# PROCEEDINGS OF THE 13th INTERNATIONAL SHIP AND OFFSHORE STRUCTURES CONGRESS

## VOLUME 1

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## COMMITTEE II.1
# QUASI-STATIC RESPONSE

## MANDATE

Concern for the quasi-static response of ship and offshore structures, as required for safety and serviceability assessments. Attention shall be given to uncertainty of calculation models for use in reliability methods, and to consider both exact and approximate methods for the determination of stresses appropriate for different acceptance criteria.

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## KEYWORDS

Quasi-static response, finite element analysis, simplified method, structural design guidelines, offshore structures, ship structures, stress concentration, tubular joints, grouted connections, buckling, monitoring, cruise ships, double hull, bulk carriers, container ships.

# CONTENTS

# 1. INTRODUCTION

The application of the finite element structural analyses had become very common in ship structural design in the 1970's, and extensive use of the method had been one of the major daily work of ship structural design in the 1980's based on certain direct calculation procedures proposed by the major ship classification societies. According to these structural design procedures, the precise evaluation of the quasi-static response of stiffened panels and girders had become possible so that considerable improvement were made in ship structural design, which had significantly reduced the number of buckling failures and ductile collapse of such structural members. Combining the use of high tensile steel with design optimization techniques based on the finite element structural analysis, considerable hull weight reduction had also been accomplished during the period.

From the late 80's to the early 90's, ship structural designers faced a relatively new problem; i.e. fatigue, which was formerly not a major issue in ship structural design in comparison with the buckling and ductile collapse problems. This should be contrasted to the offshore structural design, in which the fatigue strength evaluation at tubular connection is a critical issue from the very beginning. Fatigue cracks were first found in the side longitudinal stiffeners of relatively new oil tankers. In order to establish a rational fatigue strength evaluation, a hierarchical modeling with the use of advanced pre-/post processors is required in the finite element structural analysis, which consists of a very sophisticated procedure from global hull girder analysis to local stress analysis near the weld toe at a transverse and longitudinal connection. The selection of loading conditions is also important, because several load effects such as hull girder stresses, local internal cargo pressure, and external wave induced pressure should be properly superimposed in order to determine the characteristic value of the stress range. These are obviously one of the major issues in the structural response in these days.

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 1990's. In general these were the older ships, serving under third, fourth, or even fifth owners. They were poorly maintained, exhibited excessive corrosion and generally failed in two principal ways. Section of shell plating severely weakened by corrosion could have carried away under high sea-way loading, or corroded sections could have permitted sea water to enter the holds resulting in liquefaction and shifting of cargo. It seems that the strength evaluation of these aged ships are the new role of ship structural engineers so that aged ship structures are properly maintained, and also that excessively degraded ships are excluded from the market.

Based on the above mentioned trends, this report is organized in the following manner. In the second section of this report, the development of calculation procedures are discussed with emphasis on simplified procedures, finite element guidelines, finite element pre-processing and post-processing, and reliability methods. The detailed investigation on the commercial codes for pre-processing and post-processing are described in this report. The third section describes the recently developed structural design and approval systems proposed by several ship classification societies. The fourth and fifth sections describe the specific items relevant to offshore and ship structures, respectively. In the fourth section, structural design of tubular joints, buckling strength, and grouted connections of offshore structures are reported. In the fifth section, stress concentration of structural details relevant to fatigue analysis, and buckling problems of ship structures are reported. Full scale monitoring systems and monitoring data are also summarized. For specific type of ships, cruise ships, double hull ships, bulk carriers, and container ships are selected as the areas of major topics in this report.

As was done in the report of the previous terms, the 1997 Committee has also carried out a comparative finite element study focusing on the determination of a local stress distribution near the transverse and longitudinal connections. The numerical results are compared with the experimental results for the rational interpretation of the hot-spot stress obtained by using various finite element modeling and by experiments.

Outside of some subjects which have been traditionally addressed by Committee II.1, the subjects covered in this report stem from the interests and expertise of individual committee members. This committee in particular has been fortunate to have expertise in buckling, fatigue, offshore structures, cruise ships, double hulls, reliability analysis, and finite element modeling, and the report emphasizes these areas, possibly with the omission of other subjects pertinent to the committee mandate.

# 2 CALCULATION PROCEDURES

## *2.1 Simplified Calculation Procedures*

### *2.1.1 Simplified Calculation Procedure for Ship Strength Analysis*

The problem of the torsional stiffness of ships with large deck openings was discussed by Committee II.1 of ISSC'94 (Pegg *et al.*, 1994). In the continuation of that problem, a new simplified approach is proposed by

Wang and Chen (1994) that is based on the multi-celled thin beam theory and the transfer matrix method. A variety of decks composed of closed sections, opened sections and center deck girder are compared concerning the torsional stiffness and the natural frequencies. As expected, the most significant parameter is the ratio of the length of opened sections to the full length. An increased number of short deck openings is better than a reduced number of long openings, where the center girder can be disregarded for the torsional stiffness.

