web cut was to reduce the discontinuity in the web caused by sharp corners, thus reducing stress concentrations. Lapping the panel stiffener without a snipe is made possible by the semicircular web cut. It offers more shear area in the snipe location and productivity improvements by eliminating the snipe cutting operation.

Figure 14 shows the type of finite element model created for each design. Each model contains approximately 2,000 elements, with the element size approximating the material thickness. This allows relatively good definition of stress concentration areas. Lap connections were modeled with multipoint constraints to model the welds accurately.

Identical loads and boundary conditions, which represent bottom shell worst case hog and sag condition, were applied to the three alternative design models. Figure 15 shows a typical post-processing stress contour plot used to compare the designs.

Review of the stress plots showed reduced stresses and areas of high stress in Design 3 compared with Designs 1 and 2. Table 3 summarizes these results.

TABLE 3 Web Cut Stress Results Summary

Stress	Design 1	Design 2	Design 3	Percent Improved from Design 1
Shear Stress (psi)	17,000	17,000	12,000	29.4
Von Mises Stress (psi)	31,000	31,000	22,000	29.0
Normal "Y" Stress (psi)	21,000	20,000	16,000	23.8

Although this is not fatigue analysis, the reduction of stress range by 30 percent will approximately double the fatigue life of the structural detail. This web cut design was subsequently incorporated into our midship section design. A producibility review was performed for the three design alternatives. Design 3 was selected as the preferred, most producible connection; however, no detailed cost savings analysis was done.

It should be noted that changes in design details can be made quite easily at this stage of preliminary design. Changes made during or after detail design are very costly. This is primarily because of the impact to a three-dimensional CADAM and VIVID®[1] product model, which consists of the entire detailed ship structure and is used for material ordering, fabrication packages, and assembly packages.

[1] VIVID is a registered service mark of Newport News Shipbuilding.

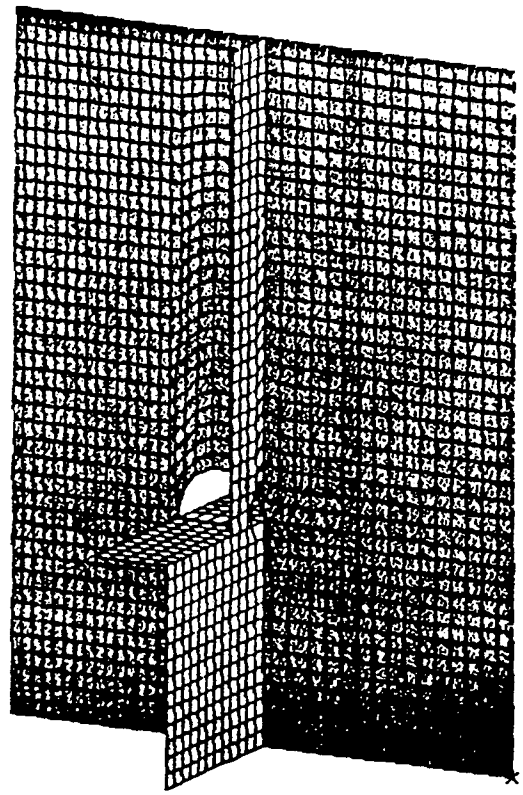


FIGURE 14 Finite element model.

CONTRACT ANALYSIS

When a ship design moves from preliminary to a contract stage additional analysis is performed on both local and global levels. Dependent upon the regulatory agency used for classification, different finite element analyses are required. ABS, for example, will require structural verification using "SafeHull." SafeHull is divided into Phase "A" and Phase "B." Phase "A" sizes scantlings and performs a fatigue assessment on detailed connections. Phase "B" consists of finite element modeling three tanks of the cargo box parallel midbody, which verifies the scantling selections of Phase "A."

In addition to classification analysis requirements, spectral fatigue analysis is often performed. This analysis subjects local structure to loads from wave spectra of intended service routes for the expected life of the vessel. The objective of this advanced analysis is both to verify the ship's global strength and to minimize future maintenance and repair problems. With proper loading, meshing, and application of boundary conditions, finite element analysis will offer design confidence and improvements.

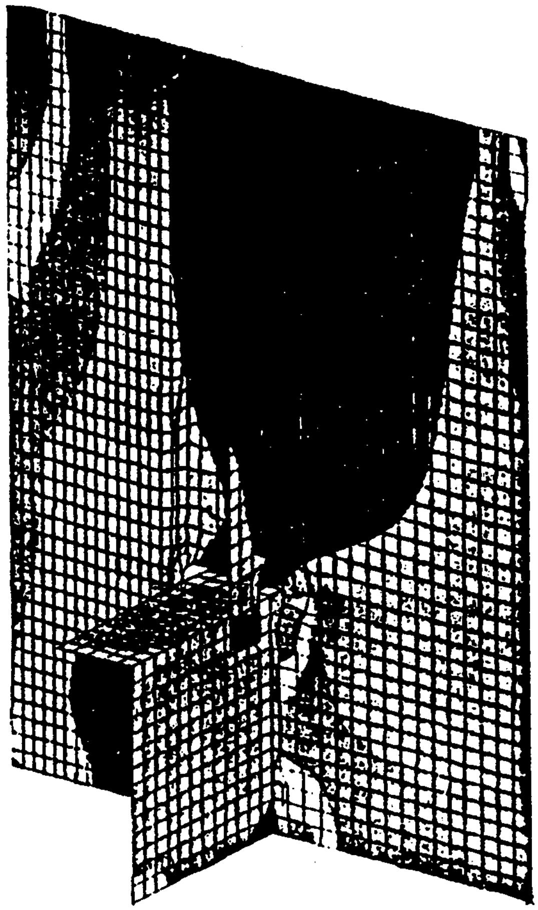


FIGURE 15 Stress contour plot.

SUMMARY

Design Engineering, Industrial Engineering, and Production Planning work together on the structural configuration of a ship to minimize cost and maximize the use of production facilities. Changes in material cost, labor rates, cargo deadweight requirements, and facility upgrades will affect the optimal steel design.

FATIGUE AND FRACTURE OF SHIP STRUCTURES

Harold S. Reemsnyder

ABSTRACT

Two state-of-the-art approaches to fatigue life and residual strength assessment that have received extensive experimental corroboration and wide application are presented. The component-test, life-assessment model applied to weldment fatigue and the Miner-Palmgren cumulative damage model are described in detail with brief mention of the Neuber (local-strain) and fracture mechanics life-assessment models. The failure assessment diagram currently used to assess the residual strength and structural integrity of cracked elements is presented in greater detail because of the lack of a single comprehensive guide. Also, the philosophy and current practice of fitness for purpose are outlined.

INTENTIONS

1. Provide a tutorial document that will introduce to, or refresh for, the reader certain tools of fatigue life and residual strength assessment.
2. Provide the reader with a document that will give guidance to the pertinent references for fatigue life and residual strength assessment and to the steps necessary to perform a residual strength assessment.

ASSUMPTIONS

1. The reader has a basic understanding of fatigue and fracture mechanics,
2. The reader knows the definition of S–N curve, cumulative damage, stress-intensity factor, J-integral, crack-tip opening displacement, and stable and unstable crack extension.
3. Another presentation in the symposium will discuss the acquisition and sources of marine load spectra necessary for fatigue life and residual strength assessment.

INTRODUCTION

A structure or structural component may fail by one or more mechanisms:

- plastic collapse
- buckling
- fatigue
- fracture
- corrosion
- stress corrosion cracking
- creep

The discussion herein is directed to the failure mechanism *fatigue* followed by either *plastic collapse* or *fracture*. Fatigue failures in structural components are always initiated at notches or stress raisers. Notches may be classified as mechanical or metallurgical. Mechanical notches include sudden changes in shape (fillets, thickness-transitions, etc.), nicks, scratches, pitting, and similar notches. Metallurgical notches include inclusions, blowholes, porosity, quench cracks, and so forth.

The fatigue life of a structural component consists of three phases:

- initiation of a crack, that is, growth and coalescence of micro-cracks to form a macroscopic crack capable of solid mechanics modeling
- growth of the macroscopic crack to a critical size
- exceeding the residual strength of the cracked element, causing either plastic collapse or fracture (partial or complete)

The relative magnitudes of the three phases depend upon the notch acuity, material, structural stiffness, and environment. Table 1 lists the various fatigue life assessment models and fatigue life phases that are, *or could be*, accommodated by the various models.

The fatigue life assessment models of Table 1 have been described and illustrated elsewhere (Reemsnyder, 1983a, 1983b). In general, the fatigue-critical details of a ship (as well as all land and marine vehicles and structures) are the weldments, as shown in Figure 1 (Jordan and Krumpen, 1984; Ramwell, 1993). Therefore, discussion of fatigue in this paper, Part I, will be limited to the fatigue of weldments. Assessment of the structural integrity, that is, exceeding the residual strength, is discussed in Part II.

PART I: FATIGUE OF WELDED JOINTS

Fatigue of Welded Joint Details

Traditionally, designs of, and design specifications for, welded joints have been based on test data, that is, the *Component Test Model* of Table 1. Therefore, parameters affecting the fatigue resistance of weldments will be discussed below in the light of test results. Compilations of fatigue test results on welded joints and discussions of the parameters affecting the fatigue resistance of weldments have been published (Gurney, 1979; Munse and Grover, 1964; Reemsnyder, 1968, 1978a; Maddox, 1991; Radaj, 1990; Munse et al., 1983), and references will be made only to particular papers when merited. The life assessment models, Table 1, used currently in a wide range of weldment fatigue design and fitness-for-purpose criteria are discussed in Reemsnyder (1982).

The fatigue-critical areas of all complex weldments (e.g., Figure 1) may be modeled by simple, axially-loaded fatigue specimens containing either butt or fillet welds. Therefore, such specimens have been the most widely used test configurations. These specimens are generally fabricated with plate thicknesses and weld sizes and processes identical to those of full-scale structures. However, the specimen width is usually limited by the load capacity of the test system to the order of 1 in. to 6 in. (25 mm to 152 mm).

TABLE 1 Fatigue Life Assessment Models

Models	Phases of Fatigue Life: Crack Initiation	Crack Propagation	Residual Strength
Analytical:			
Stress-life	✓		✓
Local strain (Neuber)	✓		
Fracture mechanics	✓	✓	✓
Experimental:			
Component test	✓	✓	✓
Structural test	✓	✓	✓

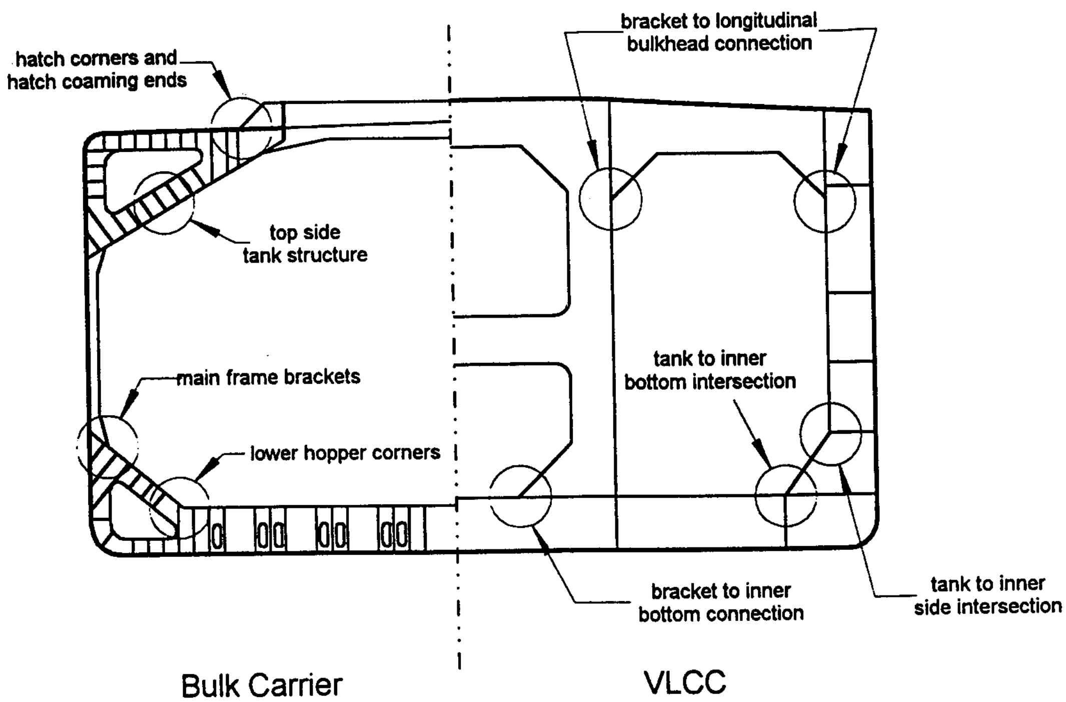


FIGURE 1 Typical fatigue-critical details in ships.

Butt Welds

Butt welds may be either transverse or longitudinal (Figures 2a and 2b, respectively). In full-penetration transverse butt welds with reinforcement intact, fatigue cracks are initiated at the weld toe, where a geometric stress raiser exists (Figure 3a). However, in the case of partial-penetration transverse welds, cracks are initiated at the weld root (Figure 3b). With the reinforcement removed, cracks are initiated either in the base metal or in the weld at inclusions or gas pockets (porosity). Cracks are initiated in longitudinal butt welds at weld-bead surface imperfections, such as ripples, or at a point of change of electrode in manual welding. Internal discontinuities, such as lack of penetration, are not critical when aligned with the principal stress.

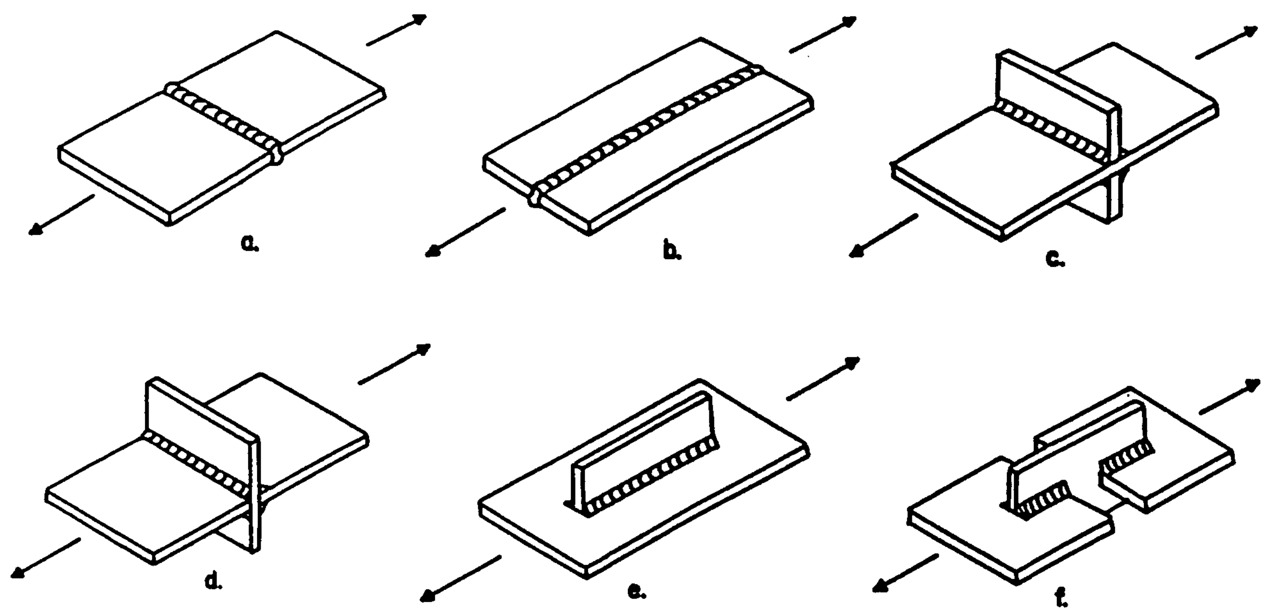


FIGURE 2 Axially loaded weldment specimens.
Source: Reemsnyder, 1978a.

The fatigue resistance of structural steels, as well as the difference in fatigue resistance between various grades, is reduced by the presence of transverse butt welds (Reemsnyder, 1974). The effect of stress ratio is also less pronounced in the case of butt welds. Removal of reinforcement from transverse butt welded carbon steel *may* increase the fatigue life by a factor of 4 or 5 (Reemsnyder, 1969). Similar increases have also been observed for butt welded constructional alloy steel (Munse and Grover, 1964). *It should be cautioned that the process used to remove the reinforcement can introduce notches that are more severe than the presence of the reinforcement.*

The fatigue resistance of carbon steel is little affected by the presence of longitudinal butt welds. On the other hand, limited tests show that longitudinal butt welds reduce the fatigue life of low-alloy, high-strength steels. This is probably due to the fact that the weld metal, in the latter case, was of a lower strength than the base material (Reemsnyder, 1974).

Fillet Welds

Transverse and longitudinal fillet welds are either non-load–carrying (Figures 2c and 2e) or load-carrying (Figures 2d and 2f). The longitudinal non-load–carrying fillet weldment of Figure 2e simulates only those cases in which the fillet weld is terminated or intermittent in a region of high stress. In transverse fillet welds, cracks are initiated at either the weld toe or root

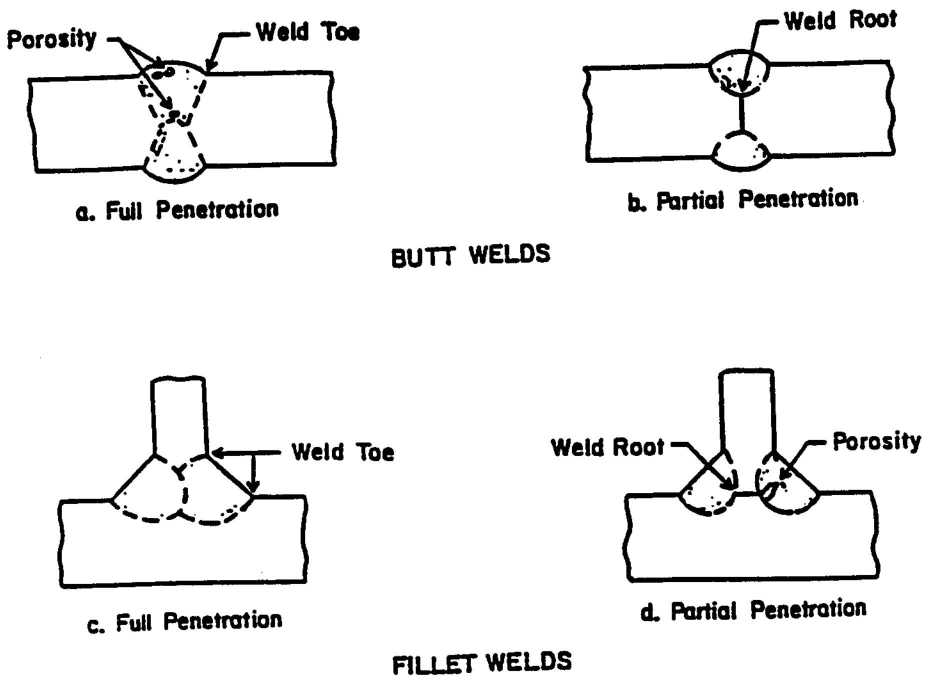


FIGURE 3 Typical weldment notches.
Source: Reemsnyder, 1978a.

(Figures 3c and 3d). Cracks are initiated at the toe of intermittent longitudinal fillet welds, where there is a severe mechanical notch.

A continuous, longitudinal fillet-welded specimen has been developed to simulate the flange-web weldment of built-up beams, axially loaded box members, or attachments (Reemsnyder, 1965a). Fatigue tests utilizing this specimen have been made for both carbon steel and constructional alloy steel (Reemsnyder, 1969; 1965b). The reduction in life of a continuous, longitudinal fillet weld is due, generally, to a notch, such as porosity (Figure 3d) or a crater due to change of electrode. On the other hand, transverse or intermittent longitudinal fillet welds (Figures 2c to 2f, inclusive) combine the detrimental effects of both external and internal notches. The fatigue strengths for a given life of transverse and intermittent longitudinal fillet welds are about one-half of those for continuous longitudinal fillet welds (Reemsnyder, 1974).

Welded Beams

Fatigue failures in manually welded built-up beams may be initiated at craters due to electrode changes in the flange-web fillet weld (Figure 4) or weld discontinuities (e.g., porosity) and propagate into both the flange and web. Fatigue failures in submerged arc, automatic-welded beams may be initiated at surface ripples or discontinuities such as porosity (Figure 3d). Most failures at beam splices are initiated at the weld toes or at discontinuities in the weld. Failures at partial-length cover plates generally are initiated at the toe of the transverse seal weld and at the ends of intermittent, longitudinal fillet-weld toes. In the case of stiffeners welded to the web, fatigue failure usually is initiated at the termination of the web-to-stiffener fillet weld. The crack propagates up into the web in the panel toward the load point and along the flange-to-web fillet weld in the panel away from the load point. When the stiffener is welded to the tension flange, failure is initiated at the toe of the fillet weld on the flange.

The fatigue resistance of rolled beams is similar to that of plain, as-rolled material but greater than that of welded, built-up beams. Built-up beams of high-strength, low-alloy steel, with fillet welds laid both manually and automatically, have been cyclically loaded; the fatigue strength at 2 million cycles was increased by about 35 percent when automatic submerged arc welding was used (Gurney, 1979). Partial-length cover plates significantly reduce the fatigue strength of welded beams, but, on the other hand, full-length cover plates have no effect. The scatter band for partial-length cover plates includes various details and butt-welded flange transitions. It has been suggested that cover plates should be extended past the theoretical cut off point a sufficient distance so that the stress at the end of the cover plate is 40 percent of the stress at its center. Intermittent fillet welds on cover plates should be avoided. Beam splices have little or no effect on the fatigue resistance of welded beams. Various types of stiffeners, some welded to the tension regions of beams and some not, have been studied. No one type of stiffener had a particular advantage over another, although welding to a tension zone in the web or flange could lead to an early fatigue failure.

Notches in Weldments

A weldment generally contains both external and internal notches (Figure 3). Notches include changes in section due to reinforcement or weld geometry, surface ripples, undercuts, and lack of penetration. In addition, welds may be subject to such internal heterogeneities as shrinkage cracks, lack of fusion, porosity, and inclusions.

Shrinkage cracks are caused by excessive restraint exerted by adjacent material to shrinkage of the weld zone on cooling. Presence of slag or scale on the surfaces to be welded may lead to lack of fusion. Porosity is caused by gas entrapped during solidification of the weld metal, excessive moisture in the electrode coating, or disturbance of the arc shield by drafts. Slag inclusions from the electrode coating are one of the most common weld discontinuities encountered. One cause is imperfect cleaning of the weld between successive passes.

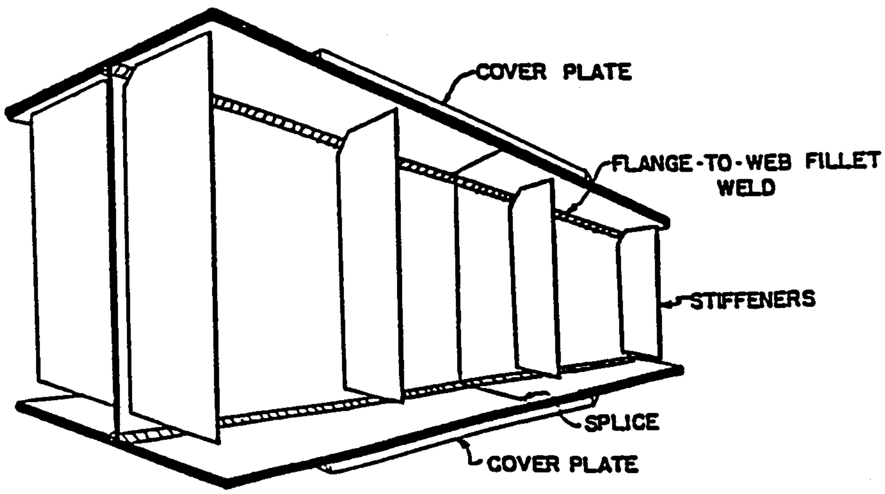


FIGURE 4 Welded built-up beam.
Source: Reemsnyder, 1974.

The importance of mechanical notching on fatigue resistance of weldments is illustrated by the fact that removal of butt weld reinforcement raises the fatigue resistance at long lives to that of the base metal (Figure 5). The stress concentrations due to weld geometry have been the subject of much study (Gurney, 1979; Green and Marlin, 1951; Sanders et al., 1965; Selby et al., 1965; Williams et al., 1970; Arockiasamy et al., 1989; Raghavendran and Fourney, 1994). The effect of reinforcement height is shown in Figure 5 for a quenched and tempered carbon steel, 0.75-in. (19-mm) thick, with a tensile strength of 114 ksi (786 MPa). Doubling the height of reinforcement (h in Figure 5) reduced the fatigue strength of 2 million cycles by approximately 67 percent (Reemsnyder, 1979).

The geometric notch severity of the weld reinforcement accounts for the frequently observed phenomenon that the fatigue strength of transverse butt welds with reinforcement intact (or of transverse fillet welds) is insensitive to tensile strength. This effect is demonstrated in Figure 6, where a data band for fatigue-test results of transverse-butt–welded, hot-rolled carbon steels; high-strength, low-alloy structural steels; quenched and tempered carbon steels; and constructional alloy steels and an additional data band for hot-rolled carbon steels are plotted (Reemsnyder, 1979). Although the tensile strengths of the steels shown in Figure 6 varied from 58 ksi to 148 ksi (400 MPa to 1020 MPa), there are no significant differences among the fatigue

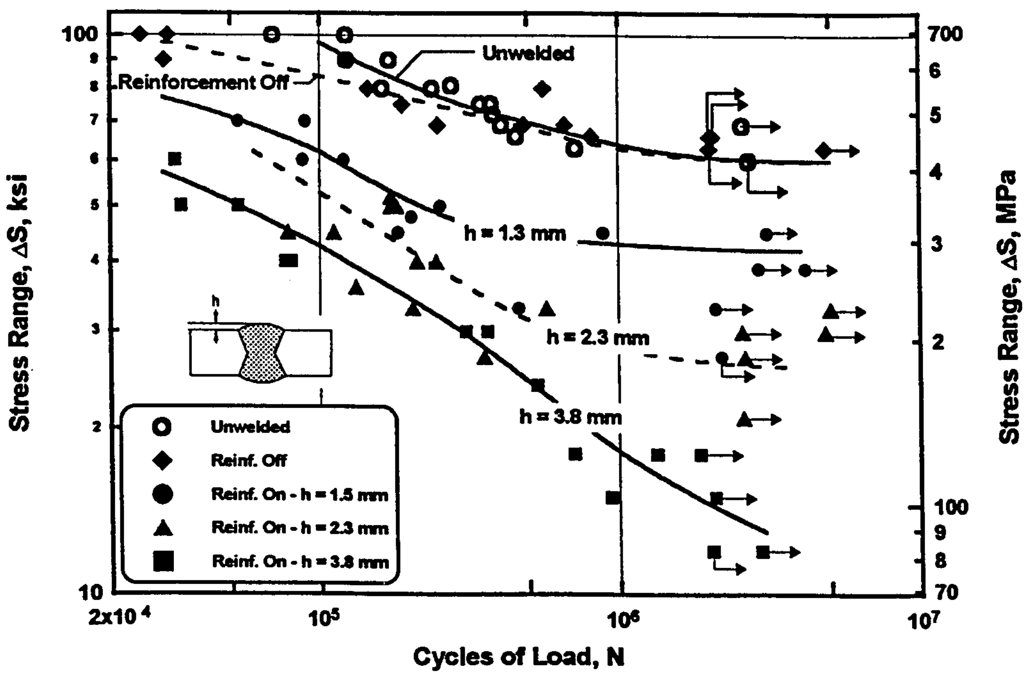


FIGURE 5 Transverse butt welds, quenched and tempered carbon steel, R = 0. Source: Reemsnyder, 1979.

strengths of the various steels. For a given investigator, there is a slight stratification with tensile strength. However, the laboratory-to-laboratory variability for a given grade is far greater. In all likelihood, the sensitivity to reinforcement geometry contributes to the significant variability seen from one investigator to another.

In a recently completed Ship Structure Committee project, a fatigue design strategy for welded ship details was presented (Stambaugh, 1994). This strategy, using nominal stress, cumulative-damage theory and experimentally determined stress-concentration factors, was developed to guide ship designers in the improvement of weld-detail fatigue life through the selection of details with lower geometric notch severities.

Experimental techniques and a lack of uniform criteria for judging the severity of internal discontinuities make it difficult to discuss, quantitatively, the effects of these discontinuities on fatigue strength. It is possible, however, to draw some general conclusions from reviews of existing tests (Gurney, 1979). Lack of penetration lowers the fatigue strength of transverse welds significantly but has relatively little effect on longitudinal welds. Porosity and slag inclusions decrease fatigue resistance in proportion to the decrease in effective weld area of transverse welds. Microstructural changes due to severe quenching concomitant with the sudden

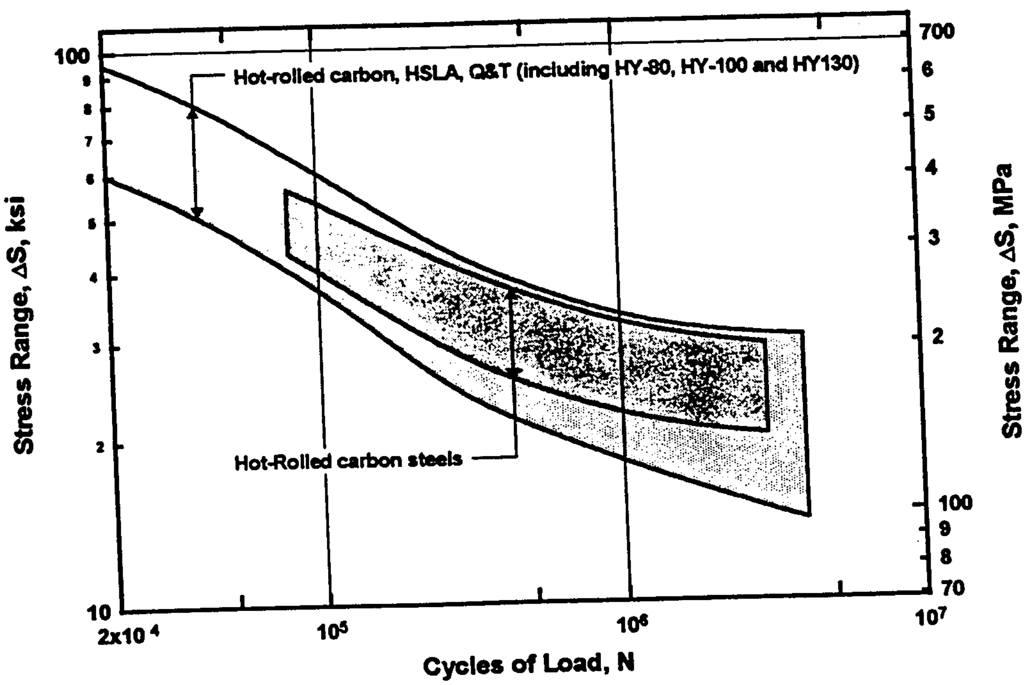


FIGURE 6 Effect of grade, transverse butt welds, reinforcement intact, R = 0.
Source: Reemsnyder, 1979.

extinguishing of the welding arc may initiate a fatigue crack at the point of change of electrode. Severe quenching also results from stray flashes and weld splatter. Such stress raisers may be reduced or eliminated by control of the welding procedure.

The presence of internal weld discontinuities can contribute to the variability observed in the fatigue testing of weldments. Wide variations in fatigue life have been observed when cracks were initiated at internal discontinuities. This variability decreases markedly for specimens in which cracking is initiated at the toe of the weld.

Typical weld discontinuity acceptance criteria are generally based on what is considered to constitute good workmanship. *Often, however, repair of discontinuities can, in some cases, be detrimental to subsequent service* (Slater, 1985). A more rational basis for the assessment of weld discontinuities is the concept of *fitness for purpose* discussed in a later section (Maddox, 1993).

Residual Stresses

Deposition of weld metal in a joint is similar to the casting of steel. Welds are subject to such internal discontinuities as shrinkage cracks, lack of fusion, porosity, and inclusions. The heat-affected zone[1] of the base metal is heated by the welding procedure to a temperature above the lower critical temperature (the temperature at which a steel changes phase), and the rate of cooling, governed by the conduction of heat away from the fusion zone, will produce metallurgical transformations similar to those produced by deliberate heat treatment.

Residual stresses due to welding are formed as a result of the differential in heating and cooling rates at various locations in the material. In addition, due to these thermal gradients, some of the material will be elastic while other regions are plastic. The interaction between these regions results in residual or internal stresses after cooling. These stresses may be quite large and will be tensile in the vicinity of the weld where their magnitude is approximately equal to the yield strength of the deposited weld metal, (Figure 7).

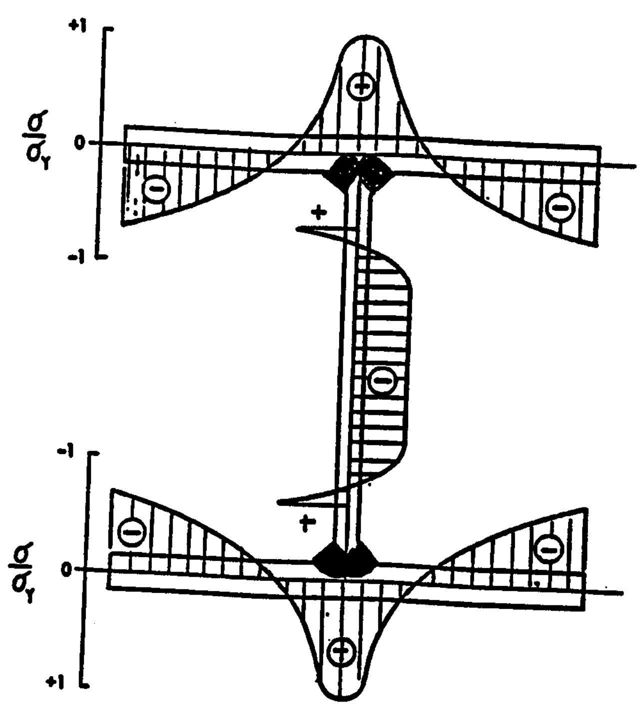


FIGURE 7 Residual stresses in a welded beam.

[1] The heat-affected zone is that portion of the base metal that has not been melted but in which the mechanical properties and microstructure have been changed by welding.

Peening. The introduction of compressive residual surface stresses at stress raisers can increase the fatigue resistance of weldments. For example, shot peening of non-load–carrying, fillet-welded carbon steel (Gurney, 1979) and butt welded constructional alloy steel (Doty, 1955) increased the fatigue strengths at 2 million cycles by 20 percent to 40 percent. The efficacy of shot peening for fatigue resistance is strongly influenced by shot size, arc height, and percent coverage (Campbell, 1971). For example, the 20 percent and 40 percent improvement in the case of quenched and tempered constructional alloy steel was achieved by peening to an arc height of 0.010 C to 0.012 C (Doty, 1955).[2] On the other hand, an improvement of only 7 percent was observed for a steel of similar tensile strength peened to an arc-height of 0.005 C to 0.007 C (Reemsnyder, 1979). Therefore, for an improvement in fatigue resistance to be significant and repeatable, shot peening must be closely controlled. The importance and selection of shot-peening parameters to optimize fatigue resistance are illustrated in Bignonnet et al. (1987). Improvements in weldment fatigue resistance due to peening are reviewed in Maddox (1985).

Hammer peening has been observed to improve the 2 million-cycle fatigue strength of carbon steel butt welds by 15 percent to 25 percent and that of fillet welds by 20 percent to 50 percent (Gurney, 1979). In contrast, hammer peening of a quenched and tempered carbon steel was observed to reduce the long-life fatigue strength by 9 percent (Reemsnyder, 1979). In general, hammer peening should not be considered the equivalent of a carefully controlled shot-peening program in the fabrication of cyclically loaded elements.

Other. Spot heating to introduce compressive residual stresses at the toes of fillet welds (Gurney, 1979) and of welded attachments in beams (Puchner, 1960) has been observed to double the fatigue strength at 2 million cycles. Proof loading also increases the fatigue resistance of weldments at lives greater than 1 million cycles. This increase is probably the result of a favorable alteration of the residual-stress distribution. TIG dressing of the weld toe can improve significantly the fatigue strength of weldments (10 percent to 50 percent) (Millington, 1973; Metal Construction, 1978; Zaczek, 1984). Also TIG remelting of in-service weld toes to remove fatigue-damaged regions has been shown to extend service life (Pyle et al., 1990). Weld-toe grinding can improve the fatigue resistance by up to 30 percent (Booth, 1986). However, gouging and/or burning during grinding must be avoided. It should be noted, however, that periodic compressive loads in variable-amplitude spectrum could eliminate the above mentioned beneficial effects of compressive residual stresses.

Thermal Stress Relief.[3] A topic of wide and continuing interest is the effect of residual stresses and their reduction through post-weld heat treatment; that is, thermal stress relief, on the fatigue resistance of weldments. Welding residual stresses are formed as a result of the differential in heating and cooling rates at various locations in the material. Due to these thermal

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[2] Almen C strip.

[3] This section on thermal stress relief is abridged from Reemsnyder (1981).

gradients, some regions of the material are elastic, whereas while others are plastic. The interaction between these regions results in residual or internal stresses after cooling. These stresses may be quite large and will be tensile in the vicinity of the weld, where their magnitude is approximately equal to the yield strength of the weld metal.

Thermal stress relief can increase the fatigue strength of weldments at lives greater than 1 million cycles, provided that the stress ratio is less than or equal to zero and notches are present in the weldment. For example, the fatigue strength of transverse butt welds with reinforcement intact at 2 million cycles has been increased by up to 12 percent and 24 percent to 33 percent for stress ratios of, respectively, R = 0 and R = 1, through thermal stress relief.[4] Such improvement has also been shown for longitudinal non-load–carrying fillet welds (e.g., attachments, gussets, etc.), where the increases in fatigue strength at 2 million cycles due to thermal stress relief for stress ratios of 0, -1, and -4 were, respectively, 15 percent, 57 percent, and 168 percent. Also the 2-million-cycle fatigue strength of transverse load-carrying fillet welds at R = 0 has been improved by 19 percent through stress relief. Fatigue tests on heavy-duty, spot-welded, single lap joints have shown that stress relief improves the 3-million-cycle fatigue strength at R = 0 and R = -1 by, respectively, 17 percent and 57 percent. On the other hand, thermal stress relief has little or no effect on the fatigue resistance of longitudinal butt welds, transverse butt welds with reinforcement removed, transverse non-load–carrying fillet welds, or longitudinal fillet welds terminating well beyond the highly stressed region.

The improvement in fatigue strength due to stress relief increases with an increase in life and with a decrease in stress ratio below zero, that is, when the stress varies from tension to compression in every cycle. Also, in this regime, for example, at R = -1 and at 2 million cycles, the benefits of stress relief increase with an increase in specimen size. The effect of specimen size on the efficacy of stress relief is probably due to the sequence of specimen preparation steps. In the case of small fatigue specimens that are sawed from a large welded plate, the residual stresses are mechanically relieved by the sawing, and subsequent thermal treatment would have no further relieving effect. On the other hand, when welding is performed after specimen blanks are sawed from the parent plates (typical of the larger specimens), the welding residual stresses are present and undiminished in magnitude in the specimens prior to thermal stress relief and/or fatigue testing. It should not be overlooked that the geometry and, therefore, the stress concentration factor of weld reinforcement may vary with plate thickness and contribute to the size effect in butt weldment fatigue resistance (Yoshida et al., 1978).

Environment. A corrosive environment can have a deleterious effect on weldment fatigue strength. Increasing concern for fatigue in high-performance ships and offshore structures has led to the fatigue testing of welded specimens in either a salt solution or sea water. Crack initiation in weldments tested at 0.1 Hz does not appear to be affected significantly by sea water. However, presence of salt water accelerates fatigue crack growth both in low-cycle and high-cycle fatigue at 0.1 Hz (Reemsnyder, 1978a).

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[4] Stress Ratio R is the ***algebraic*** ratio of minimum stress per cycle to maximum stress per cycle.

The cathodic protection of weldments in a seawater or saltwater environment restores the fatigue strength to that of air. However, the overprotection by cathodic means can in some cases lead to hydrogen embrittlement and the loss of low-cycle fatigue resistance (Reemsnyder, 1978a).

Corrosion fatigue resistance is sensitive to even small changes in test environment factors such as pH and test frequency. Very little long-life data have been developed to date, and the existing long-life data have been obtained at high test frequencies.

The effect of sea water on fatigue-crack propagation has been studied in American Society for Testing and Materials (ASTM) A36 weldments, including underwater repair welds (Matlock et al., 1987); ABS EH36, under spectrum loading (Cheng, 1985); and HY-80 weldments in low-temperature service (Ebara et al., 1981). Mathematical models for corrosion fatigue-crack initiation and propagation in marine steels have been developed for both low- and high-cycle regimes (Burnside et al., 1984).

Design of Weldments

Weldment Fatigue Design Criteria

In the past, all weldment fatigue design criteria considered, in general, the effects of maximum and minimum stress (or mean stress and stress range), material tensile strength, and the stress-concentration effect of various details on the service life of a structure. These criteria were presented in the form of *constant life diagrams* (CLD).

Construction of a Constant Life Diagram. In fatigue testing of structural connections, the stress ratio, R, may be held constant during the construction of a given S–N curve. However, it is impractical to construct one S–N curve for each value of R (or mean stress). A convenient graphical presentation of fatigue behavior is a CLD, such as the Ros Diagram (Figure 8). In Figure 8, the ordinate of a general point is the maximum stress S_{max} in a single stress cycle, and the abscissa is the minimum stress S_{min} in that cycle. The curve ABCD (Figure 8), represents the critical combination of S_{max} and S_{min} that will cause failure in N_i cycles. The coordinates of points A, B and C in Figure 8 are taken from the S–N curves for each R. The coordinates of point D are each the value of engineering tensile strength. If a given set of maximum and minimum cyclic stresses falls below the curve ABCD (Figure 8) the fatigue life will exceed N_i. If, on the other hand, the point falls on or above the curve ABCD, failure is likely to occur at or prior to N_i cycles. In Figure 8, the critical stress range S_r for a life of N_i cycles is the vertical distance between curve ABCD, and the ray R = +1. It must be remembered that the curve ABCD usually represents the mean or *50 percent survival life with a confidence of 50 percent.* Obviously, a CLD can be constructed for any value of percent survival at any confidence.

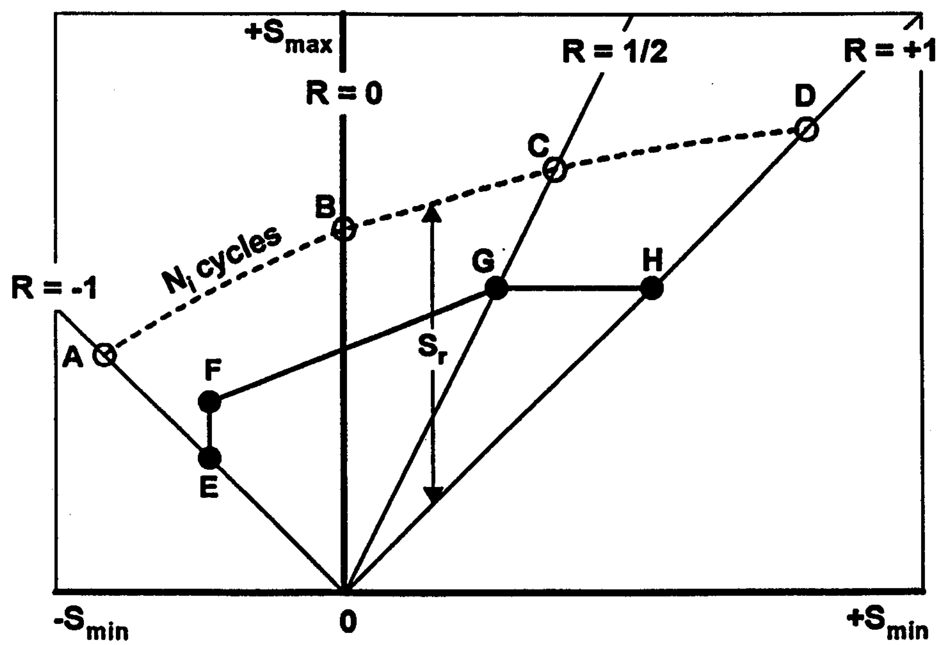

FIGURE 8 Construction of CLDs.

Use of a Constant Life Diagram. Design criteria may be modeled by the CLD as shown in Figure 8, where lines GH, EF, and FG are, respectively, the allowable tensile stress, the allowable compressive stress, and the maximum allowable cyclic stress for N_i cycles. Line FG is offset by safety factors from the dashed line ABCD, which is fitted to fatigue strengths at N_i cycles from test-data S–N curves for various stress ratios R. At one time, most major weldment fatigue criteria were presented in the form of CLDs for each material and type of weldment detail. A few criteria today still use the CLD concept presenting a CLD at some specified life for each material and detail. See, for example, AAR (1981).

A CLD is shown in Figure 9 for fatigue tests at several stress ratios for axially loaded carbon-steel butt welds (transverse, reinforcement intact).

Stress Range. Examination of weldment fatigue data (primarily, pulsating tension) when plotted as CLDs shows that the line ABC (Figure 8) is in many cases practically parallel to the ray R = +1. Also, this is obvious in Figure 9. That is, the fatigue strength is sensitive to stress range S_r but is insensitive to mean stress S_m. (The vertical distance from the ray R = +1 to the line ABC in Figure 8 is the magnitude of the stress range.) The insensitivity of weldment fatigue data to stress ratio or mean stress is illustrated in Figure 10, where there is little difference among the S–N curves for transverse butt welds, reinforcement intact, at stress ratios of R of -1, 0, and 1/2.

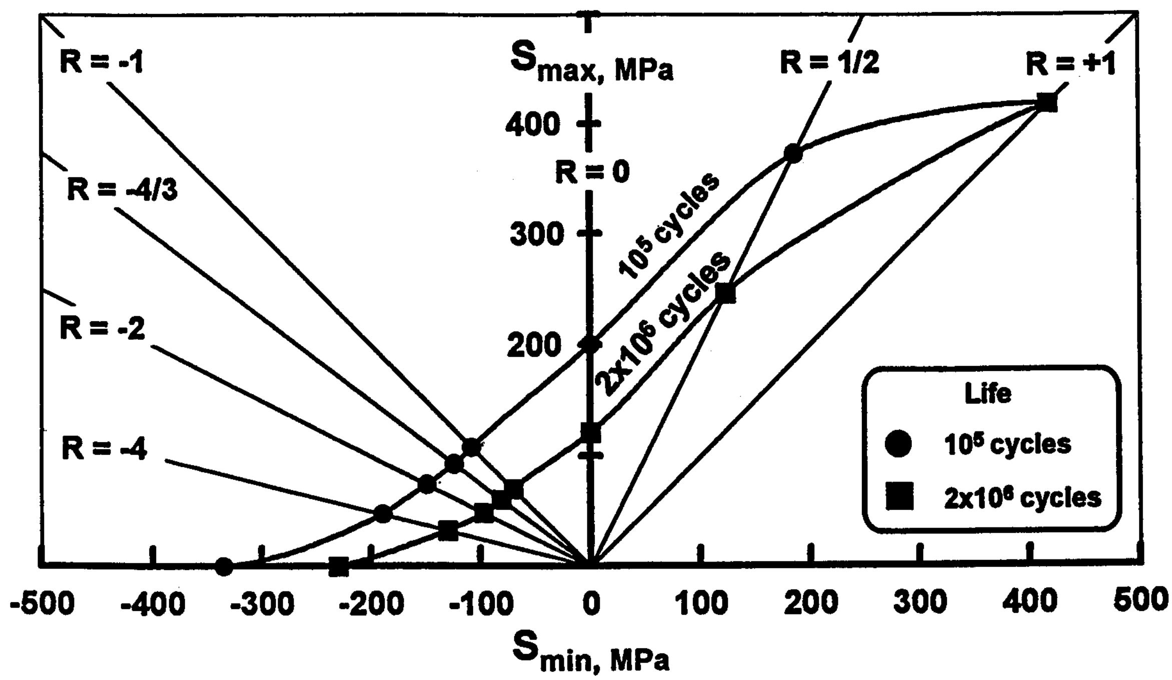

FIGURE 9 CLD, carbon steel butt welds.
Source: Reemsnyder, 1978a.

The sensitivity to mean stress decreases with an increase in notch severity and is a consequence of plasticity at the notch root. In the case of weldments, the presence of tensile residual stresses at the notch root causes plastic behavior even at low nominal stresses. The current criteria governing bridges, buildings, offshore structures, support structures of nuclear reactor containment vessels, overhead cranes, and mill buildings assume weldment fatigue resistance to be independent of mean stress and cite only stress range. However, these criteria do not show CLDs for each detail. Instead, they show sketches of various weldment configurations (called categories in the United States and classes in Great Britain) from which the designer selects the stress category (or class) closest to his detail.[5] The designer may follow one of three procedures. The designer may enter a tabular array with this stress category and the desired service life and then select the allowable stress range for the particular case. For example, see AISC (1989). Alternately, the designer may select the S–N curve of the particular stress category and estimate graphically the allowable stress range for the desired service life. See, for example AWS (1992). Finally, the designer may select the equation for the S–N curve of the particular

[5] Similar to those of figures 2 and 3.

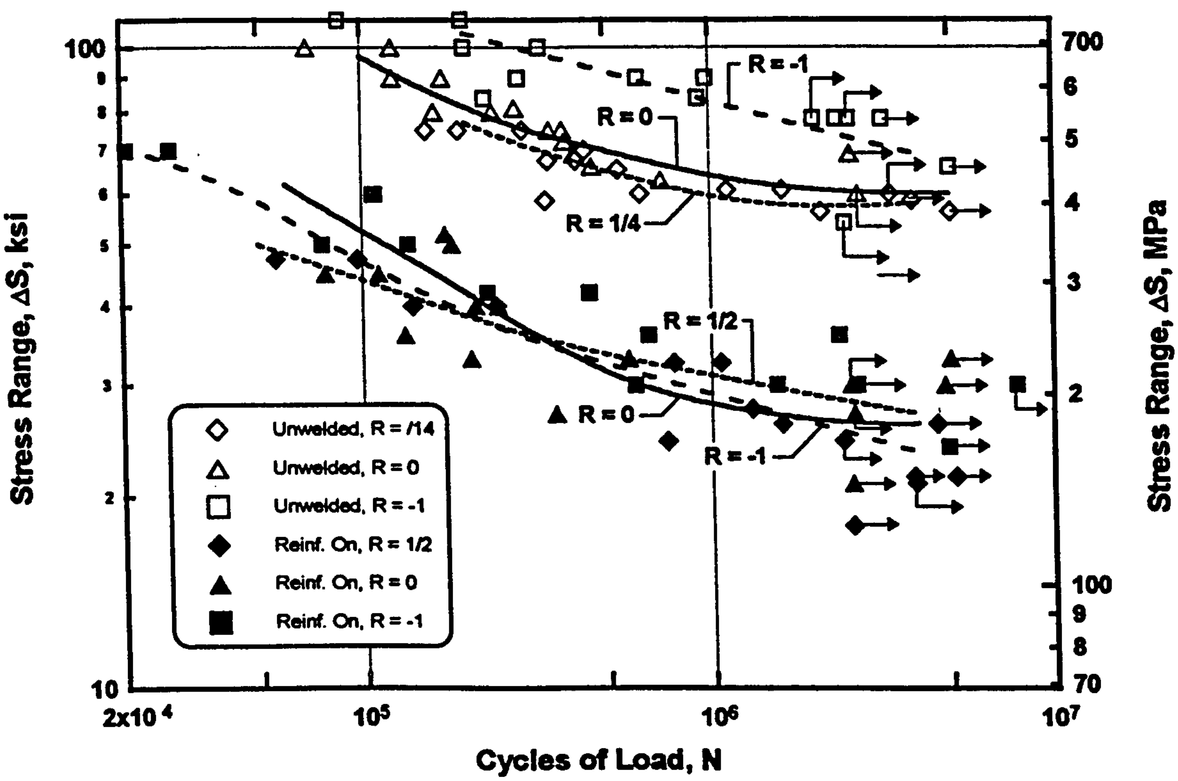

FIGURE 10 Transverse butt welds, quenched and tempered carbon steel.
Source: Reemsnyder, 1979.

stress category or class (the preferred approach) and compute the allowable stress range for the desired service life. See BSI 7608 (1993).

Stress Categories or Classes. The stress categories (United States) or classes (Great Britain) generally represent the typical weld details found in linear elements and wide plates of land and offshore structures. For example, see AWS, 1992; BSI 7608, 1993; API RP 2A, 1989; and UK DOE, 1984a.

In 1993, the British Standards Institution published their "British Standard Code of Practice for the Fatigue Design and Assessment of Steel Structures," BSI 7608 (1993),[6] which combines the British fatigue provisions for bridges, cranes, pressure vessels, and offshore structures (UK DOE, 1984a).