For preliminary design and quick strength assessment, Rigo (1992a, b) has developed an analytical algorithm based on the Fourier series expansions to solve the differential equations of orthotropic plates and cylindrical shells. Each element is composed of the shell plating with 3 layers of beams. For example, for a deck plate, the beams could be deck girders, transverse frames, and longitudinal stiffeners. Transverse frames are assumed to be equally spaced, but longitudinal stiffeners may be non-uniformly distributed. Calculations are not based on smeared-out rib properties; rather, local torsional and lateral bending stiffness are taken into account. Easy and simple modeling is the first advantage of this method. Modeling of the structures is based on geometrical considerations of the mid-ship section. The input data of the model are few, about 10 data lines for each stiffened element. The modeled structure is usually one hold length or the complete prismatic section of the ship, but local analyses of stiffened panels can also be performed. The boundaries of the global model are located at transverse bulkheads, where the stiffened panels are simply supported with end moments simulating the effects of the aft and fore ends. Concentrate loads of cargoes, deadweight, and varying pressure are considered by using a number of terms of the Fourier series. Stresses in each girder are given in the flange and at the junctions of the web with the plating and the flange. The model uncertainty has been assessed for different structures by comparison with finite element analysis (Rigo, 1992a; Sumi *et al.*, 1996). The method has demonstrated its utility and efficiency in practical application on tankers, a swath, floating structures, and many land-based structures such as lock gates, sluices, radial gates, etc. To allow an optimization process at the design stage, the method is applied to check various sets of constraints of the structural strength and serviceability requirements.

A simplified thermal elasto-plastic analysis method is proposed by Jang and Seo (1994, 1995b) to estimate the longitudinal deformations due to welding and heating. This formulation takes into account of the influence of the fabrication factor (i.e. ratio of the effective area of the beam on the heat input) on the beam curvature. Such effect was neglected in the Tusji and Okerblom formulations. The formulation has been compared with experiments and could be used to simulate the fabrication process in order to develop appropriate techniques to reduce fabrication costs. A simplified procedure to evaluate the temperature distribution, the induced thermal stress and the global deflections in a ship structure is proposed by Chen and Hu (1995). The simplification is based on a 2-D static heat conduction problem.

### *2.1.2 Simplified Methods to Evaluate the Ultimate Strength of Ship Hulls*

Since 1993, important works in the field of ultimate strength of ship girder has been achieved. In order to avoid overlaps with other reports, aspects related to the simplified global response of a ship girder are presented in this subsection, and the aspects related to precise evaluation of ultimate strength will be presented by TC III.1.

Most of the works on ultimate strength of ship hulls are usually related to strength under vertical bending. Recently, several works have been performed to provide reliable evaluation of the ultimate strength of ship hulls under combined loads. Simplified analytical approaches (Beghin *et al.* (1995); Gordo and Soares (1993,95,96); Mansour *et al.* (1995); Paik (1994), Paik and Masour (1995a,b), Paik *et al.*. (1996a)) are proposed as well as numerically based evaluations (Paik and Pedersen (1996); Hu (1995)) and reliability based approaches (Kim and Kim (1995); Wang and Moan (1995)).

A simple analytical expression Eqn. 1. for an interaction relation under vertical and horizontal bending moments is proposed by Mansour *et al.* (1995). This expression is based on numerical results obtained from one container, one tanker and two cruisers. Similar expression Eqn. 2. is proposed by Gordo and Soares (1995) after analysis of 4 tankers.

$$\left(\frac{M_V}{M_{VU}}\right) + 0.8\left(\frac{M_H}{M_{HU}}\right)^2 = 1 \qquad \text{if} \qquad \frac{M_V}{M_{VU}} > \frac{M_H}{M_{HU}} \tag{1}$$

$$\left(\frac{M_V}{M_{VU}}\right)^a + \left(\frac{M_H}{M_{HU}}\right)^a = 1 \text{ with } \quad 1.50 < a < 1.66 \tag{2}$$

where $M_V$ and $M_H$ are the vertical and horizontal bending moments, and $M_{Vu}$ and $M_{HU}$ are the ultimate vertical and horizontal bending moments, respectively. The method adopted is based on the approximate load shortening curves for stiffened plates presented by Gordo and Soares (1993). More details are presented in Gordo and Soares (1996). Effects of corrosion, residual stress and use of high tensile steel on ultimate capacity of ship hulls are discussed and quantified. A linear reduction is proposed for corrosion and a parabolic for residual stress. Uncertainties of Eqns.1. and 2. for other ship types or sizes have not been evaluated.

Concerning simplified method to evaluate the ultimate capacity of ship hulls, the more comprehensive work and valuable contribution have been achieved by Paik *et al.*. For design purposes and reliability analysis use, Paik, Thayamballi, and Jung (1996) proposes relatively simple analytical formulations representing the interaction relationship for the ultimate strength subjected to combined vertical, horizontal bending and shearing forces. Sets of interaction curves were determined empirically based on numerical results from the Idealized Structural Unit Method (ISUM) on 11 ships. For each single load component (vertical moment - sagging and hogging, horizontal moment and shearing force), maximum capacity $M_{vu}$, $M_{HU}$ or $F_U$ is determined, respectively. When combined loads are applied, the proposed interaction formula is:

$$\left(\frac{M_V}{M_{VU}\cdot F_V}\right)^{1.85}+\left(\frac{M_H}{M_{HU}\cdot F_H}\right)=1 \qquad (3)$$

where $F_V=\left(1-(F/F_U)^5\right)^{0.5}$ and $F_H=\left(1-(F/F_U)^{5.5}\right)^{0.4}$.

Determination of $M_{vu}$ is based on the work of Paik and Mansour (1995a,b) where only the vertical bending moment is considered. A credible distribution of longitudinal stresses over the hull section at the moment of overall collapse is used. Ultimate strength is calculated by integration of the assumed stress distribution. Ultimate compressive and shear strength of stiffened panels is based on empirical formulations (Paik, 1995; Paik and Lee, 1996).