BSI 7608 presents S–N curves both as mean curves through the underlying data

[6] BSI 7608 also considers weld-life improvement by both toe grinding and thermal stress relief.

$$N = \frac{C_0}{S_r^m} \tag{1}$$

and offset by two standard deviations of the data, that is, design curves,

$$N = \frac{C_d}{S_r^m} \tag{2}$$

where C_o and C_d are functions of the detail class and m is the slope of the particular S-N curve.

Equation (2) describes the S–N curve to a life of 10 million cycles. For the constant-amplitude case, it is assumed that the S–N curve is horizontal as a stress range of S_o beyond 10 million cycles, that is,

$$S_0 = S_r = \sqrt[m]{\frac{C_d}{10^7}} \tag{3}$$

The design curves of Equation (2), for BSI 7608 are shown in Figure 11 for the various classes labeled B to W.

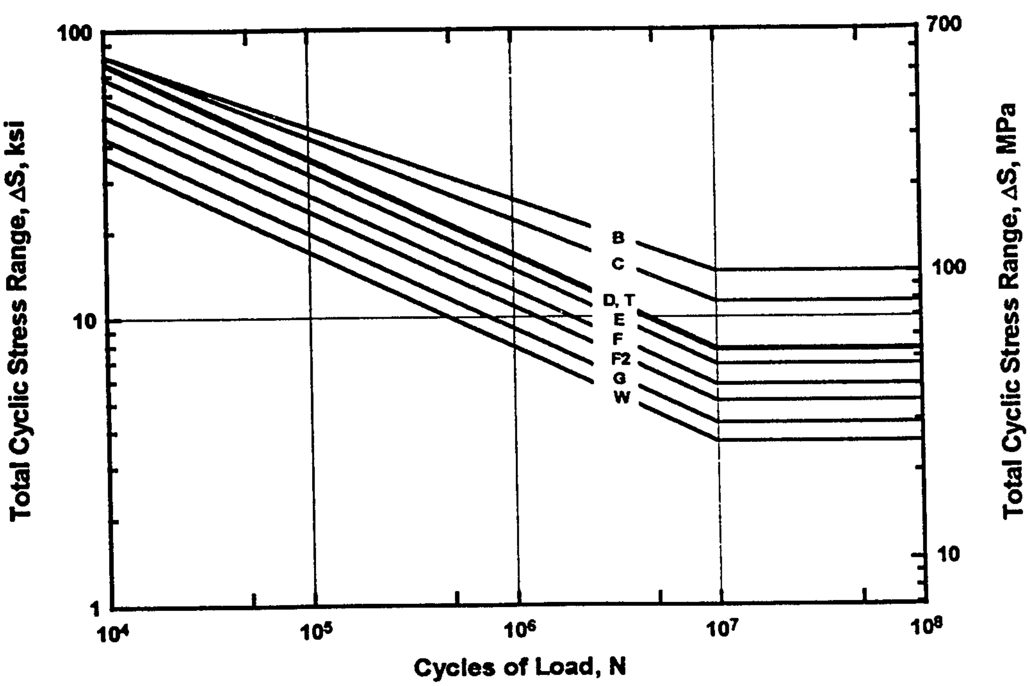

FIGURE 11 Fatigue design criteria.

For unprotected details in sea water, the S–N curves, Equation (2), are reduced by a factor of 2 on life and extended beyond a life of 10 million cycles at a slope of m. When the detail is subjected to a variable-amplitude load, stress, or strain, the S–N curves, Equation (2), are extended beyond a life of 10-million cycles at a slope of m + 2. The three criteria for design S–N curves for all of the classes in BSI 7608 are illustrated in Figure 12 for a typical class — T, the hot-spot stress. BSI 7608 acknowledges the effect of plate thickness on weldment fatigue strength through the following correction equation:

$$S_r = S_r^B \cdot \left(\frac{t_B}{t}\right)^{0.25} \tag{4}$$

where S_r is the fatigue strength of the detail under consideration and S_r^B is the fatigue strength of the detail from the basic S–N curves, Equation (2). t is the greater of 5/8 in. (16 mm) or the actual thickness of the detail and t_B is 5/8 in. (16 mm) for weldments.

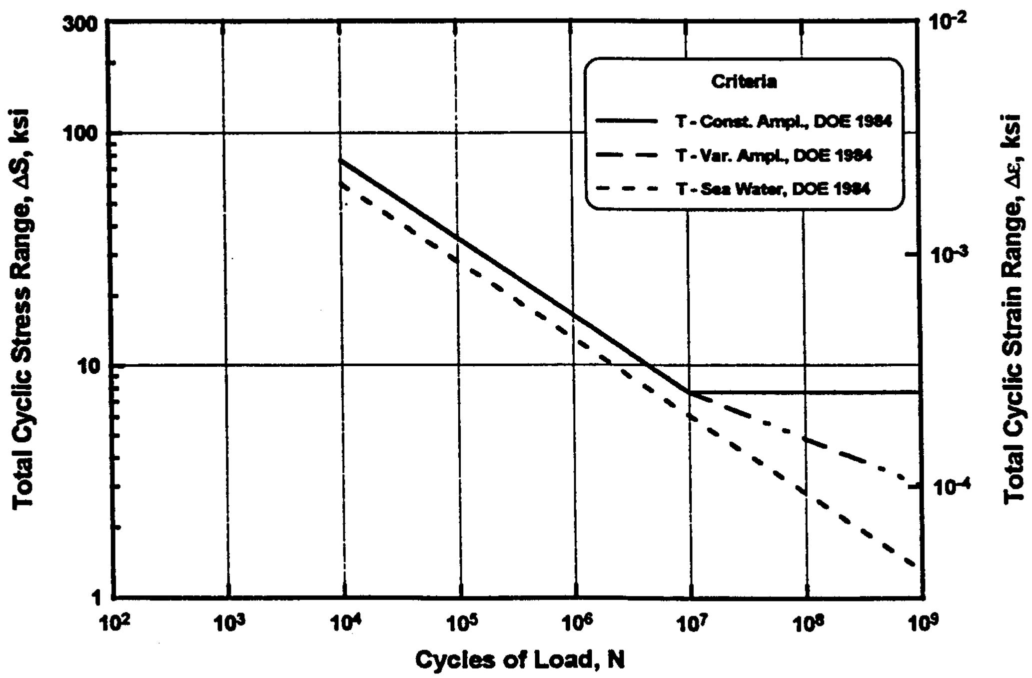

FIGURE 12 Allowable hot-spot stress range.

The American Bureau of Shipping (ABS) has published a guide for the fatigue assessment of tankers (ABS, 1993). The guide uses the design curves of the U.K. Department of Energy Guidance (UK DOE, 1984a) but without the thickness adjustment of Equation (4). The ABS guide shows typical welded details found in tankers, and assigns them to the various classes (Figure 11), and considers the effect of corrosion on increasing the stress range.

In a recent project of the Ship Structure Committee (Stambaugh et al., 1992), the fatigue data from an earlier project (Munse et al., 1983) was reanalyzed, and S–N curves for the design and analysis of ship structural details (only linear elements) were presented. Typical tubular-weldment details typical of offshore structures and the category or class for *hot-spot stress*, as shown in Figure 12 are also included in AWS, 1992; BSI 7608, 1993; API RP 2A, 1989; and UK DOE, 1984a.

Hot-Spot Stress

The concept of *hot-spot stress* is especially useful in cases where (1) the simple nominal stress required for the various categories or classes is impossible to compute because of the geometric complexities of the weldment and surrounding structure, and/or (2) the designer can not match a particular detail with those shown as categories or classes in the selected design criteria.

The use of the *hot-spot stress* allows the common treatment of many different connection geometries through the category X curves of the American Welding Society (AWS, 1992) and the American Petroleum Institute (API RP 2A, 1989) and the class T curve of the U.K. Department of Energy (UK DOE, 1984b) and *British Standard Code of Practice for the Fatigue Design and Assessment of Steel Structures* (BSI 7608, 1993). Both the X and T curves are S–N curves that express the allowable range of hot-spot stress for a given life in cycles. The hot-spot stress, widely used in the design of tubular connections of offshore platforms, is the stress (measured or computed) adjacent and perpendicular to the weld toe (Figure 13) and is the ordinate of the category **X** or the class **T** S–N curve. The API defines hot-spot stress in terms of hot-spot strain range:

> "Hot-spot strain range may be defined as that which would be measured by a strain gage element adjacent to and perpendicular to the toe of the weld after stable strain cycles have been achieved" (API RP 2A, 1989).

The DOE succinctly defines hot-spot stress:

> "The idealized hot-spot stress is defined by the greatest value of the extrapolation to the weld toe of the maximum principal stress distribution immediately outside the region where it is concentrated by the geometry of the weld . . . this idealized hot-spot stress . . . does not include the effect of the rapid increase adjacent to the weld toe" (UK DOE, 1984a).

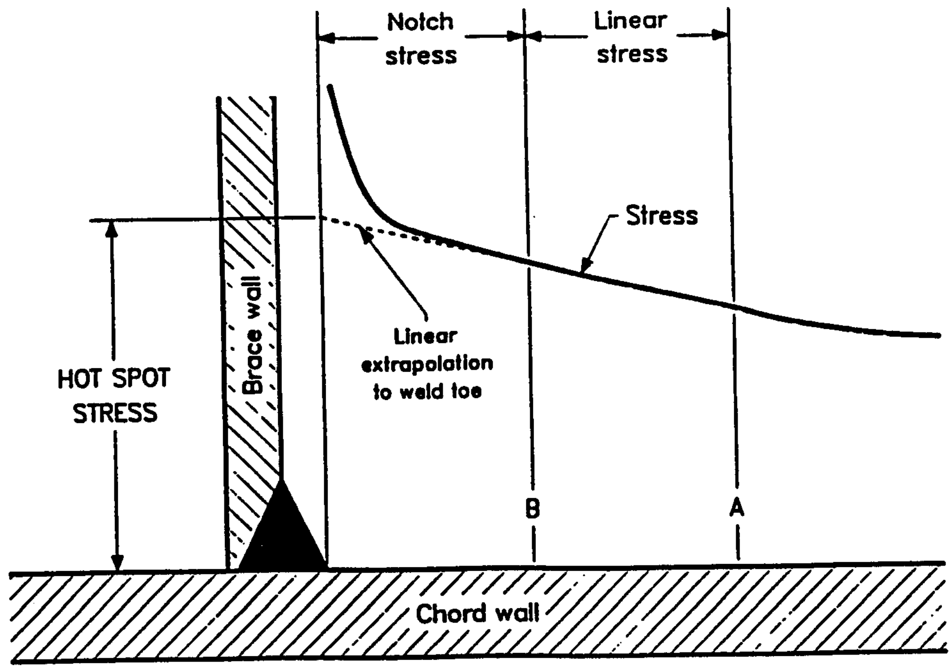

FIGURE 13 Hot-spot stress.

AWS and API Category X Curves. The background and basic test data of the AWS and API X-curves are discussed in Marshall (1974, 1975) and Rodabaugh (1980). The test data that were used by Marshall as the basis for the AWS-X curve included fatigue tests on approximately 25 small tubular joints, 6 pressure vessels, butt welded HY80 and A537, and data bands for butt welded carbon steel and A242 (Marshall, 1974). The AWS X design curve was shifted by a factor of 7 on life from the median line through the data, and Marshall assumed that, with such a shift, the AWS X-curve corresponded to the S–N curve for 97.5 percent survivals.

UK DOE and BSI 7608 Class T Curves. The UK DOE (and, subsequently, BS 7608 class T) S–N curve is expressed by Equation (1) or Equation (2). (See Figure 12.) The T-curve is horizontal beyond 10^7 cycles for constant-amplitude loading. However, for variable-amplitude loading and subsequent cumulative damage analysis, life beyond 10^7 cycles is expressed by Equation (1) or Equation (2) with a slope of $m + 2$. If the welded details are not protected from seawater, life for both $N \leq 10^7$ and $N > 10^7$ cycles is expressed by Equation (1) or Equation (2) *reduced by a factor of two on life*. Both the UK DOE Guidance and BS 7608 address the effect of thickness on weldment fatigue strength, using Equation (4).

At long lives, the T-curve for variable amplitude loads falls somewhat below the API X-curve—about midway between the API X and X′curves.[7] This deviation reflects the DOE concern

[7] X and X′ correspond to hot-spot stress at, respectively, angle changes and intersecting tubes.

that the AWS and API X-curves were based on thin specimens that did not adequately model the thicker sections used in offshore platforms (Gurney, 1983a) .

The background and basic test data UK DOE class T design curve (and of the hot-spot stress concept) are discussed in Gurney (1983a); Irvine (1983); Gurney (1983b); Loader et al. (1983); Fisher (1983); and UK DOE (1984b).

The UK DOE Guidance and BS 7608 give the equations for both the design curve and the mean S–N curve through the base data as well as the standard deviation of the base data. The T-curve is based on fatigue tests of 38, 0.63-in.-thick and 16, 1.26-in.-thick tubular joints. The T-curve was defined as the mean S–N curve through the 1.26-in. data less two times the standard deviation of the 0.63-in. data and was claimed to be the curve for a 2.5 percent risk of failure (Gurney, 1983a).

Comparison of Hot-Spot Stress Criteria with Additional Tests. The AWS X-curve was a lower bound to test data for tubular K- and T-joints summarized by (Rodabaugh, 1980). (Rodabaugh's summary was support for early AWS and API tubular joint criteria.) However, the tubular joint tests used in the development of the UK DOE T-curve and part of the United Kingdom Offshore Steels Research Program (UKOSRP) and subsequent data from tests on K- and KT-Joints that were part of UKOSRP fell below the AWS X-curve at lives above 1 million cycles (TWI, 1980; ICE, 1981; Wylde, 1980). Besides the UKOSRP, research on fatigue of offshore structures has been sponsored by the European Coal and Steel Community (ECSC) (TWI, 1978). The ECSC tests included a seawater environment for both 0.63-in.-thick and 1.26-in.-thick tubular T-joints, which also fell below the AWS X-curve at lives above 1 million cycles (Dijkstra and de Back, 1980) .

The above-mentioned test results were compared to the hot-spot stress design criteria of AWS, API, UK DOE, Det Norske Veritas, and the Norwegian Petroleum Directorate in Reemsnyder (1985). In general, the DOE and BSI 7608 criteria are lower bounds to the test data while the X-curve is nonconservative at lives greater than 10^6 cycles. The UK DOE and BSI 7608 criteria, in contrast to the AWS/API X-curve criteria, are the most comprehensive and reflect the current state of the art. The hot-spot stress concept has been extended to non-tubular welded joints (Kottgen et al., 1992).

Cumulative Damage Analysis

The fatigue-life assessment models of Table 1 were developed for constant amplitude fatigue, that is, the maximum and minimum cyclic stresses remain constant for every cycle. However, all land and marine vehicles and structures are subjected to variable-amplitude stress spectra, that is, the maximum and minimum cyclic stresses vary from cycle to cycle. For a complete fatigue-life assessment, the constant-amplitude models of Table 1 are combined with the Miner-Palmgren cumulative damage model to estimate the variable-amplitude fatigue life.

The Miner-Palmgren model assumes that

$$\sum_{1}^{m_i} \frac{n_i}{N_i} = D \tag{5}$$

where n_i is the number of cycles of application of the ith load, stress, or strain level; N_i is the fatigue life (cycles) of only the ith load, stress, or strain level, that were applied during the life of the element (i.e., constant-amplitude) from Equation (1) or Equation (2); and m_l is the number of load, stress, or strain levels. In Equation (5) failure is assumed likely if the *damage* D is close to unity. Also, it is assumed in Equation (5) that fatigue damage D is both (1) *stress-level-independent* and (2) *interaction-free*. Assumption (1) implies that the cumulative damage relation has the same form, that is, Equation (5), regardless of stress level. Assumption (2) implies that the summation of Equation (5) is independent of the order of stress levels. Neither assumption is generally valid. However, none of the other, more complicated cumulative damage models that have been proposed over the years is more accurate than Equation (5).

In many cases, only the relative frequencies of the ith load, stress, or strain levels are known. Then Equation (5) takes the form:

$$\sum_{1}^{m_i} \frac{\alpha_i \cdot N}{N_i} = 1 \tag{6}$$

where α_i is the relative frequency of the ith load, stress, or strain level, and N is the service life (cycles) of the element when subjected to m_l levels of load, stress, or strain range. Equation (6) may be rearranged to express service life N as:

$$N = \frac{1}{\sum_{1}^{m_i} \frac{\alpha_i \cdot N}{N_i}} \tag{7}$$

The *British Standard Code of Practice for the Fatigue Design and Assessment of Steel Structures* (BSI 7608, 1993), the fatigue criteria for offshore structures—American Petroleum Institute (API RP 2A, 1989), the U.K. Department of Energy (UK DOE, 1984a), and the ABS tanker guide (ABS, 1993)—give guidance in the use of the Miner-Palmgren model to assess fatigue life. Also, the British documents give guidance in *cycle-counting*—a vital part of cumulative damage analysis (BSI 7608, 1993; UK DOE, 1984a).

An alternative approach to the Miner-Palmgren cumulative damage model is the equivalent constant-amplitude, stress-range model:

$$\Delta S_{equiv} = \sqrt[\beta]{\frac{\sum \Delta S_i^{\beta}}{n}} \tag{8}$$

where ΔS_{equiv} is equivalent constant-amplitude stress range;[8] ΔS_i is the stress range at the ith load level; n is the number of load levels; and β can be 2, 3, or the slope of the S–N curve. A particular case where β in Equation (8) was taken as 2 to express the equivalent constant-amplitude range of stress-intensity factor for modeling fatigue-crack growth under spectrum loading (Barsom and Rolfe, 1987), has been extended to express equivalent constant-amplitude stress range, that is, root-mean-square stress range (Rolfe et al., 1993) as:

$$\Delta S_{RMS} = \sqrt{\frac{\sum \Delta S_i^2}{n}} \tag{9}$$

Use of the root-mean-square approach was shown to be satisfactory when characterizing fatigue-crack propagation under random loading (Barsom and Rolfe, 1987). Other investigators found that the root-mean-square approach, Equation (9), was adequate for the analysis of fatigue crack propagation in welded plates under a narrow-band random sequence representing North Sea wave loading while the Miner-Palmgren model, Equation (6), gave conservative estimates of life (Bouchard et al., 1991). On the other hand, a third group of investigators reviewing published fatigue data for welded joints found that Equation (8) with β taken as 3, that is, the root-mean-cube equivalent stress range, was superior to the Miner-Palmgren model for R = -1, high constant or varying R-ratios, and long-block load spectra (Dahle, 1994). Finally, a fourth investigator reviewing a decade of fatigue testing at TWI,[9] found the Miner-Palmgren model to be adequate and conservative except for short-block load spectra (Gurney, 1989).

The ground-vehicle industry has found that, in years of experience with blocked-load spectra, the Miner-Palmgren model is a reasonable estimator of fatigue crack-initiation life when combined with the Neuber local-strain life assessment model. The Neuber model for crack initiation can generally handle load-sequence effects that require consideration of crack-growth retardation and acceleration when modeling fatigue-crack propagation. In the latter case, the aerospace industry has confidence in the Miner-Palmgren model when combined with a retardation model. If a load spectrum is represented by a random history (e.g., wind, waves, etc.), retardation and acceleration balance one another, and the Miner-Palmgren model is adequate.

Do's and Don'ts

Good detail practice does much to improve the fatigue resistance of a welded joint subjected to repeated loads. In those case where the cyclic loads cannot be quantified for a fatigue life assessment, *good detail practice may be the only safeguard* against premature fatigue failures. Sections should be changed gradually to reduce the stress concentration effect. Joints or details with a large variation in stiffness should be avoided. High restraint in localized zones may cause

[8] ΔS and S_r are used interchangeably to denote stress range.

[9] Formerly The Welding Institute.

high secondary stresses that are not considered in the design calculations. Some Do's and Don'ts for design and fabrication of welded joints are listed below.

Do's

- Change sections gradually.
- Grind butt welds flush and smooth.
- Use butt joints instead of lap joints.
- Extend cover plates on girders well beyond theoretical cut offs.
- Streamline fillet welds.
- Give preference to a structure that will not collapse after fatigue failure of a detail.
- Locate joints where fatigue conditions are not severe.
- Use welding procedures that will eliminate gas pockets, slag inclusions, and the like.
- Avoid undercutting, cracks, spatter, and other stress raisers.
- Machine or otherwise dress weld to obtain smoothness at critical locations.

Don'ts

- Don't use joints with large variation in stiffness.
- Don't introduce high restraint in details.
- Don't use intermittent welds.
- Don't permit promiscuous striking of an arc outside of the weld area.
- Don't overweld.

Additional Reading

Fatigue design of weldments is covered in Gurney (1979), Munse and Grover (1964), Maddox (1991), Radaj (1990), Reemsnyder (1974), Heywood (1962), Osgood (1970), and in Granjon and Newman (1970). Probabilistic fatigue design of weldments is covered in Harral (1987), and in Park and Lawrence (1988). Fatigue design of aluminum weldments is covered in Sanders and Fisher (1985).

Mechanics of Crack Initiation and Propagation in Weldments

The present fatigue design criteria for welded joints are based on laboratory tests in which the failure criteria was "cycles to separation into two pieces." Although the welded specimens were large compared to typical laboratory fatigue specimens, they were small compared to welded structural joints.

The fatigue life of a test specimen, structural joint, or machine component consists of three phases: (1) *initiation* of a macroscopic crack; (2) *propagation* of the crack to a critical size; and (3) *exceeding of the residual strength* of the cracked element, causing complete fracture.

The bulk of the fatigue life consists of the first two phases. For a given weldment detail, crack-initiation life in a laboratory specimen would be similar to that of a structure. However, Phase 2 for the specimen would be considerably less than that of a structural joint. Therefore, the reliability of criteria based on complete fracture of laboratory specimens is uncertain at best.

Crack Propagation

It was shown by Signes et al. (1967) that a weld fusion zone, where the metal had either been melted or pasty during the welding process, contained a high concentration of sharp slag inclusions and other non-metallics, both as isolated inclusions and as grain boundary films. In addition, a slight undercutting was generally observed along the fusion boundary. In view of these initial crack-like flaws, considerable effort has been expended during recent years to apply fracture mechanics to weldment fatigue (Reemsnyder, 1978a; Maddox, 1970, 1974a, 1974b, 1975, 1991; Radaj, 1990; Griffiths et al., 1971; Hirt and Fisher, 1973). In this approach, it is assumed that all weldments initially contain crack-like flaws and that the significant portion of life is crack growth, that is, Phase 2 above.

Stress intensity factors K_I have been developed for a longitudinal butt weld in a residual stress field (Terada, 1976, 1983); a transverse butt weld (Lawrence, 1973); a partial-penetration fillet weld (Frank, 1971); a *three-corner* crack in a beam web-flange junction (Marek et al., 1970); transverse fillet welds, both non-load–carrying (Maddox, 1975) and load-carrying (Maddox, 1975; Skorupa et al., 1987); toe of a tee-shaped weldment in bending (Zwerneman and Frank, 1989); cruciform joints (Ferreira and Branco, 1988; Murakami, 1987); lap and offset joints (Murakami, 1987); through and surface cracks in a residual stress field (Murakami, 1987); edge-cracked ring in a weld residual stress field (Murakami, 1987); fillet welded T-joints (Tsai and Kim, 1990; Otegui et al., 1991); plate with a welded gusset (Hobbacher, 1992); semi-elliptical surface flaw in a fillet-welded T-butt (Fu et al., 1993); and surface flaws in T-plate and plate-pipe joints (To et al., 1993). Stress intensity factor solutions for spot welds were assembled in Reemsnyder (1992).

Crack propagation has been described by fracture mechanics in carbon steel base metal, weld metal and heat-affected zone (HAZ) (Maddox, 1970, 1974c), low-alloy weld metal (Griffiths et al., 1971), quenched and tempered constructional alloy steel base metal and weldment (Parry et al., 1972), quenched and tempered pressure vessel steel base metal and heat-affected zone (Socie and Antolovich, 1974), and carbon and HSLA steels with electroslag welds (Kapadia and Imhof, 1977). Fracture mechanics has been used to describe crack growth from lack of penetration in transverse butt welds (Lawrence and Munse, 1973); fillet welds (Maddox, 1974b; Yamada and Hirt, 1982a, 1982b; Makhnenko and Pochinok, 1982; Ohta et al., 1987); various discontinuities in butt welds, such as inclusions, lack of fusion, and porosity (Lawrence and Radziminski, 1970); and in stainless steel welds in the presence of residual stresses (Mills and James, 1987). Rigorously, the fracture mechanics description of fatigue-crack propagation is the sigmoidal relation:

$$\frac{da}{dN} = \frac{C' \cdot (\Delta K_I - K_{th})^{m'}}{(1 - R) \cdot K_C - \Delta K_I} \tag{10}$$

where da/dN is the crack growth rate per cycle; K_{th} and K_c are, respectively, the lower and upper asymptotes; ΔK_I is the cyclic range of stress intensity factor; and R is the ratio of the minimum to the maximum stress intensity factor per cycle. However, crack propagation in structures is usually modeled with the simple power relation:

$$\frac{da}{dN} = C \cdot (\Delta K_I)^m \tag{11}$$

Having described the cyclic range of stress intensity factor ΔK_I as a function of crack length, shape, stress range, and plate thickness, the crack length-cycles relation is then established by numerical integration of:

$$N_p = \int_{a_i}^{a_f} \frac{da}{C \cdot (\Delta K_I)^m} \tag{12}$$

where N_p is the crack propagation life; and a_i and a_f are, respectively, the initial and final crack lengths.

Crack Initiation

Fracture Mechanics. Fracture mechanics concepts may be used to estimate crack initiation in weldments (Reemsnyder, 1974, 1979). For small notch-root radii, the stress field ahead of the notch is approximately described by the stress-intensity factor. For a given range of stress-intensity factor, ΔK, the cycles to crack initiation decrease with notch-root radius until the root radius equals 0.01 in. (0.25 mm). At root radii less than 0.01 in. (0.25 mm), where crack initiation is a function only of ΔK and is independent of root radius, hot-rolled carbon steels exhibit slightly shorter initiation lives than quenched and tempered alloy steels. Typical weld-toe radii are equal to or less than 0.01 in. (0.25 mm), and therefore a plot of ΔK versus cycles to initiation may be used to predict crack initiation in weldments at lives where the notch behavior is essentially elastic (Reemsnyder, 1974, 1979).

According to one investigator, fracture mechanics models of crack growth in full-penetration butt welds adequately describe the fatigue lives of carbon and HSLA steels but underestimate the lives of quenched and tempered steels (Lawrence, 1973). This study concluded that the initiation time is relatively short for carbon and high-strength, low alloy steels but not for quenched and tempered steels. Another study found that the crack initiation time for partial-penetration butt welds in carbon steels was a significant portion of the total life (Lawrence and Munse, 1973).

The limitations of the fracture mechanics model in assessing fatigue crack initiation life have been discussed elsewhere (Reemsnyder, 1986a). It appears that the fracture mechanics model significantly *overestimates* initiation life when significant notch-root plasticity is present and/or when the notch-root radius is large relative to the notch depth, a. For example, the fracture mechanics model underestimates the notch-root stress by 5, 10 and 15 percent for ρ/a of, respectively, 0.008, 0.09, and 0.12 (Reemsnyder, 1986a).

Neuber Analysis. Strain-cycled fatigue concepts, that is, Neuber's rule and strain-controlled fatigue data, have been applied to predictions of crack initiation at the roots of notches in offshore platforms (Reemsnyder, 1986b), toes of full-penetration butt welds, and the roots of partial-penetration butt welds (Mattos and Lawrence, 1977; Lawrence et al., 1978). Such an approach not only models observed behavior adequately but also quantifies both the cyclic relaxation of mean and residual stresses and the effects of weld shape and internal discontinuities. Neuber analyses also have been successfully used to predict crack initiation in fillet-welded tees containing porosity (van der Zanden et al., 1972),[10] in butt welds and transverse non-load–carrying fillet welds (Smith et al., 1977), and T-joints in air and sea water (Bhuyan and Vosikovsky, 1989).

Total Life Model.

The complete process of fatigue-life estimation for weldments is illustrated for butt and fillet welds in, respectively, Kottgen et al. (1992) and Dahle (1994). In the total life model, the life to *crack initiation* is estimated through *Neuber analysis.* Then *fracture mechanics* is used to estimate the life to *fracture*.

Fitness for Purpose

A component in a particular structure is considered to be suitable for its intended purpose, provided that the conditions sufficient to cause failure do not develop even after one allows for accumulated damage in service. Defects that are detected by nondestructive inspection (NDI) and are less severe than those specified in the pertinent quality assurance criteria are accepted without further consideration. More severe defects detected by NDI in critical areas are not necessarily rejected. Instead, the decision to accept or reject the weldment is based on an assessment of fitness for purpose. The assessment generally includes a fracture mechanics modeling of fatigue-crack growth and fracture to estimate life. If the estimated life is greater than the desired service life (or time to next inspection period), the component is considered to be fit for the intended purpose and accepted.

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[10] The continuous, longitudinal fillet welded specimen of Reemsnyder (1965a) was used here.

Current fatigue design criteria for weldments take into account the weld-notch severity of certain welds and prohibit some weld forms, such as partial-penetration welds in cyclic tension (AWS, 1992), while allowing higher design stresses when the reinforcement is ground flush with the base metal or merges smoothly with the base metal. Also, these criteria recognize the deleterious effects of weld discontinuities and permit higher allowable stresses for certain groove and butt weld configurations when weld soundness is established by radiographic or ultrasonic inspection. However, none of the design criteria based on component tests adjusts allowable design stresses for the size, shape, and number of acceptable discontinuities. Instead, they limit discontinuity size and number through quality assurance criteria, which are not a part of the fatigue design criteria.

On the other hand, a component in a particular structure may be considered to be suitable for its intended purpose, provided that the conditions sufficient to cause failure do not develop even after one allows for accumulated damage in service. Defects that are detected by NDI and are less severe than those specified in the pertinent quality assurance criteria are accepted without further consideration. More severe defects detected by NDI in critical areas are not necessarily rejected. Instead, the decision to accept or reject the weldment is based on an assessment of fitness for purpose. If the life estimated in the assessment is greater than the desired service life (or time to next inspection period), the component is considered to be fit for the intended purpose and accepted.

NDIPD 6493

PD 6493, a document of the British Standards Institution (BSI 6493, 1991), employs the concept of fitness for purpose to provide guidance in specifying acceptance levels for weld discontinuities. A weld in a particular structure is considered to be suitable for its intended purpose, provided that the conditions sufficient to cause failure do not develop even after one allows for accumulated damage in service. Defects that are detected by NDI and are less severe than those specified in the pertinent quality assurance criteria are accepted without further consideration. More severe defects detected by NDI in critical areas are not necessarily rejected. Instead, the decision to accept or reject the weldment is based on an assessment of fitness for purpose. Two classes of defects are considered: nonplanar (i.e., porosity and slag inclusions) and planar (e.g., cracks, lack of fusion, lack of penetration, undercut). The assessment of nonplanar defects is based on fatigue tests of welded specimens containing defects, whereas that of planar defects employs fracture mechanics techniques.

For nonplanar defects, PD 6493 presents quality categories in the form of S–N curves that are lower bounds to tests of weldments containing various levels of inclusions or porosity. The required quality category is determined by the weld configuration category or by the equivalent constant-amplitude stress range computed from a variable-amplitude service load spectrum. The actual quality category is established by the maximum length of slag inclusions detected through NDI or by porosity expressed as percent of area from radiographs. The defect in the critical weld is acceptable for the intended purpose when the actual quality category is equal to or greater than the required quality category. Effects of mean stress are included in the assessment of stress-relieved welds but not in that of as-welded fabrications.

The guidelines of PD 6493 for fitness-for-purpose assessment of planar defects, such as cracks and lack of penetration, establish the required quality category in the same way as for nonplanar defects. The actual quality category is a function of the initial and tolerable defect sizes, respectively, a_i and a_m, and the material thickness. The initial defect size is determined by the NDI of the critical weldment. The tolerable flaw size times an appropriate safety factor equals the critical flaw size, a_f, sufficient to cause failure by general yielding, fracture, corrosion, and the like. The cycles of loading to achieve the tolerable flaw size is computed by integration of the crack-growth relation—da/dN versus ΔK_I—between the limits of a_i and a_m. The planar-weld defect in the critical area under study is acceptable for the intended service if (1) the actual quality category is equal to or greater than the required quality category, or (2) the cycles to achieve a tolerable flaw size are greater than the design life of the weldment.

American Society Mechanical Engineers

The American Society of Mechanical Engineers (ASME) provides a procedure for determining the acceptability of flaws detected during the periodic in-service inspection of light-water-cooled nuclear power plant components (ASME Boiler and Pressure Vessel Code, 1995). If the detected flaws exceed specified flaw indication standards, flaw growth is computed by fracture mechanics, that is, integration of the crack-growth relation–da/dN versus ΔK_I—to determine the crack length to the next inspection period or to the end of the service lifetime. The component is acceptable for continued service if (1) crack length is less than the minimum critical flaw size, a_f, for either normal or emergency and faulted conditions, or (2) the maximum stress-intensity factor at that crack length is less than either the crack-arrest fracture toughness for normal operating conditions or the plane strain fracture toughness for emergency and faulted conditions. ASME presents da/dN-ΔK_I curves for carbon and low-alloy ferritic materials in both air and light-water environments. Both surface and subsurface flaws are represented in these curves.

Other

Fitness for purpose is used in Appendix A of the American Petroleum Institute Standard 1104 (API 1104, 1983), to determine the acceptance levels of weld discontinuities in field girth-welds of oil and gas transmission pipelines. Also, the International Institute of Welding has published a draft for development on fitness for purpose (IIW, 1990).

Summary

Although the current weldment design criteria recognize the importance of weld discontinuities and limit their size, shape, and number through quality assurance criteria, they do not adjust the allowable cyclic stresses in welds for discontinuity characteristics. On the other hand, the concept of fitness for purpose recognizes that weld quality need not be constant over a particular structure or from structure to structure. Instead, the quality of the weld should be established by the role that the weld plays in the integrity of the structure and the intended use of the structure. The state of the art of the fitness-for-purpose concept is well developed and shows great promise for the design of safe, economical welded structures.

Further Reading

The influence of various parameters on the fatigue resistance of weldments is discussed in Radaj (1990), Gurney (1979), Munse and Grover (1964), Maddox (1991), Reemsnyder (1974), Osgood (1970), Pollard and Cover (1972).

Weldment fatigue tests are summarized and mean fatigue strengths at selected lives are presented in Gurney (1979), Munse and Grover (1964), Maddox (1991), Reemsnyder (1961, 1974), Spraragen and Claussen (1937), Spraragen and Rosenthal (1942), Mindlin 1968, and Munse (1978). On the other hand, the individual fatigue test results for structural steel weldments are presented in Reemsnyder (1968) in and Gurney and Maddox (1973).

A bibliography on the fatigue strength of welded joints for the years 1950 to 1971 (Larson, 1972) has been expanded and extended to the end of 1977 (Reemsnyder, 1978b).

U.S. journals containing articles on fatigue of weldments are The Welding Journal, Bulletin of the Welding Research Council, and Transactions of American Society of Chemical Engineers, American Society Mechanical Engineers and the Society of Automotive Engineers. British journals include Welding and Fabrication and its predecessors, Joining and Materials; Metal Construction and The British Welding Journal; and the International Journal of Pressure Vessels and Piping. Soviet journals include Welding Production and Automatic Welding. Other journals are Welding Research Abroad and the International Journal of Fatigue.

PART II: ASSESSMENT OF RESIDUAL STRENGTH

Structural Integrity Model

Optimized modeling of fracture-critical structural components and connections requires the application of elastic-plastic fracture mechanics. However, such applications can require sophisticated analytical techniques that require time and/or resources beyond those available to the designer. One of the first engineering tools to address this dilemma was The Welding Institute Crack-Tip Opening Displacement (CTOD) Design Curve, which was included in the first edition of the British Standards Institution fitness for purpose guidance (BSI PD 6493, 1980). The engineering tool receiving attention currently is the failure assessment diagram (FAD). This approach has been used, primarily, in the electric power industry, both in Great Britain as the R6 criteria (Milne et al., 1986), and the United States, as both the failure assessment diagram (Kumar et al., 1981) and the deformation plasticity failure assessment diagram (DPFAD) (Bloom, 1983). Both the R6 and DPFAD approaches utilize the J-integral criteria for fracture driving force and resistance (i.e., toughness). The second edition of BSI PD 6493 (1991) presents failure assessments in the form of FADs using CTOD.

Welding Institute Crack-Tip Opening Displacement Design Curve

Today, CTOD is being used in Great Britain to evaluate the toughness of steels in wide range of structures—bridges, buildings, pipelines, pressure vessels, tanks, and offshore platforms. In the United States, CTOD is used primarily for materials in pipelines and offshore structures.

The British Welding Institute developed a design curve that relates CTOD at some critical event, yield strength σ_y, nominal strain at a notch ε, and flaw size $\bar{a}$ (Burdekin and Dawes, 1971; Dawes, 1974). This empirical relation, based on wide-plate fracture tests and developed originally for pressure vessels, is expressed as

$$\Phi = \left(\frac{\epsilon}{\epsilon_y}\right)^2 \qquad for\ \frac{\epsilon}{\epsilon_y} \leq 0.5 \tag{13}$$

and

$$\Phi = \left(\frac{\epsilon}{\epsilon_y}\right) - 0.25 \qquad for\ \frac{\epsilon}{\epsilon_y} > 0.5 \tag{14}$$

where the nondimensionalized CTOD is Φ,

$$\Phi = \frac{CTOD}{2\pi\epsilon_y\bar{a}} \tag{15}$$

with the yield strain ε_y

$$\epsilon_y = \frac{\sigma_y}{E} \tag{16}$$

$\bar{a}$ is the length of a through-crack in an infinite plate equivalent in severity to that of the crack in the element under investigation. For example, the equivalent crack length a for a center-cracked panel (CCP) is

$$\bar{a} = a\left[\sec\left(\frac{\pi a}{W}\right)\right] \tag{17}$$

while that for a surface-cracked element is

$$\bar{a} = a\left(\frac{M}{\phi}\right)^2 \tag{18}$$

The secant term in Equation (15) is the finite-width correction factor for center-cracked specimens. In Equation (18), M, a function of a, c, and B, respectively, the depth and half-width of the surface-crack and the element thickness, corrects for the proximity of the front and back surfaces:

$$M = 1.13 - 0.09\left(\frac{a}{c}\right) + \left(-0.54 + \frac{0.89}{0.2 + \frac{a}{c}}\right)\left(\frac{a}{B}\right)^2 + \ldots$$
$$\ldots + \left[0.5 - \frac{1}{0.65 + \frac{a}{c}} + 14\left(1 - \frac{a}{c}\right)^{24}\right]\left(\frac{a}{B}\right)^4 \tag{19}$$

and ϕ, the shape factor of the surface flaw, is a function of a/c:

$$\phi^2 = 1 + 1.464\left(\frac{a}{c}\right)^{1.65} \tag{20}$$

Equations (17) and (18) relate the relatively large flaw (crack) size of small laboratory specimens to the small flaw in large plates typical of service. The nominal strain ε in a notched component may be computed from:

$$\epsilon = \frac{K_t\sigma + \sigma_{res}}{E} \tag{21}$$

where K_t, σ and σ_{res} are the stress concentration factor and nominal and residual stresses, respectively.

The Welding Institute design curve, Equations (13) and (14), is shown in Figure 14 along with the data band on which the design curve was based. The data band represents CCP specimens with W, L, and B of, respectively, 36, 36, and from 1 in. to 2 in. (914, 914, and from 25 mm to 50 mm) (Terry, 1974; Egan, 1974). The design curve Equations (13) and (14), was constructed relative to the wide-plate test results with a *safety factor* of 2 on flaw size $\bar{a}$. Later, the Welding Institute analyzed 73 sets of small- and large-scale fracture tests and found that, on the average, the design curve has a factor of safety of approximately 2.5 on flaw size (Harrison et al., 1979). Also, they found that the maximum allowable flaw size derived from the design curve implies about a 95 percent probability of survival with respect to the wide-plate test.

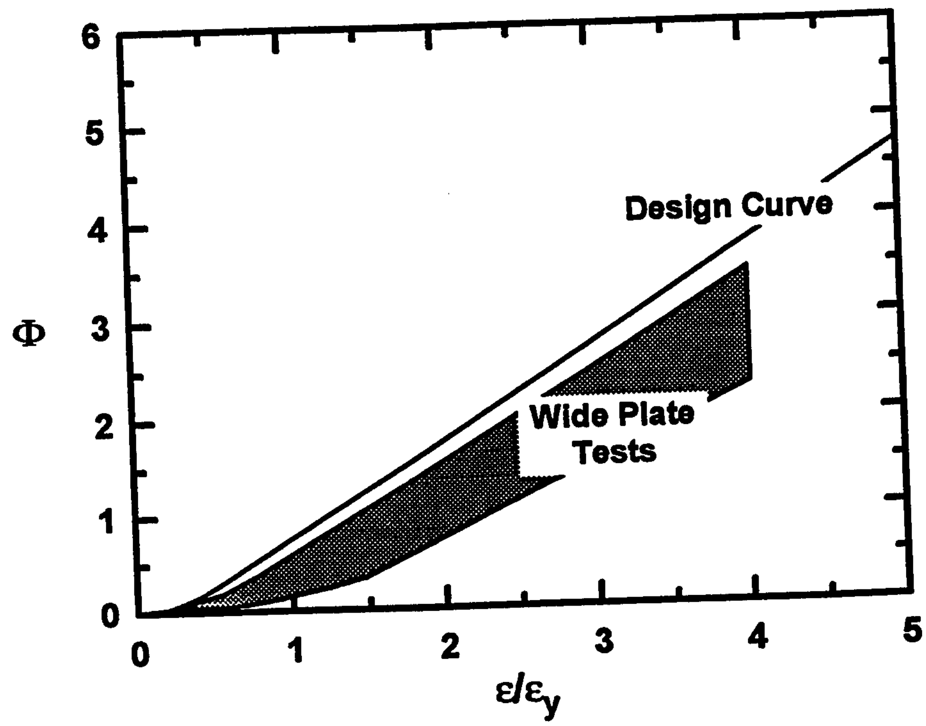

FIGURE 14 Welding Institute CTOD design curve.

The design curve can be used in the following manner. Given $\varepsilon/\varepsilon_y$ in a critical region from a stress analysis of the structure, Φ is determined from the design curve. From this value of Φ, the maximum allowable flaw size, $\bar{a}$, in the critical region may be established given the toughness CTOD of the material. Alternatively, a material with an adequate toughness CTOD can be selected for the critical region given the maximum allowable flaw size, $\bar{a}$.

NOTE: A limit analysis also should be made because the tolerable flaw size determined from the design curve analysis might be larger than that at plastic collapse.

Both the first edition of the BSI PD 6493 (1980) and the American Petroleum Institute is API 1104 (1983) use The Welding Institute's design curve as a basis for their fitness-for-purpose criteria.

NOTE: At the present time, establishment of minimum acceptable values of CTOD has been based on service experience of weldments but not other product forms and joining techniques.

Central Electricity Generating Board R6 Diagram

The U.K. Central Electricity Generating Board (CEGB) first proposed a failure assessment diagram, Figure 15, based on the two-criteria approach of Dowling and Townley (Dowling and Townley, 1975). The CEGB approach (Milne et al., 1986, 1988a; Kanninen and Popelar, 1985) addressed post-yield fracture by an interpolation formula between two limiting cases—linear elastic fracture and plastic collapse. The interpolation formula, called the failure assessment or R6 curve (Figure 15), developed from the Dugdale solution for a cracked, infinite plate, is

$$K_r = \frac{S_r}{\sqrt{\frac{8}{\pi^2}\ln[\sec(0.5\,\pi\,S_r)]}} \tag{22}$$

The right-hand side of Equation (22) may be viewed as the plastic correction to the small-scale yielding prediction. In Equation (22), the *fracture ratio*, K_r, is

$$K_r = \frac{K}{K_c} \tag{23}$$

and the *collapse ratio*, S_r, is

$$S_r = \frac{\sigma}{\sigma_c} \tag{24}$$

In Equation (23), K is the stress-intensity factor (a function of nominal stress σ, crack size a, and geometry) at fracture of the component, and K_c is the linear elastic fracture toughness of the component. In Equation (24), σ is the applied (remote) stress in the component at fracture, and σ_c is the applied (remote) stress at plastic collapse (i.e., limit stress) of the cracked component.

If a point describing the state of a component or structure falls below the R6 curve (Figure 15) the structure is safe. A point falling on or above the R6 curve represents failure. If a ray were constructed from the origin to the point, the safety factor on load is the length of the ray from the origin to the intersection of the ray and the R6 Curve divided by the length of the ray from the origin to the point, for example, OF/OW (Figure 15).

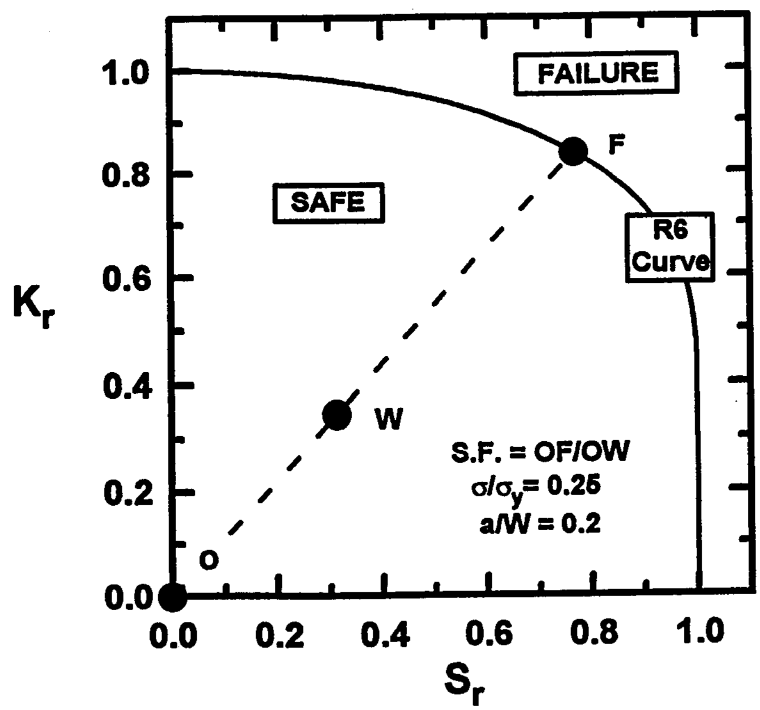

FIGURE 15 CEGB R6 curve.

The Failure Assessment Diagram

The FAD, as generally used today, is essentially a graphical model shown schematically in Figure 16. The FAD consists of two elements: (1) the failure assessment curve (FAC); and (2) the failure assessment point (FAP). The *collapse ratio*, S_r, is the ratio of the applied stress at design load to the applied stress at plastic collapse. K_r, the *fracture ratio*, is the ratio of the crack driving force (*including residual stress*) to the material toughness (K_c, J-integral or CTOD). The FAC defines the critical combination of service loads, material stress-strain properties, and geometry of the cracked member at which failure might be expected. The FAP defines the state of a member containing a flaw of given size under specific service loads. The factor of safety (on load) against failure for a given FAP (in the absence of residual stresses) is the ratio of the line segments OB/OA (Figure 16).

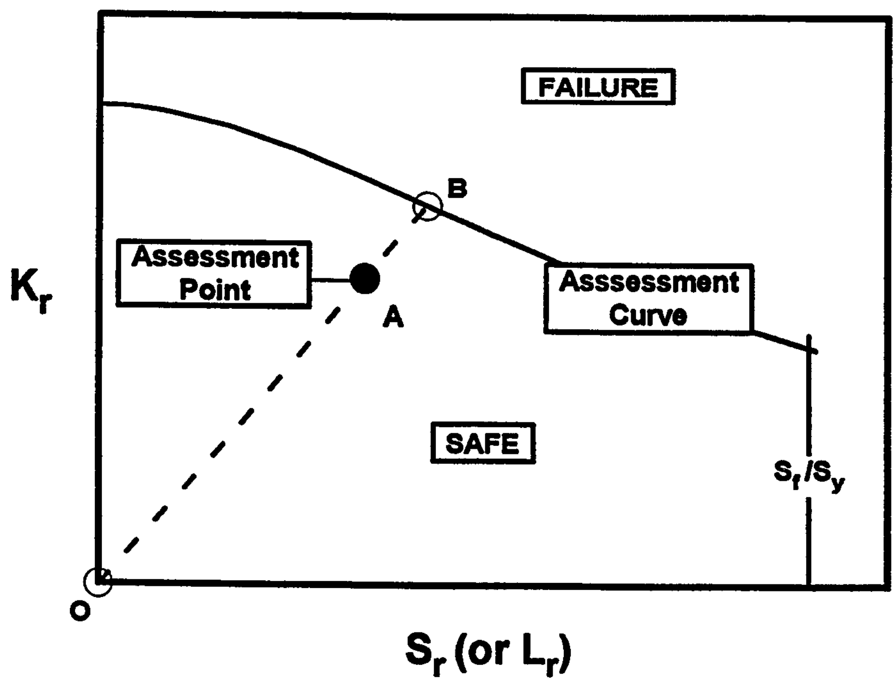

FIGURE 16 Failure assessment diagram.

Current applications of the FAD are as follows:

- CEGB R6 - Revision 3
- BSI PD 6493, Second edition
- Electric Power Research Institute/General Electric (EPRI/GE) model
- Deformation plasticity FAD
- ASME Section XI Code Case (DPFAD) for ferritic piping

Current CEGB Failure Assessment Diagrams

Chell (1979) and Bloom (1980) pointed out that K_r, Equation (23), could be interpreted as a ratio of J-integrals:

$$K_r = \sqrt{\frac{J_e}{J}} = f(S_r) \tag{25}$$

where J_e is the *elastic component* of the total J-integral, J.

The current version of the CEGB R6 FAD expresses K_r as a ratio of the linear- elastic-stress-intensity factor associated with the stress and the flaw to the J-integral toughness (Milne et al., 1986, 1988a; Kanninen and Popelar, 1985). Further, the current R6 presents three options for the FAC and three analysis categories for the FAP.

Option 1. FAC is recommended for materials with a low initial work-hardening rate or as an initial screening test or when details of the material stress-strain curve are not known. Option 1 is not recommended for materials with discontinuous stress-strain curves. The FAC is

$$K_r = (1 - 0.14 L_r^2)[0.3 + 0.7 \exp(-0.65 L_r^6)] \quad \textit{for} \quad L_r \leq \frac{\sigma_f}{\sigma_y} \tag{26}$$

and

$$K_r = 0 \qquad \textit{for} \quad L_r > \frac{\sigma_f}{\sigma_y} \tag{27}$$

where

$$L_r = \frac{\textit{applied load}}{\textit{rigid-plastic limit load}} \tag{28}$$

and σ_f, the flow stress, is the average of the yield strength, σ_y, and tensile strength, σ_u.

The Option 1 FAC, Equation (26), is an empirical fit to Option 2 FACs for a variety of steels (including an elastic perfectly plastic material) but biased toward a lower bound (Milne et al., 1988b).

Option 2. FAC is recommended for materials with a high initial work-hardening rate, (e.g., strain-aging mild steels), or for materials with a discontinuous yield point, or when the complete material true stress-strain curve is known. The FAC is

$$K_r = \frac{1}{\sqrt{\dfrac{E\epsilon}{L_r \sigma_y} + \dfrac{L_r^3 \sigma_y}{2E\epsilon}}} \qquad for \;\; L_r \le \frac{\sigma_f}{\sigma_y} \tag{29}$$

$$K_r = 0 \qquad for \; L_r > \frac{\sigma_f}{\sigma_y} \tag{30}$$

where L_r is expressed by Equation (28) and ε is the true strain for a true stress of $L_r \cdot \sigma_y$ from the material's true stress-strain curve.

The Option 2 FAC, Equation (29), was developed from expressions for J-integrals from the Electric Power Research Institute/General Electric Handbook (Kumar et al., 1981) reformulated to use actual true stress-strain curves rather than the Ramberg-Osgood model, Equation (47). Also, approximations erring on the conservative side were introduced to make the formulae geometrically independent and thus obviating the need for fully-plastic power-law solutions for each geometry (Milne et al., 1988b).

Option 3. FAC is the most sophisticated of the three options and requires a J-integral R-curve for the subject material.[11] Option 3 acknowledges the presence of stable crack extension and a concomitant increase in fracture toughness.

$$K_r = \sqrt{\frac{J_e}{J}} \qquad for \; L_r \le 1 \tag{31}$$

and

$$K_r = 0 \qquad for \; L_r > 1 \tag{32}$$

where J_e is the elastic component of the total J-integral, J.

The construction of the Option 3 FAC is identical to that of the Electric Power Research Institute/General Electric FAD and Bloom's deformation plasticity failure assessment diagram (DPFAD) described in later sections.

Appendix 8 R6, Revision 3, presents an FAC for C-Mn steels

and

[11] The R-curve is a plot of crack-driving force (K, J-integral, or CTOD) versus stable crack extension, Δa.

$$K_r = \frac{1 - 0.1 S_r^2 + 0.1 S_r^4}{1 + 3 S_r^4} \qquad for\ S_r \leq 1 \tag{33}$$

$$K_r = 0 \qquad for\ S_r > 1 \tag{34}$$

where

$$S_r = \frac{L_r \sigma_y}{\sigma_f} \tag{35}$$

The Appendix 8 FAC, Equation (33), is an empirical lower bound to Option 1 FACs for a variety of C-Mn steel plates and weld metal, finite element J analyses (option 3), and tests on CCP and compact tension specimens (Milne et al., 1988b).