Paik (1994) establishes a procedure for evaluating the possibility of hull collapse of aging bulk carriers under combined longitudinal bending and shearing force. He presents hull strength interaction curves for three standard structural damages (corrosion in plating, initial crack and dropping-off of one side-shell). Assessment of different flooding scenarios for the remaining strength shows that possibility of foundering of aging ships can be very large. To consider the cracked effects, a polynomial formula is used to give the reduced yield stress for a given initial crack length.

Growing concern of classification societies in the evaluation of the real ultimate capacity of a ship in operation is demonstrated by Beghin *et al.* (1995). The proposed RESULT computer code is a simplified iterative procedure based on the "method of components" using the Gordo-Soares' formulas for plates and stiffeners (Gordo and Soares, 1993). The same data file can be used both for RESULT and the standard rule program MARS of Bureau Veritas, in which so far the effects of horizontal bending moment, lateral pressure, shear, and torsion have not been considered yet.

Hu (1995) proposes a simplified incremental approach for the prediction of the ultimate bending moment of ship hulls. Axial shortening and strain are obtained through interpolation from a set of generic stress-strain curves (data base) which were generated using finite element analysis.

Reliability analyses on hull girder are presented by Kim *et al.* (1995) and Wang and Moan (1995). For four failure modes, Kim *et al.* (1995) give a simple design format that can produce uniform safety level for all ships. Wang and Moan (1995) use the DnV design criterion for buckling under combined stresses for reliability based design of stiffened panels. Limit state function, modeling of random variables, safety levels and uncertainty of the proposed model are discussed.

### *2.1.3 Simplified Methods to Evaluate the Ultimate Strength of Ship Components*

From a design point of view, the interaction equations are more (sometimes too much) conservative than other methods and they do not provide any information on the collapse mode. Nevertheless, they are often more convenient for implementation in design rules. This is probably the main reason why the interaction equations on the ultimate strength of stiffened panels under combined loading are continuously investigated by several authors. Ueda *et al.* (1995) present a comprehensive set of interaction relationships for the buckling, ultimate strength and fully plastic strength of stiffened and unstiffened plates subjected to biaxial and uniform shear forces. Lateral load and effects of initial imperfections should be added in the formulations before use for practical design. Benello *et al.* (1993) proposed a "modified interaction equation" to discern between failure modes as stiffener compressive failure cannot occur in sagging. Lateral pressure is considered as well as initial deflection. The proposed interaction equations are assessed by comparison with an inelastic column analysis program MLTSPAN and a Perry equation. Soares and Gordo (1996) investigate the problem of unstiffened panels and propose an interaction equation that considers biaxial compression and lateral load with the effect of initial defects. This formula is proposed for plates with slenderness larger than 1.30, otherwise von Mises equation is adopted. The model uncertainty is estimated to be within 15%.

Mikami and Niwa (1996) propose an approximate approach to predict the ultimate strength for orthogonally stiffened plates based on the orthotropic plate theory. Simple analytical formulations are presented for all collapse modes; i.e. overall buckling including transverse stiffeners, partial buckling, local buckling of longitudinals or unstiffened plates. The method is extended to the ultimate strength of non-uniformly stiffened plates with unequally spaced longitudinals having different dimensions. Jang and Seo (1995a) develop another simplified numerical approach that also covers all the collapse modes of a welded stiffened panel under uniaxial compression. Residual stress and initial distortions are considered. Ultimate strength is easily calculated as the crossing point of the plastic curve and the elastic curve that are obtained, respectively, using the virtual work principle and the energy formulation.

A simplified procedure for predicting ultimate compressive strength of unstiffened panels with complex initial deflections is proposed by Paik and Pedersen (1996). The method is based on the elastic large deflection theory and the rigid-plastic deflection analysis. It is proposed to take only one component for the deflection function. Uncertainty of the simplified procedure can be determined by reference to finite element analysis. Paik (1995c) also suggests a concept of effective shear modulus for describing stiffness and strength of a plate in edge shear. This concept is similar to the effective width used for thin-walled structures in compression. The reduction of effectiveness of the plate due to buckling is considered by replacing the shear buckling plate with an equivalent flat one. Based on the analytical results, he derives an empirical formula for a simply supported rectangular plate as a function of the average shear stress and the initial deflection.

### 2.2 *Finite Element Analysis Guidelines*

The use of the finite element method has become the standard for ship structural design. Institutionalization of the method is shown by classification societies, such as the SafeHull procedures of the American Bureau of Shipping, and the SHIPRIGHT procedures of Lloyd's Register of Shipping, and the NAUTICUS hull-design system of Det Norske Veritas, which will be separately reviewed in section 3 in our committee report. However, there are few standards or guidance that have been developed to aid the structural analyst in using this complex method of analysis. In a survey of the education of structural engineers for ship design in the United States, Yagle (1996) found that at the undergraduate level, students have a minimum exposure to the method, and that when taught at the graduate level, instruction is mixed with other applications, such as aerospace, and there is little teaching on the application of the method to ship structures.

To develop standards for finite element analysis, a series of check lists to be used in the review of these computations were developed. The check lists include the qualifications of the analysts, the type of analysis performed, the criticality of the structure being analyzed, and the computer code used. (Basu *et al.*, 1996a). The authors of this report recognize that such check lists are not a substitute for technical qualifications of analysts, but believe that such formal procedures should be incorporated when extensive analyses are performed. The check lists have been used by several government agencies in reviewing finite element analyses that have been performed under contract, and have found that the lists provide a good basis for technical review (Basu *et al.*, 1996b).