The ordinates of the FAPs are computed from:

$$K_r = \frac{K_{app}}{K_{mat}} \tag{36}$$

where K_{app} is the linear elastic stress-intensity factor concomitant with the applied load, and K_{mat} is the material fracture toughness.

Category 1 acknowledges elastic-plastic interaction of applied and residual stresses in the computation of K_{app}. Also, the initial flaw size, a_o, is used. K_{mat} is the linear elastic toughness K_{Ic} or the K-equivalent of the J-integral toughness, J_{mat}:

$$K_{mat} = \sqrt{\frac{E J_{mat}}{1 - \nu^2}} \tag{37}$$

Category 2 consists of two FAPs—one using a_o and one using $a_o + \Delta a_g$, where Δa_g is the maximum stable crack extension permitted in a J-integral fracture toughness test.

Category 3 consists of several FAPs computed using various amounts of stable crack extension, Δa, as described in a later section on the DPFAD.

In Categories 2 and 3, K_{mat} is the K-equivalent of the J-integral toughness expressed by Equation (37). In Categories 1, 2 and 3, elastic-plastic interaction of applied and residual stresses is acknowledged in the computation of K_{app}. K_{app} is increased to a magnitude such that its J-integral is equivalent to that of the combined applied and residual stresses but accommodating inelastic response.

The limit-load formulae used to compute L_r for the FAP are reviewed in Miller (1988). The options and analysis categories of CEGB R6, Revision 3, are summarized in Appendix A.

Current Fracture Criteria—PD 6493

The BSI incorporated a three-tiered (or three-level) FAD criteria using CTOD in the second edition of their fitness-for-purpose guidance (BSI PD 6493, 1991; Garwood et al., 1987; Burdekin et al., 1988).

The tiered approach to the FAD treats the problem with increasing analytical sophistication and data requirements and decreasing conservatism. The first tier or level (the least sophisticated and the most conservative) requires only an elastic fracture mechanics analysis and uses single-value estimates of fracture toughness, for example, δ_c, δ_u, or δ_m. On the other hand, the highest tier (the most sophisticated and least conservative) requires an elastic-plastic fracture mechanics analysis and uses the tearing resistance of the material (CTOD R-curve). Also, the highest tier, requires the expression of the material's true stress-strain curve. The procedure is to evaluate a given situation at the first tier and, if the FAP falls below the FAC, safe performance is concluded. If, however, the FAP falls on or above the FAC, then the investigator goes to the second tier analysis and so on. If an unsafe situation is predicted at the highest tier, a redesign or selection of a tougher material is required.

Level 1 FAC (Figure 17) is a modified version of the dimensionless CTOD design curve used in the current version of PD 6493, that is, a recasting of Equation (13) and Equation (14). Level 1 may be thought of as a screening method and includes a safety factor of about 2 on flaw size in the construction of the FAC. For the FAP (not shown in Figure 17), K_r is the square root of the ratio of the applied elastic CTOD to the material CTOD toughness or:

$$K_r = \sqrt{\frac{\delta_e}{\delta_{mat}}} \tag{38}$$

The elastic CTOD is determined from the elastic stress-intensity factor concomitant with the applied load:

$$\delta_e = \frac{K^2}{\delta_y E} \tag{39}$$

No elastic-plastic interaction of applied and residual stresses is acknowledged in the computation of K, that is, the validity of superposition is assumed.

The material CTOD toughness, δ_{mat}, may be the test determined value, δ_c, δ_u, or δ_m (ASTM E 1290, 1993; BSI 5748, 1991).

S_r is the ratio of the effective net-section stress.[12]

[12] The effective net-section stress is the net-section stress expressed in such a way that S_r is equivalent to the ratio of the applied (remote) stress to the applied (remote) stress at plastic collapse (Willoughby, 1982; Willoughby and Davy, 1989).

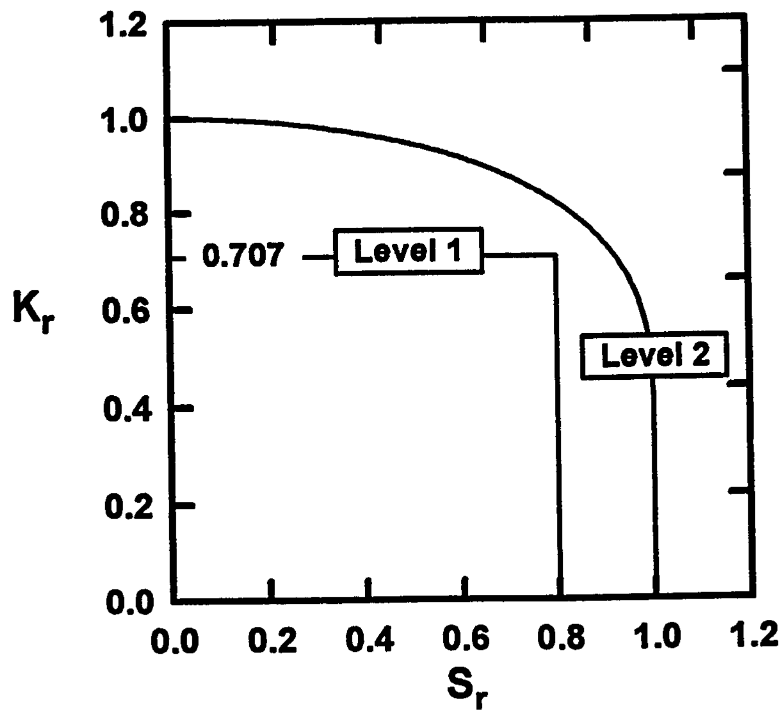

FIGURE 17 PD 6493 FADs, Levels 1 and 2.

$$S_r = \frac{\sigma_n}{\sigma_f} \tag{40}$$

Level 2 FAC (Figure 17) is identical to the original form of the R6 curve, Equation (22), based on Dugdale's strip-yield model and is expressed as an inverse function of the log-secant of S_r. The expressions for the coordinates K_r and S_r of the FAP (not shown in Figure 17) are identical to those of Level 1 except that inelastic stress redistribution due to the interaction of the applied and residual stresses is considered (in the manner of the present CEGB R6 curve).

There is no safety factor inherent to the FAC, and the latter should be considered critical in the assessment. It is believed that Level 2 will be the preferred method for most cases. However, the Level 2 model of the FAC is inaccurate for high work-hardening materials. It should be noted that, as in Level 1, only the elastic stress-intensity factor for the flawed member is needed. Level 3 is the most sophisticated of the three levels and will be used normally in the assessment of high work-hardening materials and/or stable tearing where the level 2 approach would prove too restrictive.

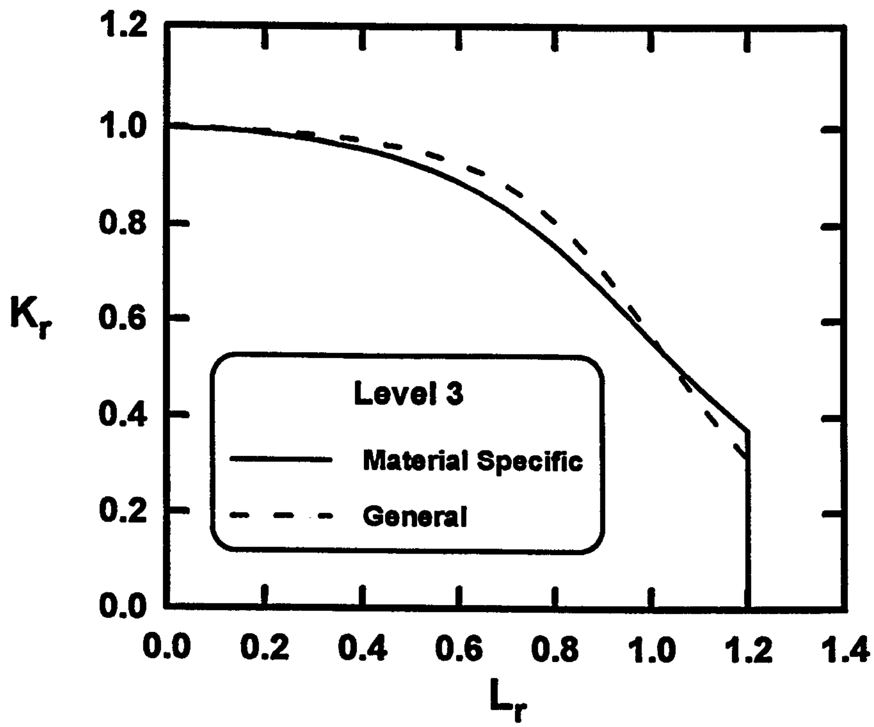

FIGURE 18 PD 6493 FAD, Level 3.

Level 3 (Figure 18) consists of two alternate criteria: (1) a general FAC similar to Option 1 of the current R6 method (Milne et al., 1986), that is, Equation (26), or (2) a material specific FAC similar to Option 2 of the current R6, that is, Equation (29).

The coordinate L_r (Figure 18) is the ratio of the net-section stress to the yield stress σ_y. Inelastic stress redistribution due to the interaction of the applied and residual stresses is handled as in Level 2.

K_r for the general FAC is a function only of L_r; for the material-specific FAC, it is a function of L_r, yield strength, and the true stress-strain curve. K_r for the FAP is identical in form to that of Level 2. There is *no* safety factor inherent to either FAC, and they should be considered critical in the assessment. It should be noted that, as in Levels 1 and 2, *only the elastic stress-intensity factor for the flawed member is needed.* The three levels for PD 6493 are summarized in Appendix B.

Electric Power Research Institute/General Electric Model

In general, the J-integral is used to rank materials qualitatively with respect to toughness when limited plastic flow is sufficient to invalidate the linear elastic approach. However, General

Electric Company, in a project sponsored by the Electric Power Research Institute (EPRI), has developed an elastic-plastic model to describe the fracture behavior of ductile materials (Kumar et al., 1981; Kanninen and Popelar, 1985). The EPRI/GE model expresses J-integral (J), and crack-tip opening displacement or CTOD (δ), as the sum of the elastic and plastic components:

$$J = J_e + J_p \tag{41}$$

and

$$\delta = \delta_e + \delta_p \tag{42}$$

The elastic component is the conventional form from linear elastic fracture mechanics, for example, for J-integral:

$$J_e = \frac{K^2(1-\nu^2)}{E} \tag{43}$$

or for CTOD (δ):

$$\delta_e = \frac{K^2(1-\nu^2)}{\sigma_y E} \tag{44}$$

The plastic components, derived from two-dimensional finite element analysis using a GE-developed incompressible, nonlinear element, are functions of the geometry and type of specimen, limit load, and the parameters of the Ramberg-Osgood description of the particular material's true stress-strain curve. The plastic components are

$$J_p = \alpha\, \sigma_0 \epsilon_0 b h_1 (S_r)^{n+1} \tag{45}$$

and

$$\delta_p = \alpha\, \epsilon_0 b d h_1 (S_r)^{n+1} \tag{46}$$

where α and n are the parameters of the Ramberg-Osgood true stress-strain curve, Equation (47), b is the depth of the uncracked ligament ($b = W - a$), and S_r is as defined in Equation (24).

The parameter h_1 in Equations (45) and (46) is tabulated for various specimen types, both plane stress and plane strain, as a function of aspect ratio a/W and strain-hardening exponent n (Kumar et al., 1981; Kanninen and Popelar, 1985; Kumar et al., 1982; Kumar et al., 1984; Kumar and German, 1987). The parameter d, Equation (46), is expressed graphically as a function of α, n, and σ_0/E for both plane stress and plane strain (Kumar et al., 1981; Kanninen and Popelar, 1985).

The Ramberg-Osgood equation[13] of the true stress-true strain curve is

$$\frac{\epsilon}{\epsilon_0} = \frac{\sigma}{\sigma_0} + \alpha\left(\frac{\sigma}{\sigma_0}\right)^n \tag{47}$$

where the yield strain ε_o is computed from the yield strength $\sigma_o = \sigma_y$,

$$\epsilon_0 = \frac{\sigma_0}{E} \tag{48}$$

The complete solutions for many geometries are listed in the EPRI/GE Handbook (Kumar et al., 1981), subsequent EPRI/GE reports (Kumar et al., 1982, 1984; Kumar and German, 1987), and the textbook by Kanninen and Popelar (1985). EPRI and Novetech jointly developed a three-volume Ductile Fracture Handbook (EPRI/Novetech, 1989, 1990, 1991) that assembles and synthesizes the elastic-plastic fracture mechanics solutions for cracked cylinders developed in EPRI-sponsored research. The first volume of the Ductile Fracture Handbook presents solutions for circumferential through-wall cracks. The second volume presents solutions for circumferential part-through-wall cracks and axial through-wall cracks. The third volume presents solutions for axial part-through-wall cracks and cracks in elbows, tees, and nozzles and flaw evaluation procedures for piping and pressure vessels.

The EPRI/GE J-integral (or CTOD) model is applied to design in one of three ways: (1) the crack driving force diagram (CDFD), (2) the stability assessment diagram (SAD), and (3) the FAD, described in the next section).

The CDFD is nothing more than a J-integral R-curve analysis analogous to that for the LEFM K R-curve analysis. In this approach, the material resistance (J R-curve) is compared to the crack driving force. The latter is a family of curves expressing J versus a for various values of load. The CDFD predicts a complete history of deformation and crack growth behavior. The CDFD may also be constructed with a family of curves showing J versus a for various values of load-point displacement.

The SAD, basically a compression of the information contained in a CDFD, utilizes the tearing modulus, T_J. T_J is the nondimensionalized slope of the J R-curve where

$$T_J = \frac{E}{\sigma_y^2}\frac{\partial J}{\partial a} \tag{49}$$

The CDFD and SAD are described and illustrated in the EPRI/GE Handbook (Kumar et al., 1981).

[13] The EPRI/GE Handbook also presents alternate estimation formulae for stress-strain curves (Kumar et al., 1981).

Electric Power Research Institute/General Electric Failure Assessment Diagram

The concept of the CEGB R6 Curve (Milne et al., 1986) can be combined with the deformation plasticity solutions of the EPRI/GE Handbook (Kumar et al., 1981) to give the FAD (Figure 19).

The FAD consists of two constructions—the FAC and the FAP. The FAC is a function of geometry, initial flaw size, and stress-strain properties of the material. The FAP is a function of geometry, initial flaw size, stable crack extension, and the J-R curve (i.e., toughness) of the material.

In the FAD, K_r, Equation (23), can be interpreted as a ratio of J-integrals. In the FAC (Figure 19):

$$K_r = \sqrt{\frac{J_e}{J}} \tag{50}$$

where J_e (a function of the geometry and initial flaw size a_0) is the elastic component of the J-integral, Equation (43), and J (a function of both geometry and material) is the sum of the elastic and plastic components:

$$J = J_e + J_p . \tag{51}$$

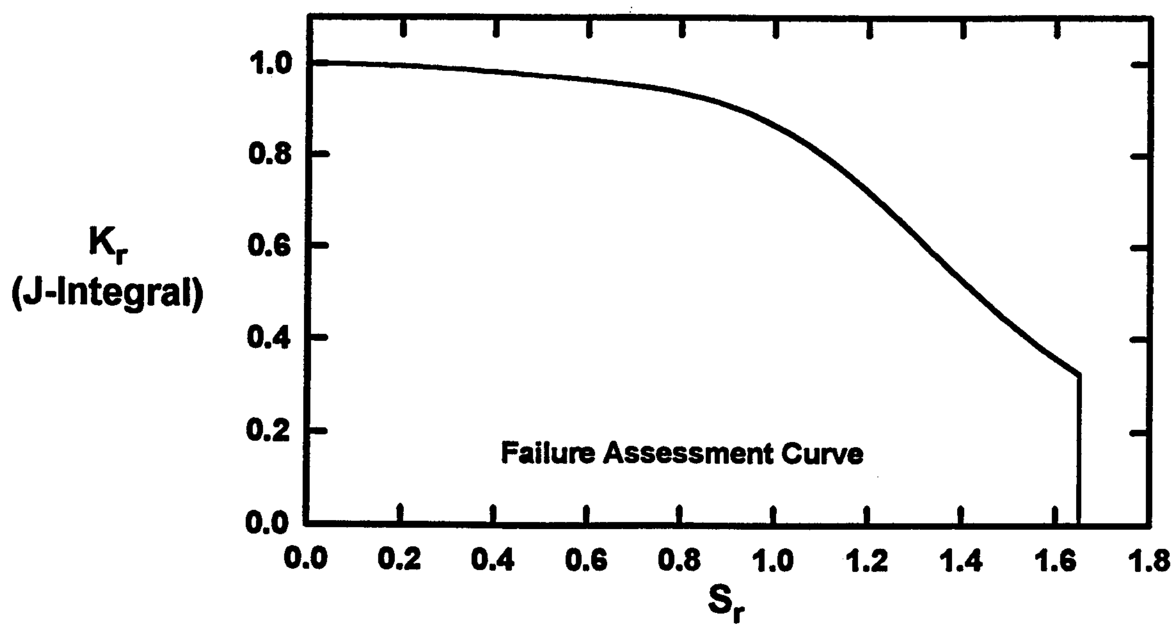

FIGURE 19 FAC.

In Equation (51), J_e is a function of the geometry, initial flaw size a_0, and the plastic zone size; and J_p, from the EPRI/GE Handbook, is a function Equation (45), of the initial flaw size; a_0, material true stress-strain curve, and ratio S_r, Equation (24). In Equation (24), $_c$ is a function of a_0.

For the FAP (Figure 20), the fracture ratio is

$$K_r = \sqrt{\frac{J_e}{J_R}} \tag{52}$$

where J_e is expressed by Equation (43) but using, for crack length, the sum of the initial flaw size and stable crack extension, that, $a_0 + \Delta a$, in the expression for the elastic stress-intensity factor K. J_R is the equation of the J-R-Curve expressing the J-Δa relation for the given material. The ratio S_r is as before, that is, Equation (24), except that σ_c is a function of $a_0 + \Delta a$.

To perform a failure assessment, a value for load (or stress) is assumed and the FAPs for several stable crack extensions, Δa, at that load (or stress) are plotted on the FAD (Figure 21). Rays from the origin to the FAC are constructed through each point, and the ratios:

$$\frac{\text{length of ray to the FAC}}{\text{length of ray to the FAP}}$$

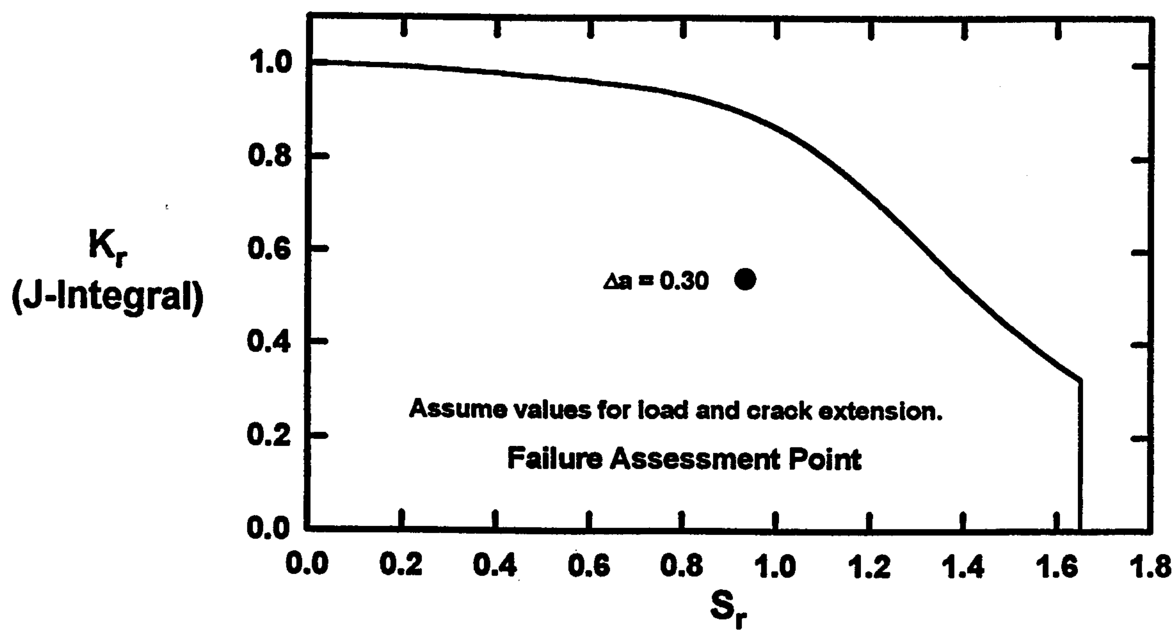

FIGURE 20 FAP.

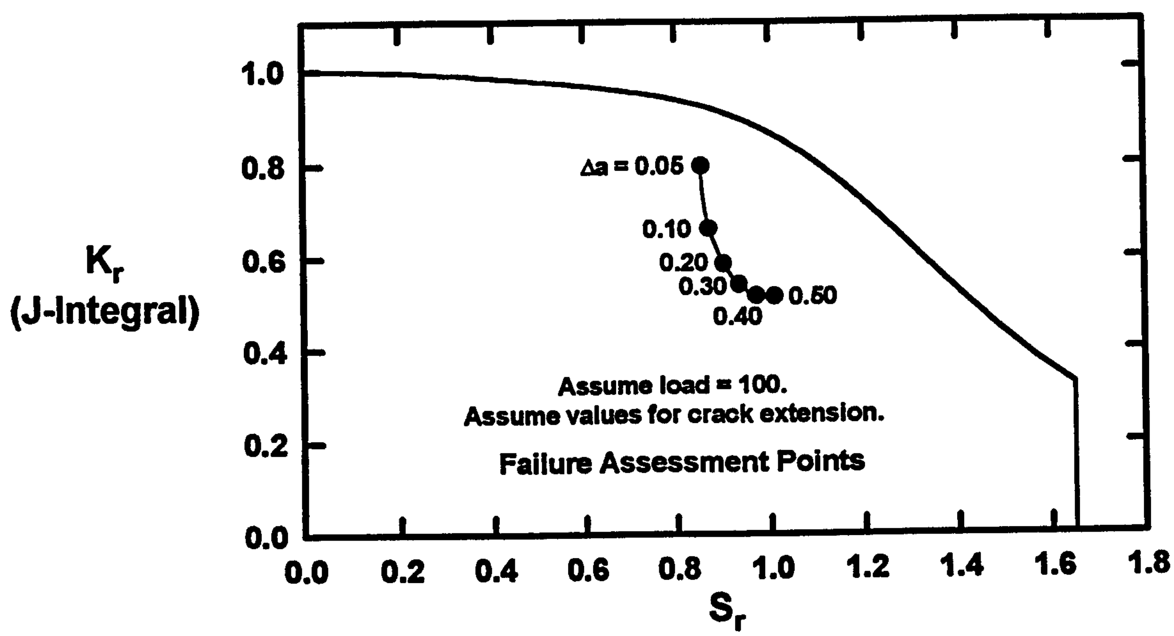

FIGURE 21 FAPs.

are determined. Instability occurs at the stable crack extension for which the ratio is a maximum, and the instability load (or stress) is that ratio times the assumed load (or stress) (Figure 22). This solution is identical to a solution using the EPRI/GE crack driving force diagram, where (Figure 23)

$$\frac{\partial J_{applied}}{\partial a} = \frac{\partial J_{R\text{-}curve}}{\partial a}. \tag{53}$$

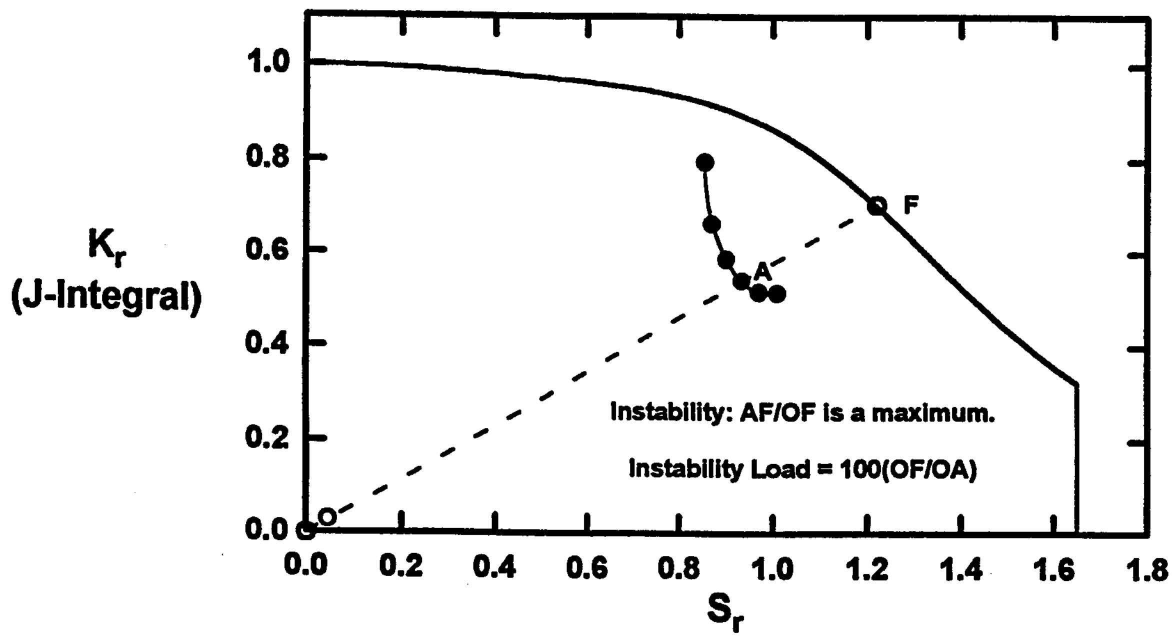

FIGURE 22 FAD, instability load.

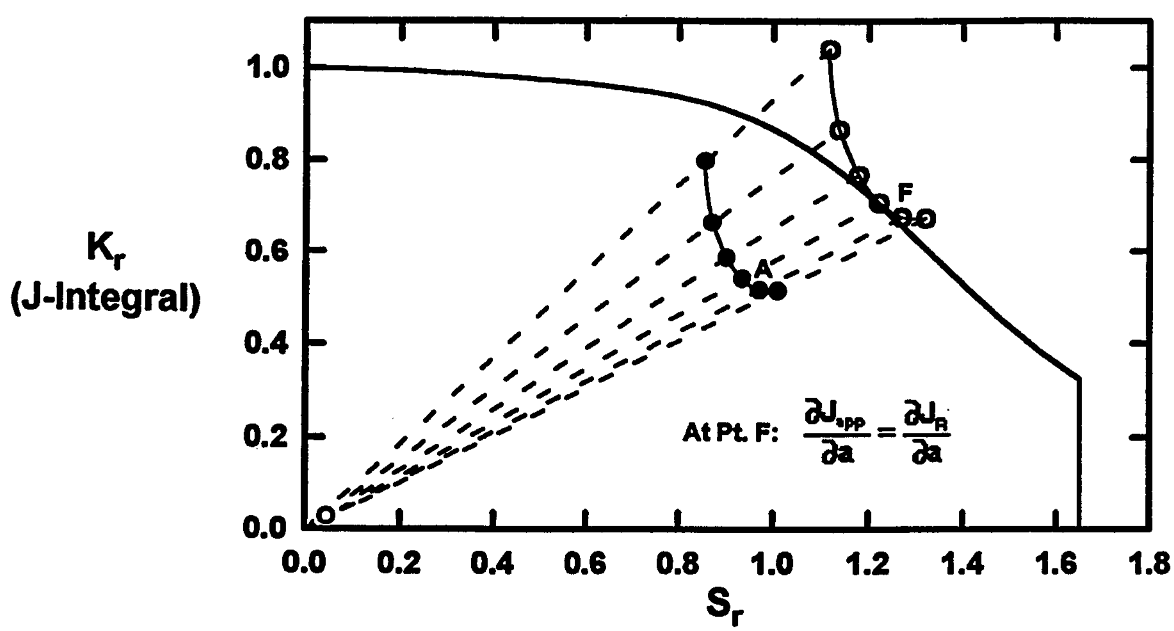

FIGURE 23 FAD, instability.

Deformation Plasticity Failure Assessment Diagram

Bloom extended the R6 concept to elastic-plastic fracture using the J-integral and called his model the DPFAD (Bloom, 1983, 1985a). In the latter approach, Bloom uses the format of the R6 curve, the plasticity solutions from the EPRI/GE model (Kumar et al., 1981; Kanninen and Popelar, 1985) and subsequent EPRI/GE reports (Kumar et al., 1982, 1984; Kumar and German, 1987), and the true stress-strain curve of the material under study (required by the EPRI/GE model). Bloom corroborated the DPFAD approach with tests on various specimen configurations (Bloom, 1983, 1985a) and extended the DPFAD to analytical models of internal semi-elliptical flaws in pressurized cylinders (Bloom, 1985b, 1986). Also, Bloom has corroborated the DPFAD approach with tests on axially flawed pressure vessels and pipes (Bloom, 1988). Recently, Bloom has shown that the FAD approach can be used reliably in structural integrity assessments for materials with stress-strain curves that do not follow the Ramberg-Osgood model (Bloom, 1994).

ASME Section XI of the Nuclear Pressure Vessel and Piping Codes and Standards includes a Code Case for the evaluation of flaws in ferritic piping using the DPFAD methodology. The Code Case (Bloom, 1989) is a simplification of the DPFAD utilizing three lower-bound FACs to describe both circumferential and axial part-through-wall flaws in ferritic piping.

Examples of Application

The structural integrity analysis of a heavy rolled H-section (with rolling residual stresses) using the levels for PD 6493 and Appendix 8, R6, is presented in Reemsnyder (1991).

The FADs of the levels for PD 6493 and Appendix 8, R6, are applied to a tubular transverse butt weld, both as-welded and stress-relieved, in Appendix D of Reemsnyder (1994).

REFERENCES BY SECTION

INTRODUCTION

Jordan, C.R., and R.P. Krumpen, Jr. 1984. Performance of ship structural details. Welding Journal 63:18–28.

Ramwell, D.C. 1993. Lessons failed to be heeded. Welding and Material Fabrication 61:186–187.

Reemsnyder, H.S. 1983a. Stress analysis. Transactions of the Society of Automotive Engineers 91:2327–2336.

Reemsnyder, H.S. 1983b. Constant amplitude fatigue life assessment models. Transactions of the Society of Automotive Engineers 91:2337–2350.

PART I: FATIGUE OF WELDED JOINTS

AAR (Association of American Railroads). 1981. Fatigue Design of Freight Cars. Manual of Standards and Recommended Practices. Chicago, Illinois: AAR.

ABS (American Bureau Shipping). 1993. Guide for Fatigue Strength Assessment of Tankers. Houston, Texas: ABS.

AISC (American Institute of Steel Construction). 1989. Specification for the Design, Fabrication and Erection of Structural Steel for Buildings. Chicago, Illinois: AISC.

API (American Petroleum Institute). 1983. Alternate Standards for Acceptability for Girth Welds. Appendix A, Standards for Welding Pipelines and Related Facilities. API Standard 1104, 16th ed. Washington, D.C.: API.

API. 1989. API Recommended Practice for Planning, Designing and Constructing Fixed Offshore Platforms. API RP 2A, 18th ed. Washington, D.C.: API.

ASME (American Society of Mechanical Engineers). 1995. Rules for Inservice Inspection of Nuclear Power Plant Components. Section XI, Division 1, Boiler and Pressure Vessel Code. New York: ASME.

AWS (American Welding Society). 1992. Structural Welding Code—Steel, 13th ed. ANSI/AWS D1.1–92. Miami, Florida: AWS.

Arockiasamy, M., D.V. Reddy, and J.S. Jani. 1989. Weld Profile and Thickness Effects on Fatigue of Tubular Welded Joints. Dallas, Texas: American Petroleum Institute.

Barsom, J.M., and S.T. Rolfe. 1987. Fracture and Fatigue Control in Structures, 2d ed. Englewood Cliffs, New Jersey: Prentice-Hall.

Bhuyan, G., and O. Vosikovsky. 1989. Prediction of fatigue crack initiation lives for welded plate T-joints based on the local strain approach. International Journal of Fatigue 11:153–159.

Bignonnet, A., H.P. Lieurade, L. Picout, and L. Castex. 1987. Shot peening of welded joints and residual stress relaxation during fatigue. Pp. 50–62 in Proceedings of the 3rd International Conference on Shot Peening held June 15–17, 1987 in Dornstadt, Germany. Deutsche Gesellschaft f. Metalkunde.

Booth, G.S. 1986. Improving the fatigue strength of welded joints by grinding—Techniques and benefits. Metal Construction 18:432–437.

Bouchard, R., O. Vosikovsky, and A. Rivard. 1991. Fatigue life of welded plate T joints under variable-amplitude loading. International Journal of Fatigue 13:7–15.

BSI (British Standards Institution). 1991. Guidance on Methods for Assessing the Acceptability of Flaws in Fusion Welded Structures, 2d ed. PD 6493. London: BSI.

BSI. 1993. Code of Practice for Fatigue Design and Assessment of Steel Structures. BS 7608. London: BSI.

Burnside, O.H., S.J. Hudak, Jr., E. Oelkera, K. Chan, and R.J. Dexter. 1984. Long-Term Corrosion Fatigue of Welded Marine Steels. Ship Structure Committee Report SSC–318. Washington, D.C.: Ship Structure Committee.

Campbell, J.E. 1971. Shot Peening for Improved Fatigue Properties and Stress Corrosion Resistance. Columbus, Ohio: Battelle-Columbus Laboratories.

Cheng, Y.W. 1985. The fatigue crack growth of a ship steel in seawater under spectrum loading. International Journal of Fatigue 7:95–100.

Dahle, T. 1994. Long-life spectrum fatigue tests of welded joints. International Journal of Fatigue 16:392–396.

Dijkstra, O.D., and J. de Back. 1980. Fatigue strength of tubular T- and X-joints. OTC Paper 3696. Pp. 177–186 in Proceedings of the 12th Offshore Technology Conference held May 5–7, 1980 in Houston, Texas. Jersey City, New Jersey: Society of Naval Architects and Marine Engineers.

Doty, W.D. 1955. Properties and characteristics of a quenched and tempered steel for pressure vessels. Welding Journal 34:425-s–441-s.

Ebara, R., H. Kino, M. Hanzawa, and H. Yokota. 1981. Corrosion fatigue crack propagation behaviour of HT 80 in 4°C sea water. Pp. 424–430 in Materials, Experimentation and Design in Fatigue—Proceedings of Fatigue '81 held March 24–27, 1981 at University of Warwick, England. Guildford, England: Westbury Press.

Ferreira, J.M., and C.M. Branco. 1988. Influence of weld and plate geometry on the fatigue strength of cruciform joints. Theoretical and Applied Fracture Mechanics 9:23–32.

Fisher, P.J. 1983. Revised fatigue guidance and the certifying authority. Paper No. 57. Pp. 57-1–57-5 in Proceedings of the 2nd International Conference on Offshore Welded Structures held November 16–18, 1982 in London, England. Abington, Cambridge: The Welding Institute.

Frank, K.H. 1971. The Fatigue Strength of Fillet Welded Connections. Ph.D. Dissertation. Bethlehem, Pennsylvania: Lehigh University.

Fu, B., J.V. Haswell, and P. Bettess. 1993. Weld magnification factors for semi-elliptical surface cracks in fillet welded T-butt joint model. International Journal of Fracture 63:155–171.

Granjon, H., and R.P. Newman. 1970. Fatigue of Welded Assemblies. Cambridge, United Kingdom: International Institute of Welding.

Green, G.J., and D.H. Marlin. 1951. Stress concentration problems in welded construction. Welding Journal 30:607–617.

Griffiths, J.R., J.L. Mogford, and C.E. Richards. 1971. Influence of mean stress on fatigue-crack propagation in a ferritic weld metal. Metal Science Journal 5:150–154.

Gurney, T.R. 1979. Fatigue of Welded Structures, 2d ed. London: Cambridge University Press.

Gurney, T.R. 1983a. Revised fatigue design rules. Metal Construction 15(1):37–44.

Gurney, T.R. 1983b. The basis of the revised fatigue design rules in the Department of Energy offshore guidance notes. Paper No. 55. Pp. 55-1–55-11 in Proceedings of the 2nd International Conference on Offshore Welded Structures held November 16–18, 1982 in London, England. Abington, Cambridge: The Welding Institute.

Gurney, T.R. 1989. Cumulative damage of welded joints, parts 1 and 2. Joining & Materials (July):320–323; (August):390–395.

Gurney, T.R., and S.J. Maddox. 1973. A re-analysis of fatigue data for welded joints in steel. Welding Research International 3:1–54.

Harral, B.B. 1987. The application of a statistical fatigue life prediction method to agricultural equipment. International Journal of Fatigue 9:115–118.

Heywood, R.B. 1962. Designing Against Fatigue of Metals. New York: Reinhold.

Hirt, M.A., and J.W. Fisher. 1973. Fatigue crack growth in welded beams. Engineering Fracture Mechanics 5:415–429.

Hobbacher, A. 1992. Stress-intensity factors of plates under tensile load with welded-on flat side gussets. Engineering Fracture Mechanics 41:897–905.

ICE (Institution of Civil Engineers). 1981. Fatigue in Offshore Structural Steels. London: Thomas Telford.

IIW (International Institute of Welding). 1990. IIW Guidance on Assessment of The Fitness-for-Purpose of Welded Structures. IIW/IIS-SST-1157-90. Cambridge, United Kingdom: International Institute of Welding.

Irvine, N.M. 1983. The concept and definition of hot-spot stress as used in the revised guidance for fatigue assessment of welded tubular joints. Paper No. 54. Pp. 54-1–54-10 in Proceedings of the 2nd International Conference on Offshore Welded Structures held November 16–18, 1982 in London, England. Abington, Cambridge: The Welding Institute.

Kapadia, B.M., and E.J. Imhof, Jr. 1977. Fatigue crack propagation in electroslag weldments. Pp. 159–173 in Flaw Growth and Fracture. STP 631. Philadelphia, Pensylvania: American Society for Testing and Materials.

Kottgen, V.B., R. Oliver, and T. Seeger. 1992. Fatigue Analysis of Welded Connections Based on Local Stresses. IIW-Document XIII-1408-91, XV-802-92. Cambridge, United Kingdom: International Institute of Welding.

Larson, C.F. 1972. Bibliography on fatigue strength of welded joints, 1950–1971. Welding Journal 51:457-s–460-s.

Lawrence, F.V., Jr. 1973. Estimation of fatigue crack propagation life in butt welds. Welding Journal 52:212-s–220-s.

Lawrence, F.V., Jr., R.J. Mattos, Y. Higashida, and J.D. Burk. 1978. Estimating the fatigue crack initiation life of welds. Pp. 134–158 in Fatigue Testing of Weldments. STP 648, Philadelphia, Pennsylvania: American Society for Testing and Materials.

Lawrence, F.V., Jr., and W.H. Munse. 1973. Fatigue crack growth propagation in butt welds containing joint penetration defects. Welding Journal 52:221-s–225-s, 232–s.

Lawrence, F.V., Jr., and J.B. Radziminski. 1970. Fatigue crack initiation and propagation in high yield-strength steel weld metal. Welding Journal 49:445-s–452-s.

Loader, D.J., P.A. Shorrock, and V.A. Staton. 1983. Implications of the new fatigue design guidance notes for the designer and platform operator. Paper No. 56. Pp. 56-1–56-10 in Proceedings of the 2nd International Conference on Offshore Welded Structures held November 16–18 in London, England. Abington, Cambridge: The Welding Institute.

Maddox, S.J. 1970. Fatigue crack propagation in weld metal and HAZ. Metal Construction and British Welding Journal 2:285–289.

Maddox, S.J. 1974a. A fracture mechanics approach to service load fatigue in welded structures. Welding Research International 4(2):1–30.

Maddox, S.J. 1974b. Assessing the significance of flaws in welds subject to fatigue. Welding Journal 53:401-s–409-s.

Maddox, S.J. 1974c. Fatigue crack propagation data obtained from parent plate, weld metal and HAZ in structural steels. Welding Research International 4(1):36–60.

Maddox, S.J. 1975. An analysis of fatigue cracks in fillet welded joints. International Journal of Fracture 11(2):221–243.

Maddox, S.J. 1985. Improving the fatigue strength of welded joints by peening. Metal Construction 17:220–224.

Maddox, S.J., 1991. Fatigue Strength of Welded Structures, 2d ed. Abington, Cambridge, United Kingdom: Abington Press.

Maddox, S.J. 1993. Recent advances in the fatigue assessment of weld imperfections. Welding Journal 72:42–51.

Makhnenko, V., and V.E. Pochinok. 1982. Fracture mechanics criteria applied in calculating the strengths of welded joints with crack-type discontinuities present. Automatic Welding 35:42–48.

Marek, P.J., M. Perlman, A.W. Pense, and L. Tall. 1970. Fatigue tests on a welded beam with pre-existing cracks. Welding Journal 49:245-s–253-s.

Marshall, P.W. 1974. Basic Considerations for Tubular Joint Design in Offshore Construction. WRC Bulletin 193. New York: Welding Research Council.

Marshall, P.W. 1975. General considerations for tubular joint design. Pp. 10–31 in Welding in Offshore Construction. Abington, Cambridge: The Welding Institute.

Matlock, D.K., G.R. Edwards, D.L. Olsen, and S. Ibarra. 1987. Effect of sea water on the fatigue propagation characteristics of welds for offshore structures. Journal of Materials Engineering 9:25–34.

Mattos, R.J., and F.V. Lawrence. 1977. Estimation of the Fatigue Crack Initiation Life in Welds Using Low Cycle Fatigue Concepts. SP-424. Warrendale, Pennsylvania: Society of Automotive Engineers.

Metal Construction. 1978. Improving fatigue strength of welded joints. Metal Construction 10(8):380.

Millington, D. 1973. TIG dressing to improve fatigue properties in welded high-strength steels. Metal Construction and British Welding Journal 4:134–139.

Mills, W.J., and L.A. James. 1987. Residual stress effects on fatigue crack growth behavior in stainless steel welds. Transactions of the American Society of Mechanical Engineers 109:336–339.

Mindlin, H. 1968. Influence of details on fatigue behavior of structures. Proceedings, American Society of Civil Engineers 94(ST-12):2679–2698.

Munse, W.H. 1978. Fatigue of weldments—Tests, design and service. Pp. 89–112 in Fatigue Testing of Weldments. STP 648. Philadelphia, Pennsylvania: American Society for Testing and Materials.

Munse, W.H., and L. Grover. 1964. Fatigue of Welded Steel Structures. New York: Welding Research Council.

Munse, W.H., T.W. Wilbur, M.L. Tellalian, K. Nicoll, and K. Wilson. 1983. Fatigue Characterization of Fabricated Ship Details for Design. SSC-318. Washington, D.C.: Ship Structure Committee.

Murakami, Y., editor-in-chief. 1987. Stress Intensity Factors Handbook. Volume 2. New York: Pergamon Press.

Ohta, A., M. Kosuge, and S. Nishijima. 1987. Fatigue crack growth in welded joints under compressive applied stress. International Journal of Fracture 33:R17–R20.

Osgood, C.C. 1970. Fatigue Design. New York: Wiley-Interscience.

Otegui, J.L., U.H. Mohaupt, and D.J. Burns. 1991. Effect of weld process on early growth of fatigue cracks in steel T joints. International Journal of Fatigue 13:45–58.

Park, S.K., and F.V. Lawrence. 1988. A probabilistic fatigue design method for weldments. Pp. 311–316 in Proceedings of the 7th International Conference on Offshore Mechanics and Arctic Engineering held February 7–12, **[YEAR]** in Houston, Texas. New York: American Society of Mechanical Engineers.

Parry, M., H. Nordberg, and R.W. Hertzberg. 1972. Fatigue crack propagation in A514 base plate and welded joint. Welding Journal 51:485-s–490-s.

Pollard, B., and R.J. Cover, 1972. Fatigue of steel weldments. Welding Journal 51:544-s–554-s.

Puchner, O. 1960. Increase by local heating of the fatigue limit of plates and beams with welded-on gusset plates. Welding Research Abroad 6:32–44.

Pyle, T., M. Pitrun, and P.H. Townsend. 1990. Removal of fatigue damage in welded joints. Pp. 811–814 in Recent Trends in Welding Science and Technology, TWR'89, S.A. David, and J.M. Vitek, eds. Materials Park, Ohio: ASM International.

Radaj, D. 1990. Design and Analysis of Fatigue Resistant Welded Structures. New York: Halsted Press.

Raghavendran, N.S., and M.E. Fourney. 1994. Stress analysis of a welded joint. Engineering Fracture Mechanics 48:619–627.

Reemsnyder, H.S. 1961. The Fatigue Behavior of Structural Steel Weldments—A literature survey. Fritz Engineering Laboratory Report No. 284.1. Bethlehem, Pennsylvania: Lehigh University.

Reemsnyder, H.S. 1965a. A new specimen for fatigue testing longitudinal fillet weldments. Proceedings ASTM 65:729–735.

Reemsnyder, H.S. 1965b. Fatigue strength of longitudinal weldments in constructional alloy steel. Welding Journal 44:458-s–465-s.

Reemsnyder, H.S. 1968. Fatigue Data for Plain and Welded Structural Steels. Bethlehem, Pennsylvania: Bethlehem Steel Corporation.

Reemsnyder, H.S. 1969. Some significant parameters in the fatigue properties of weld joints. Welding Journal 34:213-s–220-s.

Reemsnyder, H.S. 1974. Fatigue. Pp. 519–551 in Structural Steel Design, 2d ed., L. Tall, ed. New York: Ronald Press.

Reemsnyder, H.S. 1978a. The development and application of fatigue data for structural steel weldments. Pp. 3–21 in Fatigue Testing of Weldments. STP 648. New York: American Society for Testing and Material.

Reemsnyder, H.S. 1978b. Bibliography on fatigue of weldments. Welding Journal 57:178-s–182-s.

Reemsnyder, H.S. 1979. Fatigue properties of welded roller-quenched and tempered RQ steels. Transactions of the Society of Automotive Engineers 88:1905–1914.

Reemsnyder, H.S. 1982a. State of the art of design and analytical models for fatigue of weldments. Pp. 41–55 in Proceedings of the International Conference on Fitness for Purpose in Welded Construction held May 18–20, 1982 in Key Biscayne, Florida. Miami, Florida: American Welding Society.

Reemsnyder, H.S. 1982b. Evaluating the effect of residual stresses on notched fatigue resistance. Pp. 273–295 in Materials, Experimentation and Design in Fatigue—Proceedings of Fatigue '81 held in March 24–27, 1981 at University of Warwick, England. Guildford, England: Westbury Press. (Also, abstracted in Residual Stress Effects in Fatigue. STP 776. Philadelphia, Pennsylvania: American Society for Testing and Materials.)

Reemsnyder, H.S. 1985. Hot-Spot Stress Approach to Fatigue Design of Weldments. Bethlehem, Pennsylvania: Bethlehem Steel Corporation.

Reemsnyder, H.S. 1986a. Fatigue of notched steel sheet and plate in an aqueous environment. Pp. 360–371 in Proceedings of the 1986 International Conference on Fatigue of Engineering Materials and Structures held September 15–16, 1986 at University of Sheffield, England. London: Institution of Mechanical Engineers.

Reemsnyder, H.S. 1986b. Observations, Predictions, and Prevention of Fatigue Cracking in Offshore Structures: Case Histories Involving Fatigue and Fracture Mechanics. STP 918. New York: American Society for Testing and Materials.

Reemsnyder, H.S. 1992. Modelling of the Fatigue Resistance of Single-Lap Spot-Welded Steel Sheet. Annual ASTM Fatigue Lecture, IIW-Document XIII-1469-92. Cambridge, United Kingdom: Welding Institute.

Rodabaugh, E.C. 1980. Review of Data Relevant to the Design of Tubular Joints for Use in Fixed Offshore Platforms. WRC Bulletin 256. New York: Welding Research Council.

Rolfe, S.T., A.E. Henn, and K.T. Hays. 1993. Fracture Mechanics Methodology for Fracture Control in Oil Tankers. Pp. F-1–F-11 in Proceedings Ship Structures Symposium '93 held November 16–17, 1993 in Arlington, Virginia. Washington, D.C.: Society of Naval Architects and Marine Engineers/Ship Structure Committee.

Sanders, W.W., Jr., and J.W. Fisher. 1985. Development of Recommended Specifications for Fatigue Design of Aluminum Structures. Washington, D.C.: The Aluminum Association.

Sanders, W.W., Jr., A.T. Derecho, and W.H. Munse, 1965. Effect of external geometry on fatigue behavior of welded joints. Welding Journal 44:49-s–55-s.

Selby, K.A., J.E. Stallmeyer, and W.H. Munse. 1965. Influence of Geometry and Residual Stresses on Fatigue of Welded Joints. Report S.R.S. No. 297. Urbana: University of Illinois.

Signes, E.G., R.G. Baker, J.D. Harrison, and F.M. Burdekin. 1967. Factors affecting the fatigue strength of welded high-strength steels. British Welding Journal 14:108–116.

Skorupa, M., H. Braam, and J. Pru. 1987. Applicability of approximate K_I solutions towards cracks at weld toes. Engineering Fracture Mechanics 26(5):669–681.

Slater, G. 1985. The effect of repair welds on service performance. Welding Journal 64:22–29.

Smith, K.N., M. El Haddad, and J.F. Martin. 1977. Fatigue life and crack propagation analyses of welded components containing residual stresses. Journal of Testing and Evaluation 5:327–332.

Socie, D.F., and S.D. Antolovich. 1974. Subcritical crack growth characteristics in welded A537 steel. Welding Journal 53:267-s–272-s.

Spraragen, W., and G.E. Claussen. 1937. Fatigue strength of welded joints–A review of the literature to October 1, 1936. Welding Journal 16:1–44.

Spraragen, W., and D. Rosenthal. 1942. Fatigue strength of weld joints. Welding Journal 21:297-s–348-s.

Stambaugh, K.A., D.H. Leeson, F. Lawrence, and C-Y. Hou. Reduction of S–N Curves for Ship Structural Details. Ship Structure Committee Report SSC-369. Washington, D.C.: Ship Structure Committee.

Stambaugh, K.A., F. Lawrence, and S. Dimitriakis. 1994. Improved Ship Hull Structural Details Relative to Fatigue. Ship Structure Committee Report SSC-379. Washington, D.C.: Ship Structure Committee.

Terada, H. 1976. An analysis of the stress intensity factor of a crack perpendicular to the welding bead. Engineering Fracture Mechanics 8:441–444.

Terada, H. 1983. The stress intensity factor for a crack perpendicular to the welding bead. International Journal of Fracture 21:279–284.

To, S., S.B. Lambert, and D.J. Burns. 1993. A multiple crack model for fatigue in welded joints. International Journal of Fatigue 15:330–340.

Tsai, C.L., and D.S. Kim, 1990. Analysis of fatigue crack propagation behavoir in fillet welded T-joint. Engineering Fracture Mechanics 36:653–660.

TWI (The Welding Institute). 1978. European Offshore Steel Research. Abington, Cambridge, United Kingdom: TWI.

TWI. 1980. European Offshore Steels Research Seminar. Abington, Cambridge, United Kingdom: TWI.

U.K. DOE (United Kingdom Department of Energy). 1984a. Offshore Installations: Guidance on Design and Construction. London: Her Majesty's Stationery Office.

U.K. DOE. 1984b. Background to New Fatigue Design Guidance for Steel Welded Joints in Offshore Structures. Report of the Department of Energy Guidance Notes Revision Drafting Panel. London: Her Majesty's Stationery Office.

van der Zanden, A.M., D.B. Robins and T.H. Topper. 1972. Fatigue life prediction for weldments with internal cavities. Pp. 268–284 in Testing for Prediction of Material Performance in Structures and Components. STP 515. Philadelphia, Pennsylvania: American Society for Testing and Materials.

Williams, H.E., H. Ottsen, F.V. Lawrence, and W.H. Munse. 1970. The Effects of Weld Geometry on the Fatigue Behavior of Welded Connections. Report S.R.S. No. 366. Urbana: University of Illinois.

Wylde, J.G. 1980. Fatigue tests on tubular K and KT joints under axial brace loading. OTC Paper 3697. Pp. 187–195 in Proceedings of the 12th Offshore Technology Conference held May 5–7, 1980 in Houston, Texas. Jersey City, New Jersey: Society of Naval Architects and Marine Engineers.

Yamada, K., and M.A. Hirt. 1982a. Fatigue crack propagation from fillet weld toes. Proceedings American Society of Civil Engineers 108 (ST7):1525–1540.

Yamada, K., and M.A. Hirt. 1982b. Fatigue life estimation using fracture mechanics. Pp. 361–368 in Annual Proceedings of International Association of Bridge and Structural Engineering.

Yoshida, S., M. Inagaki, M. Kanao, E. Sasaki, M. Nihei, and M. Kamakura. 1978. Effect of size and frequency on fatigue properties of SM50B butt welded joints. Journal of Japan Welding Society 47(9):627–632.

Zaczek, Z. 1984. Improvement in the fatigue strength of butt welded joints by TIG remelting of weld reinforcements. Metal Construction 16:423–425.

Zwerneman, F.J., and K.H. Frank. 1989. Stress intensity factor for a tee-shaped weldment. Engineering Fracture Mechanics 32(4):561–572.

PART II: ASSESSMENT OF RESIDUAL STRENGTH

API (American Petroleum Institute). 1983. Alternate Standards for Acceptability for Girth Welds. Appendix A, Standards for Welding Pipelines and Related Facilities. API Standard 1104, 16th ed. Washington, D.C.: API.

ASTM (American Society for Testing and Materials). 1993. Standard Test Method for Crack-Tip Opening Displacement (CTOD) Fracture Toughness Assessment. ASTM E 1290. Philadelphia, Pennsylvania: American Society for Testing and Materials.

Bloom, J.M. 1983. Validation of a deformation plasticity failure assessment diagram approach to flaw evaluation. Pp. II-206–II-238 in Elastic-Plastic Fracture: 2nd Symposium Fracture Resistance Curves and Engineering Applications. ASTM STP 803, C.F. Shah and J.P. Gudas, eds. Philadelphia, Pennslyvania: American Society for Testing and Materials.

Bloom, J.M. 1980. Prediction of ductile tearing using a proposed strain-hardening failure assessment diagram. International Journal of Fracture 16:R73–R77.

Bloom, J.M. 1985a. Deformation plasticity failure assessment diagram. Pp. 114–127 in Elastic-Plastic Fracture Mechanics Technology. ASTM STP 896, J.C. Newman, Jr., and F.J. Loss, eds. Philadelphia, Pennsylvania: American Society for Testing and Materials.

Bloom, J.M. 1985b. Extensions of the failure assessment diagram approach—Semi-elliptical flaw in pressurized cylinder. Journal of Pressure Vessel Technology, Transactions American Society of Mechanical Engineers 107:25–29.

Bloom,.J.M. 1986. Extensions of the Failure Assessment Diagram Approach—Semi-Elliptical Flaw in Pressurized Cylinders, part 2. Journal of Pressure Vessel Technology, Transactions American Society of Mechanical Engineers 108:485–489.

Bloom, J.M. 1988. Validation of the deformation plasticity failure assessment diagram (DPFAD) approach. Presented at Fracture Mechanics and Plasticity in Pressure Vessels Session, ASME Winter Meeting held November 28–December 2, 1988 in Chicago, Illinois. New York: ASME.

Bloom, J.M. 1989. Appendix J—The deformation plasticity failure assessment diagram (DPFAD) approach to evaluation of flaws in ferritic piping. Presented at European Symposium on Elastic–Plastic Fracture Mechanics held October 1989 in Frieburg, West Germany.

Bloom, J.M. 1994. Deformation plasticity failure assessment diagram (DPFAD) for materials with non–Ramberg–Osgood Stress–Strain Curves. Presented at ASME Pressure Vessel and Piping Conference held June **[DATE]**, 1994 in Minneapolis, Minnesota. New York: ASME.

BSI (British Standards Institute). 1991. Fracture Mechanics Toughness Tests, Part 1: Method for Determination of K_{Ic}, Critical CTOD and Critical J Values of Metallic Materials. BSI 5748. London: BSI.

BSI. 1980. Guidance on Some Methods for the Derivation of Acceptance Levels for Defects in Fusion Welded Joints. PD 6493. London: BSI.

Burdekin, F.M., and M.G. Dawes. 1971. Practical use of linear elastic and yielding fracture mechanics with particular reference to pressure vessels. Pp. 28–37 in Practical Application of Fracture Mechanics to Pressure Vessel Technology. London: The Institution of Mechanical Engineers.

Burdekin, F.M., S.J. Garwood, and I. Milne. 1988. The basis for the technical revisions to the fracture clauses of PD 6493. Presented at International Conference on Weld Failures held November 1988 in London, England.

Chell, G.G. 1979. A procedure for incorporating thermal and residual stresses into the concept of a failure assessment diagram. Pp. 581–605 in Elastic–Plastic Fracture. ASTM STP 668. Philadelphia, Pennsylvania: American Society for Testing and Materials.

Dawes, M.G. 1974. Fracture control in high strength weldments. Welding Journal 53:369-s–379-s.

Dowling, A.R., and C.H.A. Townley. 1975. The effect of defects on structural failure: A two-criteria approach. International Journal of Pressure Vessels and Piping 3:77–107.

Egan, G.R. 1974. The application of fracture toughness data to the assessment of pressure vessel integrity. Pp. 1037–1048 in Proceedings 2nd International Conference on Pressure Vessel Technology held October 1–4, 1973 in San Antonio, Texas. New York: American Society of Mechanical Engineers.

EPRI/Novetech (Electric Power Research Institute). 1989. Ductile Fracture Handbook. Volume 1. Gaithersburg, Maryland: Novetech Corporation.

EPRI/Novetech (Electric Power Research Institute). 1990. Ductile Fracture Handbook. Volume 2. Gaithersburg, Maryland: Novetech Corporation.

EPRI/Novetech (Electric Power Research Institute). 1991. Ductile Fracture Handbook. Volume 3. Gaithersburg, Maryland: Novetech Corporation.

Garwood, S.J., A.A. Willoughby, G.H. Leggatt, and T. Jutla. 1987. Crack-tip opening displacement (CTOD) methods for fracture mechanics assessments: Proposals for revisions to PD 6493. Presented at Advanced Seminar on Fracture Mechanics held September 1987 in Ispra, Italy.

Harrison, J.D., M.G. Dawes, G.L. Archer, and M.S. Kamath. 1979. The COD approach and its application to welded structures. Pp. 606–631 in Elastic-Plastic Fracture. ASTM STP 668. Philadelphia, Pennsylvania: American Society for Testing and Materials.

Kanninen, M.F., and C.H. Popelar. 1985. Advanced Fracture Mechanics. New York: Oxford University Press.

Kumar, V., M.D. German, and C.F. Shih. 1981. An Engineering Approach for Elastic-Plastic Fracture Analysis. EPRI NP-1931, Project 1237-1. Schenectady, New York: General Electric Company.

Kumar, V., W.W. Wilkening, W.R. Andrews, M.D. German, H.G. de Lorenzi, and D.F. Mowbray, 1982. Estimation Technique for the Prediction of Elastic-Plastic Fracture of Structural Components of Nuclear Systems. 5th and 6th Semiannual Report to EPRI. Contract RP1237-1. Schenectady, New York: General Electric Company.

Kumar, V., M.D. German, W.W. Wilkening, W.R. Andrews, H.G. de Lorenzi, and D.F. Mowbray. 1984. Advances in Elastic-Plastic Fracture Analysis. EPRI NP-1931. Research Project 1237-1. Schenectady, New York: General Electric Company.

Kumar, V., and M.D. German. 1987. Further Developments in Elastic-Plastic Fracture Analysis of Through-Wall Flaws and Surface Flaws in Cylinders. Final Report No. 87-SED-017, Research Project 1237-5. Schenectady, New York: General Electric Company.

Miller, A.G. 1988. Review of limit loads of structures containing defects. International Journal of Pressure Vessels and Piping 32:197–327.

Milne, I., R.A. Ainsworth, A.R. Dowling, and A.T. Stewart. 1986. Assessment of the Integrity of Structures Containing Defects. R/H/R6, Revision 3. Leatherhead, Surrey, England: Central Electricity Generating Board.

Milne, I., R.A. Ainsworth, A.R. Dowling, and A.T. Stewart. 1988a. Assessment of the integrity of structures containing defects. International Journal of Pressure Vessels and Piping 32:3–104.

Milne, I., R.A. Ainsworth, A.R. Dowling, and A.T. Stewart. 1988b. Background to and validation of CEGB Report R/H/R6, Revision 3. International Journal of Pressure Vessels and Piping 32:105–196.

Reemsnyder, H.S. 1991. Structural integrity analysis of a heavy rolled structural section. Presented at the International Institute of Welding Conference on Fitness for Purpose of Welded Structures held October 1991 in Key Biscayne, Florida.

Reemsnyder, H.S. 1994. Engineering Approaches to Elastic-Plastic Failure Assessment. Bethlehem, Pennsylvania: Bethlehem Steel Corporation.

Terry, P. 1974. Comparison of Fracture Toughness Information Derived from the Wide Plate Test and the Deep Notch Test. GS/PROD/149/2/73/C. London, England: British Steel Corporation.

Willoughby, A.A. 1982. A Survey of Plastic Collapse Solutions Used in the Failure Assessment of Part Wall Defects. Research Report 191/1982. Cambridge, England: The Welding Institute.

Willoughby, A.A., and T.G. Davy. 1987. Plastic collapse at part wall flaws in plates. Fracture Mechanics: Perspectives and Directions. STP 1020. Philadelphia, Pennsylvania: American Society for Testing and Materials.

SOURCES

Publications of TWI (formerly The Welding Institute) may be obtained from:

Edison Welding Institute
1100 Kinnear Road
Columbus, Ohio 43212
(614) 486–9400

Publications of Her Majesty's Stationery Office may be obtained from:

HMSO BOOKS - (PC 13A/1)
Publication Centre
P.O. Box 276
London SW8 5DT
United Kingdom

Publications of the British Standards Institution may be obtained from:

American National Standards Institute
1430 Broadway
New York, New York 10018
(212) 642–4900

Documents of the International Institute of Welding may be obtained from:

The General Secretariat of the
International Institute of Welding
Abington Hall, Abington
Cambridge CB1 6AL
United Kingdom

The reports by Kumar et al. (1981, 1982, 1984, and 1987) may be obtained from:

Electric Power Research Institute
3412 Hillview Avenue
Palo Alto, California 94304
(415) 855–2011

The three-volume Ductile Fracture Handbook (EPRI/Novetech, 1991) may be ordered from:

Novetech Corporation
P.O. Box 7605
Gaithersburg, MD 20898-7605
(301) 330–1919

The ABS Guide for Fatigue Strength Assessment of Tankers (ABS, 1993) may be ordered from:

ABS Americas
16855 Northchase Drive
Houston, Texas 77060-6008
(713) 873–0700

APPENDIX A

SUMMARY—CEGB R6, REVISION 3, FAILURE ASSESSMENT DIAGRAMS

FAILURE ASSESSMENT CURVE

Option 1	General curve Low initial work-hardening rate materials Yield and tensile strengths
Option 2	Material specific curve High initial work-hardening rate materials[1] True stress-strain curve
Option 3	Same as EPRI/GE Failure Assessment Curve All Materials True stress-strain curve
Appendix 8	Carbon-manganese steels Yield and tensile strengths

FAILURE ASSESSMENT POINT

Analysis Category 1	Initiation of cracking Plastic correction for residual stresses Single assessment point J-Integral toughness
Analysis Category 2	Small amount of ductile tearing Plastic correction for residual stresses Two assessment points J-Integral toughness
Analysis Category 3	Tearing instability analysis Plastic correction for residual stresses J-Integral R-curve

[1] Strain-aging mild steels and materials with a discontinuous yield point are examples.

APPENDIX B

SUMMARY—PD 6493 FAILURE ASSESSMENT DIAGRAMS

FAILURE ASSESSMENT CURVE

Level 1	Screening Low and medium work-hardening materials Recasting of The Welding Institute Crack-Tip Opening Displacement design curve Yield and tensile strengths
Level 2	Low and medium work-hardening materials Same curve as original Central Electricity Generating Board R6 curve Yield and tensile strengths
Level 3 **General Curve**	Medium and high work-hardening materials Same curve as CEGB Option 1 Yield and tensile strengths
Level 3 **Material Specific Case**	Medium and high work-hardening materials Same curve as CEGB Option 2 True stress-strain curve

FAILURE ASSESSMENT POINT

Level 1	No plastic correction for residual stresses Single assessment point CTOD toughness
Level 2	Plastic correction for residual stresses Single assessment point CTOD toughness
Level 3	Plastic correction for residual stresses Single assessment point or tearing instability analysis CTOD toughness or CTOD R-curve

Reliability in Fatigue and Fracture Analyses of Ship Structure

Paul H. Wirsching
Alaa E. Mansour

ABSTRACT

Design factors associated with fatigue and fracture in the components of marine structures are subject to uncertainty. Reliability technology provides a formal mechanism for managing uncertainty and producing well-engineered designs. A summary of reliability methods as applied to the fatigue and fracture modes is provided herein.

PRELIMINARY REMARKS

In "conventional" design practice, uncertainties in loading and material behavior, as well as in the models used to describe stress and fatigue/fracture strength, are accounted for by factors of safety, which are chosen by relying on experience and intuition. Reliability methods, however, which employ concepts of probability and statistics, provide mathematical procedures for managing uncertainty in a rational and consistent manner, thereby promising to produce well-engineered designs.

Reliability analysis refers to the application of probabilistic and statistical methods to the design process of a system, which, in general, will consist of several components and modes of failure. Two basic problems can be formulated: (1) the reliability assessment problem, for which the goal is to compute the reliability, or an index thereof, relative to the service life for an existing or proposed design; and (2) the design problem, where a target reliability is specified as a basic requirement. Summarized here are methods that are available for performing structural reliability analyses relative to fatigue and fracture. The discussion will focus on *component* fatigue/fracture reliability analysis. Comments will be presented regarding an extension to systems. As a supplement to this paper, the reader is directed to several excellent references that, in total, provide a comprehensive treatment of modern structural reliability theory: the basic books by Benjamin and Cornell (1970); Ang and Tang (1975); Hines and Montgomery (1990); and Montgomery and Runger (1994); the more advanced works of Thoft-Christenson and Baker (1982); Shinozuka (1983); Ang and Tang (1984); Madsen et al. (1986); and Melchers (1987); and, for specific applications to ship structures Mansour (1990).

Structural reliability has been a major theme of Ship Structure Committee (SSC) research for the past ten years. A list of SSC reliability projects is given in Table 1.

TABLE 1 SSC Reliability Projects

PROJECTS IN THE SSC RELIABILITY THRUST AREA	
SSC-351	Introduction to Structural Reliability Theory
SSC-368	Probability-Based Ship Design Procedures: A Demonstration
SSC-373	Probability-Based Ship Design: Load and Load Combinations
SR-1345	Probability-Based Ship Design: Implementation of Design Guidelines for Ships
SR-1362	Probability-Based Ship Design: Synthesis of the Reliability Thrust Area Activities
OTHER SCC RELIABILITY PROJECTS	
SSC-355	Relationship Between Inspection Findings and Fatigue Reliability
SSC-363	Uncertainties in Estimating Load and Load Effects on Marine Structures
SSC-375	Uncertainty in Strength Modes for Marine Structures
SR-1344	Assessment of the Reliability of Ship Structures
SSC-378	The Role of Human Error in the Design, Construction, and Reliability of Marine Structures

Note: SR projects are still in progress. SSC reports are available through the National Technical Information Service (NTIS), U.S. Department of Commerce, Springfield, Virginia 22151.

UNCERTAINTY IN FATIGUE/FRACTURE DESIGN FACTORS

Many sources of uncertainty exist in the fatigue analysis process:

- The fatigue phenomenon is unpredictable, as evidenced by enormous statistical scatter in laboratory data, with cycles-to-failure data of welded joints having coefficients of variation (COV = standard deviation/mean) typically ranging from 30 to 40 percent and sometimes as high as 150 percent, and fracture toughness data having a relatively high characteristic COV of 15 percent. Examples of the level of uncertainties in several design factors are given in Table 2. The probability density function of cycles to failure at a given stress level is shown in Figure 1 for the DEn C-curve data. This illustrates the broad scatter in fatigue data. Comprehensive summaries of statistics on fatigue in welded joints are provided by Gurney (1979) and Munse et al. (1983).

- Extrapolation of laboratory data to a ship structure environment may require many assumptions.
- The geometry of the component (e.g., defects and discontinuities in welded joints) complicates the prediction of initiation and propagation of fatigue cracks.
- Wave loading, which produces fatigue stresses in ship structures, may not be well defined. There is likely to be significant uncertainty associated with models of the environment.
- The dynamic loads (springing and slamming) may not be accurately known.
- The oscillatory stress causing fatigue at a welded joint, produced by a force on the system, contains uncertainty in the stress analysis procedures.
- Effects of temperature, corrosion, and the like, on fatigue strength are not well known.

Facing these uncertainties, engineers must make decisions regarding the integrity of components with respect to fatigue. Therefore, a probabilistic and statistical approach utilizing recent developments in probabilistic design theory seems particularly relevant.

QUANTIFYING UNCERTAINTY IN FATIGUE/ FRACTURE DESIGN FACTORS

Relative to fatigue and fracture failure modes, the design factors would include the maximum lifetime stress, Miner's stress, modeling error for both stress and strength, fracture toughness, crack length, the parameter in the S–N model, and the like. Uncertainty in a design factor can be quantified by treating the factor as a random variable and defining its probability distribution. There are two key functions: (1) the probability density function, $f_X(x)$; and (2) the cumulative distribution function, $F_x(x)$. The random variable is X, and the probability density function (pdf) is defined so that

$$P(a<X<b)=\int_a^b f_x(x)dx \tag{1}$$

TABLE 2 Examples of Uncertainties in Fatigue/Fracture Design Factors: Some Typical Values

Design Factor		Coefficient of Variation (COV)
Modulus of elasticity		0.03
Yield strength		0.07
Ultimate strength		0.05
Fracture toughness		0.15
Damage at failure (Miner's rule)		0.30
Fatigue stress (environmental and modeling error)		0.25
Cycles to fatigue failure:	DEn C-curve	0.50
	DEn E-curve	0.63
	API X-curve	1.36

Note: Coefficient of variation of random variable X. $C_x = \sigma_x/\mu_x$, where μ_x and σ_x are the mean and standard deviation of X, respectively.

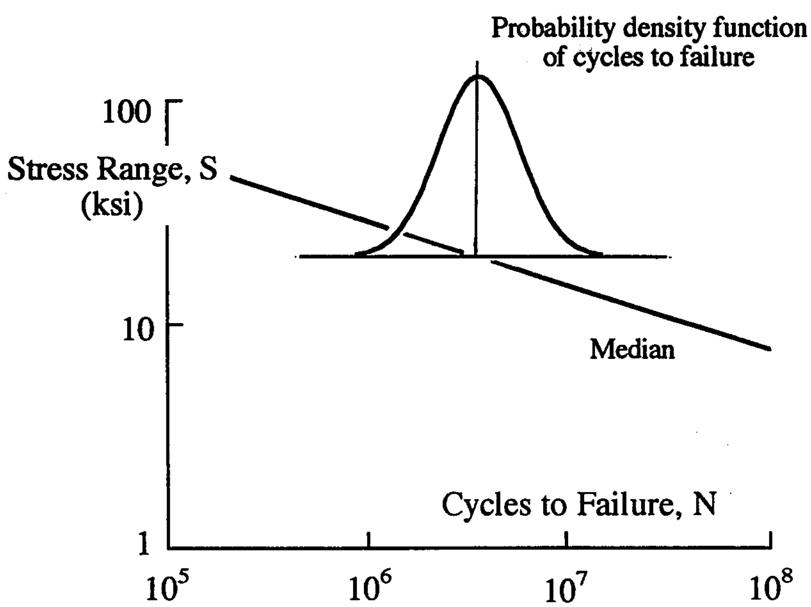

FIGURE 1 An illustration of the scatter in fatigue data based on U.K. Department of Energy E-curve.

where P(•) denotes "probability of." The pdf is demonstrated in Figure 2. The cumulative distribution function (cdf) is defined as

$$f_x(s) = \frac{dF_x(x)}{dx}, \; F_x(x) = \int_{-\infty}^{x} f_x(s)ds \tag{2}$$

The cdf is also demonstrated in Figure 2. The relationship between the pdf and cdf is

$$F_X(X) = P(X<x) \tag{3}$$

The pdf and/or cdf provide a complete probabilistic description of the design factor. Several analytical models for f_x and F_x are available. The four models that are commonly used in engineering are summarized in Table 3.

THE LIMIT STATE FUNCTION

Consider a single component or design detail. Let X denote the vector of design factors associated with a given failure mode. Assume that the distribution of each factor is defined by its pdf and cdf. The limit state function, g(X), relating to a specific failure mode, describes the relationship between the design variables for the event of failure

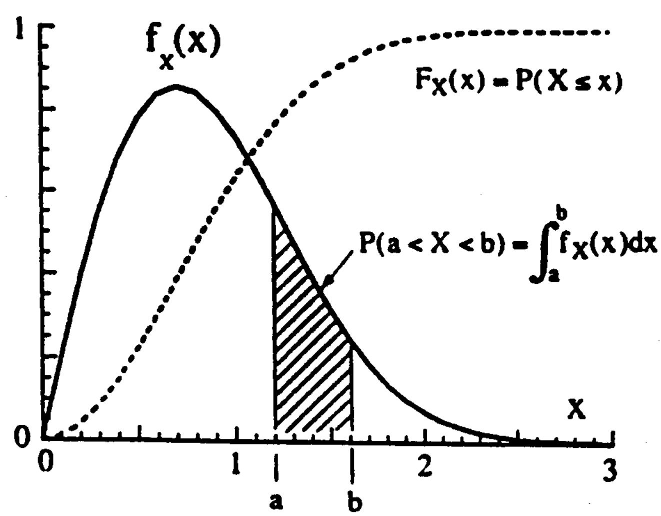

FIGURE 2 An illustration of the probability density function (pdf) and the cumulative distribution function (cdf).

Table 3 Commonly Used Models in Structural Reliability

		Mean (μ_X)	Standard Deviation (σ_X)
Normal	$f_X = \frac{1}{\sqrt{2\pi}\sigma} \exp\left[- \frac{(x - \mu)^2}{2\sigma^2}\right]$	μ	σ
Lognormal	$f_X(x) = \frac{1}{\sqrt{2\pi}\sigma_Y x} \exp\left[- \frac{(\ln x - \mu_Y)^2}{2\sigma_Y^2}\right]$	$\exp\left[\mu_Y + \frac{\sigma_Y^2}{2}\right]$	$\mu_X \sqrt{e^{\sigma_Y^2} - 1}$
Weibull	$F_X(x) = 1 - \exp\left[- \left(\frac{x}{\beta}\right)^{\alpha}\right]$	$\beta\Gamma\left(\frac{1}{\alpha} + 1\right)^{*}$	$\mu_X \alpha^{1.08}$
Extreme value distribution; Type 1 of Maxima (EVD)	$F_X(x) = \exp\{- \exp[- \alpha(x - \beta)]\}$	$\beta + \frac{0.577}{\alpha}$	$\frac{1.283}{\alpha}$

$^{*}\Gamma(\cdot)$ = gamma function.

$$\begin{aligned} g(X) &\leq 0 \ \rightarrow \textit{Failure} \\ g(X) &> 0 \ \rightarrow \textit{Safe} \end{aligned} \tag{4}$$

The function g(X) is also called the performance function, the response function, or the failure function.

The limit state is defined as

$$g(X) = 0 \tag{5}$$

This defines the boundary between the safe and failed regions in design parameter space.

Example: Let K_c denote the fracture toughness and $K = YS\sqrt{\pi a}$ denote the stress intensity factor for a brittle member under tensile stress, where S is stress, Y is the geometry factor, and a is the crack length. The event of failure is the event when $K > Kc$. The limit state function is

$$g = K_C - K = K_C - YS\sqrt{\pi a} \tag{6}$$

Example: Let N be the number of cycles to failure due to fatigue of a welded joint. The S–N fatigue curve is defined by $NS^m = A$, where m and A are the empirically determined fatigue strength exponent and coefficient, respectively. Let N_S denote the service (or design) life of the joint. The event of failure is $N < N_S$. The limit state function is

$$g = N - N_S = AS^{-m} - N_S \tag{7}$$

The probability of failure is defined as

$$P_f = P[g(X) \leq 0] \tag{8}$$

From probability theory,

$$P_f = \int_{\Omega} f_x(x)dx \tag{9}$$

where Ω is the region of failure and $f_x(x)$ is the joint probability density function of the random design factors. In general, this is a very difficult analytical (and numerical) operation, and simpler forms amenable to numerical analysis are indicated.

MODERN RELIABILITY METHODS

Modern structural reliability theory originated in a landmark paper by Alfred Freudenthal entitled "The Safety of Structures," which appeared in the 1947 Transactions of the American Society of Chemical Engineers (Freudenthal, 1947). But, serious efforts to apply probability and statistics to structural design did not really begin until the publication of another key paper by Freudenthal et al. (1966). This paper laid the mathematical foundation for modern probabilistic design theory, and a number of numerical algorithms have been developed over the last three decades. A summary is provided in Table 4.

MVFOSM Analysis

The first serious attempt to apply probabilistic methods to the development of a design code was made by C.A. Cornell in 1969, when he proposed the MVFOSM concept (Cornell, 1969). Limit state functions that are "complicated" can be represented by the first terms of a Taylor's series expansion about the mean. The mean, μ_g, and standard deviation, σ_g, of the limit state function can be approximated, and the safety index is defined as μ_g / σ_g

$$\mu_g = g(\mu),\ \sigma_g = \sqrt{\sum_{i=1}^{n} \left(\frac{\partial g}{\partial X_i} \right)_{\mu}^{2} \sigma_i^2} \tag{10}$$

Although the concepts were employed to derive probability-based design requirements for the code of the American Concrete Institute, it was discovered that reliability estimates depended upon the mechanical formulation of the limit state function. This "mathematical" difficulty was later overcome by the Hasofer-Lind generalized safety index. MVFOSM gives reasonably accurate results if the limit state is linear, and it continues to be useful in providing "quick and dirty" estimates of the safety index for components.

The Generalized Safety Index

The lack of an invariance problem associated with MVFOSM analysis was solved in the landmark paper by Hasofer and Lind (1974). The scheme is to transform all of the basic variables to reduced variables having zero mean and standard deviation of unity. The safety index is then defined as the minimum distance from the origin of the reduced coordinates to the limit state function in reduced coordinates. This measure of reliability was proved to be independent of the mechanical formulation of the subsequent development of structural reliability estimates for components. All of the other methods of fast probability integration described in the following section are essentially refinements of the Hasofer-Lind safety index concept.

TABLE 4 A Summary of Structural Reliability Methods (for Probability of Failure Estimates of Components and Systems)

ANALYTICAL METHODS

1. Direct evaluation of the probability-of-failure integral
2. Normal and lognormal formats
3. Mean value first-order second-moment (MVFOSM) (Cornell, 1969)
4. Hasofer-Lind generalized safety index (Hasofer and Lind, 1974)
5. First-order reliability methods (FORM)
    a. Limit states represented by tangent hyperplanes at design points in transformed standard normal space (Madsen et al., 1986)
    b. Rackwitz-Fiessler algorithm (Rackwitz and Fiessler, 1978)
6. Second-order reliability methods (SORM)
    a. Limit states represented by hyperparaboloids at design points in transformed standard normal space (Madsen et al., 1986)
    b. Wu/FPI algorithm (Wu and Wirsching, 1987)
7. Advanced mean value (AMV) method (Wu et al., 1990)

MONTE CARLO SIMULATION

1. Direct Monte Carlo
2. Importance sampling (Shinozuka, 1983)
3. Domain-restricted sampling (Harbitz, 1986)
4. Adaptive sampling (Bucher, 1988)
5. Directional sampling (Bjerager, 1990)

FORM

The Hasofer-Lind analysis requires that only the mean and standard deviation of each variable be considered and, therefore, ignores distributional information, even if it is available. A method proposed by Paloheimo and Hannus (1974) suggests that nonnormal distributions be transformed into standard normals by requiring that the distribution functions of the basic variable and the standard normal variate be equal. The transformation of the ith variable having cdf F_i is

$$F_i(x_i) = \Phi(u_i) \tag{11}$$

where Φ is the standard normal cumulative distribution function. An example of a limit state function involving two random design factors is provided in Figure 3. The limit state g′ is shown in terms of the transformed variables u_i, as are the contours of the joint probability density function. The safety index, β, is defined as the minimum distance from the origin of the transformed coordinates to the limit state. The minimum distance point is called the most probable point (MPP) because it is the point on g′ having the largest value of $f_{u1\ u2}$. The probability of failure is

$$p_f = \Phi(-\beta) \tag{12}$$

This probability is exact in the case where g′ is linear.

The Rackwitz-Fiessler algorithm was proposed as an efficient computational method for FORM (Rackwitz and Fiessler, 1978). An additional refinement to this procedure, the Chen-Lind algorithm has been proposed (Chen and Lind, 1983). FORM can produce significant errors in probability-of-failure estimates when g′ is nonlinear. It is difficult to predict *a priori* the expected errors in probability estimates. This problem had led to the development of SORM.

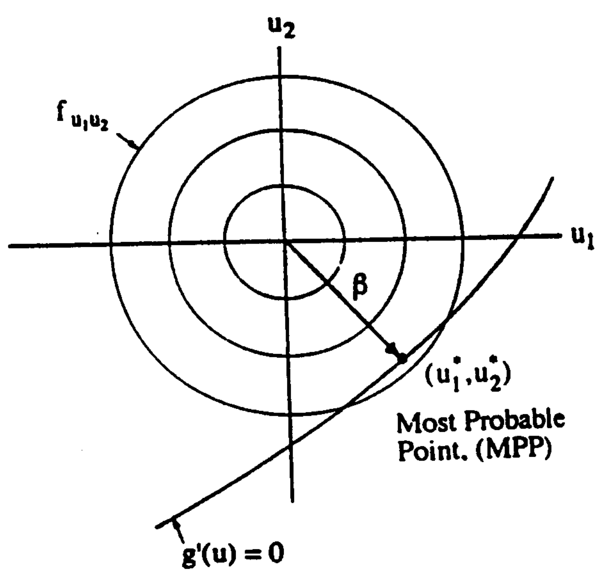

FIGURE 3 An illustration of FORM analysis.

SORM

It was found that FORM produces errors whose magnitudes are difficult to predict in advance. This observation led to the development of SORM methods. A number of SORM algorithms have been proposed (Ditlevsen, 1979; Fiessler et al., 1979; Tvedt, 1983; Breitung, 1984). These methods rely on FORM for the transformation to standard normal space, and then model the curvature of g′. Wu and Wirsching (1987) proposed a method (called the Wu/FPI algorithm) that, it is argued, is more robust and accurate because it avoids some of the mathematical pitfalls associated with transformation to standard normal space. It has been demonstrated that Wu/FPI can consistently produce point probability estimates within 5 percent of the exact value.

A model using SORM to perform reliability analysis for fatigue failure, formulated as a time-dependent, first-passage problem, has been discussed by Marley (1991) and Marley and Moan (1992). The process combines the fatigue and fracture failure modes. Fatigue failure occurs when the instantaneous stress exceeds the ultimate strength of the fatigue-weakened component.

AMV Method

A practical limitation on FORM and SORM, as described above, is that the limit state function must have an explicit closed form. The g-function, $g = R - S$, where R is strength and S is stress, is considered to be an explicit function. But, in many cases, the relationship between the variables is defined by a numerical algorithm (e.g., finite element analysis), which often requires significant computer time for a function evaluation. The problem is that FORM/SORM may require 500 to 1,000 function evaluations. An algorithm, the AMV method, has been developed that performs accurate reliability analysis with a minimum of function evaluations (Wu et al., 1990; Wu, 1994).

The AMV method works as follows. Let X be a vector of design variables, all with known distributions. The response variable, $Z = Z(X)$, is approximated as a linear function from a Taylor's series expansion about the mean. Coefficients of the X's are the partial derivatives of Z, which are approximated by evaluating Z using deviations from the mean. Discrete points on the distribution function are computed using a SORM routine. Because the cdf is smooth, a good curve can usually be constructed with six to eight points. This solution is called the mean value (MV) solution and, in general, it must be assumed to have significant error resulting from the linear approximation. To improve the quality of the solution, the MPPs at each point where the cdf is evaluated are substituted into Z to obtain an improved value of Z corresponding to the probability level. From extensive study, it has been found that this solution generally provides an accurate estimate of the cdf (see, for example, Wirsching et al., 1991). The number of function evaluations is minimal. For a cdf approximated by eight points, the total number of function evaluations will be $(n + 1) + 8$.

The AMV method is the "heart" of a probabilistic finite element code (NESSUS) that was developed for the National Aeronautics and Space Administration by the Southwest Research Institute to solve complicated design problems associated with space propulsion systems. The AMV method is considered to be a significant contribution to computational reliability analysis.

Efficient Monte Carlo Simulation

As computer capabilities have increased and computer costs have decreased, the use of Monte Carlo simulations for structural reliability analysis has gained new respectability. It has also helped that efficient methods, principally importance sampling, have been developed. The importance sampling concept has been discussed by Shinozuka (1983). Variations on the basic importance sampling concept have been proposed. These include domain-restricted sampling (Harbitz, 1986), directional sampling (Bjerager, 1990), and adaptive sampling (Bucher, 1988). In summary, these methods can produce probability-of-failure estimates having narrow confidence intervals for small sample sizes. The bad news is that (1) all require an estimate of the probability of failure, and (2) their efficiency sharply decreases as the number of variables increases.

Commercial codes are available that automate the reliability methods described above. Four of the better known codes are listed in Table 5. All of the codes will perform probability-of-failure calculations for a component using FORM, SORM, and other methods (e.g., FPI has convolution integral option using the fast Fourier transform). They all have system reliability capability. They all can perform direct Monte Carlo simulations and can do importance sampling by directional or adaptive simulation.

The Lognormal Format

An important special case, where a closed-form expression is available for computing the probability of failure, is the lognormal format. This approach has been found to be extremely useful in fatigue and fracture reliability analysis (Ang and Tang, 1975; Wirsching, 1984; Hines and Montgomery, 1990).

Consider a redefinition of g. Define the failure condition as g ≤ 1. For example,

$$g = \frac{K_c}{YS\sqrt{\pi a}}\ , \ g = \frac{AS^{-m}}{N_S} \tag{13}$$

Assume that g(X) is a multiplicative function of the design factors, X_i,

$$g(X) = B\prod_{i=1}^{K} X_i^{a_i} \tag{14}$$

where B and a_i are constants. Let Z = ln g.

TABLE 5 Reliability Analysis Software

Program	Description	Contact
CALREL	A general-purpose structural reliability analysis program.	Ken Wong Department of Civil Engineering University of California Berkeley, CA 94720
PROBAN	Developed and marked by Det norske Veritas, PROBAN is a general structural reliability analysis code.	Veritas Sesam Systems 1325 South Dairy Ashford - Suite 100 Houston, TX 77077
FPI	FPI was developed at Southwest Research Institute under contract to NASA/Lewis to produce a probabilistic structural analysis code having both nonlinear structural behavior and dynamic response capabilities.[a]	Y. T. Wu Southwest Research Institute P. O. Drawer 28510 San Antonio, TX 78228-0510
STRUREL	A computer package for probabilistic rèliability analysis; principal author is Prof. R. Rackwitz at the University of Munich; part of the program is available for WINDOWS.	RCP GmbH Attn: Dr.-Ing. S. Goilwitzer Barer Strasse 48/111 D-800 Muchen 40, Germany

[a]Includes AMV.

$$Z = \ln B + \sum_{i=1}^{K} a_i \ln X_i \tag{15}$$

Assume that all X_i have lognormal distributions. Because X_i is lognormal, it follows that all ln X_i are normal. The failure condition is ($g \leq 1$) or ($Z \leq 0$). It can be shown that $p_f = \Phi(-\beta)$ where $\beta = \mu_Z/\sigma_Z$ and

$$\mu_Z = \tilde{Z} = \ln\left[B\prod_{i=1}^{K}\tilde{X}_i^{a_i}\right] \tag{16}$$

$$Z = \ln B + \sum_{i=1}^{K} a_i \ln X_i \tag{17}$$

$$\sigma_Z^2 = \ln\left[\prod_{i=1}^{K}\left(1 + C_i^2\right)^{a_i^2}\right] \tag{18}$$

where the tilde indicates median values and C_i is the COV of X_i. In terms of the mean and standard deviation of X,

$$\tilde{X} = \frac{\mu_x}{\sqrt{1 + C_x^2}} \quad , \quad C_x = \frac{\sigma_x}{\mu_x} \tag{19}$$

Example 1: A Comparison of Reliability Methods

Consider a structural component subjected to tensile loading. The maximum lifetime stress, S, and the fracture strength R, are random variables as defined in Table 6. The event of failure is (S > R), and the probability of failure is $p_f = P(S > R)$. Determine p_f.

A summary of the results of the reliability analyses is given in Table 6. Clearly, the MVFOSM value of β does not provide an accurate estimate of risk. Note that an exact solution is possible in this special case of two random variables. Adaptive importance sampling used in FPI is a very efficient simulation scheme. To obtain the results given in the table, 52 simulations and calculations of the limit state function were performed. To obtain the same confidence interval for p_f (see Note C, Table 6) by direct Monte Carlo would require a total of 280,000 simulations.

TABLE 6 Example 1: Summary

Variable	Distribution	Mean	Standard Deviation
R	Weibull	60	5
S	EVD	35	4
Method		**Safety Index, β**	**Probability of Failure, p_{ff}**
MVFOSM		3.900	--
FORM [a]		3.111	0.932×10^{-3}
SORM [a]		2.989	1.40×10^{-3}
Exact [b]		2.9931	1.38×10^{-3}
Adaptive importance sampling [b]		2.977	1.45×10^{-3} [c]

Notes:

[a]FPI (Southwest Research) solution.
[b]Solution by numerical integration.
[c]95 percent confidence intervals on p_f, are ± 10 percent.

Example 2: The Lognormal Format

Consider the butt-welded joint shown in Figure 4. A random load, Q(™), is applied. The Miner's stress has a median of $\tilde{Q}$ = 4.0 k and a coefficient of variation, C_Q = 0.25, reflecting uncertainties in the environment, as well as stress modeling error. A joint must be designed (find the minimum thickness, t) so that the probability of failure is less than p_0=10^{-4} in a service life of two million cycles. The fatigue strength of the butt weld is given as NS^m=A. The scatter in fatigue data is accounted for by modeling the fatigue strength coefficient, A, as a lognormally distributed random variable. From available data (U.K. Department of Energy C-curve; Gurriey 1979), $\tilde{A}$ = 1.25E11 (ksi units based on range, CA = 0.50, and m = 3.5).

Solution: The event of failure is $N < N_0$.

Following the rules for the lognormal format, the limit state function is

$$g(U) = \frac{N}{N_0} = \frac{AS^{-m}}{N_0} \tag{20}$$

Note that A and S are the two random variables. The safety index is (tildes indicate mean values and Cs are COVs)

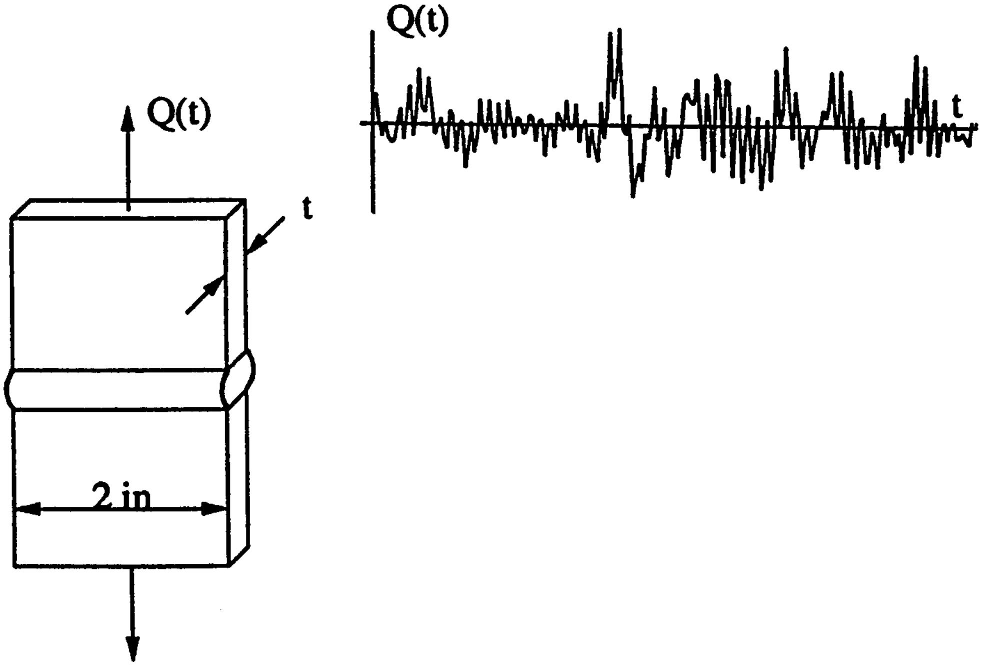

FIGURE 4 Example 2: Welded joint subjected to random loading.

$$\beta = \frac{\ln[\tilde{A}\tilde{S}^{-m}/N_0]}{\sigma_{\ln g}} \tag{21}$$

where

$$\sigma_{\ln g} = \sqrt{\ln[(1 + C_A^2)(1 + C_S^2)^{m^2}]} \tag{22}$$

Note that $C_S = C_Q$. Making the substitutions, we get an expression for the safety index as a function of S:

$$\beta = \frac{\ln[1.25E11(\tilde{S})^{-3.5}/2E6]}{0.9827}$$

Now the target safety index is

$$\beta_0 = -\Phi^{-1}(p_0) = -\Phi^{-1}(10^{-4}) = 3.72$$

Letting $\beta = \beta_0 = 3.72$, it follows that

$$\tilde{S} = 8.25 \text{ ksi}$$

But, $\tilde{S} = \tilde{Q}/2t$ and, therefore,

$$t > 0.24 \text{ in}$$

for a safe design.

Example 3: Fatigue Reliability Analysis with Implicit Performance Functions Using the Advanced Mean Value Method

For the fracture mechanics fatigue model, a preexisting crack experiences subcritical crack growth under the action of an oscillatory and generally variable amplitude stress process. A model, for which the distribution of stress ranges is Weibull and the crack growth law is the Paris equation, is summarized in Figure 5. Cycles to failure, N (given all relevant parameters), are obtained by a numerical integration of the crack growth law. The equation for cycles to failure, an implicit function that requires numerical integration, is

$$N = \frac{1}{CB^m S_e^m} \int_{a_0}^{a} \frac{dx}{G(x) Y^m(x) (\pi x)^{m/2}} \tag{23}$$

where

$$G(a) = \frac{\bar{S}_0^m(a)}{S_e^m} \tag{23a}$$

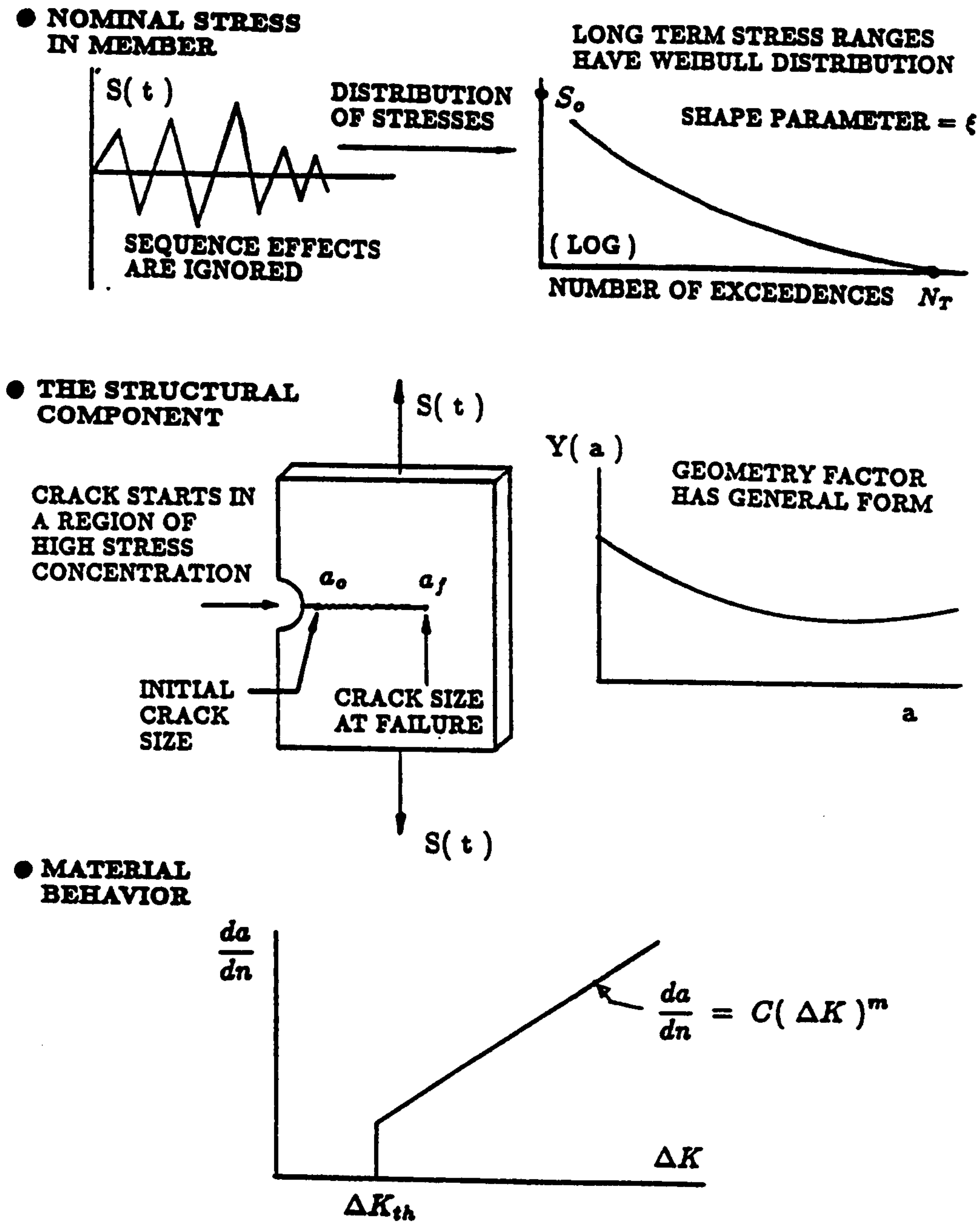

FIGURE 5 Example 3: Fracture mechanics fatigue model.

$$\bar{S}_0^m = \int_{S_0(a)}^{\infty} s^m f_s(s)ds \tag{23b}$$

$$S_0(a) = \frac{\Delta K_{th}}{Y(a)\sqrt{\pi a}} \tag{23c}$$

where C and m are the Paris coefficient and exponent, respectively; S_e is Miner's stress; B models stress uncertainty; f_s is the pdf of the stress range; ΔK_{th} is the threshold stress intensity, Y is the geometry factor; and a is crack length. Details of this model are provided in Wirsching et al., 1991. The problem is to construct the cdf of N given the random design factors of Table 7.

TABLE 7 Data for Example 3, Illustration of AMV

Variable	Note	Distribution	Parameters [a,b]
ξ	c	Constant	1.0
S_0	c	EVD	(276, 27.6) MPa
N_s	c	Constant	1×10^8
ΔK_{th}		Normal	(5.5, 0.55) MPa$\sqrt{\text{m}}$
m	e	Constant	3
C	d	Weibull	$(6.9 \times 10^{-12}, 9.5 \times 10^{-13})$MPa
a_0		Lognormal	(0.25, 0.02) cm
a_f		Constant	2.54 cm
K_t	f	Constant	3.3
ν	f	Constant	11.2 cm
δ	f	Constant	1.13 cm

Notes:

[a]All values are in SI system.

[b]For random variables, numbers in parentheses are (μ,σ).

[c]Model for long-term Weibull distribution of stresses; parameters are ξ = Weibull shape parameter, S_0 = stress that occurs, on average, once every N_s cycles; N_s = service life:

$$E(S^m) = S_0^m [\ln N_s]^{\frac{-m}{\xi}} \Gamma\left[\frac{m}{\xi} + 1\right]$$

$\Gamma(\bullet)$ = Gamma function

[d] C = Paris coefficient
[e] m = Paris exponent
[f] Geometry factor, $Y(a) = (K_t - 1)e^{-\gamma a^{\delta}}$, where K_t = stress concentration factor.

The efficient AMV method produces the solution shown in Figure 6. The AMV solution not only matches the Monte Carlo one very well but, as shown by other studies, produces accurate cdf estimates in the low-probability regions. For the Monte Carlo solution, 90 percent confidence intervals at the two extreme points of the cdf are approximately ±0.20 on Z, the standard normal variate.

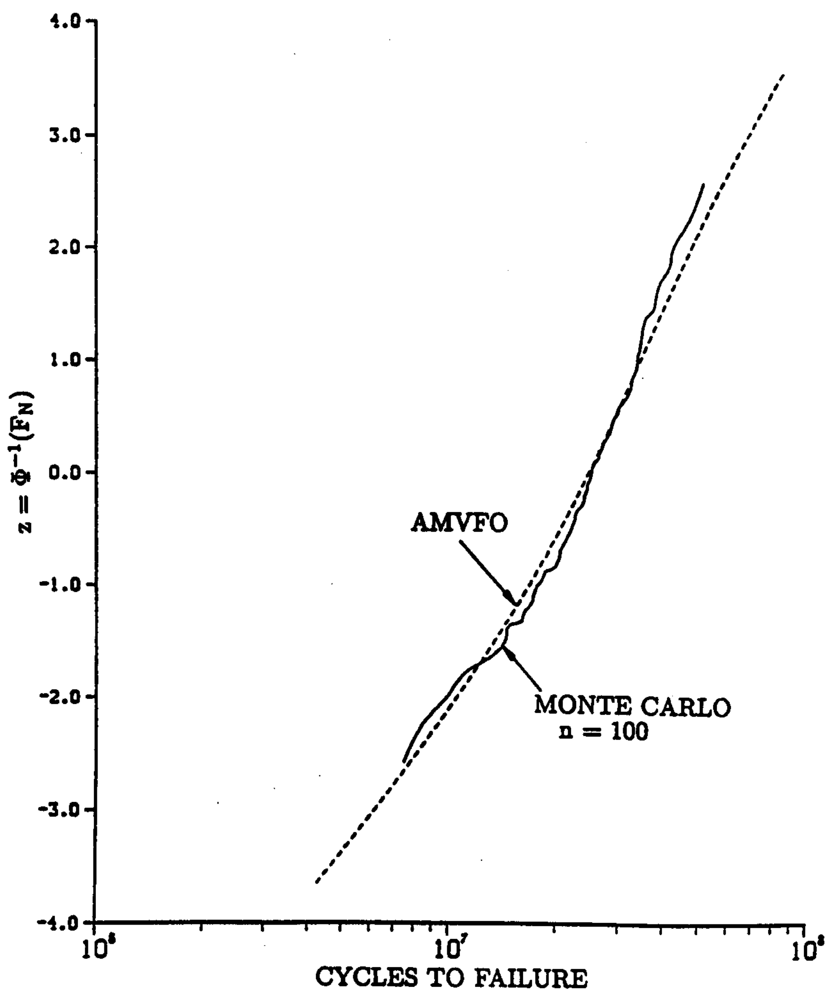

FIGURE 6 Example 3: Distribution function of cycles to failure—the AMV solution.

Acceptable Risk Levels

Reliability methods cannot be employed without first answering the question, "What is an acceptable level of risk?" This issue is always open for debate. Some examples are presented here for reference only.

One committee charged with the responsibility of developing design criteria for fixed offshore platforms has proposed the following: (1) a maximum allowable probability of failure of 10^{-5}/year for failures resulting in a great loss of life or a high potential for environmental damage and (2) a maximum allowable probability of failure of 10^{-3}/year for failures that would result in a small risk to life and have a low potential for environmental damage.

A.S. Veritas Research (1991) published the figures given in Table 8. Again, the intent was to apply these target reliabilities to offshore structures.

TABLE 8 Specified Target (Annual) Failure Probabilities

	Failure Type[a] (safety index in parentheses)		
Failure Consequences[b]	1	2	3
Not serious	10^{-3}(3.09)	10^{-4}(3.71)	10^{-5}(4.26)
Serious	10^{-4}(3.71)	10^{-5}(4.26)	10^{-6}(4.75)
Very serious	10^{-5}(4.26)	10^{-6}(4.75)	10^{-7}(5.20)

NOTE:

[a]Failure Type 1 = Ductile failure with reserve strength capacity resulting from strain hardening; 2 = Ductile failure with no reserve capacity; 3 = brittle fracture and instability.

[b]Not serious = A failure implying small possibility for personal injuries; the possibility for pollution is small, and the economic consequences are considered to be small. Serious = A failure implying possibilities for personal injuries/fatalities or pollution or significant economic consequences. Very serious = A failure implying large possibilities for several personal injuries/fatalities or significant pollution or very large economic consequences.

SOURCE: A.S. Veritas Research (1991).

Mansour (1990) published the results of a survey of the reliability relative to hull collapse (See Figure 7). Past successful practice can be used as a guideline for design specifications. For reference, $\beta = 4$ in Figure 7 implies a $p_f = 3.2 \times 10^{-5}$, and $\beta = 6$ implies $p_f = 1.0 \times 10^{-9}$. The latter figure is recommended by the Federal Aviation Administration (1988) for catastrophic loss of aircraft.