In the development of the standards for finite element analysis, authors have noted the difficulty that they have had in obtaining sample analyses to put in a guidance manual. To provide proper guidance, examples of good and bad practice in the analysis of typical ship structures, as well as validation of results through full scale and model testing is needed. However, for other than short analyses, usually done as sample problems, the resources of companies or government organizations are required for extensive such analyses. These sponsors of analyses have not been willing to release what they consider to be proprietary information for use in analysis guidelines. A mechanism is needed for the exchange of sample computations in a manner that will protect the confidentiality of the source.

### 2.3 *Pre-Post Processing*

In the recent decade, finite element method has become a common tool, and is sometimes used as a black box in the ship structural design. With the recent development of computer hardware, it is also possible to carry out finite element analysis for large scale structures such as whole ship structures.

In such background, there is a strong demand to promote the efficiency of the design and analysis process, because it is directly related to the cost of the products. For this reason, not only the cost of the analysis but also that of the pre-processing, for example mesh generation, must be reduced. Similarly, it is also needed to promote the efficiency of the post processing such as the management of the analysis result and the procedure of the re-analysis for a better design. This section describes the recent trend of pre-processing and finite element model generation based on product model. Adaptive methods are also reviewed as the post-processing.

### 2.3.1 *The Result of Investigation on Commercial Pre-Processors*

With the development of computer hardware and CAD technology, a lot of commercial pre-processors have been developed, and many engineers begin to use them in practice. The results of questionnaire about these commercial pre-processors are shown in Tables 1, 2 , and 3.

The questionnaires were carried out in July, 1995 by Kawamura and Sumi of Yokohama National University, Japan. The questionnaire sheets were send to about 25 CAE companies in Japan, and 15 companies replied on 23 pre-processors. The items of the questionnaire are,
(1) the pre-processor name and the developer,
(2) interface with external CAD systems and with solvers,
(3) mesh generation algorithms for 2-D, 3-D curved surface and 3-D solid,
(4) hardware supports, and
(5) other functions.
It is desirable for pre-processors to have an interface with external CAD systems, because it is easy to define the complex configuration, such as ship hull structures, by using CAD systems. As shown in the tables, many pre-processors have an interfaces with high-performance CAD system such as I-DEAS and Pro/Engineer, or with IGES which is a standard format for exchanging geometry between CAD systems. Many pre-processors also have an interface with solvers such as ANSYS, ABAQUS, MSC/NASTRAN, MARC etc.

In triangle mesh generation for 2-D plane or 3-D curved surface, and in tetrahedral mesh generation for 3-D solid, the most widely used algorithm is frontal method such as advancing front method. In this method, elements are generated from the boundary of the domain. The advantage of this method is that it is applicable to the constrained problem such as composite domain. In short, it is possible to generate mesh independently for each subregions of the composite domain, after the boundary of each subregion is divided taking account of the compatibility between the connected subregions. It is, however, difficult to implement this method to 3-D tetrahedrization. Delaunay triangulation is also used by many pre-processors, because it has been theoretically studied by many researchers and is easy to implement its basic algorithm.

In mesh generation for 2-D plane or 3-D curved surface, quadrilateral element is better than triangle elements from the view point of accuracy. Few years ago, most of the pre-processors are able to generate quadrilateral or hexahedral mesh only by using semi-automatic method with graphical user interface. But in the last several years, fully automatic quadrilateral mesh generator has become practical. The method adopted by many commercial pre-processors are Triqua-mesh method, Paving method (Blacker, 1991) and modified advancing front method (for example Zhu, 1991). In hexahedral mesh generation in 3-D solid, none of the pre-processors seems to have fully automatic method. Hexahedralization of 3-D solid is a subject to be studied in the near future.

### 2.3.2 *Recent Development of Mesh Generation Algorithms*

It can be said that the automation of 2-D mesh generation has already been accomplished. But algorithms of the mesh generation are still improving, and new algorithms are still being proposed especially in 3-D mesh generation. This section reviews the recent development of mesh generation algorithms.

Lo (1995) proposed automatic mesh generation algorithm for intersecting surfaces. In his algorithm, first each surface is given as a set of triangle elements. Then intersection between surfaces is found as loops of line segments, and the new triangular elements are regenerated locally around each of the intersection loops. In ship structures, mesh generation for curved surface is important, but relatively few algorithms are published for curved surface. One can refer the surface mesh generation algorithms from his reference which are applicable to ship structures. Ohtsubo *et al.* (1992) also described the mesh generation algorithm for parametric surfaces based on the frontal method.

Shimada (1993) developed a new mesh generation algorithm named bubble mesh generation. The algorithm consists of a new node placement technique and the conventional Delaunay triangulation. In the node placement stage, each mesh node is modeled as a bubble with a point mass and viscous damping, and the interacting forces like van der Waals forces between bubbles are defined. After the bubble configuration is obtained by numerically solving the governing equation of motion to yield the static force balance, the new nodes are placed at the center of bubbles. This algorithm is reasonable for generation of well-shaped triangles. Kawamura *et al.*(1994a) also proposed a new node placement technique to generate quadrilateral elements around stress concentrated area. The technique is applicable to curved surfaces.