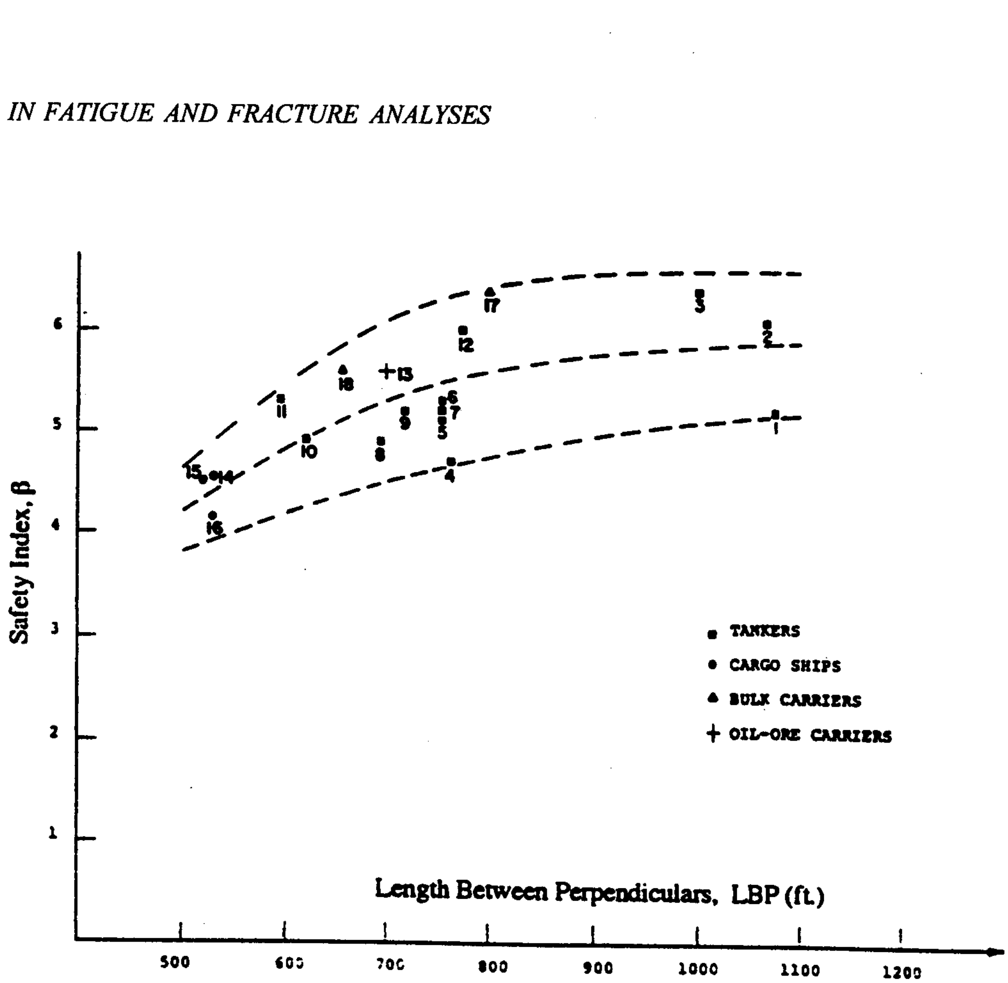

FIGURE 7 Safety index relative to hull collapse for 18 ships.

Source: Mansour, 1990.

Probability-Based Fatigue Design Criteria

Reliability methods can be employed to establish design criteria. The most popular approach is to use reliability technology to derive safety factors for safety check expressions. This approach was employed in a load and resistance factor design (LRFD) (Ravindra and Galambos, 1978). Melchers (1987) provides a summary of the codes used to implement a partial safety factor format. He also presents the mathematics of deriving partial safety factors. Mansour (1993) demonstrates derivation of reliability-based design criteria for ships. Munse et al. (1983), using the Weibull format, and Ayyub and White (1987); White and Ayyub (1987); and Jiao and Moan (1992), using advanced methods, develop reliability-based design criteria relative to fatigue and fracture.

$$E(S^m) = S_0^m[\ln N_s]^{-\frac{m}{\xi}}\Gamma[\frac{m}{\xi} + 1]$$

The lognormal format can be used very conveniently to derive a safety check expression for fatigue, as demonstrated by (Wirsching and Chen, 1988). Employing Miner's rule, fatigue damage at a detail, under random, environmentally induced stresses, is at the end of the service life:

$$D = \frac{N_S S_e^m}{A_0} \tag{24}$$

where S_e is Miner's stress, N_s is the service live, m is the fatigue strength exponent, and A_0 is the fatigue strength coefficient of the design curve.

The safety check expression is

$$D_0 \le \Delta_0 \tag{25}$$

where the target damage level is

$$\Delta_0 = \frac{\lambda\tilde{\Delta}}{\tilde{B}^m \exp(\beta_0\ \sigma_{\ln g})} \tag{26}$$

where λ is the scatter factor that defines the design curve relative to the median S–N curve. The following form of X implies that the design curve lies to the left (safe side) of the median curve by two standard deviations on a log basis. This is standard practice in the marine industry:

$$\lambda = \exp[\{\ln(1 + C_A^2)^4\}^{\frac{1}{2}}] \tag{27}$$

B denotes stress uncertainty and includes both modeling error and uncertainties in the environmental loading, and the median, denoted as B, indicates bias in the stress analysis; Δ is the uncertainty in the performance of Miner's rule; the median, denoted as $\tilde{\Delta}$, quantifies bias in Miner's rule. And,

$$\sigma_{\ln g} = \sqrt{\ln\left[(1 + C_A^2)(1 + C_\Delta^2)(1 + C_\beta^2)^{m^2}\right]} \tag{28}$$

The Cs denote COVs. Finally, β_0 is the target safety index.

Example 4: Target Damage Level Using the Lognormal Format

Statistics that are typical of a welded joint are given in Table 9. Note that $C_A = C_N$ implies $\lambda = 2.57$. This means that the design curve lies to the left of the median by a factor of 2.57. Using Equation (26), the target damage level can be derived as a function of the target safety index. The relationship is plotted in Figure 8.

TABLE 9 Statistics for Fatigue Design Variables

Variable	Value
Fatigue strength exponent, m	3.00
COV of fatigue strength coefficient A, C_A	0.50
Damage at failure, Δ	
Median, Δ	1.00
COV, C_Δ	0.30
Stress uncertainty, B	
Median, B	1.00
COV, C_B	0.25

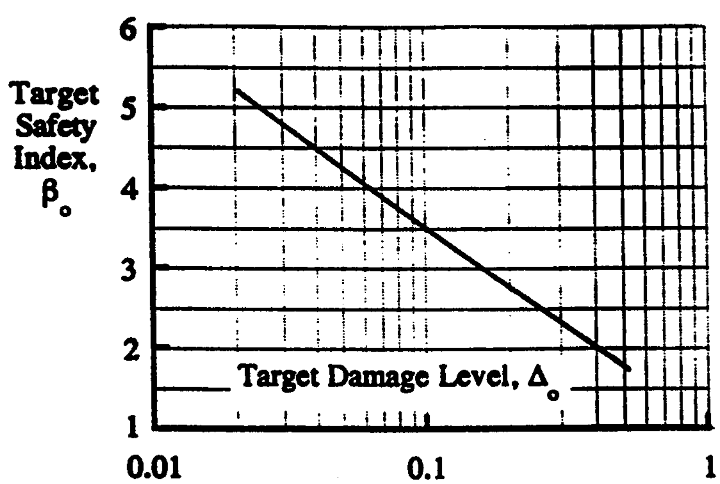

FIGURE 8 Example 4: Target damage level as a function of the target safety index.

A Note on System Reliability

It is generally extremely difficult to perform a reliability analysis or to develop reliability-based design criteria for a structural system of several components. An overview of the state of the art can be obtained from Ang and Tang (1984); Madsen et al. (1986); Thoft-Christensen and Murotsu (1986); Melchers (1987); and Karamchandani (1990). The fundamental mathematical problem is to define the failure modes in a highly redundant system.

The computer codes cited in Table 5 all have system capability. They are programmed to input the system in a fault-tree format. In general, this is not amenable to analysis of redundant systems.

In some systems, it can be assumed that the failure modes are independent, and an estimate of the probability of system failure is then

$$P_S = 1 - \prod_{i=1}^{k} (1 - p_i) \tag{29}$$

where p_i represents the probability of failure of the ith component and k is the number of components.

FATIGUE/FRACTURE RELIABILITY AND MAINTAINABILITY ANALYSES

In the structural system of a ship, fatigue cracks typically grow from welded joints. A fatigue-weakened structure is vulnerable to fracture under extreme loads, but system reliability can be improved by a maintenance program of periodic inspection and repair. However, maintenance is expensive and, ultimately, the challenge to designers is to specify a design, inspection, and repair strategy that will minimize total expected life cycle costs.

The maintenance problem in marine structures has been studied by several investigators (Mardindale and Wirsching, 1983; Stahl and Geyer, 1984; Madsen et al., 1987; Bannon and Harding, 1989; Torng, 1989; Jiao et al., 1990; Ximenes and Mansour, 1991; Torng and Wirsching, 1991; Banon et al., 1991; Kung and Wirsching, 1993). The physical problem is described in Figure 9. In general, the system can consist of both parallel and series structures. A random stress (produced principally by wave loading) can generate a fatigue crack and, under operational stresses, the crack can grow and eventually fail. Or, if the structure experiences an extreme load, a member can fracture. Failure of a member does not necessarily mean collapse for a redundant structure, as the load is shed to the other intact members.

The fatigue/fracture reliability and maintainability process is extremely complicated, and an analytical solution is not viable. Generally, simulation is required. Typical results of an analysis would be as shown in Figure 10. A simulation program developed by Kung and Wirsching (1993) has been used in a number of applications.

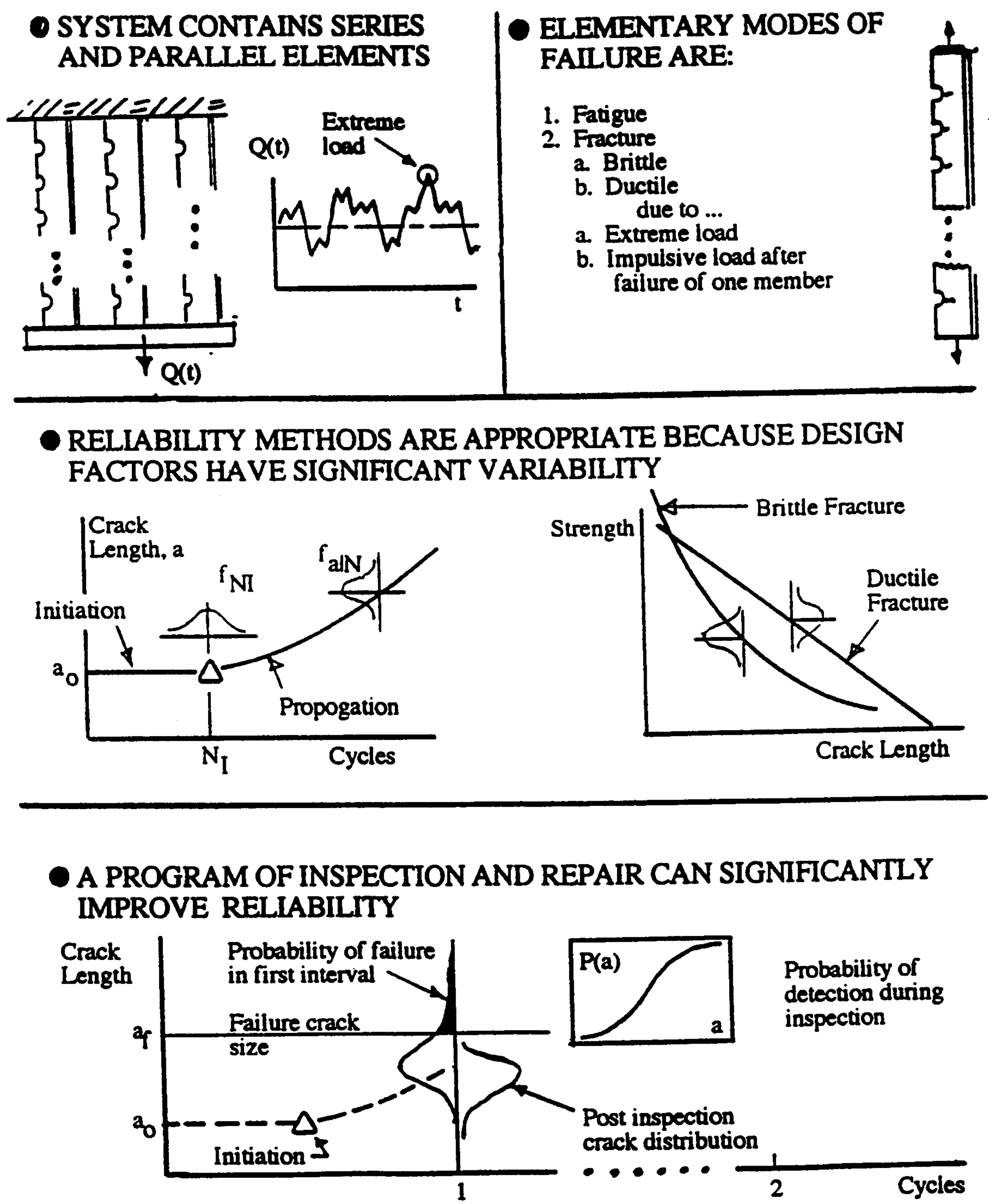

FIGURE 9 A description of the fatigue and fracture reliability and maintainability process.

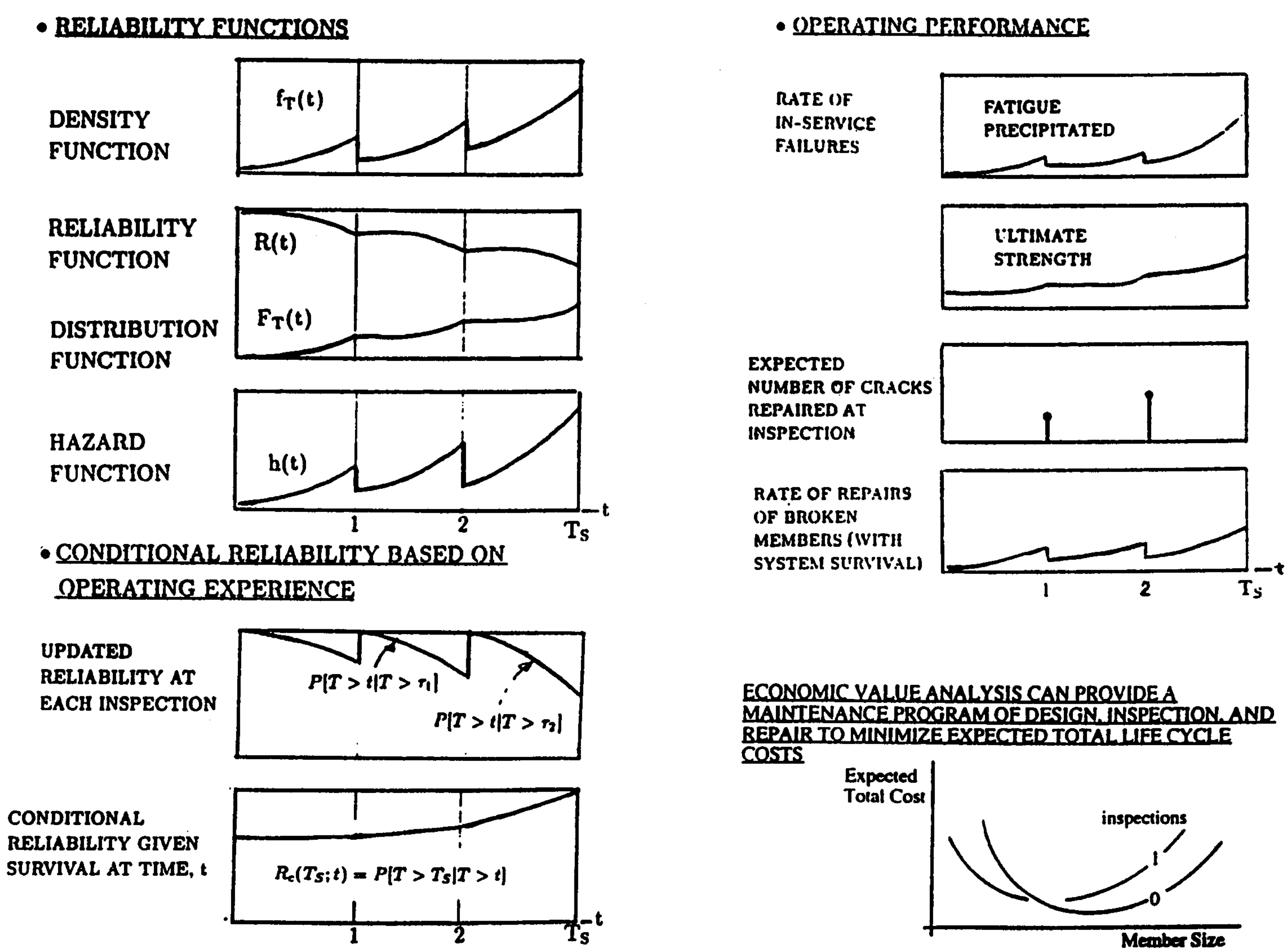

FIGURE 10 Results of reliability analysis of the fatigue and fracture reliability and maintainability process.

SUMMARY: BENEFITS AND LIMITATIONS OF STRUCTURAL RELIABILITY METHODS

There are two basic functions for structural reliability analysis. The first is to perform a reliability assessment of an existing design. This design may be an operational structure, or it may be on the drawing board. The second function is to use reliability methods to derive safety-check expressions for design criteria documents. Other important functions are to perform failure analysis, to compare alternative designs, to develop a strategy for design and maintenance of aging structures, and, in general, to manage uncertainty in structural design. This paper has provided a summary of reliability methods that could be employed for any of these functions, with a focus on the fatigue failure model.

The comments that follow are generic to reliability analysis and are not necessarily restricted to the fatigue and fracture failure modes. General benefits to be gained from employing reliability methods are:

- Statistical data are used explicitly in engineering analysis and design.
- Reliability (or probability of failure) is a more meaningful index of structural performance that is the factor of safety.
- Reliability information is useful in making design decisions, as well as decisions to repair or redesign after a failure.
- Rational comparisons can be made between competing designs on the basis of reliability.
- An optimal design of a system results when each component is designed to the same reliability level, assuming that the consequences of failure are the same.

Consider using reliability technology to develop probability-based design criteria. Establishing a uniform level of risk throughout the provisions of a design code and, therefore, throughout a structure, promises to provide improved structural design. Specific advantages are as follows:

- A more efficiently balanced design results in weight savings and/or an improvement in reliability. Experiences with the probability-based AISC/LRFD indicate weight savings in structural components from 10 percent to 30 percent.
- Because of an improved perspective of the overall design process, development of probability-based design procedures can stimulate important advances in structural engineering.
- The codes become a living document. They can easily be revised periodically to include any new sources of information and to reflect additional statistical data on design factors.
- The partial safety factor provides a framework for extrapolating existing design practice to new ships, where experience is limited.

The limitations of reliability analysis are as follows:

- Insufficient data exist for almost all of the design factors, such as fatigue and fracture strength, environmental loads, modeling error in transforming the environmental loads to design stresses, errors in the strength models (e.g., Miner's rule), and so on.
- Because of some lack of confidence in the data, risk estimates must be considered only as a "best estimate." Confidence intervals on actual risk must be considered to be "broad."
- Reliability analysis is relatively complicated, for example, it is easier to compute the margin of safety than it is the probability of failure of a component.
- Structural system analysis, particularly for redundant structures, is extremely complicated.

REFERENCES

Ang, A.H-S., and W. Tang. 1975. Probability Concepts in Engineering Planning and Design, Vol. I. New York: John Wiley and Sons.

Ang, A.H-S., and W. Tang. 1984. Probability Concepts in Engineering Planning and Design. Vol. II. New York: John Wiley and Sons.

A.S. Veritas Research. 1991. Guidelines for the Use of Reliability Methods. Report No. 91-2000. Hovik, Norway: A.S. Veritas Research.

Ayyub, B.M., and G.J. White. 1987. Reliability-conditioned partial safety factors. ASCE Journal of Structural Engineering 113:279–294.

Banon, H., C.A. Cornell, and S.J. Harding. 1991. Probabilistic combination of forces in tension leg platform tethers. ASCE Journal of Structural Engineering 117:1532–1548.

Banon, H., and S.J. Harding. 1989. A methodology for assessing the reliability of tension leg platform tethers. ASCE Journal of Structural Engineering 115(9):2243–2260.

Benjamin, J.R., and C.A. Cornell. 1970. Probability, Statistics, and Decision for Civil Engineers. New York: McGraw-Hill.

Bjerager, P. 1990. On computation methods for structural reliability analysis. Structural Safety 9:79–96.

Breitung, K. 1984. Asymptotic approximations for multinormal integrals. ASCE Journal of the Engineering Mechanics Division 110:357–366.

Bucher, C.G. 1988. Adaptive sampling—An iterative fast Monte Carlo procedure. Structural Safety 5:119–126.

Chen, X., and N.C. Lind. 1983. Fast probability integration by three parameter normal tail approximation. Structural Safety 1:269–276.

Cornell, C.A. 1969. A probability-based structural code. Journal of the American Concrete Institute 66:974–985.

Ditlevsen, O. 1979. Generalized second moment reliability analyses. Journal of Structural Mechanics 7:435–451.

Federal Aviation Administration (FAA). 1988. System Design and Analysis. FAA Advisory Circular (AC) No. 25.1309-IA. Washington, D.C.: FAA.

Fiessler, B., H.J. Neumann, and R. Rackwitz. 1979. Quadratic limit states in structural reliability. ASCE, Journal of the Engineering Mechanics Division 105:661–676.

Freudenthal, A.M. 1947. Safety of structures. Transactions of the ASCE 112:125–180.

Freudenthal, A.M., J.M. Garrelts, and M. Shinozuka. 1966. The analysis of structural safety. ASCE Journal of the Structural Division 92:267–325.

Gurney, T.R. 1979. Fatigue of Welded Structures, 2nd Edition. New York: Cambridge University Press.

Harbitz, A. 1986. An efficient sampling method for probability of failure calculations. Structural Safety 3:109–115.

Hasofer, A.M., and N.C. Lind. 1974. Exact and invariant second-moment code format. ASCE Journal of the Engineering Mechanics Division 100:111–121.

Hines, W.W., and D.C. Montgomery. 1990. Probability and Statistics in Engineering and Management Science, 3rd Edition. New York: John Wiley and Sons.

Jiao, G., and T. Moan. 1992. Reliability-based fatigue and fracture criteria for welded offshore structures. Engineering Fracture Mechanics 41:271–282.

Jiao, G., T. Moan, and M.J. Marley. 1990. Reliability analysis of TLP tether system. Pp. 65–71 in Proceedings of the 10th International Conference on Offshore Mechanics and Arctic Engineering held May 7–10, 1990 in Houston, Texas. New York: American Society of Mechanical Engineers.

Karamchandani, A. 1990. New Methods in System Reliability. Ph.D. dissertation. Stanford University, California.

Kung, C.J., and P.H. Wirsching. 1993. Fatigue and fracture reliability and maintainability of tlp tendons. ASME Journal of Offshore Mechanics and Arctic Engineering 115:137–141.

Madsen, H.O., S. Krenk, and N.C. Lind. 1986. Methods of Structural Safety. Englewood Cliffs, New Jersey: Prentice Hall.

Madsen, H.O., R. Skjong, A.G. Tallin, and F. Kirkemo. 1987. Probabilistic fatigue crack growth of offshore structure with reliability updating through inspection. Pp. 45–56 in Proceedings, Marine Structural Reliability Symposium, SY-23, Society of Naval Architects and Marine Engineers held October 5–6, 1987 in Arlington, Virginia. Jersey City, New Jersey: Society of Naval Architects and Marine Engineers.

Mansour, A.E. 1990. An Introduction to Structural Reliability Theory. SSC-351. Washington, D.C.: Ship Structure Committee.

Mansour, A.E. 1993. Probability-Based Ship Design Procedure: A Demonstration. SSC-368. Washington, D.C.: Ship Structure Committee.

Marley. M.J. 1991. Time Variant Reliability Under Fatigue Degradation. Ph.D. dissertation. University of Trondheim, Norway.

Marley, M.J., and T. Moan. 1992. Time variant formulation for fatigue reliability. Pp. 161–168 in Proceedings of the 11th International Conference on Offshore Mechanics and Arctic Engineering held in Calgary, Alberta, Canada. New York: American Society of Mechanical Engineers.

Martindale, S.M., and P.H. Wirsching. 1983. Reliability based progressive fatigue collapse. Journal of Structural Engineering 109(ST8):1792—1810.

Melchers, R.E. 1987. Structural Reliability Analysis and Prediction. Chichester, United Kingdom: Ellis Horwood Limited.

Montgomery, D.C., and G.C. Runger. 1994. Applied Statistics and Probability for Engineers. New York: John Wiley and Sons.

Munse, W.H., T.W. Wilbur, M.L. Tellalian, K. Nicoll., and K. Wilson. 1983. Fatigue Characterization of Fabricated Ship Structural Details for Design. SSC-318. Washington, D.C.: Ship Structure Committee.

Paloheimo, E., and M. Hannus. 1974. Structural design based on weighted fractiles. ASCE Journal of the Structural Division 100:1367–1378.

Rackwitz, R., and B. Fiessier. 1978. Structural reliability under combined random load sequences. Journal of Computers and Structures 9:489–494.

Ravindra, M.K., and T.V. Galambos. 1978. Load and resistance factor design for steel. ASCE Journal of the Structural Division 104:1337–1354.

Shinozuka, M. 1983. Basic analysis of structural safety. ASCE Journal of Structural Engineering 109:721–740.

Stahl, B., and J.F. Geyer. 1984. Fatigue reliability of parallel member systems. ASCE Journal of Structural Engineering 110(STIO):2307–2323.

Thoft-Christenson, P., and M. Murotsu. 1986. Applications of Structural Systems Reliability Theory. Berlin, Germany: Springer-Verlag.

Thoft-Christensen. P., and M.J. Baker. 1982. Structural Reliability Theory and Its Applications. New York: Springer-Verlag.

Torng, T.Y. 1989. Reliability Analysis of Maintained Structural Systems Vulnerable to Fatigue and Fracture. Ph.D. dissertation. The University of Arizona, Tucson.

Torng, T.Y., and P.H. Wirsching. 1991. Fatigue and fracture reliability and maintainability process. ASCE Journal of Structural Engineering 117(12):3804–3822.

Tvedt, L. 1983. Two Second Order Approximations to the Failure Probability. Veritas Report RDIV/20-004-83. Oslo, Norway: Det norske Veritas.

White, G.J., and B.M. Ayyub. 1987. Reliability-based fatigue design for ship structures. Naval Engineers Journal 99:135–149.

Wirsching, P.H. 1980. Fatigue reliability in welded joints of offshore structures. International Journal of Fatigue 2:77–83.

Wirsching, P.H. 1984. Fatigue reliability of offshore structures. ASCE Journal of Structural Engineering 110:2340–2356.

Wirsching, P.H., and Y.N. Chen. 1988. Considerations of probability-based fatigue design criteria for marine structures. Marine Structures 1:23–45.

Wirsching, P.H., T.Y. Torng, and W.S. Martin. 1991. Advanced fatigue reliability analysis. International Journal of Fatigue 13:389–394.

Wu, Y.T. 1994. Computational methods for efficient structural reliability and reliability sensitivity analysis. AIAA Journal 32:1717–1723.

Wu, Y.T., H.R. Millwater, and T.A. Cruse. 1990. An advanced probabilistic structural analysis method for implicit performance functions. AIAA Journal 28:1663–1669.

Wu, Y.T., and P. H. Wirsching. 1987. A new algorithm for structural reliability estimation. ASCE Journal of Engineering Mechanics 113:1319–1336.

Ximenes, M.C.C., and A.E. Mansour. 1991. Fatigue system reliability of TLP tendons including inspection updating. Pp. 203–209 in Proceedings of the 10th International Conference on Offshore Mechanics and Arctic Engineering held in Stavager, Norway. New York: American Society of Mechanical Engineers.

Deep Draft Ship Inspection—Factors That Affect Fatigue and Fracture Prevention

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ABSTRACT

Much of the practical real-world work with material failures depends on early detection of the failure before it causes catastrophic harm. The metallurgist and ship designer need to understand the difficulties of the inspection of ship structures and capabilities of the inspector in order to properly strategize for the prevention of failures. This paper discusses those concerns and outlines some of the strategies that are currently under consideration. By combining the practical realities of the marine inspectors with the theoretical knowledge of metallurgists and designers, a more fracture-resistant and maintainable hull is foreseen for the post-Oil Pollution Act of 1990 generation of ships.

INTRODUCTION

When I was invited to talk to this group, I asked, "What should I discuss?" I was told this talk was intended to "level the playing field" for the audience, that is, to have all the audience go into tomorrow's discussions with a common understanding of what is done and, more importantly, not done, during the hull inspection of a deep draft ship. Further, the audience needs to understand what to expect and what not to expect from a structural inspection. I understand most members of the audience are very high level technical people. My talk is decidedly not highly technical.

Inspection is more a black art than a science. It does not require a masters or doctorate level education, but it does require a solid understanding of how the structure gets its strength, how loadings are applied, and how the structure responds to those loadings. Those are all clues that the inspector uses to "solve" the problem of where the failures are hidden. Once the ship is designed and the slide rules, french curves, and T-squares are put away, the inspectors—be they acting on behalf of the owner, operator, shipyard, class society, or regulatory body—have one of the largest impacts on major fatigue mitigation during the life of the ship. As engineers, you

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[1]The comments and opinions expressed in this paper are those of the authors, and not necessarily those of the United States Coast Guard or the Department of Transportation.

are being asked to develop tactics to prevent fatigue and fracture development throughout the life of the ship. To do so effectively, you must understand the inspection process in order to make realistic decisions on how best to mitigate the initiation of fractures.

PERSPECTIVE ON INSPECTION

In order to get the best general perception of the inspection task, picture yourself in a large gymnasium. The compartments in a given ship may be larger or smaller than that, but they are on that scale. The inspector usually enters this compartment via a ladder from the main deck. He is typically wearing coveralls, armed with a flashlight, and hopefully an atmosphere monitor, a hammer, pen, and inspection book. Often the only available light source is the natural light coming from a few 350-mm-diameter tank-washing openings in the deck. Usually the tank has not been staged for repairs. Now, given those conditions, consider that the inspector is tasked with being able to find a 25-mm crack on the framing as far away as the back corner of the gymnasium.

Clearly, this results in less than perfect results. Again, inspection is not a science. There are many variables that affect the probability that an inspector will find a fracture. Despite the enormous odds, once the Coast Guard Inspector signs an inspection book, he or she "owns" any fractures that may have been missed. Such fractures may manifest themselves prior to the next inspection. Our inspectors never lose sight of the fact that they have a stake in the safety of the vessel's crew, and the structural integrity of the vessel, as long as their names are on the vessel's inspection book.

THE VARIABLES IN INSPECTION

As I said there are many variables that enter the inspection "equation." The ship structural designer and the metallurgists need to understand these variables and keep them in mind throughout their decision making.

The first variable is the inspector. The requirements for the inspector or surveyor vary widely depending on whom they work for and who is requiring the inspection. Qualifications for inspectors employed by the vessel owner or operator may be met by experience from years of operating ships, contracting repairs, or working in shipyards. Often the responsibility for preinspecting a ship in preparation for a shipyard repair period is placed on the crew of the vessel. While they have lots of experience at sea, they may have no specific inspection training. In the past few years societies such as the National Association of Marine Surveyors have been formed, and a certification program for commercial marine surveyors has been developed. The classification societies usually hire surveyors that already have a significant amount of experience and then provide additional training.

The Coast Guard generally attempts to use either officers who have been trained in shipboard naval engineering or who are maritime academy graduates. However, their backgrounds often vary widely from those fields. Early in their inspection career they are sent to 2 months of inspection school. At their field unit they are given hands-on training and must demonstrate their abilities after many inspections as a trainee with a qualified inspector, prior to being qualified to do inspections on their own. As they progress, they are sent to additional 1- or 2-week schools for specialized training concerning mobile offshore drilling units, wooden boats, nondestructive inspection, fiberglass boats, and others topics. There is no typical background for an inspector, but it should include, as a minimum, a lot of experience with ship structures.

Another variable is the environment of the inspection. The environment for doing the inspection is not conducive to assisting the inspector. The inspection is usually conducted either during a rafting while the ship is at sea, or in a shipyard. There is no climate control. In the winter the bulk of clothing hampers the inspector; in the summer the empty cargo tank becomes a solar heated oven, limiting the amount of time an inspector can effectively and safely perform his or her tasks. Prior to entering the tank, the inspector must check the gas- free certificate to ensure there is adequate oxygen to sustain life and that there are (or were when the compartment was tested) no explosive levels of gasses.

In order to find the fractures or wastage in a ship's hull, the inspector must first see it. Lighting in a tank is often a feast or famine situation, with some bright lights in a few locations and shadows over much of the area. In general, the lighting in a tank does little good other than assisting the inspector in finding his way through and over the structure framing; the failures must be found with a flashlight.

Inspection fatigue is an omnipresent consideration. Climbing down into and up out of 20 or 30 tanks of a large tanker is quite tiring by itself without the added factors of required equipment, heavy clothing, or hot weather and the gymnastics of maneuvering over, under, and through the framing and staging in the vessel.

Available time is another factor affecting inspection quality. Nowhere is the concept of "time is money" more evident than in the operation of large ships. Rafting done at sea provides some cushion from the downtime crunch, but it still must be completed within the available time on the voyage. Turnarounds in port are kept to the bare minimum. The costs associated with cleaning a tank, gas-freeing it, and sitting at the pier while it is inspected are prohibitive. Gas-freeing is rarely done unless a tank is actually leaking at the time. Ideally a repair plan should be developed before the ship arrives in the yard; every day in the yard is a day of lost revenue. The inspector is constantly faced with the conflict of doing a thorough job versus time constraints. Frequently inspectors are responsible for a number of ships and must apportion their time accordingly. In addition to the actual time inspecting, they must coordinate with the shipyard personnel and must get all the written reports in a timely manner.

Coatings for tanks vary widely and can either assist an inspector or hide problems. In the best situation, the coatings are light and allow the cargo to runoff well when the tank is washed. Often a crack can stand out quite well with this type of coating, as heat causes the oil to slow-seep out of the cracks in the coating well after cleaning. In other cases the coatings may not harden, leaving a coating that flows or stretches over cracks and prevents them from being seen.

When the tank is inspected, the degree of "cleanliness" is highly variable. Sometimes the cleaning leaves a layer of sludge on the bottom of the tank that makes finding cracks on the bottom very difficult. In those cases the inspector can either require the tank to be cleaned further, causing delays, or do the best he or she can with the given conditions. Walking through the sludge may stir up vapors from the residue that may have settled on the bottom. In such instances the gas-free certificate may not be an accurate representation of the actual condition in the tank, especially if the tank has not been properly ventilated. When in doubt, the inspector should require additional tank cleaning, but this is not looked upon as a minor request.

CONDUCTING THE INSPECTION

After all these environmental factors are considered, the quality of the inspection still hinges on a visual inspection to detect discontinuities in shape, color, or shadows. This is where the black art comes in. Hopefully the inspectors are looking from a distance of 2 m or less. Given the time constraints, the magnitude of the area to cover, and the temperature and atmospheric environment, they cannot feasibly look at every linear centimeter of weldment. They cannot check every square meter of plating that shows some corrosion. They cannot chip away the coating everywhere it is broken in order to see if there is a crack in the framing beneath. Over time, inspectors learn where to expect to find problems. They develop a sense as to when a deflection affects strength, and when it is merely a cosmetic problem that need not be repaired. There are heuristic rules for cropping and renewing or leaving failures as is, but they leave much to judgement. For example, the Coast Guard Marine Inspector relies on Navigation and Inspection Circular (NVIC) 7-68, "Notes on Inspection and Repair of Steel Hulls." It allows wastage in excess of 25 percent in localized areas "if the condition of the adjacent material is sufficiently good to maintain an adequate margin of strength . . ." (USCG, 1968). This allows the experienced inspector with a real feel for the structure to make judgement calls, but it is dangerous when used improperly by the inexperienced person.

Many of the deep draft structural inspections are done prior to entering the shipyard while rafting during a ballast voyage. For this, the tanks are washed after the cargo is discharged. After the tank is considered safe for entry, members of the crew and the inspector enter the tank and get in a raft. As water is pumped into or out of the tank, the team maneuvers around the periphery of the tank and inspects the hull at arms length for each level. If the inspector is not able to attend the rafting, he or she must inspect the vessel early in the repair period to ensure that the work list is developed early enough for the yard to be able to complete the work in time. In the past the inspector climbed the framing up the side of the tank, holding on with one hand while hammering, marking failures with chalk or spray paint, or writing notes with the other. Now the inspectors are no longer permitted to free climb beyond 2 m. The classification societies have required enhanced surveys on tankships and bulk carriers that require the compartment to be staged so that the entire structure is within arms' reach of the inspector. Unfortunately this only applies to every other Coast Guard inspection, and it still leaves smaller unclassed tank barges and inland tankships for the inspectors to manage as best they can.

In general, the first tank in a given ship takes the longest. The inspector must do a very thorough inspection to learn how the framing is arranged, how this particular vessel hull fails, and how to efficiently cover the necessary area. The second tank takes slightly less time. By then trends start to develop. On subsequent tanks the inspector is generally able to go directly to the problem areas and knows where to look for structural failures. Careful examination of the external hull envelope will aid in locating internal problems such as tugboat or piling damage.

Generally our requirements call for a large tankship hull structure to be inspected twice in 5 years, every other one coinciding with a class society survey.

During construction, the inspector is called upon to perform many functions. Although the overall plan review is done at the Marine Safety Center, the on-site inspector has final acceptance authority. The inspector must witness operational testing of equipment and verify that construction and installation of the various piping and electrical systems are in accordance with the regulations, class requirements, or owners specification. Because they cannot feasibly visually inspect every linear meter of weld in the structure, much reliance is placed on the weld procedure and welder performance qualifications.

RECENT DEVELOPMENTS

Even with the "black art" nature of inspection, we have been looking at means to make it less costly, to reduce the regulatory impact without reducing safety, and to focus on methods to improve the inspection process.

After the unexpectedly high volume of cracking was found on the Trans-Alaska Pipeline Service (TAPS) tankers, the U.S. Coast Guard published NVIC 15-91, "Critical Area Inspection Plans" (CAIPs) (USCG 15-91, 1991). CAIPs required operators of certain vessels to record and track the occurrence of structural failures and to develop repair methodologies that address the root causes of the failures, rather than just the symptoms. Internal structural examination intervals for TAPS tankers were shortened from twice in 5 years, to an annual requirement. For the Attigun Pass design class, the interval has been reduced to 6 months. The CAIPs program has provided significant results. TAPS vessels operators have analyzed the causes of failures and have made structural modifications and/or employed innovative repair techniques, such as weld peening, to reduce the occurrence of structural failures. Other operators are conducting fracture mechanics analyses to determine appropriate repair methodologies.

Following the cracking found on the TAPS tankers the question was asked, "How often must we look at these hulls to ensure a crack does not propagate to a critical length before it could be found during an inspection?" Professor Rolfe of the University of Kansas (Rolfe et al., 1993), looked at the growth rate of a 50-mm surface crack at a rathole in a longitudinal member of a TAPS vessel. He found that, should the 50-mm crack be missed during one inspection, it could feasibly grow to about 375-mm prior to the next scheduled inspection. Under the right circumstances, a 375 mm crack could provide a catastrophic failure. With further studies such as this, and a better understanding of the probability of detecting a given flaw, realistic inspection decisions for specific vessels can be made.

Maritime Regulatory Reform (MRR) is intended to reduce the differences between the American Bureau of Shipping (ABS) rules and the U.S. Coast Guard regulations to make compliance with the regulations less costly for the industry. The U.S. Coast Guard is also constantly under pressure to reduce its size, do more with less, and still prevent marine disasters from occurring. The U.S. Coast Guard is currently working with ABS to have the ABS take over some parts of the inspection program which, in general, both organizations are currently doing. By turning over some of the inspection process to the ABS, the work hours saved can be reinvested in doing more thorough inspection of the small percent of the vessels that cause the most problems. More effort can be expended in overseeing the foreign vessels that are coming into the country that are not directly subject to most of our laws but are subject to international treaties.

An additional approach that is being developed is the idea of Safety Partnerships or a Streamlined Inspection Process. By establishing ISO 9000-type procedures or being registered by ISO 9000, a company with a good maintenance and operation history can work with the local Officer in Charge Marine Inspection to be able to obtain permission to self-inspect their ships in some aspects, reducing much of the U.S. Coast Guard inspections to random checks rather than verification of 100 percent compliance. This also will take much of the labor out of efforts that achieve little return on investment and free up the resources to spend more time on the real problem vessels.

However, there is a down side of both MRR and the Safety Partnerships. Even if the partner fulfills the required inspection procedures, in most cases it will result in having one less set of eyes looking at the structure. We are banking on setting up the procedures and checks and balances to make this effect as small as possible, but it cannot be ignored.

The International Safety Management (ISM) code is a requirement for certification of shipping companies' management of the safety program, much the same as ISO 9000 certifies a company's quality system. It is hoped that with ISM certification the Coast Guard will be able to further redirect its efforts from the routine safety checks, which the company will perform, to reorient of efforts more tightly on the biggest problems.

UPCOMING INSPECTION TECHNIQUES

As I stated before, the inspector used to climb the framing of the vessel in order to inspect the framing of the vessel. Now safety concerns prohibit that. In some cases the Tanker Structure Cooperative Forum allows for climbing the framing if a water bottom is provided for a cushion. Even then it does not allow climbing above 6 m above the water (TSCF, 1986). Fall safety devices are similar to the equipment used by mountain climbers; the inspector hangs from a rope secured to the overhead or uses a rope as a safety should he or she fall. The "Spider" and "Stageaway" systems are platforms that raise and lower the inspector similar to a window washer on a large building (Holzman, 1992). Each system has its benefits and liabilities; while they get the inspector closer to the structure they may be expensive, cumbersome, or time consuming. Each is fully usable now, but their use may depend on the limiting factors for the inspection under consideration.

While we are focusing our efforts on the areas with the greatest failure prevention potential, we are also attempting to better understand the uncertainties in the inspection process. The Ship Structure Committee is currently conducting three projects related to inspections. Project SR-1355, "Inspection of Marine Structures"—conducted by Professor Demsetz at the University of California, Berkeley—examined the variables associated with the probability of finding an existing failure in a structure. It will be followed by Project SR-1375, "Detection Probability Assessment of Visual Inspection of Ships," that is intended to provide some experimental numbers to these probabilities. Project SR-1365, "Optimal Strategies for Inspection of Ships for Fatigue and/or Corrosion Damage," will focus the inspection effort into the most likely areas of failure when limited resources are available. Through a better understanding of these human factors it is anticipated that inspections may produce better results in the future with a net smaller effort expended. A fourth related project being carried out by the SSC is Project SR-1356, "Strength Evaluation of Pitted Plate Panels." Through this project we are hoping to obtain an inspector-friendly decision tool to use in the field to evaluate severely pitted plates on ship hulls. Currently the inspector must make a decision to repair, replace, or leave as is, based on a minimal amount of information, his or her own personal experience, and some very general heuristics.

The Coast Guard's research and development center has been looking into high technology applications for inspection. They include remote controlled lights, remote controlled cameras, advanced visual inspection with heads up display video for remote evaluation, polarized light, laser ultrasonics, acoustic emission resonance, and infrared mapping. All require further evaluation or development. The research and development center also demonstrated the feasibility of a marine portable inspection unit, a handheld computer in which the inspection books are already formatted and vessel information is provided. With this the inspector can enter the information while still in the tank. This may not help in the inspection of the tank, but, by reducing the preparation and follow-up administrative times significantly, the inspector is freed up to spend more time on the inspection. Where these techniques can best be applied, and how reliable they will be, remains to be seen (Hansen, 1994).

It is in the ship owner's and operator's best interests to have an inspection program tailored for the vessel. You cannot afford to wait until a class society surveyor, Coast Guard Marine Inspector, or shipyard foreman discovers a failure. By then the failure has probably grown, the repair contract is already signed, or the shipyard departure date is set. The repair will cost much more in time and money. Ship Structure Committee report SSC-332, "Guide for Ship Structural Inspections," laid out the ingredients and steps necessary to plan a life cycle inspection program for a vessel during the design process (Basar and Jovino, 1990). The greatest cost savings and labor efficiencies can be made by establishing an inspection plan during the initial design of the vessel. This report has been converted into the format for an American Society for Testing and Materials Guide and is currently under review for adoption.

CONCLUSION

If I've gotten one concept across it is that the inspector should be considered a last line of defense. Economics have in the past—and will to a greater extent in the future—prohibited the type of detailed frequency of inspection that would be required to find every crack before it can begin to grow. Most likely there will be fewer inspectors looking at a hull than has been the case in the past. The environment and variables that work against the inspector make the likelihood of missing something important too great to ignore. CAIPs, inspection strategies and the economic incentives to focus on the most likely areas of failure should help stem the rate of potentially catastrophic failures, but it will not eliminate them.

Tomorrow there will be a separate work group to deal with inspection-related issues. However, I urge all the work groups to keep the inspector in mind in making your decisions. To ensure success, your recommendations must take into account the inspection processes.

REFERENCES

Basar, N., and V. Jovino. 1990. Guide for Ship Structural Inspections. Ship Structure Committee Report SSC-332. Washington, D.C.: Ship Structure Committee.

Hansen, K. 1994. Technology Evaluations for Coast Guard Marine Inspections. Presented at the 43rd Defense Working Group on Nondestructive Testing. Held November 1994 in Indianapolis, Indiana. Washington, D.C.: Department of Defense.

Holzman, R. 1992. Advancements in Tankship Internal Structural Inspection Techniques. Berkeley, California: University of California.

USCG (U.S. Coast Guard). 1968. Navigation and Vessel Inspection Circular (NVIC). NVIC 7-68. Washington D.C.: USCG.

USCG. 1991. Critical Area Inspection Plans. NVIC 15-91. Washington, D.C.: USCG.

Rolfe, S., A.E. Henn, and K. Hays. 1993. Fracture mechanics methodology for fracture control in oil tankers. Pp. F-1–F-11 in Ship Structures Symposium '93, held November 16–17 1993 in Arlington, Virginia. Jersey City, New Jersey: Society of Naval Architects and Marine Engineers.

TSCF (Tanker Structure Cooperative Forum). 1986. Guidance Manual for the Inspection and Condition Assessment of Tanker Structures. Witherby, London, England: TSCF.

Consideration of Loads for Fatigue Assessment of Ship Structures

Y.N. Chen and Y.S. Shin

ABSTRACT

Technical issues associated with loads obtained for the purpose of fatigue assessment are discussed in this paper. The topic of discussion is focused specifically on spectral fatigue analysis and linear elastic fracture mechanics based crack-growth analysis. In order for the frequency domain formulation and the associated probability analysis to be valid, load analysis presented in this paper is largely limited to be linear. Some nonlinear features that should also be taken into consideration in the process of fatigue assessment are also discussed. This includes nonlinear roll motion and nonlinearity brought forth by intermittent application of loads, such as wetting of side shell in the splash zone by waves. The discussion emphasizes the dominant loads relevant to particular regions of the ship structure, as well as the eventual constitution of the set of loading. A simple, linear rule for load combination is proposed.

INTRODUCTION

Fracture and fatigue are major limit states that are becoming more prominent in ship structural design practice. These phenomena have gone from being minor nuisances in the good old days, when most ships were built of mild steel, to major failure modes that can drive the design. Their significance goes beyond design, as they also play a prominent role in the planning and execution of inspection and maintenance. The authors witnessed first hand the emergence of fatigue assessment as an integral part of the restated Rules at the American Bureau of Shipping (ABS) known as the SafeHull system. In this undertaking, the consideration of fatigue failure and the criteria that aim to alleviate it are recognized as essential components of structural integrity.

While much of the scientific and engineering basis of fatigue that is built into the SafeHull system can be regarded as existing and well-established technology, the implementation of such into a design guidance is far from trivial. Primarily, there were two areas that commanded the greatest effort in that endeavor; namely, calibration and the treatment of loads. It is the topic of loads that the authors will address here.

The discussion presented in this paper begins with the examination of the load effect premise for a fatigue analysis. In this regard, the authors advocate the acceptance of a linear theory; not universally, but as a rule. In so doing, the positive aspects of a frequency domain, linear analysis can be retained and imbedded into the classical stochastic analysis also known as spectral fatigue initiation analysis. Similar advantages also prevail in a linear elastic fracture-mechanics–based crack-propagation analysis. This is not to say that nonlinearity should be categorically excluded from consideration in a fatigue analysis. It simply indicates that, in most cases, a linear analysis is sufficient.

The conventional classification of primary, secondary, and tertiary load components, restated in the context of fatigue, is next discussed within the analytical framework of structural (or stress) analysis as a prerequisite to fatigue analysis. The notion of dominant load (a deterministic statement of Turkstra's rule) for a specific region of the structure and associated load components sets the tone of the all important issue of load combination, which is presented last in this paper. On this basis, general guidelines are presented for the types of loads that need to be considered in various regions of a ship. Certain important issues that could easily slip by the attention of an analyst are specifically addressed, such as how to treat the intermittent wetting of the side shell near the mean water line. Also addressed in this paper, is the representation of the wave environment by the joint probability of significant wave height and characteristic period employed in spectral fatigue analysis. The authors' advocacy for the use of the whole-range of wave periods is particularly important for inertia-dominated (as contrasted with those of drag-dominated) structural response.

The paper also gives a series of "how to's" regarding the treatment of various loads. The intention is not to detail the dynamic and hydrodynamic bases, which are outside the scope of this paper. Rather, they serve as identification of the degree of sophistication that can be regarded as adequate.

How to put the load component together is addressed in the section on load combination. Here distinction is made between limit states driven by multiple environmental effects and those driven by a single one, viz., waves. In connection with the latter, considered relevant in the domain of fatigue considerations, a simple method using the well-known notion of correlation coefficients is proposed.

PREMISE OF LOAD EFFECTS IN FATIGUE ASSESSMENTS

In both fatigue-crack initiation and crack propagation, the most commonly used analytical formulations are those based on the cumulative damage scheme. The first issue that arises is how nonlinearity of the stress-range exceedence curve affects the basic notion of cumulative damage. It is probably safe to state that the conceptual cycle-by-cycle counting of damage incurred remains valid in both the linear and nonlinear regimes of the driver (i.e., the load effects). However, nonlinearity of loading brings in the importance of loading sequence, which effectively precludes the application of the principle of superposition in most cases.

In a recent paper, Winterstein (1988) considered nonlinearity of the type that can be represented by a model in the form of a cubic Hermitian polynomial. A remarkable feature of his finding was that nonlinearity (at least the cubic type) has a far less profound effect in fatigue (initiation) damage than in the extreme values. Chen and Thayamballi (1991) further suggested that, even in the linear regime, owing to the frequency content of the loading, severity in extreme value does not imply severity in fatigue damage. The key to illustrate such a phenomenon can be cast in terms of a familiar notion, that is, the Weibull shape parameter. In Winterstein's model, the Weibull shape parameter for the long-term distribution tends to decrease from the corresponding value associated with the linearized model. On the other hand, there are types of nonlinearities that may raise the Weibull shape parameter in the long-term distribution of the stress range. A typical example that belongs to this category (i.e., the second kind) is the nonlinear roll motion. However, nonlinear roll reduces the extreme value enough to negate the effect of higher Weibull shape parameter. Still, there is a third kind of nonlinearity stemming from the intermittent (thus nonlinear) occurrence of the phenomenon. A typical example is the long-term effect of slamming on the stress range that can be translated in terms of fatigue damage. In the following paragraph the last category of nonlinearity will be discussed in some detail.

In a systematic examination of slamming records in a study funded by ABS, Boitsov et al. (1994) suggest that the stress range, S, relevant to fatigue can be represented by the relations:

$$S = C_1 h \qquad if \qquad 0 \le h_{CR} \tag{1a}$$

$$S = C_1 h + C_2 h^{\zeta}(h - h_{CR}) \qquad if \qquad h \ge h_{CR} \tag{1b}$$

in which h is the wave height and h_{cr} is a critical wave height beyond which slamming occurs. Boitsov further suggests that the parameter ζ be chosen as unity (i.e., a quadratic model for the part attributed to impact). For the sake of simplicity here, this parameter is (conservatively) taken to be zero; thus effecting a segmental linear model. With this simplification, the long-term probability of exceedence can be idealized as (see Figure 1)

$$x = \frac{X_{CR}}{S_{CR}} s \qquad if \qquad 0 < x < X_{CR} \tag{2a}$$

$$x = \left[\frac{\epsilon X_{20}}{S_{20}(\alpha + \epsilon)}\right]\left\{s + \frac{\alpha S_{20}(1 - \epsilon)}{\epsilon}\right\} \qquad if \qquad X_{CR} < x < \infty \tag{2b}$$

In these expressions, x is an exceedence parameter defined as $x = \log_{10}(1 - F)$ with F denoting the long-term distribution. Other quantities are defined as below:

X_{20} = value of X in a ship's design lifetime
S_{20} = value of s associated with X_{20} on the exceedence curve without impact
X_{CR} = a threshold value of X below which impact would not occur

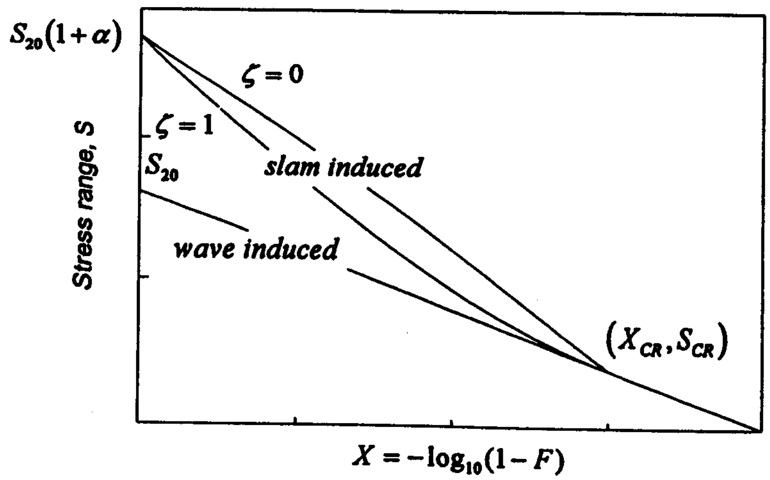


FIGURE 1 Stress range exceedence including slamming.

α = fraction of increase in long-term stress range, i.e.,$s(X_{20}) = (1 + \alpha)S_{20}$

and

$$S_{CR} = \frac{S_{20}}{X_{20}} X_{CR}$$

$$\epsilon = 1 - \frac{X_{CR}}{S_{20}}$$

The probability density function, $f = dF/ds$, can be obtained by differentiating Equations (2), leading to:

$$f_s(s) = \lambda e^{-\lambda s} \qquad if \quad 0 \le s \le S_{CR} \tag{3a}$$

$$f_s(s) = \mu \exp\left(-\mu s - \mu \, \alpha \, S_{20} \frac{1-\epsilon}{\epsilon} \right) \qquad if \qquad s \ge S_{CR} \tag{3b}$$

in which

$$\lambda = \frac{X_{20}}{S_{20}} \ln(10)$$

$$\mu = \frac{\lambda \epsilon}{\alpha + \epsilon}$$

Now consider the case of fatigue initiation governed by a one-segment S–N curve, that is, $NS^m = K$, the cumulative damage incurred, $D(\alpha)$, can be obtained by straight-forward integration. If the ratio $\Re$ is defined as:

$$\Re = \frac{D}{D(\alpha = o)}$$

It can be shown easily that $\Re$, the ratio that measures the increase in fatigue damage attributed to this type of nonlinear effect, is given by:

$$\Re = \frac{\gamma(m+1, \lambda S_{CR})}{\Gamma(m+1)} + \left(\frac{\lambda}{\mu}\right)^m \left[1 - \frac{\gamma(m+1, \mu S_{CR})}{\Gamma(m+1)}\right] \exp\left[\frac{-\mu\,\alpha\, S_{20}(1-\epsilon)}{\epsilon}\right] \qquad (4)$$

where $\gamma(\ ,\ )$ and $\Gamma(\ )$ are the incomplete and complete gamma functions, respectively. It is interesting to note that, of the two S–N parameters, the negative slope, m, enters the preceding expression, while the parameter K does not. Moreover, the threshold parameter, ε, explicit in Equation (4) as well as implicitly imbedded in μ and S_{CR}, also plays an important role in $\Re$.

In order to provide some feel for the numerical order, consider the specific case of $X_{20} = 8$ (i.e., 1- $P = 10^{-8}$), $S_{20} = 140$ N/mm^2, and $0.5 < \alpha < 1$. The threshold value of X (i.e., X_{CR}), is believed to be in the order of 2 (Boitsov et al., 1994; Ochi, 1964; and Stiansen and Mansour, 1975), meaning that slamming occurs about 1 percent of the time. Equation (4) then yields 19 percent to 46 percent increase in fatigue damage as a result of slamming (see Figure 2). This is a very modest increase, indeed, considering the corresponding increase in extreme value being in the range from 50 percent to 100 percent. Parallel computation using the quadratic model (i.e., via numerical integration) results in a range of increase in fatigue damage from 6 percent to 15 percent corresponding to the same increase in extreme value. This is not surprising considering that the exceedence relation, Equation (2) (and the quadratic equivalence), strongly suggests the resemblance of a Weibull distribution with its shape parameter substantially lower than unity.

The preceding analysis, plus the fact that only moderate to low values of the stress range are important in a fatigue analysis, is a strong argument that linearity can be assumed in the loading formulation for the purpose of fatigue assessment. This premise further implies that the use of classical stochastic approach (or spectral analysis) in crack initiation analysis that is built

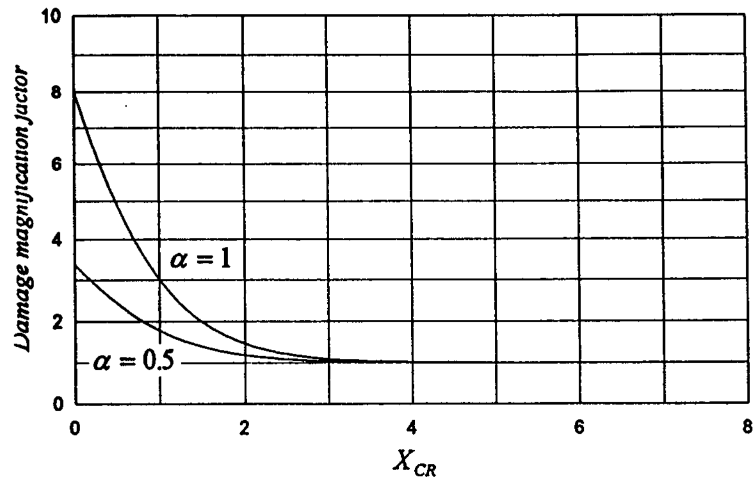


FIGURE 2 Damage magnification factor versus slamming threshold exceedence parameter, X_{CR}.

around notions of unit wave height stress-range transfer functions and superposition, and the application of linear elastic fracture mechanics for crack-propagation analysis is justified.

ANALYTICAL FRAMEWORK

The analytical approaches taken in the process of fatigue assessment will evidently have a bearing on the strategy of the load's determination. In the analysis of fatigue-crack initiation, for example, the primary method of analysis can be loosely termed the spectral method. This is meant to refer to the many considerations, especially those related to the load determination strategy, stemming from the spectral analysis without limiting oneself to the specific numerical algorithm. For example, the well-known "simplified" method, utilizing a Weibull distribution to fit the long-term distribution of the stress range can be conceptualized within the context of a spectral formulation (Chen and Mavrakis, 1988). Another example that can be used to support this notion is the ABS Tanker Fatigue Guide (ABS, 1993b; see also ABS 1995 Steel Vessel Rules), which has the appearance of a "simplified" method, but its acceptance criteria were determined with lengthy calibration using spectral analysis.

A more complex situation is the type of analysis that stands between the establishment of the loads and the determination of the stress transfer function, that is, the stress analysis. The types of loads that should be imposed on the structural model in the stress analysis depend upon the type or level of stress analysis. To this there may not always be a straight, simple answer. On the surface of it, the traditional notions of primary, secondary, and tertiary classification of loads, so fundamental to naval architecture, would cease to be a meaningful criterion of ranking. For example, when the target of fatigue analysis is the weldment centered at a connection between a side longitudinal near the ladened water line, the transverse frame, and the connecting flat-bar, hull-girder loads become secondary, and pressure (external and/or internal) becomes primary. In the case of misalignments, local bending, a usually tertiary effect, is the dominant factor. Thus, the key word is "dominant load," which really does not have a conflict with the essence of the traditional classification. In any case, fatigue is a localized problem, and the analyst's attention must be focused on the immediate neighborhood of the fatigue point in question. In this connection, the term "far field" could be a mere 100 mm away from the specific hot spot. In terms of the stress analysis itself, quite often for the longitudinal detail cited in the foregoing, a beam model may suffice (versus two-dimensional or three-dimensional finite element analysis), provided that the two-dimensional nature of the detail assembly that is not reflected in a beam model is properly compensated (calibrated). On the other hand, even the most complex, sophisticated finite element model does not in itself guarantee that the resulting stresses reflect the most adverse situation in terms of fatigue damage, unless the loads imposed cover the full range of cases.

LOADS AND LOAD EFFECTS FOR FATIGUE ASSESSMENT

Various regions in a ship obviously are subjected to different types of loads and load effects. The unique characteristics of loads are briefly outlined below.

Closed Deck Structures

Loading to be considered in a fatigue analysis for structural details in this region is primarily uniaxial loads produced from hull-girder bending. Examples are deck plating and deck longitudinals plus their connections to adjacent structure, such as web frames, transverse bulkheads, and the like. This is probably the least complex situation insofar as the loading formulation is concerned. Typically it refers to the deck structure of tankers or similar configurations.

Open Deck Structures

This region includes the deck structure of container vessels and, possibly, the deck structure of bulk carriers and all-hatch cargo ships. Away from the deck openings, plating and longitudinals and their assembly structures can be treated as in Item 1 above. Hatch corners present a frequent and severe problem of fatigue, and they require special attention.

Hatch-corner fatigue failures are commonly related to wave-induced torsional effects, whether it is the dominant load effect or it enhances other load effects, such as wave-induced hull-girder bending. Primarily there are two contrasting scenarios that will make the hatch corners around open decks vulnerable to torsion. The first is the restrained warping-induced longitudinal stress that adds to the extreme fiber bending (including both vertical and horizontal hull-girder bending) stress. Such a contribution to the global load effect can be significant at the junctions where a relatively rigid section meets a weaker one. A typical example that exhibits such a discontinuity of geometric property is the first hatch opening forward of the aft-mounted deck house. The second mechanism that could cause problems at a hatch corner is attributed to excessive distortion of the hatch opening. This action tends to open up or to squeeze the hatch corner assembly, which may result in fracture. A typical example of such an occurrence is at the forward hatch openings, where maximum distortion usually occurs.

Unlike hull-girder bending, restrained warping-induced stresses and hatch diagonal distortion do not have a ready made one-to-one correspondence to their driving force, that is, the torsional moment and horizontal shearing force. Thus, such load effects cannot be directly applied, as in the case of bending moments, as controlling parameters that could be maneuvered to maximize the fatigue causing stress. This is a situation typical of any statically indeterminate structure. In a recently completed project that measured a containership's hatch diagonal distortion, close correlation was observed between roll motion and distortion (Chen, 1994). This offers a possibility that the most adverse wave condition regarding distortion can be quickly determined without first going through the elasticity solution. As for the effect of restrained warping, there is no such shortcut available, and a large number of load cases must be used and stress analysis results examined. One possibility is to employ the "opened thin-walled torsion bar" approach to simplify the stress analysis into one that is one dimensional, as it is done at ABS using the computer program DYSOS.

Side-Shell Structures

This region refers to the submerged portion of the side shell and its longitudinals and their connections to adjacent structures. The portion exposed to air can be treated in a similar manner as deck structures. Particular attention must be paid to the neighborhood near the stillwater line, where external pressure is applied intermittently. In addition, hull-girder bending and internal pressure exerted by liquid cargo or ballast water, both static and dynamic, should be applied where applicable. For a bulk carrier, the granular cargo may also exert loading, both normal to the side shell and tangential to it. In a fully loaded configuration, vertical bending may not have much effect on the side structure near the water line, as it is close to the neutral axis. However, its effect becomes more significant for a very light ballast loading case and for the deeper, submerged portion of the side shell. In any case, bending-load effects should be applied in addition to pressures.

If one focuses on the area above the mean water line, it is wetted intermittently by the incident waves. Pressure in that zone can be determined by some selected hydrodynamic theory. Direct reading of the hydrodynamic pressure results would suggest that maximum hydrodynamic pressure will occur above the mean water line. This implies that the most fatigue damage should be incurred above the mean water line. However, reality indicates otherwise. Most recently, Witmer and Lewis (1994) presented a detailed catalog of fatigue-crack occurrences. A rule-of-thumb was actually suggested that the most frequent occurrence of cracks (i.e., the location on the side shell that is most vulnerable to cracks) is about four bays (of longitudinals) below the mean water line. Payer and Fricke (1994) suggest disregarding the negative portion of the water pressure. With a numerical algorithm derived on this basis, results similar to the trend recorded in Witmer and Lewis were obtained. Similar numerical formulation also was reported earlier by Barltrop and Brooking (1990). An analytical treatment along this line was also reported by Friis-Hansen and Winterstein (1995).

Along the line of reasoning of negative-pressure truncation, a simple and practical correction factor can be derived. On this basis, one can postulate that, if the draft and the elevation of the field point are denoted by D and z, respectively, there exists a critical wave height, h*, defined as:

$$h^* = 2(D-z)$$

such that, if y denotes the pressure head,

$$y = C_p h \qquad \text{if} \qquad 0 \le h \le h^* \tag{5a}$$

$$y = y* + \frac{1}{2}C_p(h-h^*) \qquad \text{if} \qquad h* < h < \infty \tag{5b}$$

where $y^* = h^*/2$ and C_p is a known coefficient. The situation is shown schematically in Figure 3.

With the given relation, which transforms the wave height h, to the pressure head, y, the probability density function for the pressure head can be obtained through a probability transformation of the Rayleigh density function, which governs the wave height; that is, for $0 \le y \le 2C_p\, y^*$:

$$f_Y(y) = \frac{16y}{C_p^2 H_s^2} \exp\left[-8\left(\frac{y}{C_p H_s}\right)^2\right] \tag{6a}$$

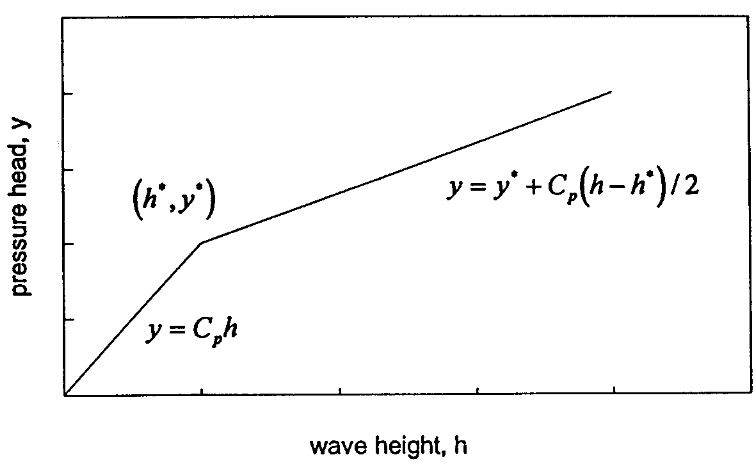


FIGURE 3 Pressure head on side shell below MWL.

For $2C_p\, y^* \le y \le \infty$:

$$f_Y(y) = \frac{32\,(2y-y*)}{C_p^2 H_s^2}\exp\left[-8\left(\frac{2y-y*}{C_p H_s}\right)^2\right] \tag{6b}$$

The *2nd* moment of the density function is given by

$$m_2 = \left(\frac{C_P H_S}{\sqrt{8}}\right)^2 \left\{\gamma(2,u^*) + \frac{2y^*}{(C_P H_S)^2}e^{-v^*} + \frac{\sqrt{8}y^*}{2C_P H_S}\left[\frac{\sqrt{\pi}}{2} - \gamma\left(\frac{3}{2},v^*\right)\right] + \frac{1}{4}[1-\gamma(2,v^*)]\right\}$$

where $\gamma(\ ,\ )$ is the incomplete gamma function, and

$$u^* = \frac{32y^{*2}}{H_S^2}$$

$$v^* = \frac{8\,(4C_P-1)^2 y^{*2}}{C_P^2 H_S^2}$$

A pressure-correction factor can be introduced as:

$$\kappa = \sqrt{\frac{m_2}{\lim\limits_{y^*\to\infty} m_2}}$$

leading to:

$$\kappa = \left\{\gamma(2,u^*) + \frac{2y^*}{(C_P H_S)^2}e^{-v^*} + \frac{\sqrt{8}y^*}{2C_P H_S}\left[\frac{\sqrt{\pi}}{2} - \gamma\left(\frac{3}{2},v^*\right)\right] + \frac{1}{4}[1-\gamma(2,v^*)]\right\}^{\frac{1}{2}} \tag{7}$$

This reduction factor can be applied to either the pressure obtained directly from the computed hydrodynamic (unfactored) pressure or as a reduction factor to the significant wave height in the wave-scatter diagram. In the latter application, the resulting reduction factor is shown, parametrically with respect to the depth below the mean water line, designated as "MWL" in Figure 4, below.

The reduced wave data can then be applied in conjunction with the unfactored pressure transfer function. It is interesting to note that the factored pressure range (peak to trough) profile

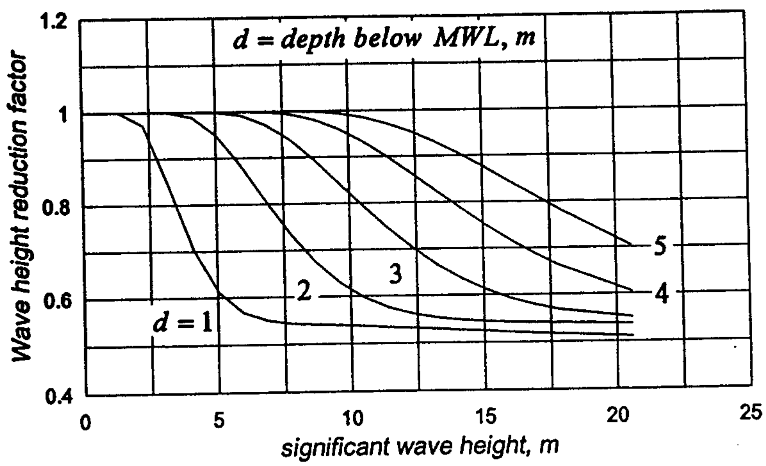


FIGURE 4 Pressure reduction factor applicable to significant wave height.

along the depth below the mean water line has a relative maximum approximately between 4 m and 5 m below the stillwater line; a marker insensitive to the draft, D, and significant wave height, H_s.

Bottom Structures

The load effects relevant to fatigue of the bottom and double-bottom structures are, in a way, similar to the side shell structures. In other words, hull-girder bending, as well as external and internal liquid cargo pressure and bulk loads, where applicable, are to be included in the loading terms. The noted exception is the intermittent application of the external pressure discussed in Item 3. The relative dominance of the hull-girder loading is much more profound in the bottom structures. While hydrostatic pressure here is high, the hydrodynamic pressure without slamming is generally fairly modest and ceases to be a dominant load effect.

Wave Environment

In the spectral fatigue analysis, the wave data required are the joint probability of wave parameters, such as significant wave height and wave characteristic period (e.g., average period or zero up-crossing period). These wave data are commonly presented as the wave-scatter diagram. The selection of wave data for the purpose of fatigue analysis depends on the anticipated areas of operation. From the classification society's standpoint, most large vessels are normally classed for unrestricted service. On that basis, the nominal North Atlantic wave environment is usually selected. For a site-specific application or for a trade route known to be more severe than the North Atlantic, the wave-scatter diagram for a specific wave environment or trade route should be used.

The inclusion of wave-period variation is particularly important for ships (or any floating structure) for which motion and loads are strongly frequency dependent. Each seastate (a cell in the scatter diagram) is typically represented by a mathematical spectrum, such as the Bretschneider for open ocean conditions or the JONSWAP for fetch-limited cases. The use of spectra is an integral part of the spectral fatigue analysis.

The North Atlantic Ocean is an area for which a considerable amount of wave data exists. Measured data are considered the most reliable source as long as they are for sufficient years of record; typically, for records 20 years or longer are considered desirable. Ochi's spectral family (Ochi, 1978) and H-family (ABS, 1980) belong to this category. Other sources of wave data are hindcast wave and observed wave data. The U.S. Navy's Spectral Ocean Wave Model (SOWM) is one of the best known hindcast data sources, and it is presented in a scatter diagram form (Chen and Chen, 1979). In recent years the hindcast model has been improved for directionality and extended to include the southern hemisphere. The model become known as the Global Spectral Ocean Wave Model (GSOWM), which is being used by the U.S. Navy for ocean wave forecasts (Chen, 1982). Visually observed data include those compiled by Hogben and Lumb (1967), BMT Global Wave Statistics (Hogben, 1985), and also the well-known Walden data that are used by many classification societies.

ABS uses the H-family wave spectra as a reference in the vessel's longitudinal strength assessment by the dynamic loading approach (DLA) (Liu et al., 1992). So far as the vertical bending effects are concerned, in the context of the most probable extreme values, the H-family data is very reliable, as the prediction based upon this data ensemble has been calibrated by full-scale measurement known in the U.S. maritime industry as the ABS's five-ship instrumentation program (Little and Lewis, 1971). However, for fatigue applications, the emphasis is not the tail of the joint probability distribution. A prerequisite is the unbiased joint probability of characteristic wave period and (significant) wave height. The H-family data do not meet this requirement and must not be used in fatigue application as pointed out by Chen and Thayamballi (1991).

For fatigue analysis, wave directionality and the kinetic energy spread also play an important role. Wave directionality refers to the probability distribution of the frequency of occurrence of the wave heading; and energy spread refers to the instantaneous spread of energy about a particular wave heading. The former can be incorporated only if such information is available. The latter can generally be treated with a mathematical form, such as the n^{th}-cosine law. In some specific cases of fatigue assessment, such as the Trans-Alaska Pipe Service trade ships that sail between Southern California (and on occasion, Panama) and Valdez, Alaska, the wave direction is biased and wave directionality is a dominant effect. However, since the wave directionality data are normally difficult to come by, equal probability of wave heading can be assumed in fatigue assessment of vessels designed (and/or classed) for unlimited service. The wave energy spread is tied to the notion of "confused seas." In a fatigue analysis, because the most damage is incurred at low and moderate seas that are known to be "confused," the use of directional spectra should be the norm rather than the exception. If directional spectra information is not available, it is commonly accepted that the cosine-squared law be applied.

Wave scatter diagrams as a working tool can be found in Chen and Chen (1979), Chen (1982), Hogben and Lumb (1967), Hogben (1985), and Liu et al. (1992); they will not be illustrated in this paper.

Ship Motion and Acceleration

Motion analysis of the vessel precedes the calculation of loads applied in the stress analysis. Based on the discussion of the premise of load effects presented earlier in this paper, linear strip theory seakeeping analysis will suffice. The only exception is the effect of nonlinear roll motion. Such considerations are especially important in the handling of hydrodynamic pressure on the side shell and in acceleration that is used for internal cargo loads (e.g., internal oil or ballast water pressure and bulk loads).

Typically the roll motion (as in other transfer functions) is calculated for a range of wave frequencies varying from 0.2 rad/sec to 1.8 rad/sec, with a frequency increment of 0.05 to 0.10. Wave heading is considered with an increment of 15 to 30 degrees covering the range from head sea to following sea of a unit amplitude wave. An example of the roll transfer function is shown in Figure 5. Nonlinearity has a more profound effect on roll motion than others, partly due to viscous-related damping, as shown in Figure 6. Wave-induced damping, alone, is not adequate to accurately predict roll motion. Damping due to viscous lift and cross-flow drag of the ship hull, bilge keel, appendages, and control fins needs to be accounted for in the analysis. Details of the topic can be found in the works by Ikeda et al. (1978), Schmitke (1977) and others, such as Ogilvie and Tuck (1969).

Once the ship motions are calculated, local accelerations at many locations need to be calculated to determine the inertial load resulting from cargo and light ship weight. The following will apply:

$$\vec{A} = -(\vec{R}\times\vec{\Theta})\,\omega_e^2 + \vec{a}$$

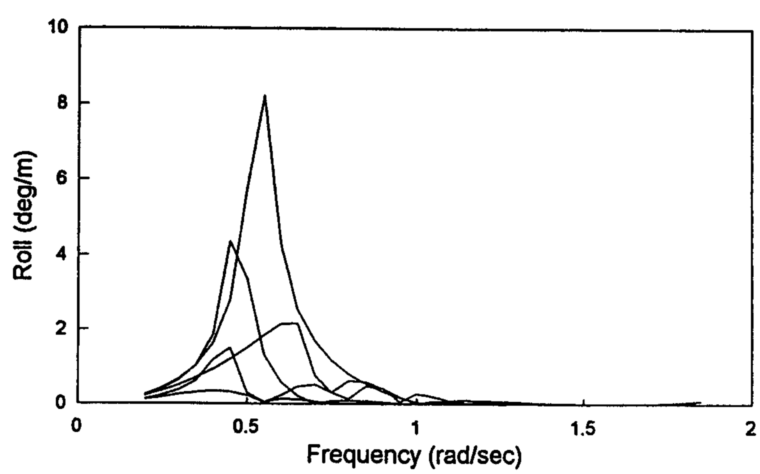


FIGURE 5 Typical roll motion transfer functions.

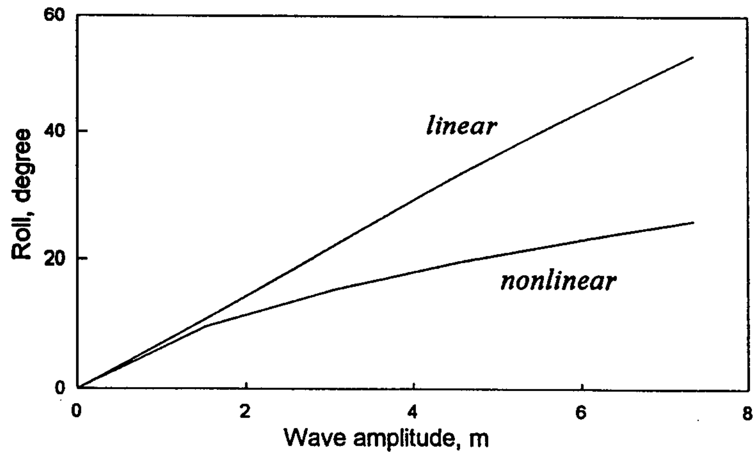


FIGURE 6 Nonlinearity of roll motion (schematic).

where $\vec{A}$ is the acceleration vector, $\vec{R}$ is the position vector from the center of gravity to the point of interest, $\vec{\Theta}$ is the rotational motion vector, ω_e is the encounter frequency, and $\vec{a}$ is the acceleration at the center of gravity due to translatory motion. The accelerations in all three directions are needed in a global coordinate system to calculate inertial load to be applied to the structural FE model to achieve a dynamic equilibrium with other external load effects driven by the waves.

External Wave Pressure

The hydrodynamic pressure distributions over the surface of the wetted hull section need to be determined for global and local structural analysis. The external pressure consists of four components; namely, the incident wave pressure exerted by plane, progressive waves; the diffracted wave pressure due to the presence of the ship body held in the wave train; the radiated wave pressure due to the ship oscillating in otherwise calm water; and the quasi-static pressure due to the change in vertical displacement of the ship hull.

First, the incident wave pressure is represented by:

$$P_I = -i\rho g a \exp[ky - ik_0 z\sin\mu + ikx\cos\mu]e^{i\omega t}$$

where g, a, k, μ are gravitational acceleration, wave amplitude, wave number, and wave heading angle, respectively. In the present paper (as in the convention of the ABS ship motion computer program), the variable x is positive pointing toward the bow, y is positive upward, and z is positive pointing toward starboard.

The diffraction pressure and radiation pressure are given by:

$$P_D = \left(i\omega + U\frac{\partial}{\partial x}\right)\phi_D(x,y,z;k,\mu)$$

$$P_R = \left(i\omega + U\frac{\partial}{\partial x}\right)\sum_{m=1}^{6} \zeta_m \phi_R^m(x,y,z)$$

where U is the ship speed, ϕ_D is the diffraction potential, ϕ_R is the radiation potential in the m^{th} (of six) degrees of freedom. Both ϕ_D and ϕ_R are obtained as solutions to the three dimensional fluid boundary value problem which can be solved by the three-dimensional source and sink distributions method. For two-dimensional strip method, approximate body boundary condition can be applied, and a more simplified expression for pressure can be obtained. Some caution should be taken to properly include the effect of so-called "m term" in the formulation. Such an effect has been studied in Ogilvie and Tuck (1969), and Salvesen and Lin (1994). These hydrodynamic pressures are used to calculate the wave exciting force and added mass and damping coefficients to solve the equations of motion.

The quasi-static pressure due to body motions is given by:

$$P_S = -\rho g(\zeta_2 + x\zeta_6 - z\zeta_4)$$

where ζ_2, ζ_4, and ζ_6 denote the heave, roll, and pitch motions. Hence the resultant hydrodynamic pressure is given by:

$$P_T = P_I + P_D + P_R + P_S$$

An example of pressure distribution along the girth of the midship section is shown in Figure 7. The pressure amplitude can be as much as three times that on the weather side, according to Kim (1982). It should also be noted that the linear theory provides only the pressure distribution up to the mean waterline. Therefore, the pressures under the wave crest and trough need to be corrected to reflect more realistic behavior of the pressure near the waterline. The side pressure near the waterline thus must be specially treated as its intermittent characteristics have been described earlier in this paper.

Cargo Loads and Inertia Load

For commercial ships, the cargo load is one of the major components of the total load, and it has to be calculated accurately to predict the stress transfer function and fatigue life. Two major types of vessels, tankers and bulk carriers, are discussed in this section.

Tankers

The fluid pressure in cargo tanks and water ballast tanks, if not empty, needs to be determined and applied to the finite element method (FEM) structural analysis model. The

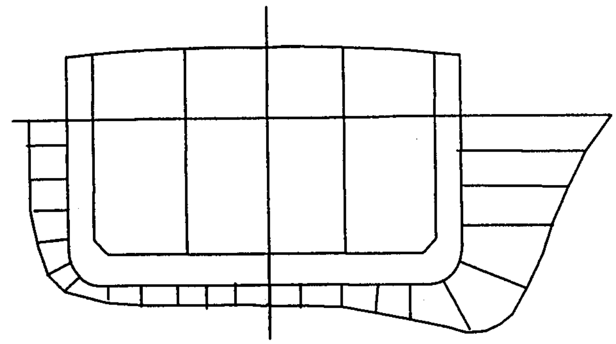


FIGURE 7 External total (hydrostatic and dynamic) pressure distribution.

internal tank pressure is composed of quasi-static and dynamic pressure components. The static component originates from gravity, considering the instantaneous rolled and pitched position of the ship. The dynamic component is due to the acceleration of the fluid caused by the ship motion in six degrees of freedom as discussed in the previous section. The total instantaneous internal tank pressure (quasi-static + dynamic) for each of the tank boundary points at the instant of time of interest can be calculated by:

$$P_t = P_0 + \rho h_1 \sqrt{(g_x + A_x)^2 + (g_y + A_y)^2 + (g_z + A_z)^2}$$

where P_t is the total instantaneous internal tank pressure; P_0 is the vapor pressure that is usually taken as the value of the pressure relief valve setting; ρ is the density of the liquid cargo or ballast; h_t is the total pressure head defined by the height of the projected fluid column in the direction to the total instantaneous acceleration vector, A_x, and so forth, are the instantaneous acceleration components relative to the vessel's axis system at a tank boundary point.

It should be noted that the time-dependent pressure component can be determined by subtracting the static component of pressure from the total instantaneous pressure,

$$P_d = P_t - (P_0 + \rho g h_0)$$

where h_0 is the internal pressure head when the vessel is in the upright position. Special attention needs to be paid to the case of double-side and double-bottom ballast tanks as to whether they are separated at the center line by water tight bulkheads (the J-tank) or not (the U-tank). For fatigue analysis, phasing between the roll, pitch, and accelerations should properly be taken into account. Impact load due to tank sloshing for partially filled cargo or ballast tanks is not discussed in this paper; the reader is referred to ABS (1992).

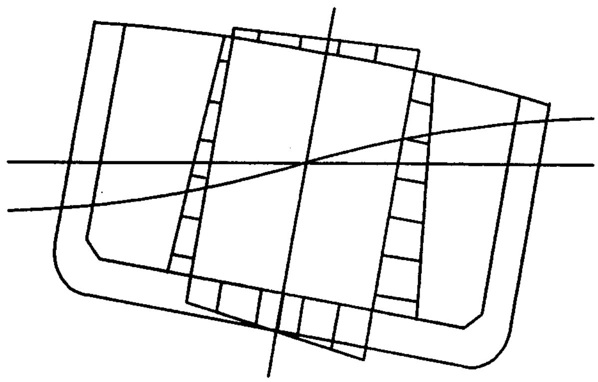


FIGURE 8 Internal tank pressure for tankers.

Bulk Carriers

The load from a bulk (granular) cargo is more complicated than that of the liquid cargo since both the coarseness of bulk cargo and rigidity of the tank structure can affect the bulk pressure. Contrary to the behavior of liquid cargo, the bulk cargo load is affected by the "angle of repose" and the presence of the tangential component of the load. In this paper a semi-empirical formulation is suggested as a practical engineering estimate.

The bulk load, expressed as force per unit area, can be decomposed into quasi-static and dynamic components. The quasi-static component is attributed to gravity, considering the instantaneous rolled and pitched position of the tank. The dynamic component is due to the acceleration from the ship's motion.

The quasi-static load is further decomposed into normal and tangential components relative to the structural surface. The normal bulk load on the panel of unit area is given by

$$N_S = \rho g h_e \left[\cos^2(\alpha - \theta) + (1 - \sin\alpha_0)\sin^2(\alpha - \theta)\right]$$

and the tangential component of the bulk load is given by:

$$T_s = -\rho g h_e \sin\alpha_0 \sin(\alpha - \theta)\cos(\alpha - \theta)$$

where ρ is the density of the bulk cargo, h_e is effective "head" of the bulk cargo, that is, the height of the projected bulk cargo column in the direction of gravity at the inclined vessel position; α_0 is the angle of repose. The preceding formulations are in accordance with the International Maritime Organization (IMO) publication "Code of Safe Practice for Solid Bulk

Cargoes." α is the angle measuring the slope of structural surface relative to the horizontal surface ($\alpha \rightarrow \pi/2$), and θ is the roll or pitch angle. These bulk load components are shown in Figure 9.

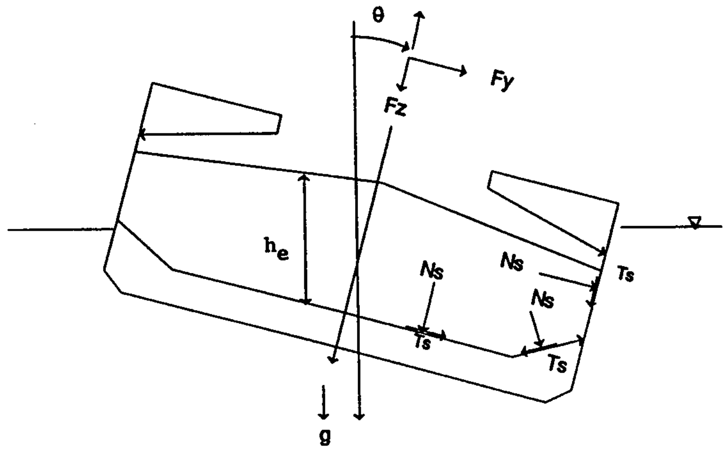


FIGURE 9 Normal and tangential components of static load exerted by bulk cargo.

The dynamic normal and tangential load components due to transverse acceleration are given respectively by:

$$N_T = C_N \rho A_t b \left[\cos^2(\pi/2 - \alpha) + (1 - \sin\alpha_0)\sin^2(\pi/2 - \alpha)\right]$$

$$T_T = C_t \rho A_t b \sin\alpha_0 \sin(\pi/2 - \alpha)\cos(\pi/2 - \alpha)$$

where b is horizontal distance from the centerline of the cargo hold to the point of interest on the hold boundary, A_t is the local transverse acceleration in ship fixed coordinate system. C_N and C_t are the reduction factors due to transverse acceleration for normal and tangential components, respectively. These factors, in general, depend on the type of bulk cargo and need to be determined based on reliable test data. If no such data are available, 0.35 for ore and 0.6 for grain may be used for both C_N and C_t. Similarly, those two load components due to vertical acceleration can be calculated (ABS, 1993a; Krivanec and Bea, 1992).

The total bulk load can be obtained by summing the quasi-static and dynamic components. The time-dependent part of the bulk loads can be obtained by subtracting the initial static components from the total load.

Bending and Torsion

Bending moments, both vertical and horizontal, are the global load effects traditionally considered as the dominant loads for assessing the global strength of ships. In the context of this paper relevant to fatigue, only the wave-induced dynamic bending is referred to. For a ship with a large opening like a containership, torsional loads also must be taken into consideration. In this regard, both the torsional moment about a certain reference point and the horizontal shear should be included. In what follows, the term "torsional couple" refers to the combined effect of the computed torsional moment and the added torsional moment, due to translating the sectional horizontal shearing force from the point of reference during computation to the sectional shear center.

The wave-induced load consists of the contribution of inertial (of the ship mass) load and of external pressure. These forces are superimposed (with proper phasing). The analytical method for calculation of wave-induced loads associated with a linear strip theory is well known, and it will not be further discussed in this paper.

A typical set of vertical bending moment transfer functions is shown in Figure 10. Vertical bending moment is always maximum in either head sea or following sea and is reduced as the ship turns toward beam sea. The vertical bending moment is employed as a criterion to gauge the adequacy of a ship's (global) longitudinal strength in classification rules. Such a notional value (or pair of values distinguishing hogging and sagging) has been unified by the International Association of Classification Societies (IACS).

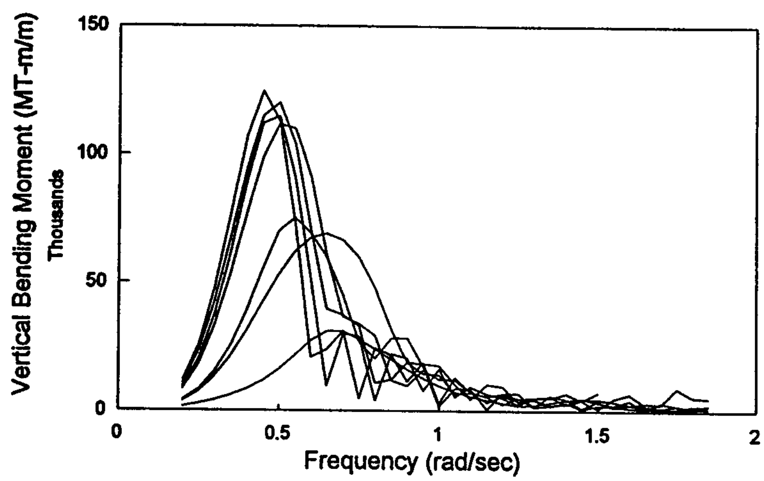


FIGURE 10 Vertical bending moment transfer functions.

For fatigue analysis, vertical bending is to be combined with other load effects as outlined earlier in this paper. In doing so, proper phase relation (see the following section on load combination) is essential. Thus, whenever feasible, all load effects should be obtained from a direct analysis instead of relying on rules-specified values. The latter should be employed only if necessary.

Similarly, typical sets of transfer functions associated with the horizontal bending moment, the torsional moment, and horizontal shear force are shown in Figures 11 to 13. The amplitude of the horizontal bending and torsional moment normally peaks in oblique seas. The spatial maxima of the torsional moment normally are located near the quarter lengths of the ship, whereas those of the bending moments normally occur in the midship region.

Hogging and sagging vertical bending moments are, in principle, not identical. However, linear theory is unable to produce the difference. This difference can be significant for

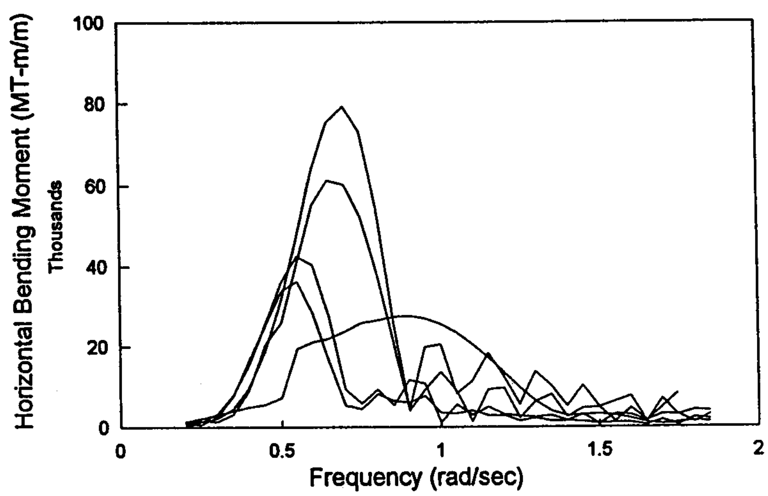


FIGURE 11 Horizontal bending moment transfer functions.

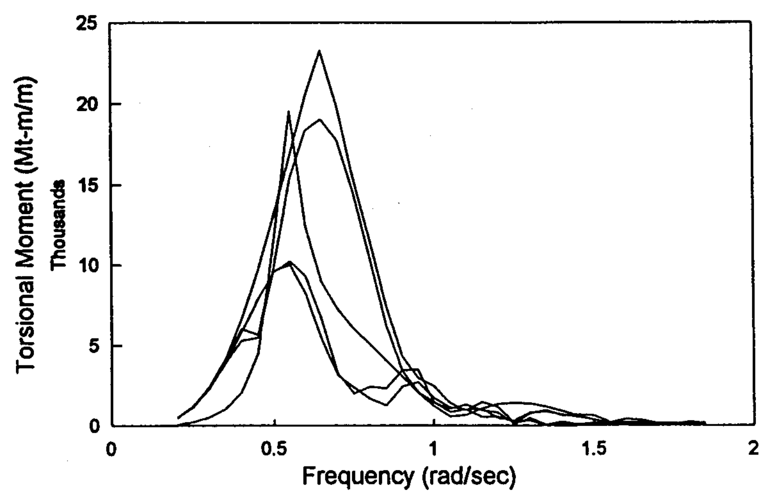


FIGURE 12 Torsional moment transfer functions.

non-wall–sided ships having large bow flare. Such an effect also becomes more apparent for modern designs of fine-form containerships and car carriers with low block coefficients. An example illustrating the difference is shown in Figure 14. Several analytical methods (Dawkins, 1986; Jensen and Pedersen, 1981) have been available to calculate the hogging and sagging bending moments using a nonlinear perturbation method. Also nonlinear time domain solution by Salvesen and Lin (1994) exhibits the difference clearly. Nevertheless, in the realm of fatigue analysis, where the emphasis is placed in the low to moderate seastates, such a distinction is typically of secondary importance.

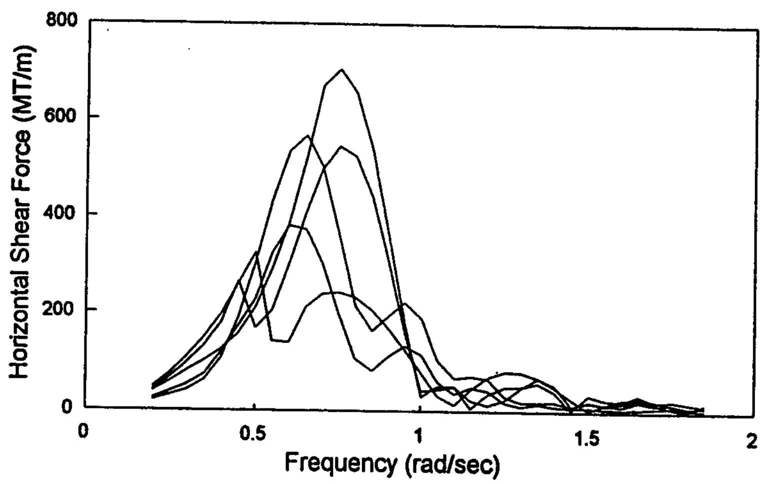


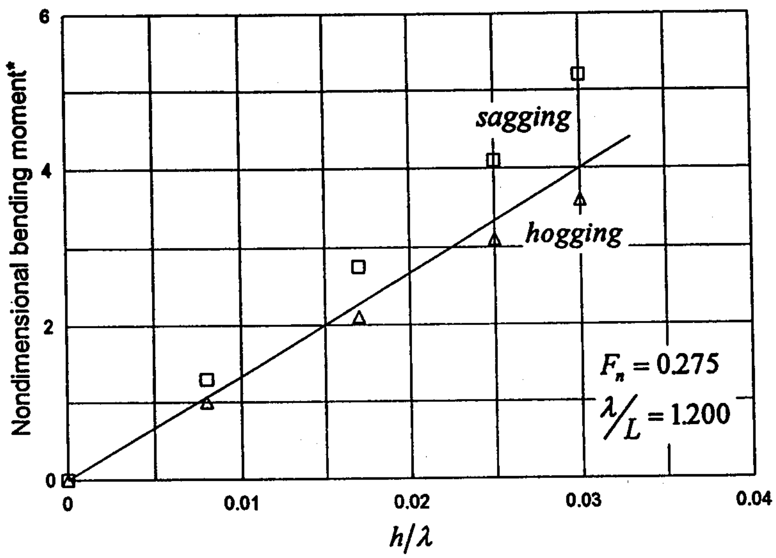
FIGURE 13 Horizontal shear transfer functions.

Loading to Finite Element Method Approach for Fatigue Analysis

When full spectral fatigue analysis is to be performed, first a stress transfer function, which represents the response due to a wave of unit amplitude from a given direction and for a range of frequencies, has to be calculated through particular structural analysis. The sophistication of analysis needed, for example, beam theory or finite element procedures, depends in part on the type of structural detail, type of loading and the stress gradients present. If the FEM approach is to be taken for purpose of obtaining the stress transfer functions, all of the dynamic load components of external pressure, cargo load, and inertial load on the light ship have to be applied to the FEM model.

If a partial model is to be used for FEM, then the shear forces and moments have to be applied also to the boundary ends of the model with proper phase. For proper phasing, often real and imaginary parts of the loads and accelerations are computed and applied separately. In a physical sense, the real and imaginary parts correspond to two wave systems that are 90 degrees out of phase relative to one another. With these two cases, a complete cycle of stress time history can be defined.

When all the dynamic load components are applied to the FEM model, care must be taken to assure that dynamic equilibrium is achieved within acceptable tolerance. The unbalanced forces in three global directions for each load case should be checked. For head-sea conditions—for which vertical force is the most dominant—as a rule of thumb, the unbalanced force should not exceed 1 percent of the ship weight. For oblique and beam-sea conditions, generally higher tolerance may be acceptable since balancing in the transverse direction could be a daunting task. This unbalanced load comes from many sources: certain limitations of linear theory; inconsistent



$$*\text{Nondimensional } BM = \frac{BM \times 10^{-4}}{\rho g B L^3}$$

FIGURE 14 Hogging and sagging bending moments.

methods used in calculating the hydrodynamic coefficients and wave diffraction force not by direct pressure integration; viscous-related damping linearized in the equation of motion, inconsistent modelling between the hydrodynamic and structural models; and other reasons. Furthermore, hydrodynamic pressure calculated from the coarse mesh hydrodynamic model often needs to be interpolated to obtain the pressure at the nodes of the structural FEM, which is much finer. An appropriate interpolation scheme should be used. Care again needs to be taken in the interpolation near the regions where the dynamic pressure changes sign.

There are no easy solutions to such balancing problems. What is needed is the development of a consistent, nonlinear, time-domain method using combined potential and viscous flow solution. This may be possible in the future. In the meantime, the unbalanced force should be minimized by a practical method of applying well-distributed additional accelerations over the entire model.

Load Combination and Fatigue Spectrum

The preceding discussions have been focused on the individual load effects that need to be considered in a fatigue assessment. It should be noted that, working together as a set of load effects, each load effect in the combination is usually less than the corresponding individual most-probable extreme value. A functional relationship (combination law) exists among these load effects that will produce the greatest (the most adverse) structural response, say, in stresses. In this regard, the scope of the ensuing discussion is limited to cases where all load effects are driven by but one environment disturbance; namely, the waves. For load combinations that are driven by multiple disturbances, such as waves, wind, earthquakes, and the like, more complex formulations, such as the theory of coincidence and the theory of vector outcrossing, may be required.

Various load effects can be combined only in terms of a common denominator. For example, consider a stiffened plate panel representing a portion of the side shell. Load effects, such as wave-induced vertical bending moment and pressure, cannot be combined directly. They have to be first mapped into some quantity, such as stress. In particular, the selection of the location and stress component can be guided by the location and potential orientation of the fatigue crack. However, once these combination coefficients are determined, they can be applied to load factors as direct multipliers of loads. The remaining parameter that governs the load combination is the phase of these load-effect variables. For this reason, combination of load effects should be at the level of the stress transfer function.

From a deterministic viewpoint, one could easily visualize the combination of two vectors (or complex variables) to form a third. Let these complex variables be denoted by $\boldsymbol{F_1}$, $\boldsymbol{F_2}$, and the resultant, $\boldsymbol{F_3}$, respectively. Moreover, let the polar angles of these variables be identified by ϕ_1 to ϕ_3, respectively. The well-known cosine law and the projective property in a Cartesian vector space are given by, respectively,

$$|F_3|^2 = |F_1|^2 + 2|F_1||F_2|^2\cos(\phi_2 - \phi_1) + |F_2|^2 \tag{8}$$

$$|F_3| = |F_1|\cos(\phi_3 - \phi_1) + |F_2|\cos(\phi_3 - \phi_2) \tag{9}$$

These are two forms of vector combinations, but they both represent the same combination. This basic geometric notion can be carried over to the spectral analysis formalism provided that one defines the inner product as:

$$[A,B] = \int_0^\infty A\bar{B}\,d\omega \tag{10}$$

where A and B are complex variables and $(\bar{\ })$ denotes the complex conjugate of ( ). This inner product operator is, of course, a bilinear operator.

Now let G_j be defined as:

$$G_j = H_j[S(\omega)]^{1/2}$$

where H_j is the transfer function of the variate X_j, and S denotes the ordinate of a wave spectrum. Moreover, if G_i, G_j, and G_k are defined similarly associated with the variates X_i, X_j, and their combination X_k such that

$$G_k = G_i + G_j \tag{11}$$

by the elementary property of the bilinear operator

$$[G_k, G_k] = [G_i, G_i] + \{[G_i, G_j] + [G_j, G_i]\} + [G_j, G_j]$$

By definition of the correlation coefficient and standard deviation, this relation becomes

$$\sigma_k^2 = \sigma_i^2 + 2\rho_{ij}\sigma_i\sigma_j + \sigma_j^2 \tag{12}$$

This is the spectral counterpart of the cosine law shown in Equation (8). The major difference between this pair of relations is that the contribution of the phase has been averaged out in the spectral form of the "cosine law" in a manner that defines the inner product. Equation (12), while applicable to the standard deviations exactly, can also be applicable to the variates themselves, approximately, as a potential law of load combination.

Next, it can be shown again by the elementary properties of a bilinear operator that

$$[G_k, G_i] = \sigma_i^2 + [G_j, G_i]$$

and, by way of symmetry,

$$[G_k, G_i] + [G_i, G_k] = 2\sigma_i^2 + 2\rho_{ij}\sigma_i\sigma_j$$

Since the left-hand side is equal to $2\rho_{ki}\varsigma_k\varsigma_i$, it follows that

$$\rho_{ki}\sigma_i\sigma_k = \sigma_i^2 + \rho_{ij}\sigma_i\sigma_j$$

Replace the index i by j, and add the sides to the preceding, it follows that

$$\sigma_k(\rho_{ki}\sigma_i + \rho_{kj}\sigma_j) = \sigma_i^2 + 2\rho_{ij}\sigma_i\sigma_j + \sigma_j^2$$

Since the right-hand side is equal to ς_k^2 it finally leads to:

$$\sigma_k = \rho_{ki}\sigma_i + \rho_{kj}\sigma_j \tag{13}$$

Equation (13) is the spectral counterpart of the projective property given by Equation (9). This relation can be generalized to cover the situation in which the $(n+1)^{th}$ variate, representing the combination of variates $X_1, X_2, \cdots, X_n$ such that, if

$$G_{n+1} = \sum_{i=1}^{n} G_i$$

then

$$\sigma_{n+1} = \sum_{i=1}^{n} \rho_{n+1,i}\sigma_i \tag{14}$$

Once again, it should be noted that Equations (12) and (14) are two alternative forms of load-effect combination relationships, but they are identical. The geometric meaning of this pair of relations is identical to that of Equations (8) and (9). It is evident that, for the combination of more than two random variates, the form shown in Equation (14) is preferred. The coefficient of combination, given by the specified correlation coefficients $\rho_{n+1,1}$, $\rho_{n+1,2}$, $\cdots$, $\rho_{n+1,n}$, can be used as partial-load factors.

While the notion of load-effect combination considered in this paper is intimately tied to phase, the exact phase relationships are lost after integration through the frequency range in the definition of the inner product. However, the statistical meaning of the phase remains in this formulation. This in fact relaxed the requirement of the one-to-one correspondence that maps a particular load effect to a particular stress component. If the analysis is linear or quasilinear (e.g., beam-column) the mapping can be written as

$$S = CX$$

where S and X are column vectors for stress range and load effect, respectively, and C is the mapping (full) matrix. In a linear system, the matrix C becomes diagonal, that is, the various load effects are decoupled and the transformation becomes

$$S_j = C_j X_j, \qquad j = 1,2,\cdots,n \tag{15}$$

Here the constants C_j are frequency dependent. Finally, if the correlation coefficients are used as load factors in the manner shown in Equation (14), the C_j's are independent of the frequency. For example, the stress-range transfer function due to external pressure differs from the factored external pressure only by a constant. (This is considered an approximation. It implies that the local stresses are not affected by the far-field pressure.) For deck structures, the stress transfer function can be approximated by scaling the vertical bending moment transfer functions. The process of scaling can be done with a single stress analysis load case. The notion of scaling the load-effect transfer function has been in use at ABS in the FLECS family of computer program modified for spectral fatigue analysis of ship structures (e.g., FLESS and FLIP). This has become a standard practice in the ABS DLA fatigue analysis.