Table 1 : Results of questionnaire about commercial pre-processors (1)

| Preprocessor Name Developer (Country) | Term and Language for the Development | Interface with External CAD Systems | Interface with Solvers | 2DMesh Generation (Triangle/ Mixed Tri & Quad/ Quadrilateral) | Mesh Generation for 3DCurved Surface (Triangle/Mixed Tri&Quad/ Quadrilateral) | Mesh Generation for 3D Solid (Tetrahedron/ Mixed Tetra & Hexa/ Hexahedron) | Hardware Support | Other Functions |
|---|---|---|---|---|---|---|---|---|
| ADINA-IN AUI<br>ADINA R&D Inc.(USA) | 1 year<br>C<br>FORTRAN | Pro/Engineer, IGES, DXF | ADINA | Mixed : Divide and Conquer Method<br>Quad : not clear | Mixed: Divide and Conquer Method<br>Quad : Semi-Automatic (with GUI) | Tetra : not clear<br>Hexa : Semi-Automatic (with GUI) Mapping Method | SUN SPARCstation, SGI IRIS IRIX, HP9000/700, IBM RS6000, Windows NT, DEC OSF/1 | |
| preprocessor for ASTEA RATCHET<br>Research Center of Computational Mechanics Inc (JAPAN) | 4 years<br>FORTRAN | | ASTEA RATCHET | Tri : Semi-Automatic (with GUI)<br>Mixed : Semi-Automatic (with GUI) | | | | |
| ATLAS<br>CRC Research Institute Inc (JAPAN) | 5 years<br>FORTRAN<br>C | IGES | ABAQUS, ADINA, ANSYS, MARC, MSC/NASTRAN, FINAS, DYNA3D, ATLAS/SOLVER, DINAS, MAGNA, NEPTUNE | Mixed : Mapping Method | Mixed : Mapping Method | Hexa : Mapping Method | SUN SPARCstation, SGI IRIS IRIX, HP9000/700, DecStation Ultrix, SONY NEWS | Automatic mesh generation by Delaunay triangulation/ Hexahedral mesh generation by extruding 2D mesh (will be implemented) |
| CADCEUS/FEM<br>Nihon Unisys, Ltd (JAPAN) | 3 years<br>C<br>FORTRAN | CADCEUS, IGES | MSC/NASTRAN, MARC, LS-DYNA3D | Tri:Semi-Automatic(with GUI) (Advancing) Front method<br>Mixed : Triquamesh Method<br>Quad : Semi-Automatic(with GUI)/Mapping Method/ Triquamesh Method | Tri:Semi-Automatic(with GUI) (Advancing) Front method<br>Mixed : Semi-Automatic(with GUI)/Triquamesh Method<br>Quad : Semi-Automatic(with GUI)/Mapping Method/ Triquamesh Method | Tetra : (Advancing) Front method<br>Mixed : Semi-Automatic(with GUI)<br>Hexa : Semi-Automatic(with GUI)/ Mapping Method | SUN SPARCstation | |
| C-Mold modeler<br>Advanced CAE Technology(USA) | 9 years<br>C | C-MOLD modeler I-DEAS, Pro/Engineer, IGES | ANSYS | Tri : not clear | Tri : not clear | | SUN SPARCstation, SGI IRIS IRIX, HP9000/700, IBM RS6000, DEC OSF/1 | |
| COMPOSIC<br>FRAMA SOFT+ CSI (FRANCE) | 7 years<br>C | IGES | COMPOSIC | Tri : (Advancing) Front method<br>Mixed : (Advancing) Front method<br>Quad : Modified Advancing Front Method | Tri : (Advancing) Front method<br>Mixed : (Advancing) Front method<br>Quad : Modified Advancing Front Method | | SUN SPARCstation, SGI IRIS IRIX, HP9000/700, IBM RS6000, DecStation Ultrix | |
| DISPLAY III<br>Engineering Mechanics Research Corp (USA) | 15 years<br>FORTRAN<br>C | I-DEAS, Pro/Engineer, IGES, MICRO-CADAM, CATIA, AUTO CAD, etc | NISA series( (NISA II, P-ADAPT, 3D-FLUID, NISAOPT, DYMES, EMAG), ANSYS, MSC/NASTRAN, MARC | Tri:Semi-Automatic(with GUI) /Modified Triquamesh Method<br>Mixed : Semi-Automatic (with GUI)/ Modified Triquamesh Method<br>Mixed : Semi-Automatic(with GUI)/Mapping Method/ Modified Triquamesh Method | Tri:Semi-Automatic(with GUI) /Modified Triquamesh Method /Modified Delaunay Method<br>Mixed : Semi-Automatic(with GUI)/Modified Triquamesh Method<br>Mixed : Semi-Automatic(with GUI)/Mapping Method/ Modified Triquamesh Method | Tetra : Semi-Automatic(with GUI)/Octree Method/Paving Method/Advancing Front Method<br>Mixed : Semi-Automatic(with GUI)<br>Hexa : Semi-Automatic(with GUI)/Extrusion of 2D Mesh/ /Mapping Method | SUN SPARCstation, SGI IRIS IRIX, HP9000/700, IBM RS6000, DecStation Ultrix, DEC VAX VMS, MS/DOS | Graphcial User Interface is improved/ NISA is executable from DISPLAY III |
| FEMAP<br>Enterprise Software Products (USA) | 9 years | IGES, AUTO CAD, DXF | ABAQUS, ADINA, ANSYS, MARC, MSC/NASTRAN, SAP IV, COSMOS, STARDYNE, ALGOR, CAEFEM | Tri : Mapping Method<br>Mixed : not clear<br>Quad : Mapping Method | Tri : Mapping Method<br>Mixed : not clear<br>Quad : Mapping Method | Hexa : Extrusion of 2D Mesh/ Mapping Method | SUN SPARCstation, SGI IRIS IRIX, HP9000/700, IBM RS6000, Windows 3.1/NT | |