The combination algorithm represented by Equations (14) and (15) is deceptively simple. Certain subtle points in their application can be observed from the following numerical example.

Consider a stiffened panel taken from the side shell of a tanker. The panel is situated on a patch 10 m to 15 m from the starboard bottom (covering five bays of longitudinals) with a width of 5 m. The panel is considered to be simply supported all around. A FEM is created, and the point chosen for calibration is at the top of the stiffener flange at the geometric center of the panel. The participating load effects are external pressure, vertical hull-girder bending, and horizontal hull-girder bending. A unit problem is first solved by applying a uniform pressure to the panel of intensity, 1 N/mm^2 (101.97T/m^2). The resulting von Mises stress at the reference point was found to be 916.48 N/mm^2 (ignore the fact that this is greater than yield). It follows that C_P = 8.988 $N/mm^2/T/m^2$. The other stress coefficients for the bending effects are simply the reciprocals of the section modulii, that is, C_V = 1.107E-5 and C_H = 7.553E-5. The unit for the bending coefficients is $N/mm^2/T\text{-}m$. Explicitly, the amplitude of the combined stress-range transfer function analogous to Equation (14) can now be written as:

$$S_C = \rho_{CV} C_V M_V + \rho_{CH} C_H M_H + \rho_{CP} C_P P$$

in which the stress coefficients, C's, are as given; and M_V, M_H, and P are the vertical bending moment, horizontal bending moment, and external hydrodynamic pressure, respectively. Although this expression does not exactly follow Equation (14), if the combined stress range is obtained from this expression, the corresponding root mean square value will satisfy Equation (14). Thus, this is not necessary, but it is sufficient, to ensure that the phase relationship of the various participating load effects is properly incorporated.

The correlated coefficients can be shown to be ρ_{CV} = 0.8016, ρ_{CH} = 0.7910, and ρ_{CP} = 0.9991 under the wave system of period 12 at beam sea. The beam sea stress transfer function coefficients can be obtained as:

$$\begin{aligned}
\rho_{CP} C_P &= 0.9991 \times 8.988 = 8.980 \\
\rho_{CV} C_V &= 0.8016 \times 1.107 \times 10^{-5} = 0.8874 \times 10^{-5} \\
\rho_{CH} C_H &= 0.7910 \times 7.553 \times 10^{-5} = 0.5974 \times 10^{-4}
\end{aligned}$$

One such set of coefficients is to be determined for each wave heading angle. These are the load factors to be applied to the boundary force resultants and the lateral pressure acting on the panel. Finite element analysis then determines the detailed stress ranges locally.

SUMMARY AND CONCLUDING REMARKS

This paper is written aiming at addressing the important aspects related to loads within the framework of fatigue analysis. The selection of the key issues is consistent with the theme of the symposium and the charges to the authors. In this regard, the emphasis of this paper is on the conceptual aspects underscoring "what constitutes a credible set of loading in the context of fatigue analysis." In addressing this particular question, issues such as linearity versus nonlinearity in the analysis that leads to the determination of the load effects, the structural analysis framework, are first discussed. Regarding nonlinearity, certain minimum considerations that are judged necessary in the context of fatigue analysis are identified. These effects include nonlinear roll (physical), truncation of hydrodynamic pressure acting on the side shell, and the effect of roll on internal quasi-static pressure (geometric). Other effects, such as side impact and bottom slamming, are considered less significant based on the present understanding of the subjects, while the final verdict awaits further study.

Another effect that could be of considerable importance but is not addressed at great length is wave directionality. A part of the reason for its lack of emphasis in this paper is that wave data carrying the directionality information are extremely rare. Recently emerging hindcast models, such as the GSOWM, may contribute in this regard. From a classification standpoint, vessels classed for worldwide unlimited services may not warrant consideration of wave directionality. In specific applications, such as fatigue consideration for a floating production, storage, and off-load system (FPSO), and for specific trade routes, such as TAPS, where wave directionality is well documented, its consideration is favored.

The authors are in favor of, for the purpose of fatigue analysis, employing the complete, unbiased, joint probability of significant wave height and characteristic period (viz., the wave-scatter diagram) over the more familiar design wave concept that is very prominent in the offshore engineering discipline. The basis for that argument rests on the fact that most floating structural responses are inertia-dominated rather than drag-dominated, as in the case of seabed supported structures. To this end, a full-fledged spectral fatigue analysis is preferred. For ease of application, design criteria, such as the recently completed SafeHull, ABS uses a "simplified" fatigue approach built around the representation of the long-term stress-range distribution by a Weibull distribution. In such a case, calibration of its parameters, by way of spectral analysis, has to be performed.

Some typical hydrodynamic formulations that can be applied in the course of load analysis are presented for the purpose of exemplifying what is considered acceptable. For this purpose, in-depth presentation of the hydrodynamic theory is not within the scope of this paper, nor should it be construed that these are the only acceptable formulations. In most situations, structural elements around fatigue-sensitive points are subjected to loadings attributed to multiple load effects. The paper rounds up the discussion on load analysis with a proposed algorithm for load combination in the realm of fatigue analysis. The geometric meaning of the proposed rule of combination is also given.

ACKNOWLEDGMENT

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REFERENCES

ABS (American Bureau of Shipping). 1993a. Analysis Procedure Manual for the Dynamic Loading Approach for Bulk Carriers. New York: American Bureau of Shipping Report.

ABS. Guide for the Fatigue Strength Assessment of Tankers. New York: ABS.

ABS. 1992. Scantling Requirements of Oil Tight Bulkheads of Partially Filled Tanks. American Bureau of Shipping, Technical Notes. Jersey City, New Jersey: ABS.

ABS. 1980. ABS/SHIPMOTION Program Theory. ABS Technical Report OE-5046-2. New York: ABS.

Barltrop, N.D.P., and M. Brooking, 1990. Structural analysis of FPS. Proceedings, 6th International Conference on Floating Systems held December 10–11 in London, England. London: IBC Technical Services Ltd.

Boitsov, G.V., O.M. Paliy, and M.A. Koudrin. 1994. A Study on Local Hydrodynamic Loads due to Wave Impacts Against Ship Bow (Slamming) and an Evaluation of the Slamming Effects Upon the Ship Hull Fatigue Endurance. Technical Report No. ABS/KR-3-01. New York: American Bureau of Shipping.

Chen, H. 1982. Effect of wave spectral shape and directional variability on the design and analysis of marine structures. Presented at BOSS Conference held May 1982 at Massachusetts Institute of Technology, Cambridge, Massachusetts.

Chen, Y.N. 1994. Full-Scale Measurement of Hatch Distortion, Motion, and Stress on Board the APL C-10 Containership PRESIDENT KENNEDY. ABS Technical Report No. RD-94001. New York: American Bureau of Shipping.

Chen, H., and H.H. Chen. 1979. The implementation of the 20 year hindcast wave data in the design and operation of marine structures. Paper No. 3664 in Proceedings Offshore Technology Conference held April 30–May 3 1979 in Houston, Texas. Jersey City, New Jersey: Society of Naval Architects and Marine Engineers.

Chen, Y.N., and S.A. Mavrakis. 1988. Closed-form spectral fatigue analysis for compliant offshore structures. Journal of Ship Research 32(4):297–304.

Chen, Y.N., and A.K. Thayamballi. 1991. Consideration of global climatology and loading characteristics in fatigue damage assessment of ship structures. Paper VI-D-1 in Proceedings Marine Structures Inspection, Maintenance, and Monitoring Symposium held March 18–19, 1991 in Arlington, Virginia. Jersey City, New Jersey: Society of Naval Architects and Marine Engineers.

Dawkins, R.A. 1986. Practical prediction methods for short and long term sagging and hogging moments. Presented at International Conference; Computer Aided Design, Manufacture, and Operations in the Marine and Offshore Industry, held September 1986 in Washington, D.C.

Friis-Hansen, P. and S.R. Winterstein. 1995. Fatigue damage in the side shell of ships. Journal of Marine Structures 8(6):631–655.

Hogben, N. 1985. Global wave statistics. British Maritime Technology, May.

Hogben, N., and F.E. Lumb. 1967. Ocean Wave Statistics. London: Her Majesty's Stationery Office.

Ikeda, Y., Y. Himeno, and N. Tanaka. 1978. Components of rolling damping of ship at forward speed. Journal of the Society of Naval Architects of Japan 143:113–125.

Jensen, J.J., and P.T. Pedersen. 1981. Bending moments and shear forces in ships sailing in irregular waves. Journal of Ship Research 25(4):243–251.

Kim, C.H. 1982. Hydrodynamic loads on the hull surface of a seagoing vessel. Presented at STAR Symposium, held April 19 1982 in Honolulu, Hawaii. Jersey City, New Jersey: Society of Naval Architects and Marine Engineers.

Krivanec, C.E., and R.G. Bea. 1992. Analysis of Static and Dynamic Pressures in the Holds of Bulk Cargo Carriers. Report to the American Bureau of Shipping. Berkeley, California: University of California.

Liu, D., J.S. Spencer, T. Itoh, S. Kawachi, and K. Shigematsu. 1992. Dynamic load approach in tanker design. Transactions of the Society of Naval Architects and Marine Engineers 100:143–172.

Little, R., and E. Lewis. 1971. A statistical study of wave-induced bending moments on large oceangoing tankers and bulk carriers. Transactions of the Society of Naval Architects and Marine Engineers 79:117–168.

MARIN (Marine Institute of The Netherlands). 1993. Analysis of Cross Flow Drag Coefficient from Segmented Model Test Results. Wageningen, Netherlands: Marine Institute of The Netherlands.

Ochi, M.K. 1964. Prediction of occurrence and severity of ship slamming at sea. Pp. 214–243 in Proceedings, 5th Symposium on Naval Hydrodynamics, held September 1–12, 1964 in Arlington, Virginia. Arlington, Virginia: Office of Naval Research.

Ochi, M.K. 1978. Wave statistics for the design of ships and ocean structures. Transactions of the Society of Naval Architects and Marine Engineers 86:47–76.

Ogilvie, T.F., and E.O. Tuck. 1969. A Rational Strip Theory of Ship Motion: Part I. Department of Naval Architecture and Marine Engineering Report No. 013. Ann Arbor, Michigan: University of Michigan.

Payer, H.G., and W. Fricke, 1994. Rational dimensioning and analysis of complex ship structures. Transactions of the Society of Naval Architects and Marine Engineers 102:395–418.

Salvesen. N., and W.M. Lin. 1994. Large Amplitude Motion and Load Program. SAIC Report 94/1110. Annapolis, Maryland: Science Applications International Corporation.

Schmitke, R.T. 1977. Prediction of wave-induced motions for hullborne hydrofoils. Journal of Hydronautics 11.

Stiansen, S.G., and A.E. Mansour. 1975. Ship primary strength based on statistical data analysis. Transactions of the Society of Naval Architects and Marine Engineers 83:214–243.

Winterstein, S.R. 1988. Nonlinear vibration models for extremes and fatigue. Journal of Engineering Mechanics 114(10):1772–1790.

Witmer, D.J., and J.W. Lewis. 1994. Operational and scientific hull structure monitoring on TAPS trade tankers. Transactions of the Society of Naval Architects and Marine Engineers 102:501–533.

Current and Future Directions in U.S. Shipbuilding

Paul A. Blomquist

ABSTRACT

The liberty ship failures of the World War II era spurred the investigation into the relationships between base metal properties, design, and workmanship, with the goal of improved reliability of ships. That goal was realized, and fortunately for the shipbuilding industry, the value of that effort was not seen as merely the solving of one problem, but as a means to continuously improve the entire process of ship construction. Since the formation of the Ship Structure Committee, the knowledge gained by a wide-ranging research program has found extensive and repeated use in many phases of ship design concepts, material properties and fabrication (Evangelista, 1992).

MATERIALS DEVELOPMENT FOR NAVAL AND COMMERCIAL SHIPS

When considering the development of specific steels for shipbuilding, most of the recent progress has taken place in materials for naval combatant ship fabrication. The reasons for this are obvious, especially the high cost associated with alloy development. The comparatively long time required for the testing and verification of a new alloy interferes with the shipbuilder's need to minimize design and build cycle time. With time being a major issue in the acquisition of a new vessel, ship designers must work within a context of proven materials, and seldom is a large vessel design predicated on growth of technology in the area of alloy development. Further, it is not profitable for a steel producer to develop new materials for commercial shipbuilding, considering the risks involved and the fact that global competitive pressure on the steel industry has forced American steelmakers to drastically reduce the scope of research facilities that were once arguably the best in the world.

Since the service requirements of combatant vessels are viewed as more severe, developmental testing of any specific alloy has been carried out to a much greater degree than the equivalent commercial product, and a new alloy may wait through years of testing before approval for use. Finally, allowable variations in product chemistry and processing practices may be more highly constrained than those allowed in the production of material to commercial standards.

While some may argue that whole structure for control of materials and fabrication methods for military vessels is too restrictive, proponents point out that the expected performance of a material or vessel is generally achieved, and few "surprises" have occurred. The recent program to develop and improve the high strength low alloy steels of 80 and 100 ksi yield strength under Mil-S-24645 is a case in point. These alloys are of some benefit to the shipbuilder since they can be fabricated more easily and reliably than the materials they replace. An unfortunate example of the opposite of this is the issue raised in the last decade, when one specific alloy grade and its developers were criticized in the national media (Wall Street Journal, 1984). Without debating the merits of those reports, the fact is clear that material development in the commercial sector is a risky business.

NAVAL VERSUS COMMERCIAL SHIP FABRICATION

Military Standards and the fabrication documents governing the practices for building combatant ships are comparatively more rigorous than their commercial counterparts and are generally more detailed. In contrast, materials that are allowed in the commercial arena generally are more loosely characterized than those required by combatant vessel specifications, and frequently, materials produced to other specifications may be substituted with greater latitude. Rules for building commercial ships provide more opportunity for innovation to the builder. This should not be construed to mean that commercial shipbuilding is in any way uncontrolled; nonetheless, a shipyard has a greater choice of methods, especially in the arena of high-deposition processes, than those available to the builder of naval combatants.

With this greater freedom, greater responsibility is placed on designer and builder of commercial vessels, and while there is greater room for creative approaches to improved production, there is a somewhat greater risk. Further, when sourcing materials to build a ship, a specific grade may not be readily available, and an "equivalent" material specification may be used. Typically, such substitution offers a savings of time and money, is quickly approved, and the substitute material performs as well as the originally specified material. In some cases, the substituted grade may not have the same testing requirements as the specified grade. Typically, the additional testing is performed to "upgrade" the substitute material to original requirement.

Conceptually, commercial vessels work in a less demanding service environment than their naval counterparts. But is this really true? Certainly, the idea of building to survive combat conditions supports that argument. But combat aside, commercial vessels are in more continuous service, spend little time in port, and may be in operation for a far longer period than a combatant; thus, the commercial ship may be subjected more cycles of stresses. Further, unexpected weather conditions may impose loads greater than those anticipated in the original design, or maintenance may be compromised, allowing more exposure to simultaneous mechanisms of damage.

FABRICATION—THE PRESENT

All ship fabrication in the United States, be it commercial or naval, requires that welders and welding procedures be qualified prior to use, so that intended weld metal and base metal heat-affected zone properties are achieved, and necessary quality levels are attained. All shipbuilding codes explicitly or implicitly require 100 percent visual inspection, and nondestructive testing of welds in critical areas or components. These items are fairly easy to document and regulate. American shipbuilders concede that the more difficult area to control is the detail of how well pieces fit together and how and when each weld is made. Fit-up inaccuracy usually implies that root openings of weld joints are greater than desired, requiring additional weld metal, which dramatically increases distortion, residual stresses, and increased the risk of weld-metal defects, such as slag inclusions, lack of fusion, porosity, and cracking. Excessive gaps may also encourage welders to use heat inputs greater than those allowed for certain materials, with risk of degrading required base metal properties.

Fabrication documents, both naval and commercial, place limits on the maximum root opening which may be welded "in place" without making some kind of repair. For fillet welds, a typical maximum gap is 3/16-in., which may be "repaired" by increasing the design fillet size by an amount which will ensure that the actual fillet weld throat is equal to the throat of the design weld size. Gaps greater than 3/16-in. require either weld metal buildup of one or both of the joint edges (up to a certain allowed maximum) or cropping out a portion of one of the members and inserting a patch of new material of dimensions such that the new gap is now within limits. If buildup is performed, it must be completed before the builtup members are joined together, so that shrinkage may occur freely to relieve residual stresses. Nonetheless, shrinkage will be resisted by base metal to a certain degree, and warpage may still occur. Patching reduces the amount of welding, but necessitates that instead of one joint, as originally intended (e.g., a fillet weld), two joints must be welded; one of them a full penetration butt weld. It is obvious that the cost to repair excessive root openings is considerable; what is less obvious is that openings that fall within the currently allowed tolerance, while incurring less cost to correct, still have an impact on the overall distortion of ship structural assemblies, which are made into larger units and then must be joined to other large assemblies. Typically, extra material is added to one edge of each major unit; scribing and trimming this stock adds time and cost to a critical phase of the shipbuilding cycle and may, itself, be a source of further inaccuracy. Certainly one of the areas of greatest attention in shipyard improvement programs is developing the ability to cut all plate "neat," or to the final dimension (including exact allowances for accumulated weld shrinkage) at the initial numerically controlled burning operation.

Even when parts fit together ideally, changes in welding sequence (the pattern with which welds are made with respect to other welds), or changes in the number of passes or welding parameters (such as amperage or voltage), can affect the degree to which one panel distorts compared with a similar panel. Thus, welding distortion and residual stresses are never assumed to be repeatable or predictable at present.

There can be considerable cost to correct welding-induced distortion (Kirk et al., 1995), and there are many efforts to characterize and control the elements which cause it. A frequently used method is restraint: this may be in the form of strongbacks across joints, restraining fixtures, or weight blocks, among others. These methods can be expected to result in high levels of residual stress since they force stretching of the cooling weld, and, since these restraints may be placed in varying positions as parts are fit, it is again difficult to predict the true pattern of residual stresses in a finished structure. Alternative methods such as precambering, often called "back-setting," are being explored. In such presetting, a part is set up for welding in an initial position, offset from the actual dimension by the amount of weld shrinkage expected. Weld shrinkage then moves the part into the required final position. Intuitively, this approach should give low values of residual stresses. Obviously, all of the variables, especially fit-up, weld parameters, and welding sequence must be controlled to the extent necessary for the expected direction and amount of shrinkage to occur.

While backsetting is being increasingly used, the major areas of distortion control have been welding sequence (Doerksen, 1993), such as welding all vertical joints in a structure before welding horizontal joints (ANSI/AWS, 1993), and the increasing use of semiautomatic and mechanized welding processes and equipment. Both mechanization and semiautomatic welding offer generally lower heat input and smaller welds compared to manual welding and improved uniformity, consistency, and therefore predictability, but usually only to a small degree.

Additionally, American shipbuilders have been investing in accuracy control programs to improve the quality and consistency of fit-up. One area of concern is that hot-rolled structural shapes are allowed dimensional variation that is so large that ship structures using them cannot be fit accurately enough. Some foreign yards are currently fabricating their own tee sections so that dimensional accuracy of structural shapes can be controlled. It has been said that fit-up tolerances of plus or minus 0.5 mm are achieved in many foreign shipyards (Jensen, 1995).

Inaccuracy is a major obstacle to large-scale implementation of robotic welding. A further problem is the lack of conceptual design standards that favor automation. Intersections of structure may have several weld joints with difficult access for manual welding, requiring the use of mirrors and arduous welding positions. Joints such as this defy automation, and they are significant sites for rework. Rework, of course, can increase distortion and residual stresses. It has been stated that a major competitive advantage of foreign shipyards is superior design (Daidola, 1994).

While this is an easy statement to make, one sees little concrete representation of what constitutes advantageous design with corresponding examples of bad design. In fact, the whole issue of improving productivity often stumbles on a sort of "technology trap," in which technology growth alone is envisioned as the entire solution to a problem for which the technical aspects (compared with, say, the management aspects) may form only a small part of the whole. In a recent analysis of the production of tees by I-beam deflanging, it was noted that only 4 percent of the total cost of the operation was actual burning time (Blomquist, 1995), and thus, any significant increase in cutting speed would have a small effect on total cost. In that particular case, material handling was seen to be a major element, and strategies aimed at reducing handling would have a much bigger payoff.

A major challenge in domestic shipyards is simply keeping the welding arc going. "Operator factor," sometimes called arc factor, is the percentage of arc time to total time charged to the job. Actual comparisons of domestic versus foreign operator factors do not appear in the literature. Mechanization can increase operator factor, but another aspect of the technology trap is the assumption that the "new machine" will always be operated at a higher arc factor than the process it replaces. This is mostly true, but if low arc factor is the result of poor management, the benefits of improved technology may not be fully realized.

A third aspect of the technology trap is the potential for merely relocating costs versus really reducing costs. For instance, the current generation of "dumb robots" available in the United States requires so much programming effort that only highly repeatable parts can be cost-effectively processed. Welding cost would be relocated as programming cost for the vast majority of other operations. Reduction in programming time through the use of macros (preprogrammed subroutines) and simulations and real-time seam tracking can help, but these methods are still not demonstrably reliable or widely available in America. Complex details and difficult access compound the problems of increasing automation.

This should not be construed as an argument against technology: in a well-managed operation, as other costs are controlled, the process cost becomes a more significant part of the whole, and technical improvements to process will have greater impact. Certainly design and planning play an important part. If the work can be kept simple, accessible, and as much as possible in the ideal position (flat or horizontal) for welding, all processes, including manual, mechanized, and robotic will benefit. As design tools become more capable and easier to use, it may finally become possible to analytically determine an accurate cost to produce a certain detail of a ship in addition to the more traditional aspects such as strength, or fatigue life. Currently, design software can't assess producibility, but there needs to be the ability to make rapid "what if" iterations of a given design so that the best of several alternatives can integrate all of the functional needs of a vessel with the economic need to build it quickly and, yes, cheaply.

GLOBAL COMPETITION—THE FUTURE

To be competitive in the future world shipbuilding market, U.S. yards must cut costs and improve delivery schedules. Cost control will require greater automation; many overseas shipyards have either accomplished or are in the process of implementing this strategy. Obstacles to automation, such as poor quality of fit-up, will require greater emphasis on accuracy. Smarter robots with real-time adaptive control and adaptive joint-fill strategies may ease this transition, but such technology is still too expensive to be within reach, and reliability has not yet been demonstrated. A number of robot development programs aimed at closing this gap are currently underway. At the same time, ship designs need to accommodate greater automation, by using simpler details, with greater accessibility than previously afforded.

Materials need to be tolerant to fabrication by high-productivity processes, but as mentioned earlier, material development will most likely take place only in a context of consortium sponsorship or government funding. If material development can occur, welding filler metal development must be carried out in parallel. Too often, material development projects have given us improved steels that can be more reliably fabricated, but development of filler metals is an afterthought, and the full advantage of improved fabricability is seldom realized in production.

Certainly, increased automation is the single most likely change to occur in American shipbuilding in the next few years. Large-scale integration of panel fabrication and robotic welding equipment is planned or underway in several major U.S. yards. This degree of automation will make possible a level of uniformity and quality that has been heretofore unachievable and can have a positive impact on control of distortion and residual stresses. To achieve this, the accuracy problems must be solved, but the ultimate solution will probably be a blending of improved accuracy and smarter robots (Reeve and Rongo, 1995).

The implications of a move to large-scale robotic welding need to be considered in the light of some studies that show that automatic welds have lower fatigue lives than those produced manually. Otegui (1989) demonstrated that transversely loaded groove tee joints welded with manual shielded metal arc welding (SMAW) performed better that those welded with the submerged arc welding (SAW) process. As reported, manual welds had a more erratic contour at weld toes, allowing multiple initiation sites for small cracks that were randomly oriented and, thus, took longer to link up into major cracks. The automatic or mechanized welds typically had smooth weld edges and minor undercutting (well within visual inspection acceptance criteria), but crack-initiation sites were all aligned within this undercut (due to the uniformity) and, thus, the development of substantial cracks occurred at lower stresses and fewer cycles than experienced by the manual welds. Currently, most U.S. shipyards use some kind of mechanized welding process for welding longitudinal stiffeners to decks; not all shipyards use SAW as tested by Otegui. Further, the welds tested by Otegui were full penetration groove tee welds, not fillet welds with intentionally unwelded root areas, which comprise the greatest length of any particular type of weld seen in ships. Finally, loading for these welds is usually longitudinal, not transverse.

The envisioned expansion of automation will include the welding of components that are loaded transverse to welds. Welding methods for joining the intersections of webs, bulkheads, and other structure are currently not automated, and this application is a major focus for increased use of robotic welding. Otegui suggests that automatic welding may make use of oscillation of the arc to interrupt the continuity of the weld toe. Many robotic welding implementations currently use through-arc seam tracking, which oscillates the arc through a distance to find the joint sidewalls and, thus, adaptively compensates for minor variations in weld path. A downside to such oscillation is that fillet-weld leg length is usually forced to be greater than otherwise achievable and may exceed design size, increasing risk of distortion and higher residual stresses.

In contrast to this, Brooke (1988) demonstrated that small, mechanized laser welds made from one side, performed better in fatigue than larger SMA welds made from two sides. As above, only one type of weld was compared to only one other type of weld, and a limited number of tests were performed, although more loading modes were evaluated.

Clearly, some review and additional research into loading modes and failure patterns is needed. A thorough evaluation of the fatigue performance of robotically produced gas–metal arc welds should be made because this is expected to be a major growth area in shipbuilding.

Even given the considerations for fatigue performance, robotic welding will offer a significant improvement in overall uniformity and weld metal quality. The positive benefit of this uniformity is that weld shrinkage will be repeatable and therefore predictable, so that realistic shrinkage allowances can be factored into detail designs. Consistent quality will reduce rework and provide for better performance of structures in general. As "first-time quality" increases, the need for rework inspection is reduced, so that inspection costs can be reduced, and inspection can focus on more critical areas.

SUMMARY

The irony that pervades most of this discussion is that many overseas shipbuilders have already resolved the issues of design for producibility and accuracy control. These improvements stand on their own in allowing good ships to be built to aggressive schedules at reduced cost. But because they provide more nearly ideal weld joint positions and fit-up accuracy, they enable the increased use of automation, and they allow welding sizes to be kept closer to design sizes, reducing distortion and the buildup of residual stresses. The economic message, "improve or go out of business," is now providing the impetus for American shipbuilders to look beyond the traditional to solve what were once only viewed as technical problems.

To accomplish all of this, we need to make major changes in the way we design and build ships, and all of the active shipbuilders in the U.S. are committed to this type of improvement as a necessity for continued operation. In summary; the relationship of design, material, and process must become more integrated, interactive, and flexible than it has been in the past.

REFERENCES

ANSI/AWS (American National Standards Institute/American Welding Society). 1993. Guide for Steel Hull Welding. ANSI/AWS D3.5-93. Miami, Florida: American Welding Society.

Blomquist, P. 1995. Tee-beam manufacturing analysis: Producibility of panel stiffening elements. In Proceedings of the 1995 Ship Production Symposium held January 25–27, 1995 in Seattle, Washington. Jersey City, New Jersey: Society of Naval Architects and Marine Engineers.

Brooke, S.J. 1988. High power laser beam welding of T joints in ship production. Transactions of the Northeast Coast Institution of Engineers and Shipbuilders 104(3).

Daidola, J.C., J. Parente, and W.H. Robinson. 1994. Hull Structural Concepts for Improved Producibility. Ship Structure Committee Report SSC-377. Washington D.C.: Ship Structure Committee.

Doerksen, R. 1993. Weld Shrinkage Study. National Shipbuilding Research Program, Transportation Research Institute. NSRP No. 0397. Ann Arbor: University of Michigan.

Evangelista, J., ed. 1992. The silent partner of progress: The Ship Structure Committee. Surveyor 23(2).

Jensen, H.K. 1995. Total Transportation Systems Corporation. Personal communication to Paul A. Blomquist. January 25, 1995.

Kirk, M., C. Conrardy, and P. Michaleris, 1995. Control of Residual Stresses and Distortion in Thin Section Panel Fabrication—Review of Phase I Studies. MT project M0649, NJC TDL No. 93-04, EWI Project J8662. Columbus, Ohio: Edison Welding Institute.

Otegui, J.L., D.J. Burns, and U.H. Mohaupt. 1989. Fatigue crack initiation from defects at weld toes in steel. Journal of Pressure Vessels and Piping 38:385–417.

Reeve, R., and R. Rongo. 1995. Shipbuilding robotics and economics. Proceedings of the 1995 Ship Production Symposium held January 25–27, 1995 in Seattle, Washington. Jersey City, New Jersey: Society of Naval Architects and Marine Engineers.

Wall Street Journal. January 16, 1984. How U.S. Steel reassured customers even while it reformulated T-1 steel. p. 14.

PAPERS PRESENTED AT THE FABRICATION AND REPAIR WORKSHOP

Catalogue of Builder's Practice for Fabrication

Young-Min Lee

INTRODUCTION

Recently hull structures have become larger and more complex, and the operating conditions in seagoing have been worse. Structures are to be constructed to resist the design load that is normally greater than the actual load. However, cracked structures may fail at the lower load rather than the design load. These failures are mainly due to a fracture.

Many studies indicate that both design fault and steel quality were the primary factors that contributed to the brittle fracture in the structure. The improvement of structural materials and design/fabrication practice have made the higher allowable stress to be used in the structural design. These factors have rather increased the possibility of fracture failure due to high stress, as the fracture still remains a main problem for designers to solve.

Most of the faults made during fabrication are controlled and corrected as proper method in accordance with the builder's own practice. In this regard, the review of current builders' practices is very important and remarkable for prevention of fracture in ship structure.

The data have been subdivided into a number of different groups that are identified by the activity performed during design and fabrication. A total of seven builder's practice groups were established and considered specially pertinent to the fracture in ship's fabrication. These groups are based on Samsung Heavy Industry's practice due to limited information about other builders' practices.

PRESENTATION FORMAT

The catalogue includes the following information for each item in the fabrication of ship structure:

- sketch/details illustrating the items of fabrication fault causing the fracture
- lists of the tolerance limits and descriptions about the correction or repair method of each fault
- some considerations on prevention of fracture in the future

TABLE 1 List of Builders' Practice Grouping

Group No.	Description of Practice Group
1	Material
2	Welding
3	Details (misalignment and gap)
4	Cutting
5	Fabrication of built-up member
6	Assembly block dimension tolerance
7	Deformation

TABLE 2 Index of Activities in Fabrication—Group 1, Material

Object No.	Title	Fig. No.
1	Steel grade selection at design stage	1
2	Steel handling	2
3	Steel flaw (pit/flaking)	3
4	Lamination	4

TABLE 3 Index of Activities in Fabrication—Group 2, Welding

Object No.	Title	Fig. No.
1	Shape of bead	5
2	Preparation of weld/preheating	6

TABLE 4 Index of Activities in Fabrication—Group 3, Details (Misalignment and Gap)

Object No.	Title	Fig. No.
1	Misalignment (1) Between components on each side of a through-going component (2) In block/section joint (3) In joints between longitudinal profiles	7
2	Gap (1) Overlap brackets (2) Before welding of overlap (3) Before welding of fillet joint (4) Before welding of joint between bracket/intercostal and stiffener (5) Before Welding of profiles to webs in way of cutout	7–8
3	Considerations on prevention of fracture	10

TABLE 5 Index of Activities in Fabrication—Group 4, Cutting

Object No.	Title	Fig. No.
1	Notch/dimension	11

TABLE 6 Index of Activities in Fabrication—Group 5, Fabrication of Built-up Member

Object No.	Title	Fig. No.
1	Flanged longitudinal	12
2	Flanged bracket	

TABLE 7 Index of Activities in Fabrication—Group 6, Assembly Block Dimension Tolerance

Object No.	Title	Fig. No.
1	General structure (flat/curved)	13

TABLE 8 Index of Activities in Fabrication—Group 7, Deformation

Object No.	Title	Fig. No.
1	Standard range of various areas (including measurement method of deformation)	14

<table>
<tr><td colspan="2">GROUP ITEM : MATERIAL
OBJECT NO. 1: STEEL GRADE SELECTION (AT DESIGN STAGE)</td><td>GROUP NO. 1</td></tr>
<tr><th>DETAILS</th><th colspan="2">CORRECTION/REMARKS</th></tr>
<tr><td>Design policy for the application of HT steel (location, ratio, etc.)</td><td colspan="2">The main purpose for adopting higher tensile (HT) steel is to reduce hull scantlings by its material factor having the equivalent strength. Its advantages can be obtained where high tensile stresses mainly occur.

However, reduced scantlings with HT materials cannot act as the equivalent of mild steel for corrosion, buckling, and fatigue strength.</td></tr>
<tr><td colspan="3">CONSIDERATIONS ON PREVENTION OF FRACTURE
1. HT steel application for standard design is for longitudinal members in way of the cargo tank region, especially upper and lower parts, where hull girder stress is mainly governed.
2. Structural members related to local strength are recommended to be mild steel with thicker scantlings.
3. Forebody, engine room, after hull, and deckhouse construction are also recommended to be mild steel to increase the inertia against impact, vibration, and so forth.</td></tr>
</table>

FIGURE 1 Material—Steel grade selection.

GROUP ITEM : MATERIAL OBJECT NO. 2: STEEL HANDLING	GROUP NO. 1

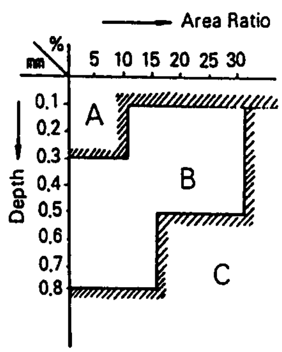
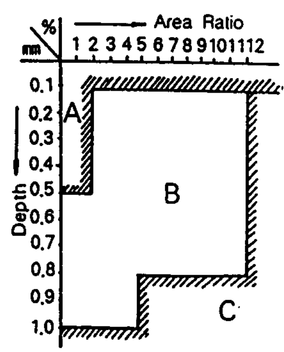
ITEMS	DESCRIPTION
MILL (THICKNESS) TOLERANCE	- Actually builders are not concerned with the mill tolerance because the builder uses steel that has received certification issued and inspected by a classification society.
CONTROL OF STEEL GRADE	- Steel grades, such as HT32/HT36 or mild steel having grade of A, B, D, or E are distinguished by identification letters, marks, or colors in order to provide the materials as designed in accordance with the shipbuilder's own method.

CONSIDERATIONS ON PREVENTION OF FRACTURE

1. Builders also have to check the mill tolerance as per guidance for allowance issued by classification societies.
2. Builders have to establish the procedure for application of steel as having the correct scantlings as defined in the design stage.

FIGURE 2 Material—Steel handling.

GROUP ITEM : MATERIAL **OBJECT NO. 3:** SURFACE FLAW (PIT/FLAKING)	**GROUP NO. 1**

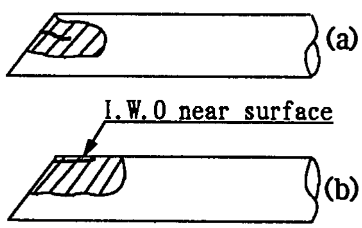
DETAILS	**CORRECTION/REMARKS**
• Grade of pit  • Grade of surface flaking 	1. Grade A is to be considered so slight that any repair is unnecessary. 2. The area ratio of pit denoted as D in the abscissa means the percentage of pitted areas where surface appearance is unsatisfactory for practical use. 3. Repair method of surface flaw where : d = depth of defects t = plate thickness $d < 0.07t$ or $d \leq 3mm$ - removed by grinding $0.07t \leq d \leq 0.2t$ of $d > 3mm$ - grinding followed by welding

CONSIDERATIONS ON PREVENTION OF FRACTURE

1. Surface flaws make the material that is used equivalent to a lesser scantling. Its result is that the life of the ship shall be shortened, even if the plate has a corrosion margin.
2. The current tolerance for repair seems to be too excessive in allowance. It is recommended that the tolerance be reduced.
3. Surface flaws on the outside of a ship will be smoothly ground, regardless of A, B, or C grade, for good looks.

Figure 3 Material—Surface Flaw (pit/flaking).

GROUP ITEM : MATERIAL OBJECT NO. 4: LAMINATION	GROUP NO. 1

DETAILS	CORRECTION/REMARKS
1. LOCAL LAMINATION 	In cases where the range of lamination is limited, it can be chipped out and built up by welding as shown in (a). In cases where the range of lamination is limited and near the plate surface, it is preferable to make the built- up welding as shown in (b).
2. SEVERE LAMINATION	It is recommended to exchange the plate locally. The standard minimum breath of the plate to be exchanged: Shell and Strength Deck under large constraint - 1600 mm not under large constraint - 800 mm other structural members - 300 mm

CONSIDERATIONS ON PREVENTION OF FRACTURE

1. In way of the fillet joint of a hopper top plate between the hopper sloping bulkhead and longitudinal bulkhead, the higher tensile stress in the vertical direction occurs during the fully loaded condition of the cargo tank, and also excessive welding at both surfaces of the plate are to be provided. For the prevention of lamination on that detail, it is recommended that "z-quality" steel be used at the design stage.
2. It must be carefully examined whether the procedure is acceptable or not in cases where the degree of the lamination is more severe and defective.

FIGURE 4 Material—Lamination.

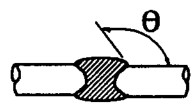
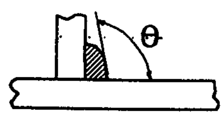
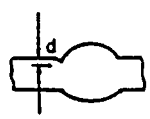
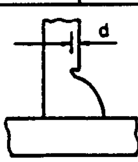
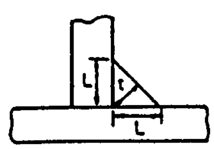
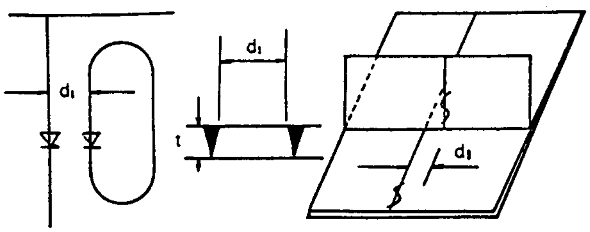
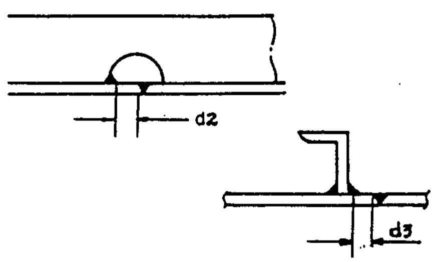
GROUP ITEM : WELDING **OBJECT NO. 1:** SHAPE OF BEAD					**GROUP NO. 2**
ITEM	**DETAILS**			**TOLERANCE LIMITS**	**REMARKS**
Over-lap	Butt			$\theta \geq 90°$	In cases where θ is less than 90°, it shall be repaired by grinding or welding to make $\theta \geq 90°$. (Carefully avoid short bead for higher tensile steels.)
	Fillet				
Under-cut	Butt		Skin plate and face plate within 0.6L amidship	$d \leq 0.5$ (over 90mm continuous)	To be repaired by using a fine electrode. (Carefully avoid short bead for higher tensile steels.)
			Other	$d \leq 0.8$	
	Fillet			$d \leq 0.8$	
Leg length and throat length				$L \geq 0.9 \times$ (specified leg length) $t \geq 0.9 \times$ (specified throat thickness)	In cases where "L" or "t" is less than tolerance limits, weld over it. L = leg length t = throat thickness
CONSIDERATIONS ON PREVENTION OF FRACTURE Above faults in shape of weld beads could cause fracture in the structure by the separation of a solid body into two or more parts under a mechanical load. For the solution of this problem, good workmanship and quality control by the builder, as well as careful inspection, would be required.					

FIGURE 5 Welding—Shaping of bead.

GROUP ITEM : WELDING **OBJECT NO. 2:** PREPARATION OF WELDING/PREHEATING		**GROUP NO. 2**
DETAILS	**Tolerance Limit**	**REMARKS**
1. PREPARATION OF WELDING o Distance between two butt welds 	$d_t \geq$ 30mm $d_t >$ 250mm	at $t <$ 15mm at $t \geq$ 15mm
o Distance between butt and fillet welds 	d2 > 5mm d3 > 10mm d2,d3 ≥ 0mm	for main structure for other structure
2. PREHEATING		
Kind of steel	Limit of atmospheric temperature	Min. preheating temperature
- Grade A,B,D,E of mild steel - TMCP 50 HT ($C_{eq} \leq$ 0.36%	T ≤ −5°C	Min. 50°C
- TMCP 50 HT ($C_{eq} \leq$ 0.36%	T ≤ −5°C	Min. 50°C

CONSIDERATIONS ON PREVENTION OF FRACTURE

1. Preheating:
   - When the C_{eq} of touched metals are different, the higher C_{eq} has to be taken.
   - When the atmospheric temperature at immediate vicinity of weld is below −5°C, welding shall be prohibited.

FIGURE 6 Welding—Preparation of Welding

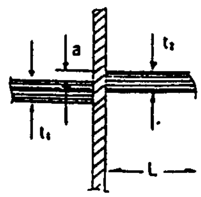
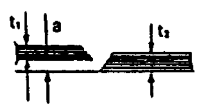
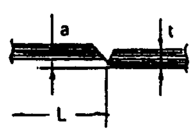
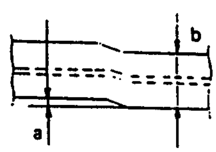
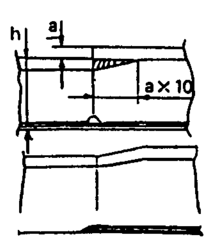
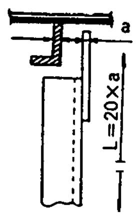
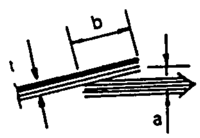
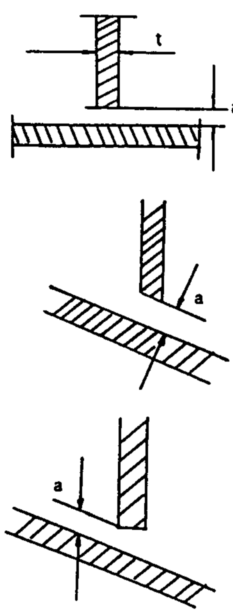
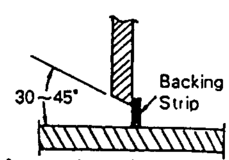
GROUP ITEM : DETAILS **OBJECT NO. 1:** MISALIGNMENT				**GROUP NO. 3**
Object		**Detail**	**STD Range**	**Corrections/Remarks**
(1) Misalignment between components on each side of through-going component		 $t_1 \leq t_2$	$a \leq t_1/2$ but $a \leq t_1/3$ to be applied for strength member	(1) $a \geq t_1/2$ Position to be adjusted. Adjustment length $L \geq 30 \times a$ (2) $t_1/3 < a \leq t_1/2$ (In the case of a strength member) The leg length to be increased by 10% of misaligned amount.
(2) Misalignment in block/section joint. (Shell plating, deck, bottom, inner bottom, bulkhead, etc.)		 $t_1 \leq t_2$	Others $a \leq 0.2t_i$, but max. 3mm	The plates are to be readjusted.
Misalignment in joints between longitudinal profiles.	Web of longitudinal T and bulb		$a \leq 0.2t$, but max. 3mm	If $a > 0.2t$ or $a > 3mm$, position to be adjusted. Adjustment length $L \geq 30 \times a$
	Flange of T-profiles		$a \leq 0.4b$, but max. 8mm	If $a > 0.04b$ or $a < 8mm$, position to be adjusted. Flange to be released not less than $30 \times a$.
	Height of T-bar, L-bar, and bulb		$a \leq 3$	(1) $3 < a \leq 6mm$ Make a smooth shape by buildup. (2) $a > 6mm$ Cut the web off to $10 \times a$, and weld after knuckling the flange.

FIGURE 7 Details—Misalignment.

<table>
<tr><td colspan="3">GROUP ITEM : DETAILS
OBJECT NO. 2: GAP</td><td>GROUP NO. 3</td></tr>
<tr><th>Object</th><th>Detail</th><th>STD Range</th><th>Corrections/Remarks</th></tr>
<tr><td>(1) Overlap brackets</td><td>
</td><td>a ≤ 3mm</td><td>(1) 3mm < a ≤ 6mm
Adjust before welding
(2) a > 6mm
Release and adjust in length L ≥ 20 × a.</td></tr>
<tr><td>(2) Gap before welding</td><td>
</td><td>a ≤ 3mm
b = 2t + 25mm
(b ≥ 50mm)</td><td>(1) If 3mm <a ≤ 5mm, increase leg length.
(2) If a > 5mm, overlap to be knuckled.</td></tr>
<tr><td>(3) Gap before welding of fillet joint</td><td>Fillet Weld

</td><td>a ≤ 3mm</td><td>(1) When 3mm <a ≤ 5mm, the weld length is to be increased as much as the increase of gap opening exceeds 3mm.
(2) When 5mm <a ≤ 16mm, welding with bevel preparation.
- Welding with bevel preparation to make bevel edge of web to 30° ~ 45°; attach the backing strip, and remove it after welding.


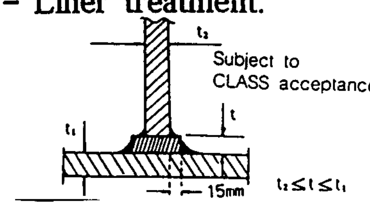
- Liner treatment.

</td></tr>
</table>

FIGURE 8 Gap.

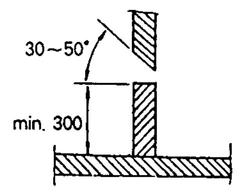
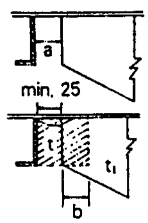
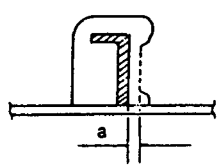
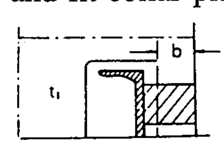
<table>
<tr><td colspan="3">GROUP ITEM : DETAILS
OBJECT NO. 2: GAP</td><td>GROUP NO. 3</td></tr>
<tr><th>Object</th><th>Detail</th><th>STD Range</th><th>Corrections/Remarks</th></tr>
<tr><td>(3) Gap before welding of fillet joint</td><td></td><td>a ≤ 3mm</td><td>(3) When a > 16mm
—Partial renewal.
</td></tr>
<tr><td>(4) Gap before welding of joint between bracket/ intercostal and frame, beam, stiffener, etc.</td><td></td><td>a ≤ 3mm</td><td>(1) If 3mm <a ≤ 5mm, the weld-leg length to be increased as much as the gap opening exceeding 3mm.
(2) When 5mm < a <10mm, fit the collar plate as shown in the detail:
$t \geq t_1$
b =b2t + 25mm (min. 50mm)</td></tr>
<tr><td>(5) Gap before welding of profiles to webs, floors, etc. in way of cutout.</td><td></td><td>a ≤ 3mm</td><td>If a > 3mm, then apply the following, subject to class acceptance:
(1) When 3mm <a ≤ 5m, weld length to be increased as much as the gap opening exceeds 3m.
(2) When 5mm < a ≤ 10mm, nib to beveled, and built up by welding as for gap openings.
(3) When a > 10mm, cut off nib and fit collar plate.

b = b2t + 25mm (min. 50mm)</td></tr>
</table>

FIGURE 9 Gap.

GROUP ITEM : DETAILS OBJECT NO. 3: MISALIGNMENT AND GAP	GROUP NO. 3

CONSIDERATIONS ON PREVENTION OF FRACTURE

- It has been reported that the most of fractures in the ship structure occur due to misalignment between components on each side of a through-going component within a high-stress–concentration area.

- Especially, the important joints, described as follows, for which the connection details are to be indicated in the drawing must have careful attention in both fabrication and inspection.

- The important joints within the high-stress–concentration area are as follows:

(1) Inner bottom plate between hopper sloping bulkhead and hopper side girder

(2) Hopper top plate between hopper sloping bulkhead and longitudinal side bulkheads

(3) Joints between double bottom floor and transverse bulkhead

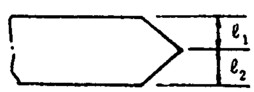
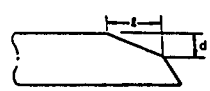
(4) Joints between transverse bulkhead stringer and side stringer in side tanks

(5) Joints between transverse stool side plate and corrugated bulkhead (or inner bottom plate)

(6) Joints between toe part of transverse web and back side of web

It is recommended to specify and control the connection details as described above with allowable tolerances in the approval drawings subject to satisfaction of the supervisor from the ship's owner and classification society.

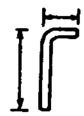
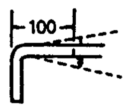
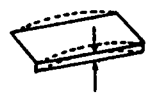
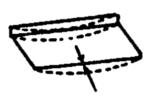
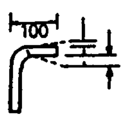
FIGURE 10 Misalignment and gap.

GROUP ITEM : CUTTING **OBJECT NO. 1: NOTCH/DIMENSION**					**GROUP** **NO. 4**
Section	**Items**			**Standard Range**	**Tolerance Limits**
Notch	Free edge	(1) Upper edge of sheer strake			Indentation ≤ 0.5mm
		Longitudinal and transverse strength			Indentation ≤ 1.0mm
		Others			Indentation ≤ 3.0mm
	Weld groove	Butt Weld	Shell and upper deck		Indentation ≤ 2.0mm
			Others		Indentation ≤ 3.0mm
		Fillet weld			Indentation ≤ 3.0mm
Dimension	Straight-ness of plate edge	Both sides submerged arc welding		±0.4mm	±0.5mm
		Manual welding, semi-automatic welding		±1.0mm	±2.5mm
	Depth of groove	Deviation of 11 mm and 12mm between designed depth and actual depth 		±1.5mm	±2.0mm
	Length of taper	Deviation of 1 mm between designed length and actual length  (ℓ compared with correct size)		±0.5 d	±1.0 d
	Size of member	General members compared with correct size		±3.5mm	±5.0mm
		Especially for the depth of floor and girder of double bottom compared with correct size		±2.5mm	±4.0mm
		Breadth of face bar compared with correct size		±2.0mm	-3mm +4mm

CONSIDERATIONS ON PREVENTION OF FRACTURE

1. The notches on the members under high tensile stress such as sheer strake, strength deck, and side shell have to be controlled strictly.
2. If the notches are unavoidable and excessive of the tolerance limits, the edge of the member should be ground smoothly.

FIGURE 11 Cutting—Notch/dimension.

GROUP ITEM : FABRICATION OF BUILT-UP MEMBER OBJECT NO. 1: FLANGED LONGITUDINAL AND BRACKET					GROUP NO. 5
Section	**Subsection**	**Item**	**Standard Range**	**Tolerance Limits**	**Remarks**
Flanged Longi-tudinals	Breadth of flange and web		±3.0mm	±5.0mm	Compared with correct size
	Angle between flange and web		±2.5mm per 100	±5.0mm per 100	Compared with template
	Curvature or straightness in the plane of flange		±10mm	±25mm	per 10 m in length
	Curvature or straightness in the plane of web		±10mm	±25mm	per 10 m in length
	Breadth of flange		±3mm	±5mm	Compared with correct size
	Angle between flange and web		±3.0mm per 100	±5.0mm per 100	Compared with template

CONSIDERATIONS ON PREVENTION OF FRACTURE
The deviation between designed and actual dimensions in the fabrication of the built-up member shall be the member that has less stiffness.
It may be that unexpected loads are to be transferred to neighboring primary members, which should be one of the causes of a fracture in the primary member.
The tolerance limits should be kept for the fabrication of built-up members.

FIGURE 12 Fabrication of built-up member—Flanged longitudinal and bracket.