Table 2 : Results of questionnaire about commercial pre-processors (2)

| Preprocessor Name Developer (Country) | Term and Language for the Development | Interface with External CAD Systems | Interface with Solvers | 2DMesh Generation (Triangle/ Mixed Tri & Quad/ Quadrilateral) | Mesh Generation for 3DCurved Surface (Triangle/Mixed Tri&Quad/ Quadrilateral) | Mesh Generation for 3D Solid (Tetrahedron/ Mixed Tetra & Hexa/ Hexahedron) | Hardware Support | Other Functions |
|---|---|---|---|---|---|---|---|---|
| FIDAP<br>Fluid Dynamics International Inc (USA) | 12 years<br>FORTRAN<br>C | IGES | FIDAP,<br>ANSYS, MARC | Quad : Paving Method | Quad : Paving Method | Hexa :<br>Extrusion of 2D Mesh/ Mapping Method | SUN SPARCstation, SGI IRIS IRIX, HP9000/700, IBM RS6000, DecStation Ultrix, Windows 3.1/NT, Dec Alpha OSF, CRAY, CONVEX, NEC EWS, NEC SX, CDC/MIPS, Fujitsu | 3D automatic mesh generation will be implemented |
| GEOSTAR<br>Structural Research & Analysis Corp.. (SRAC) (USA) | 6 years<br>C | GEOSTAR, Pro/Engineer, IGES, DXF | ANSYS, MSC/NASTRAN, SINDA, ALGOR | Tri:Semi-Automatic(with GUI) /(Advancing) Front Method<br>Mixed :<br>(Advancing) Front Method<br>Quad : Mapping Method /Modified Advancing Front Method | Tri:Semi-Automatic(with GUI) /(Advancing) Front Method<br>Mixed : Semi-Automatic(with GUI)/(Advancing) Front Method<br>Quad : Mapping Method /Modified Advancing Front Method | Tetra : Semi-Automatic(with GUI)/(Advancing) Front Method<br>Hexa : Semi-Automatic(with GUI)/Extrusion of 2D Mesh | SUN SPARCstation, SGI IRIS IRIX, HP9000/700, IBM RS6000, DecStation Ultrix, DEC VAX VMS, Windows 3.1/NT, EWS 4800 | |
| HyperMesh<br>Altair(USA) | 6 years<br>C | I-DEAS, Pro/Engineer, IGES, AUTOCAD, etc | ABAQUS, ANSYS, MSC/NASTRAN, MARC, Solvers developed by Altair | Tri :<br>(Advancing) Front Method<br>Mixed :<br>(Advancing) Front Method<br>Quad : Semi-Automatic(with GUI)/Modified Advancing Front Method(conditionally) | Tri :<br>(Advancing) Front Method<br>Mixed :<br>(Advancing) Front Method<br>Quad : Semi-Automatic(with GUI)/Mapping Method /Modified Advancing Front Method | Mixed : Semi-Automatic(with GUI)<br>Hexa :<br>Extrusion of 2D Mesh/ Mapping Method | SUN SPARCstation, SGI IRIS IRIX, HP9000/700, IBM RS6000, DecStation Ultrix, Windows 3.1/NT, etc | |
| ICEM CFD, CAE<br>CDS (Control Data Systems Inc) (USA) | 10 years<br>C, C++ | ICEM DDN, IGES, PLOT3D, VDA-FS, SET, DXF | ANSYS, Airflo3d, C-MOLD, CFD-ACE, CFDS-FLOW3D, Conca DSMC_Sandia, Datex, Fastu, Fidap, Fire, Flotran, Fluent, MSC/NASTRAN, PHOENICS, etc | Tri : Delaunay Method<br>Mixed : Paving Method<br>Quad : Paving Method (Conversion to triangles is available.) | Tri : Delaunay Method<br>Mixed : Paving Method<br>Quad : Paving Method (Conversion to triangles is available.) | Tetra : Delaunay Method<br>Mixed : Delaunay+ Automatic Block Generation<br>Hexa : Semi-Automatic by automatic Block Generation (generation of T & Y-junctions) | SUN SPARCstation, SGI IRIS IRIX, HP9000/700, IBM RS6000 | Semi-automatic hexahedral mesh generation by automatic block generation (will be implemented)/ CFD, structural Analysis /Speedy mesh generation for large scale models |
| I-DEAS<br>SDRC (Structural Dynamic Research Corporation) (USA) | 15 years<br>FORTRAN<br>C | I-DEAS<br>IGES | ABAQUS, ANSYS, ADINA, MSC/NASTRAN, MARC, I-DEAS universal file | Tri : Triquamesh Method<br>Mixed : Mapping Method<br>Quad : Semi-Automatic(with GUI)/Mapping Method /Triquamesh Method | Tri : Triquamesh Method<br>Mixed : Mapping Method<br>Quad : Semi-Automatic(with GUI)/Mapping Method /Triquamesh Method | Tetra : Triquamesh Method<br>Hexa: Semi-Automatic(with GUI)/Extrusion of 2D Mesh /Mapping Method /Triqumesh Method | SUN SPARCstation, SGI IRIS IRIX, HP9000/700, IBM RS6000, Windows 3.1/NT, DEC OSF/1, Hitachi3050 SONY NEWS | Automatic conversion of Quadrilateral to Triangles |
| I-FEM<br>Intergraph Corporation (USA) | 10 years<br>C | I-EMS, IGES, DXF, STEP, CATIA, CADDS | ABAQUS, ADINA, ANSYS, MSC/NASTRAN, MARC, ASKA, CMOLD, MOLDFLOW, COSMIC, CSA/NASTRAN, GT-SIRUDL | Tri : Automatic<br>Mixed : Automatic<br>Quad : Semi-Automatic (with GUI) | Tri : Automatic<br>Mixed : Automatic<br>Quad : Semi-Automatic (with GUI) | Tetra : Automatic<br>Mixed : Semi-Automatic (with GUI)<br>Hexa: Semi-Automatic(with GUI)/Extrusion of 2D Mesh /Mapping Method | SUN SPARCstation, SGI IRIS IRIX, Clipper(UNIX), TD(PC)+, Solarisx86 | |
| Mechanica Structure<br>RASNA(USA) | 8 years<br>C | Mechanica Structure, Pro/Engineer, IGES, DXF | Mechanica Structure | Tri:Semi-Automatic(with GUI) /(Advancing) Front Method<br>Mixed :<br>(Advancing) Front Method<br>Quad : Semi-Automatic(with GUI)/Triquamesh Method/etc. | Tri:Semi-Automatic(with GUI) /(Advancing) Front Method<br>Mixed :<br>(Advancing) Front Method<br>Quad : Semi-Automatic(with GUI)/Triquamesh Method/etc | Tetra : Semi-Automatic(with GUI)/etc.<br>Mixed : Semi-Automatic(with GUI)<br>Hexa : Semi-Automatic(with GUI) | SUN SPARCstation, SGI IRIS IRIX, HP9000/700, IBM RS6000, Windows 3.1/NT, Dec Alpha | |