GROUP ITEM : ASSEMBLY BLOCK DIMENSION TOLERANCE OBJECT NO. 1: GENERAL STRUCTURE (FLAT/CURVED)					GROUP NO. 6
No.	**ITEMS**	**FLAT BLOCK**		**CURVED BLOCK**	
		Standard Range (mm)	Tolerance Limit (mm)	Standard Range (mm)	Tolerance Limit (mm)
1	Breadth/length	±3	±5	±4	±6
2	Diagonal	±3	±5	±4	±6
3	Height	±2	±3	±4	±6
4	Level	±3	±5	±4	±6
5	Up-tightness	±3	±5	±4	±6
6	Flatness	±2	±3	±4	±6
7	Girth length	±2	±3	±3	±5
8	Distance between internal members	±2	±3	±3	±4
9	Straightness of the internal member	±2	±3	±3	±4
10	Reference line for dimensioning	±1	±2	±1	±2
11	Angle for longitudinals	±3	±4	±4	±6

EXAMPLE—DOUBLE BOTTOM BLOCK IN HOLD

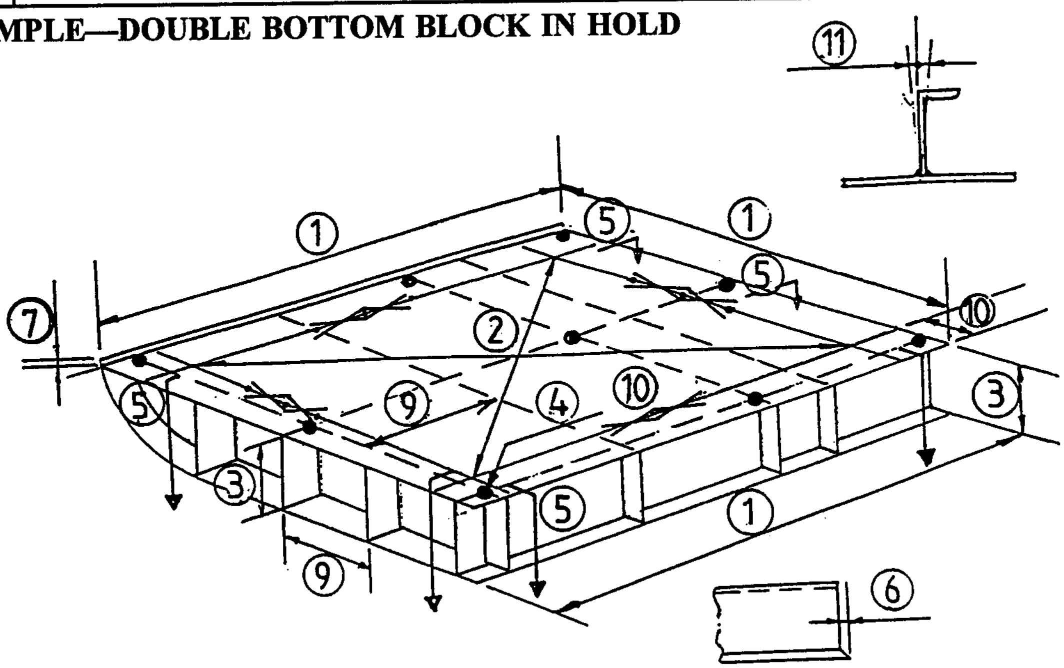
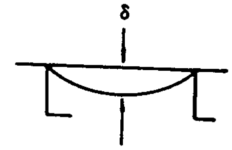
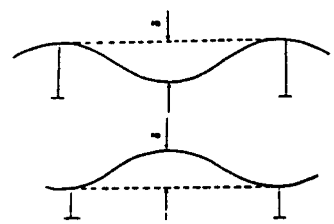
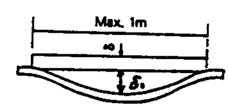
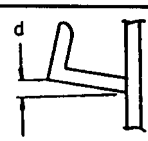
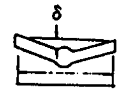


FIGURE 13 Assembly block dimension tolerance.

<table>
<tr><td colspan="3">GROUP ITEM : DEFORMATION
OBJECT NO. 1: STANDARD RANGE OF VARIOUS AREAS</td><td>GROUP NO. 7</td></tr>
<tr><th>AREA</th><th>ITEM</th><th>STANDARD RANGE</th><th>REMARKS</th></tr>
<tr><td rowspan="3">Shell plate</td><td>Parallel part (side shell)</td><td>±6mm</td><td rowspan="13">For length of frame, longitudinal, and stiffener space, none of the above is applicable for over 2 m length.


Remark:
When measuring the deformation (δ) one of the following methods shall be applied:
(1)

- Frame, longitudinal, beam, or other stiffeners

(2)


- A straight 1-m-long rule to be placed on the plate. The rule should be placed on the plate at two points with at least 0.3 m distance from each other and δ converted into δ_1 corresponding to 1 m of measuring length.</td></tr>
<tr><td>Parallel part (bottom shell)</td><td>±6mm</td></tr>
<tr><td>Fore and aft part</td><td>±9mm</td></tr>
<tr><td>Double bottom tank-top plate</td><td></td><td>±6mm</td></tr>
<tr><td>Bulkhead</td><td>Longitudinal bulkhead, transverse bulkhead, and swash bulkhead</td><td>±9mm</td></tr>
<tr><td rowspan="2">Strength deck</td><td>Parallel part (0.6L amidship)</td><td>±6mm</td></tr>
<tr><td>Fore, aft part/bare part</td><td>±9mm</td></tr>
<tr><td rowspan="2">House wall</td><td>Outside wall</td><td>±6mm</td></tr>
<tr><td>Inside/covered part</td><td>±9mm</td></tr>
<tr><td>Unfairness of internal member after weld (beam, longi-tudinal stiffener)</td><td>
</td><td>d ≤8mm</td></tr>
<tr><td rowspan="3">Angular distortion of weld joint after weld</td><td>Skin plate between 0.6L amidship</td><td>

δ ≤ 7mm
Span of frame or beam</td></tr>
<tr><td>Fore and aft shell plate and transverse strength member</td><td>≤ 8mm</td></tr>
<tr><td>Others</td><td>≤ 9mm</td></tr>
</table>

FIGURE 14 Standard range of various areas.

Substantial Repairs and Maintenance of Hull Structures

Kuniaki Ishida

ABSTRACT

After disasters happened in many aging bulkers and tankers in 1990 or so, due to vital defects of hull structures, it was widely recognized that reliable inspection, good repairs, and proper maintenance are essential to keep ships safe and that elimination of substandard ships from the shipping market tends to be emphasized.

In this connection, poor state control has been highlighted, and the enhanced hull survey by the major classification societies, in accordance with the guidelines of the International Association of Classification Societies (IACS), is being pushed forward. In such circumstances, it seems to be generally thought that the system for ideal hull inspection, repairs, and maintenance to keep ships safe have been established.

But, in the eyes that have observed ship repairing to the bottom, the current repairs and maintenance reveal not enough compilation of know-how for repairs and maintenance to eradicate the disasters of ships.

Therefore, the speaker will make some proposals for substantial repairs and maintenance of hull structures, showing some examples.

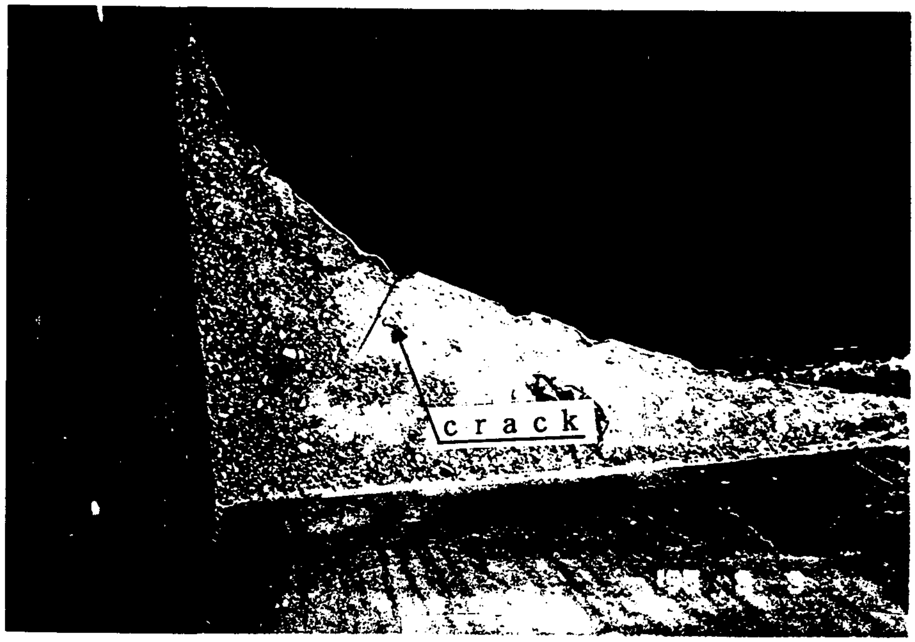
SUBSTANTIAL REPAIRS

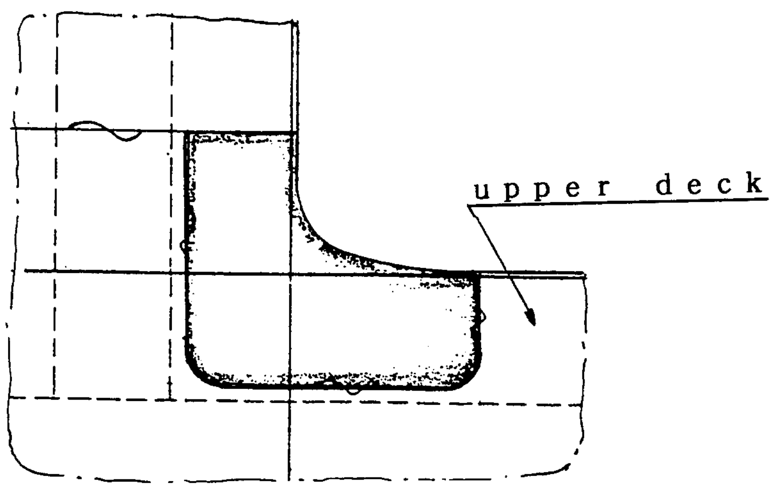
Usual ways of repairing a damaged hull structure are to restore the structure to the original condition, to restore it with thicker plate, to restore it with additional reinforcement, to modify its shape to avoid stress concentrations, to modify it to increase its scantlings, and so on. Substantial repairs require something additional to the said usual ways of repairs.

Figure 1 shows a crack in the deck plating at a hatch opening corner of a bulk carrier. The corner of the opening was elliptical to reduce stress concentrations along the edge at the corner.

Cracking at hatch opening corners is so familiar to ship repairers that it is normally repaired in accordance with the usual ways of repairs to renew the subject part of the plating to the original condition or to restore the subject part with a thicker plate.

But it was found by the shipyard that the crack was initiated at one of scraped spots at the elliptical corner, where the edge had been scrubbed by the wires for grabs to discharge





Usual Repair Method

FIGURE 1 Crack in upper deck at hatch corner.

cargoes.

In substantial repairs to the damage, additional countermeasures are required to prevent the edge from scrubbing with wires, in addition to the usual repairs to restore the plating with thicker plate. One example is to fix a round bar against rubbing apart from the opening edge (but not along the opening edge).

Figure 2 shows an example of repairs to damaged hold frames of bulk carriers recommended by Lloyd's Register of Shipping. The cracks in these constructions initiate at the end of the face plate or bracket toe due to stress concentration. And, in this repair, it is naturally required to apply a good quality of fillet welding in the subject area. By good quality fillet welding, it is usually meant to avoid defects, such as undercut, uneven surface or slug inclusion, and similar defects. But, in substantial repairs in this example, dressing beads on normal fillet welds shall be provided in limited areas around the bracket toe and the end of the face plate, as shown in Figure 3, considering reduction of stress concentration at the spots liable to initiate cracking.

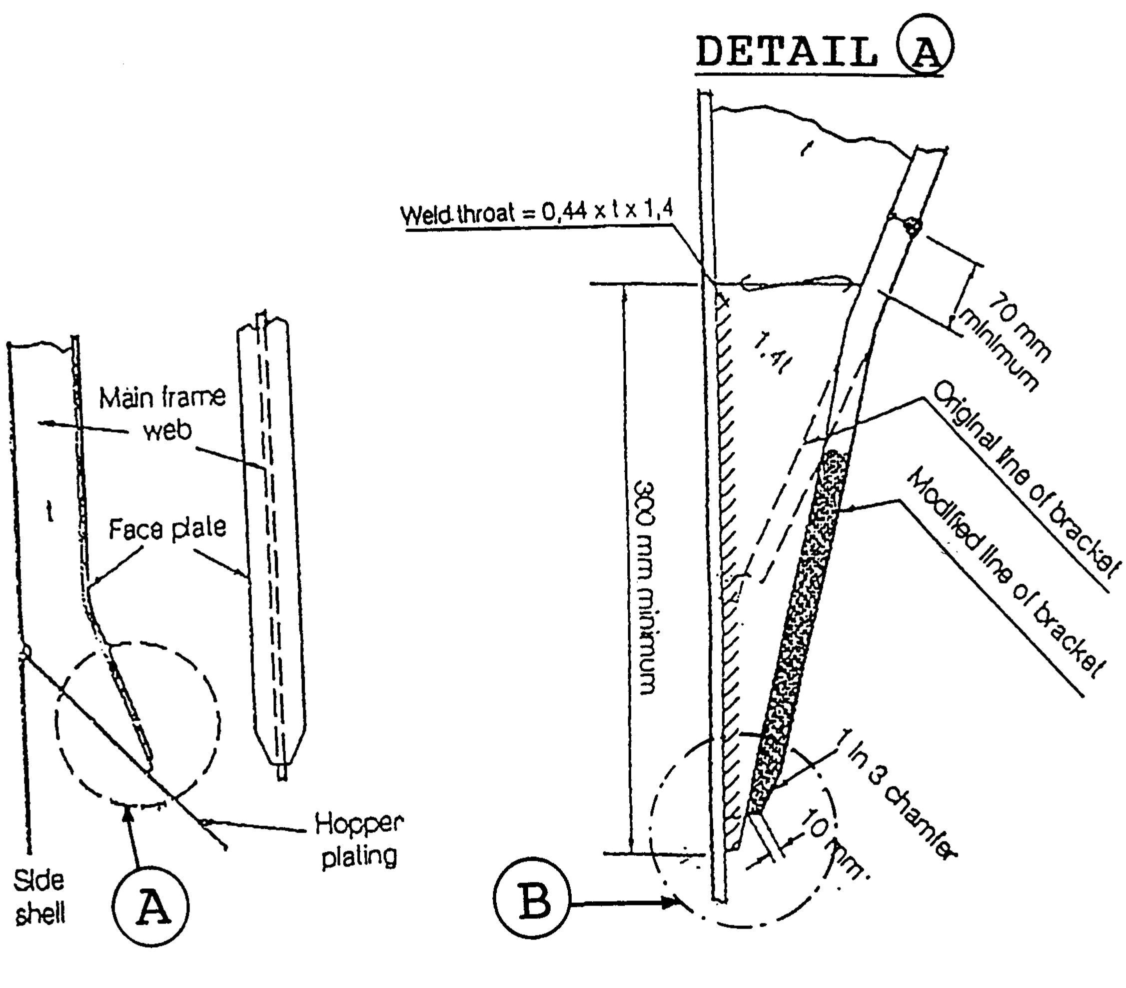


FIGURE 2 Lloyds' recommendation for repairs to damaged hold frame of bulk carriers.

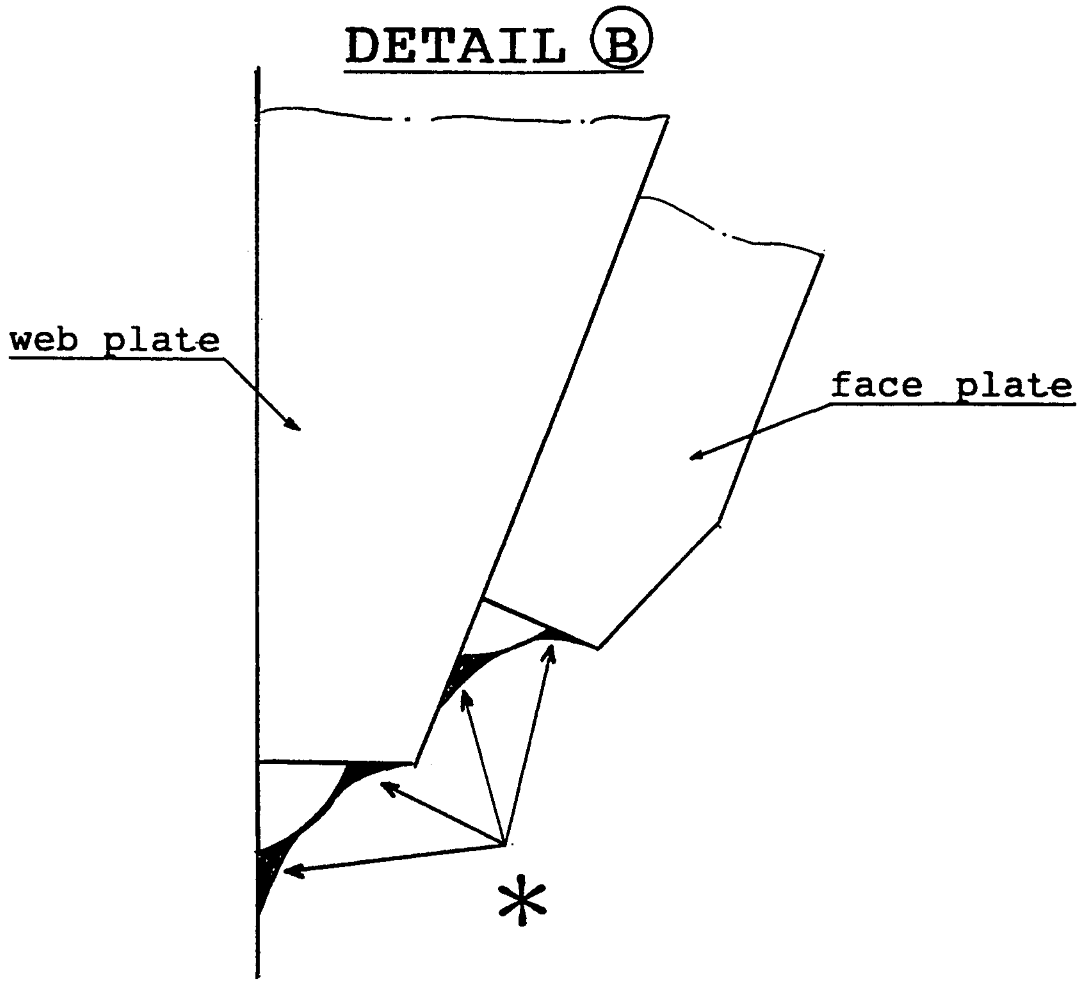


FIGURE 3 Dressing bead on fillet weld at the edge of hold frame.

Figure 4 shows a deck longitudinal of a 10-year–old bulk carrier, which proves that 10 years are enough to build up such damages in vital structural members (Wada, 1992a).

In substantial repairs, in order to avoid an unexpected disaster of the ships and to avoid repetition of costly repairs, special attention shall be paid to painting, including surface preparation in the subject area after completion of steel work.

In substantial repairs, scrupulous instructions and orders for repairing, as shown by the above examples, should be added to the usual ways of repairs on the basis of technical know how obtained through ship repairing.

PREVENTIVE MAINTENANCE

There are two functions of preventive maintenance. One is for a decrease in expenses of repairs, and the other is for preventing hull structures from unexpected or serious damages.

The one is commonly understood as a economical way of maintenance, such as maintenance of paint in advance instead of steel renewal after obvious damages, as shown in Figure 4. The other is not so common in ship repairing and requires advanced application of structural theories.

Figure 5 shows the patterns of buckling in transverse bulkhead plating in a wing tank of an ore carrier (Wada, 1992b). In the ballasted condition, slight deformation could be observed in a few panels, as shown in the left figure. During the return voyage, in the fully loaded condition, distinctive shear buckling could be observed, as shown in the right figure. Then the ship moved to a shipyard for drydocking after discharging the cargo when the buckling had almost disappeared.

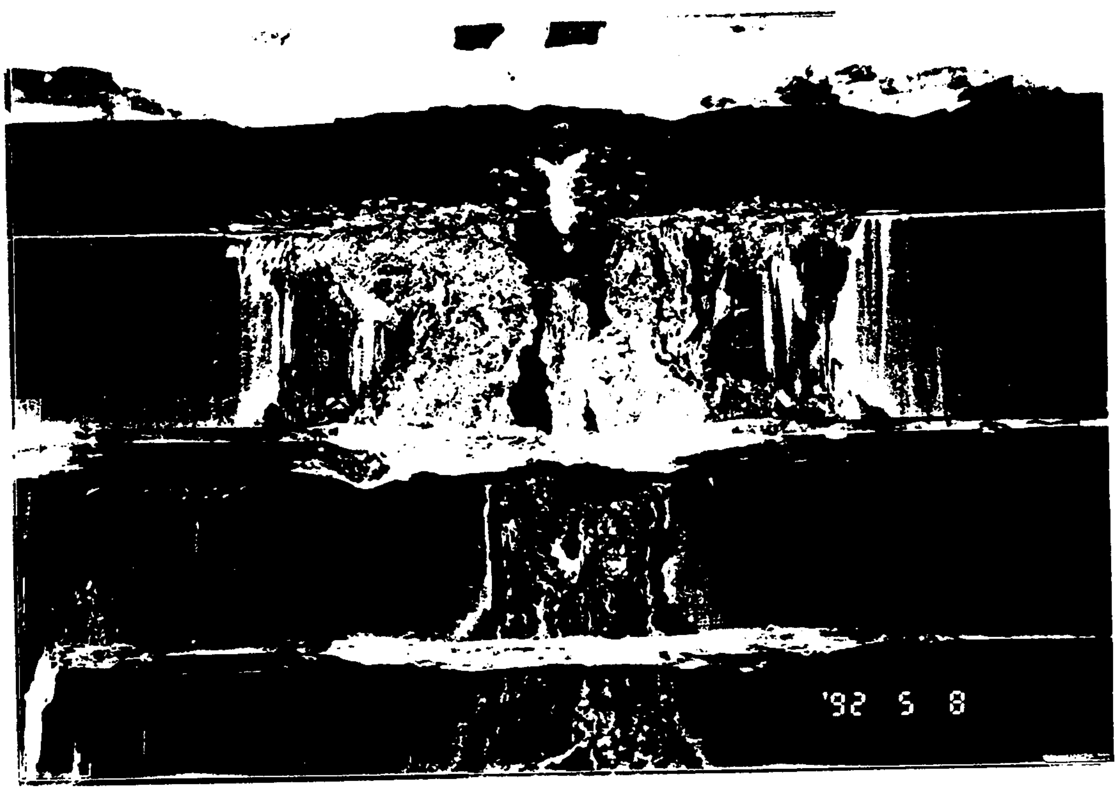


FIGURE 4 Corrosion of deck longitudinal in way of welded joint.

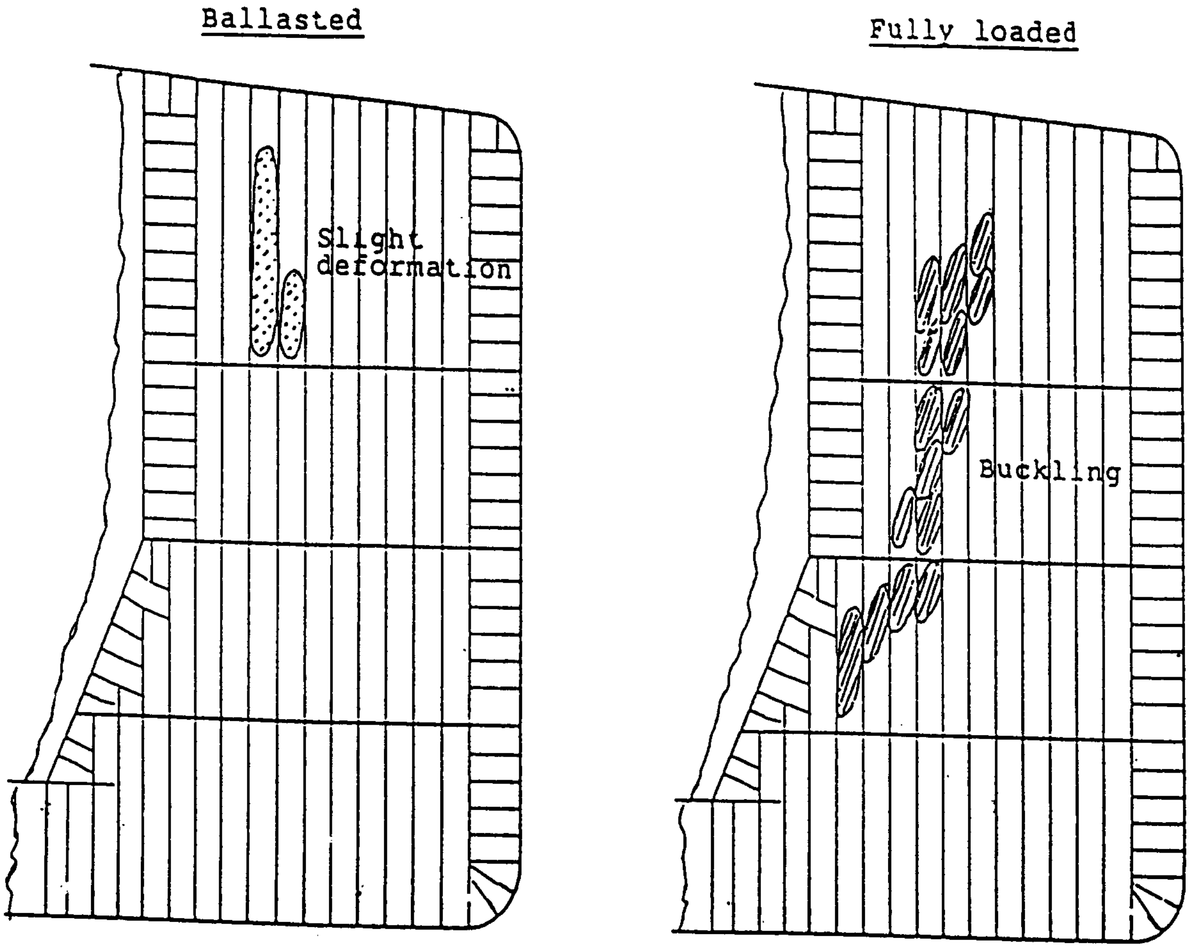


FIGURE 5 Buckling of transverse bulkhead plating in wing tank of an ore carrier.

Once such shear buckling occurs in a small area of the bulkhead plating due to local wastage, it tends to spread like dominoes to other areas without wastage. And, due to shear deformation of the wing tank, buckling of the web plates of transverses in the wing tanks may follow, as shown in Figure 5a (Wada, 1992c).

If the buckling is not inspected during the fully loaded condition, and is left untouched without knowing the phenomena, repeated elastic buckling causes deterioration of paint in the area where corrosion of the plating will be escalated. The buckling will be advanced to plastic buckling when the bulkhead and transverse may have to be repaired on a large scale.

In preventive maintenance, when the ship comes to a dock, the bulkhead plating with slight deformation shall be reinforced with stiffeners as shown in Figure 6, where heavy deformation has been found at fully loaded condition.

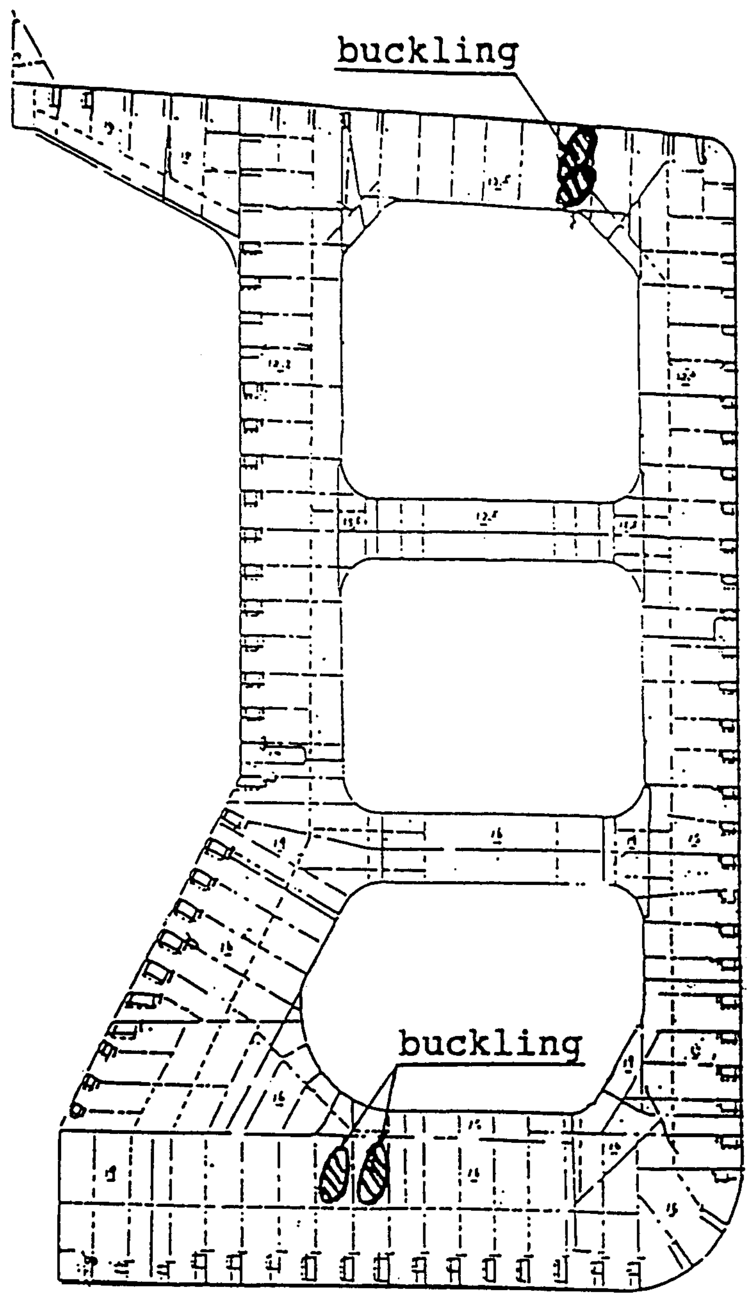


FIGURE 5a Buckling of transverse web plate.

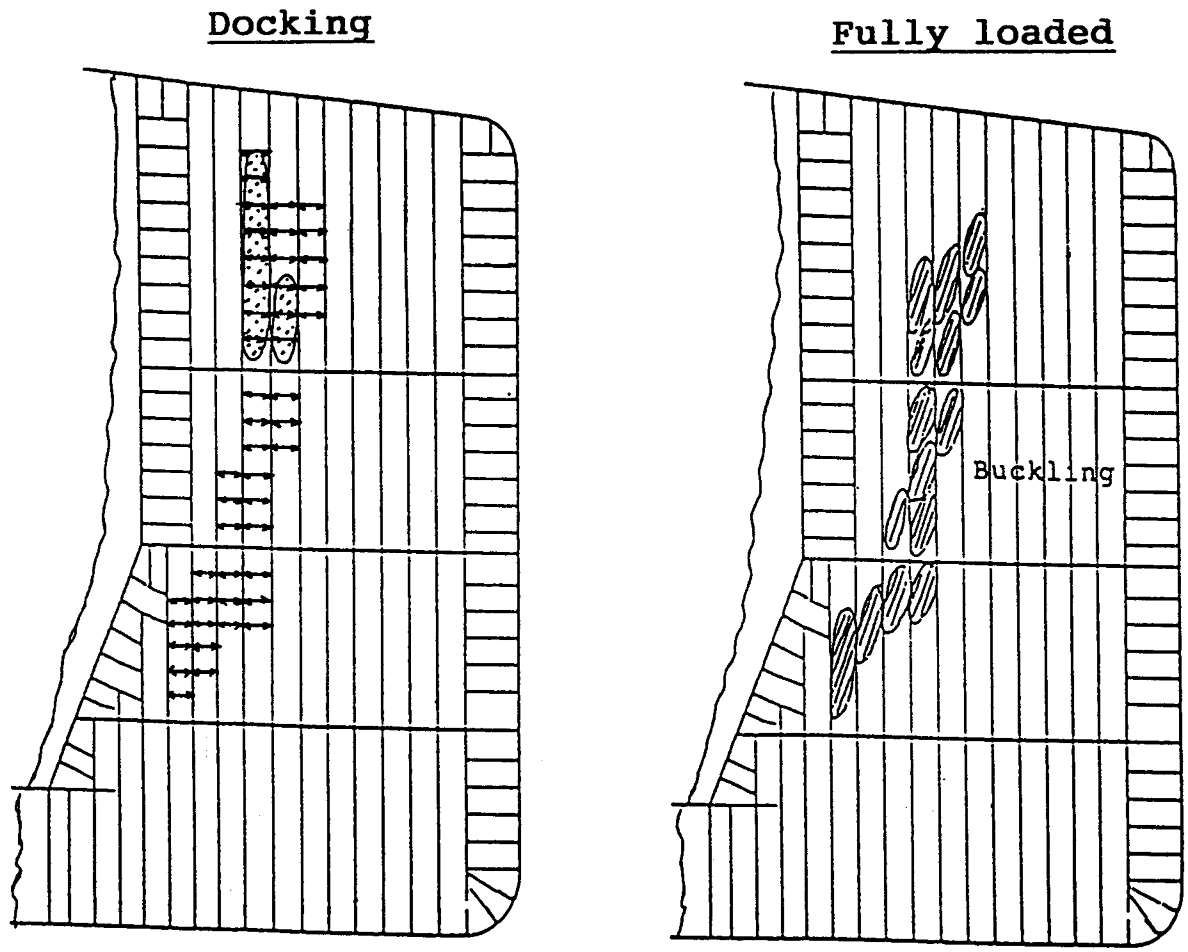


FIGURE 6 Reinforcement of bulkhead plating with stiffeners at docking.

ESTABLISHMENT OF MAINTENANCE SCHEDULE

It is essential for substantial maintenance to establish a maintenance schedule and to fulfill it seriously. This means that maintenance of a ship should be done in accordance with the maintenance schedule, and maintenance instructions should be prepared by the shipyard, based on the designed life span and results of calculation of hull strength (Wada, 1992c).

Recently, it has been possible through computer-aided design to specify areas sustained under high stress. As a consequence, it has been possible to apply advanced treatments on the spots in the process of building ships, such as dressing beads on fillet welds (as shown in Figure 7), to eliminate stress concentrations.

But as time goes on, the surface of the welds with dressing beads corrodes unevenly, as shown in Figure 8, to an extent that the initially intended stress level cannot be maintained.

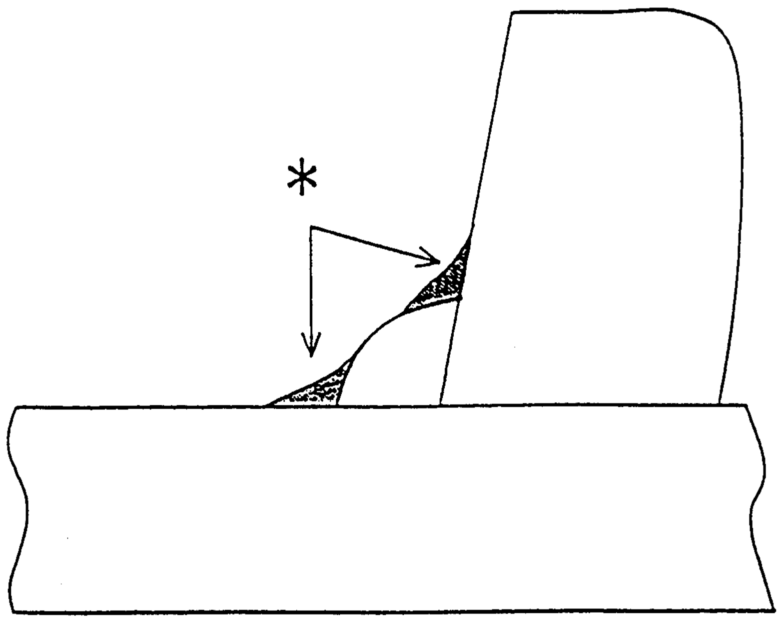


FIGURE 7 Dressing bead on fillet weld.

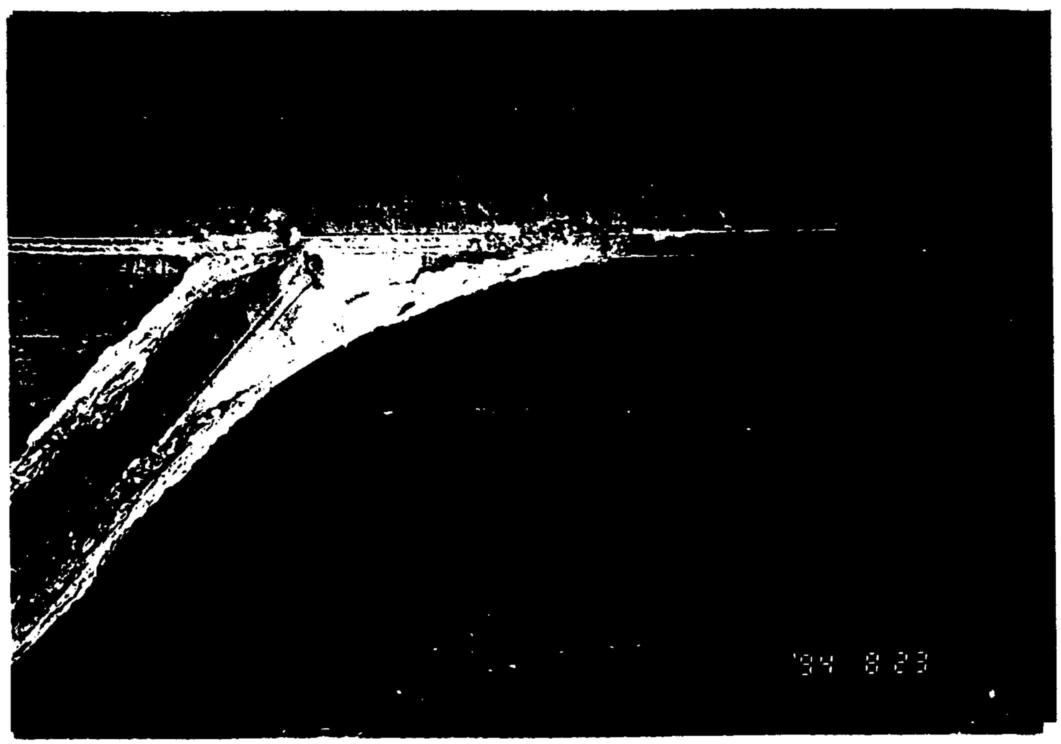


FIGURE 8 Corrosion in way of fillet weld.

Therefore, it is required for adequate maintenance schedule to instruct timely inspection and proper remedy in time for areas in question. By establishing and fulfilling such maintenance schedules, safe voyages can be guaranteed. Moreover, it will become possible to save the maintenance cost over the life of a ship.

ACKNOWLEDGEMENTS

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REFERENCES

Wada, Takao. 1992a. Design of hull structures with wider scope. Presented at the 14th Brazilian Congress of Maritime Transport and Shipbuilding—SOBENA'92 held October 5–8, 1992 in Rio de Janeiro, Brazil.

Wada, Takao. 1992b. Practical design of hull structures. Presented at the 5th International Symposium on Practical Design of Ships and Mobil Units held May 17–22, 1992 in Newcastle Upon Tyne, United Kingdom.

Wada, Takao. 1992c. Some consideration on current hull maintenance. Memorandum of Lecture for meeting organized by The Japanese Hull Insurers Union, held September 7, 1992 in Tokyo, Japan.

Repair Rules of Thumb

David J. Witmer

INTRODUCTION

Over the years, certain repair procedures, methodologies, and philosophies have achieved a "rule of thumb" status. That is to say they have become, among many owners and regulatory agencies, almost "standard procedures." It is not the intent of these notes to recommend any particular repair procedure, regardless of its achieved or perceived status. Rather, it is to stimulate discussion among marine professionals regarding the basic technical soundness of such procedures, in spite of their common usage and, in many cases, common acceptance in ship repair.

Specific repair procedures deemed worthy of reexamination include:

- repair of cracks in longitudinal stiffeners
  - vee and weld
  - insert
- shell and deck plating inserts
  - full-plate width
  - specified minimum size, for example, 12 in. by 12 in., or 18 in. by 18 in.
  - circular
- use of doublers
- welding against the sea
- peening

REPAIR OF CRACKS IN LONGITUDINAL STIFFENERS

One particularly interesting rule of thumb involves criteria used to determine whether a crack in a longitudinal stiffener (in most cases, a side-shell longitudinal stiffener) should be veed out and rewelded or inserted. This rule of thumb says that if the length of the crack is less than one-half the depth of the stiffener web, it *may* be rewelded. If, on the other hand, the length of the crack exceeds one-half the depth of the web, it is to be inserted.

Is there any technical merit to establishing a repair procedure based on whether or not a fracture length exceeds half the depth of the longitudinal? What is significant or critical about this 0.5d point on the web? See Figure 1.

There is nothing particularly critical or significant about this point and, other repair considerations notwithstanding—for example, bracket-toe geometry if the fracture is in way of the toe of a bracket as they often are—veeing out and rewelding are perfectly acceptable, regardless of fracture length.

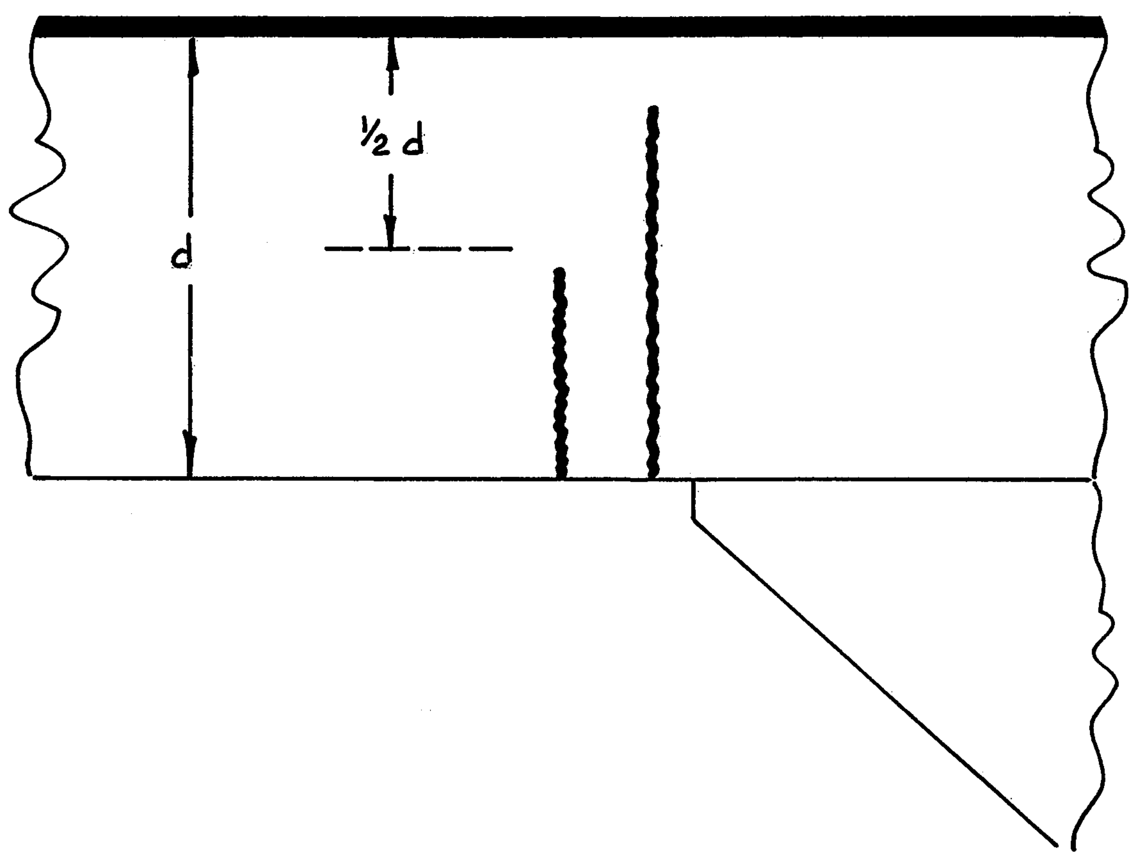

FIGURE 1 Repair of cracks in longitudinal stiffeners.

SHELL AND DECK PLATING INSERTS

Figure 2 illustrates various sizes and shapes of shell and deck plating inserts. The "X" indicates a structural defect (for example, a plate lamination or fracture) requiring a portion of the plate to be renewed by inserting. I remember when working as a field surveyor for the American Bureau of Shipping in New Orleans during the late 1970s, all plating inserts were required to be a full-plate width (from existing seam to existing seam) and, typically, from an existing butt to a point well beyond the defect. Was this sound ship repair or overkill? Later, it became perfectly acceptable to specify a minimum size for a plating insert, say, 18 in by 18 in. or 12 in. by 12 in. Was this enough? Should inserts always originate from an existing butt or seam, or is it okay to position them on the body of a plate? And, finally, what about circular inserts? After all, we think nothing about welding 6-in.-diameter sounding pipe caps in the bodies of deck plates.

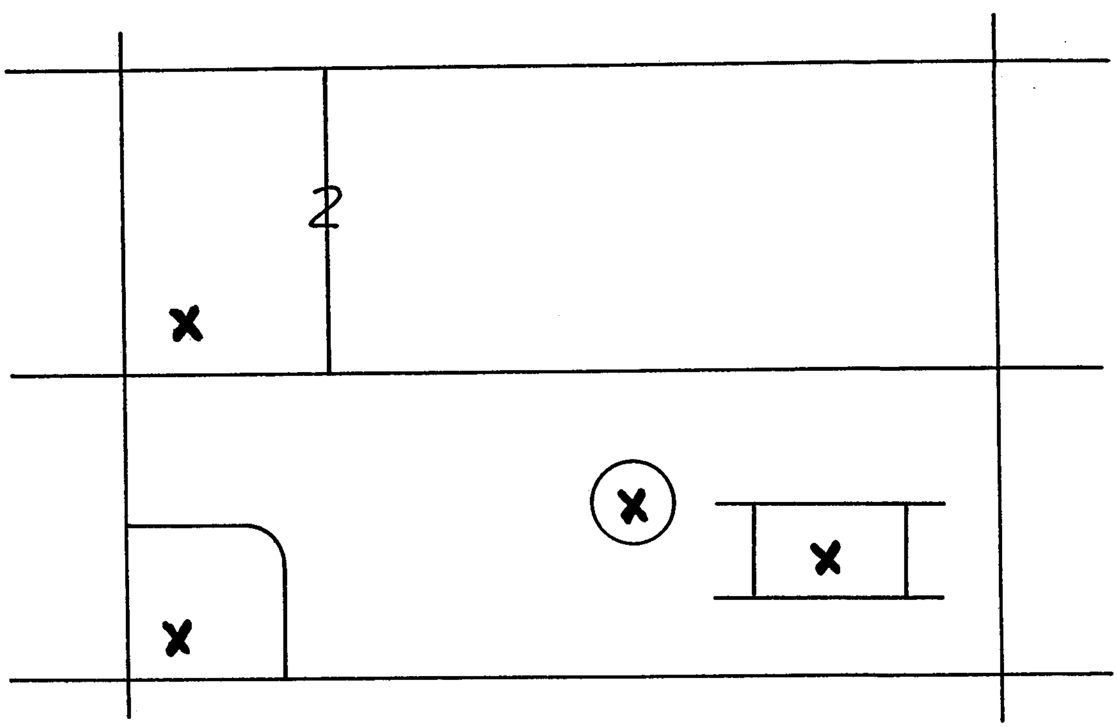

FIGURE 2 Deck/shell inserts.

It should be acceptable practice to insert in way of a structural defect using circular inserts as small as 6 in. in diameter. Anything larger would be at the owner's discretion, depending, of course, on the size of the defect. If rectangular inserts are used, the corners should be radiused (6-in. minimum). If an existing butt or seam is not handy, there is nothing wrong with inserting within the body of the plate.

The key to a successful insert seems to be in establishing and following a sound welding procedure (including the joint welding detail) and not in the size of the insert per se.

USE OF DOUBLERS

In general, the use of doublers as a repair option is not sound practice on tankers. This philosophy is based on the danger of trapping gases between the doubler and the parent plate. But what about in the wing tank bulkheads on bulk carriers?

As an example, if the wing (ballast) tank bulkhead were punctured by shoreside cargo grabs, could we not lay a doubler plate over the puncture and fillet weld it continuously around its perimeter. What is the technical reason for not doing so?

This is a repair option that warrants further discussion.

WELDING AGAINST THE SEA

When discussing welding against the sea, we are really talking about fillet welding to the inside of a shell plate that is located below the vessel's waterline. Is this acceptable practice or, as another rule of thumb states, must we either drydock the vessel or fit a vacuum box to the exterior of the hull in way of the welding? See Figure 3.

Most owners are, at one time or another, faced with this very question. Of particular interest is a Canadian engineering company that developed welding procedures for use on icebreakers.

Fillet welding to the inside of a shell plate located below the waterline is perfectly acceptable repair practice for plating thicknesses greater than 0.75 in. Research has proven that the cooling rate of the weld metal deposit is *not* affected for such plates. Fillet welding *should not be* a problem for plates greater than 0.5 in., although 0.75 in. would be no problem at all.

The American Bureau of Shipping has developed procedures for welding against the sea.

PEENING

Time does not permit a detailed discussion of peening in ship repair, although this procedure can, if properly carried out, extend the inservice life of a weldment.

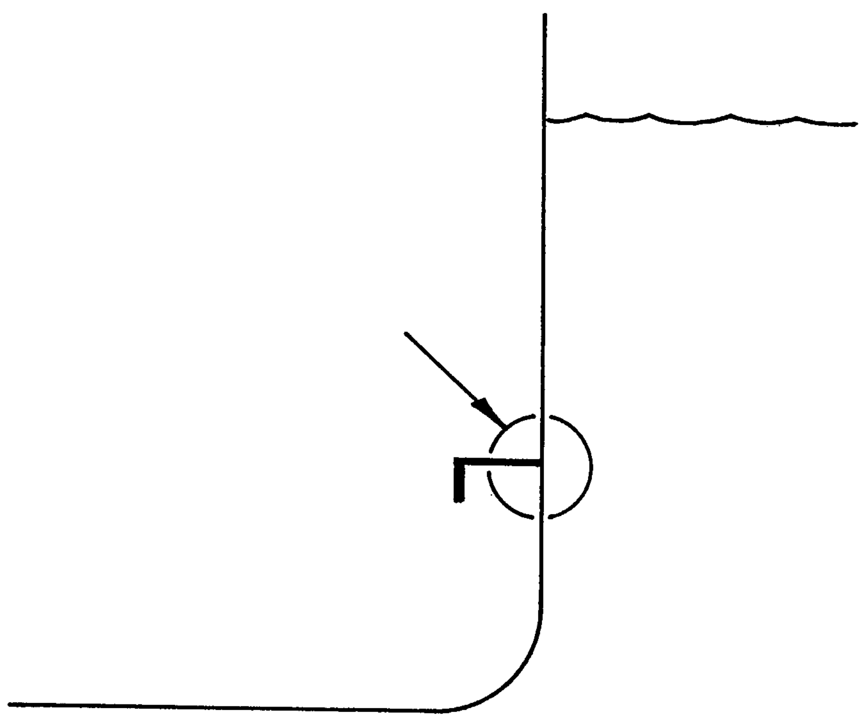

FIGURE 3 Welding against the sea.

GLOSSARY FOR PART II

ABS	American Bureau of Shipping
AFASAD	Air Force Aeronautical Systems Division
AMV	advanced mean value
API	American Petroleum Institute
ASIS	Association for Structural Improvement of Shipbuilding Industry
ASME	American Society of Mechanical Engineers
ASTM	American Society for Testing and Materials
BL	bottom longitudinal
BSI	British Standards Institution
CAIP	critical area inspection plan
CEGB	United Kingdom Central Electricity Generating Board
CCP	center-cracked panel
CDF	cumulative distribution function
CDFD	crack driving force diagram
CDR	cumulative damage ratio
CIMS	computer integrated manufacturing for shipbuilding
CLD	constant life diagrams
COV	coefficient of variation
CTOD	crack-tip–opening displacement
CVK	center vertical keel
DH	double-hull tanker
DISAM	discrete analysis method
DLA	dynamic loading approach
DPFAD	deformation plasticity failure assessment diagram
DWT	deadweight ton
EPRI	Electric Power Research Institute
FAA	Federal Aviation Administration
FAC	failure assessment curve
FAP	failure assessment point
FEA	finite element analysis
FEM	finite element method
FMEA	failure mode and effect analysis
FORM	first order reliability method

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FPSO	floating production, storage, and off-load system
FRS	fracture record sheet
GRT	gross registered ton
GSOWM	Global Spectral Ocean Wave Model
HAZ	heat-affected zone
HFDB	hull fracture data base
HSLA	high strength low alloy steel
HT	higher tensile steel
HTS	high tensile steel
IACS	International Association of Classification Societies
IIW	International Institute of Welding
IMO	International Maritime Organization
ISO	International Standards Organization
JSQS	Japan Shipbuilding Quality Standard
LBP	length between perpendiculars
LEFM	linear elastic fracture mechanics
LRFD	load and resistance factor design
LOA	length over all
MARINE	Mitsubishi Advanced Realtime Initial Design and Engineering System
MARPOL	Marine Pollution convention of the International Maritime Organization
MD	mid-deck tanker
MDWT	thousand deadweight ton
MPP	most probable point
MRR	Maritime Regulatory Reform
MV	mean value
MVFOSM	mean value, first order, second moment
MWL	mean waterline
NDE	nondestructive examination
NDI	nondestructive inspection
NTIS	National Technical Information Service
NVIC	Navigation and Inspection Circular
OPA 90	Oil Pollution Act of 1990
PDF	probability density function
RAO	response amplitude operator
RRDA	rapid response damage assessment
SASMIT	Strength Analysis System of Midship Part of Tanker
SOLAS	safety of life at sea
SMAW	shielded metal arc welding
SORM	second order reliability method
SOWM	U. S. Navy's Spectral Ocean Wave Model
SR RAO	stress range, response-amplitude operator
SSC	Ship Structure Committee
TAPS	Trans-Alaska Pipeline Service
TMCP	thermo-mechanical controlled process

TSCF	Tanker Structure Cooperative Forum
TWI	The Welding Institute
UK DOE	United Kingdom Department of Energy
UKOSRP	United Kingdom Offshore Steels Research Program
VLCC	very large crude carrier
WP/HD	working parties on hull damages