Table 3 : Results of questionnaire about commercial pre-processors (3)

| Preprocessor Name Developer (Country) | Term and Language for the Development | Interface with External CAD Systems | Interface with Solvers | 2DMesh Generation (Triangle/ Mixed Tri & Quad/ Quadrilateral) | Mesh Generation for 3DCurved Surface (Triangle/Mixed Tri&Quad/ Quadrilateral) | Mesh Generation for 3D Solid (Tetrahedron/ Mixed Tetra & Hexa/ Hexahedron) | Hardware Support | Other Functions |
|---|---|---|---|---|---|---|---|---|
| MENTAT-II<br>MARC Analysis Research Corporation(USA) | about 20 years<br>C, C++ | I-DEAS, Pro/Engineer, IGES, ACIS SAT | MARC, MARC/Design | Tri : Delaunay Method<br>Quad : Overlay Method | Tri : Delaunay Method<br>Quad : not clear | Tetra : Delaunay Method | SUN SPARCstation, SGI IRIS IRIX, HP9000/700, IBM RS6000, DecStation Ultrix, DEC VAX VMS, Windows NT, DEC OSF/1, NEC EWS, SONY NEWS | Developing automatic 3D mesh generation |
| MSC/ARIES<br>MSC(USA) | 12 years<br>C, C++ | Modeler is included, IGES, DXF, STEP, ACIS SAT, CATIA | MSC/NASTRAN, MSC/EMAS, MOLDFLOW, ADAMS | Tri : Semi-Automatic(with GUI)/not public<br>Mixed : Semi-Automatic(with GUI)/not public<br>Quad : Semi-Automatic(with GUI) | Tri : Semi-Automatic(with GUI)/not public<br>Mixed : Semi-Automatic(with GUI)/not public<br>Quad : Semi-Automatic(with GUI) | Tetra : Semi-Automatic(with GUI)/not public<br>Mixed : Semi-Automatic(with GUI)<br>Hexa: Semi-Automatic(with GUI)/Extrusion of 2D Mesh | SUN SPARCstation, SGI IRIS IRIX, HP9000/700, IBM RS6000, DecStation Ultrix, DEC OSF/1, Windows NT(from 95) | Examining automatic 3D hexahedral mesh generation |
| MSC/PATRAN<br>MSC(The MacNeal-Schwendler Corporation) (USA) | about 20 years<br>C<br>FORTRAN | Modeling Systems of MSC, Pro/Engineer IGES, CADDS5, CATIA, Euclid3, Unigraphics | ABAQUS, ANSYS, MSC/NASTRAN, MARC, DYNA | Tri:Semi-Automatic(with GUI) /(Advancing) Front Method<br>Mixed : Semi-Automatic(with GUI)<br>Quad : Semi-Automatic(with GUI)/Mapping Method /Paving Method | Tri:Semi-Automatic(with GUI) /(Advancing) Front Method<br>Mixed : Semi-Automatic(with GUI)<br>Quad : Semi-Automatic(with GUI)/Mapping Method /Paving Method | Tetra : Semi-Automatic(with GUI)/Octee Method /(Advancing) Front Method<br>Mixed : Semi-Automatic(with GUI)<br>Hexa : Semi-Automatic(with GUI)/Extrusion of 2D Mesh /Mapping Method) | SUN SPARCstation, SGI IRIS IRIX, HP9000/700, IBM RS6000, DecStation Ultrix | Developing automatic 3D hexahedral mesh generation based on Advancing Front Method |
| MYSTRO<br>FEA(UK) | 12 years<br>C<br>FORTRAN | DXF | LUSAS | Tri : Regular and Irregular<br>Mixed : Transition and Irregular(automatic mesh generation based on division numbers of edges)<br>Quad : Regular and Irregular | Tri : Regular and Irregular<br>Mixed : Transition and Irregular(automatic mesh generation based on division numbers of edges)<br>Quad : Regular and Irregular | Tetra : Regular<br>Mixed : Transition for 1 direction(automatic mesh generation based on division numbers of edges and surfaces)<br>Hexa : Regular | SUN SPARCstation, HP9000/700, DecStation Ultrix, DEC VAX VMS, PC | Mesh gradation requirement is handled by weighting to the background nodes |
| PAM-GENESRIS<br>(only for setting analysis conditions)<br>ESI (FRANCE) | | only mesh information is readable I-DEAS, PATRAN, HYPERMESH, (NASTRAN, ABAQUS, and DYNA format files are readable) | PAM-CRASH(SAFE), PAM-STAMP | | | | SUN SPARCstation, SGI IRIS IRIX, HP9000/700, IBM RS6000, Dec Alpha | |
| PRE2D,3D (preprocessor for ASTEA MACS)<br>Research Center of Computational Mechanics Inc. (JAPAN) | 3 years<br>C<br>FORTRAN | ASTEA MACS | | Tri:Semi-Automatic(with GUI)<br>Mixed : Semi-Automatic(with GUI)<br>Quad : Semi-Automatic(with GUI) | | Tetra:Semi-Automatic(with GUI)<br>Mixed : Semi-Automatic(with GUI)<br>Hexa : Semi-Automatic(with GUI) | | |
| QUEF<br>QUINT(JAPAN) | 8 years<br>F, C | | Solvers of QUINT | Quad : Mapping Method | Quad : Mapping Method | Hexa : Mapping Method | SUN SPARCstation, SGI IRIS IRIX, HP9000/700, IBM RS6000,Dec Alpha, Macintosh | |
| preprocessor for Quick Therm<br>Research Center of Computational Mechanics Inc. (JAPAN) | 2 years<br>Quick Basic<br>Quick C<br>Visual Basic | Quick Therm | Quick Therm | Mixed : Delaunay Method | | Hexa : Extrusion of 2D Mesh | Windows 3.1/NT | |

As shown in the previous subsection, semi-automatic mesh generation method is adopted in many commercial pre-processors. In most of the semi-automatic methods, the complex domain is first subdivided into simple subregions, which are then meshed separately using conventional mapping methods. Because the each mesh in the subregion imposes constraints on the numbers of subdivisions on the boundary, each subregion must satisfy the compatibility with the adjacent subregions. It is time consuming to satisfy the compatibility in the domain with many subregions, because the user always have to set the edge subdivisions manually with graphical user interface. Tam and Armstrong (1993) indicated an effective method to satisfy these constraints. The method is based on an integer programming technique, and can be applied to 2-D, 3-D curved surface and 3-D mesh generation.

Though a lot of mesh generation algorithms has been proposed by many researchers, there exists no completely automated algorithm for 3-D hexahedral mesh generation. Price and Armstrong (1995) proposed a new method for subdividing a solid into topologically simple subregions which is suitable for meshing with hexahedral elements. This method is based on the Euler's equation corresponding to the topological property of a solid, and uses medial surface to define the subregions. The application of this method is limited by the topology of a solid, but it is one of the good solutions for 3-D automatic hexahedral mesh generation.

The 2-D Delaunay triangulation is widely used, because it produces well-shaped triangles (small angles in triangles are avoided) and satisfies the max-min angle criterion. In 3-D tetrahedral mesh generation by the Delaunay triangulation, however, the criterion is not satisfied and the method sometimes produces sliver elements. To avoid this drawback, Joe (1991) first proposed a new mesh generation method based on local transformation technique, which is named max-min solid angle triangulation. The local transformation technique is also described by Kettunen and Forsman(1995). Furthermore, Golias and Tsiboukis(1994) proposed the mesh refinement procedure with similar transformations and the node relaxation technique. It is expected that such local transformation technique is adopted to 3-D tetrahedral mesh generators based on Delaunay triangulation. In Delaunay triangulation, the original Watson's method is a standard algorithm and widely applied to 3-D problem. This algorithm itself is, however, not always applicable to the domain which is constrained by the surface mesh, adjacent objects with different material properties, and degenerate node configurations. Wright (1994) indicates a good algorithm to avoid illegal cases when the Delaunay algorithm is applied to these situations. This kind of algorithm is called constrained Delaunay triangulation, which is also an important algorithm to construct reliable 3-D mesh generators.

Some improvements in advancing front method are proposed by Jin and Tanner (1993) and George and Seveno(1994). Both of them pointed out the classical difficulties in advancing front method such as the difficulty in controlling the element size or the inefficiency of searching process. George and Seveno proposed a control space generated by the Delaunay triangulation, and succeeded in controlling the size of generated element. They also proposed a neighborhood space based on the tree data structure to search nodes and edges effectively.

Frey *et al.*(1994) and Yeker and Zeid(1995) described the 3-D mesh generation algorithm based on the decomposition of the space by the collection of cubes. These algorithms can be regarded as the improvement of the octree method. In octree method, the domain is enclosed in a cube which is recursively subdivided into octants. Because cubes are not subdivided in the proposed method, the new algorithms are simpler than the octree method.

Recently, new analysis method named Element-Free Galerkin Method, by which analyst does not need to generate mesh of the domain, has been proposed by Belytchko *et al.*(1994). In Domain Composition Method proposed by Charlethworth *et al.* (1994), it is permitted to generate overlapped mesh between subregions of the analysis domain. Though these methods are very interesting, they are still in a research stage at the moment. It is expected that the scale of problems will become larger, and 3-D solid analyses will be carried out for more complex geometry as the computing capability develops. Therefore, the mesh generation, especially 3-D hexahedral mesh generation, becomes more important, and worth studying in the next several years.

### 2.3.3 *Finite-Element Model-Generation Based on Product Model*

For the improvement of productivity of shipbuilding industries, there is a trend toward the use of 3-D CAD systems. At the same time, standards of product data have been gradually developed for the exchange of digital data among different organizations. With the advancement of these computer technologies, the need is growing for the research on the Finite Element Modeling associated with the Product Model. Nomoto and Takechi (1993) and Aoyama *et al.*(1995) implemented a structural analysis system based on a product definition system which was originally developed for the manufacture of ship structures. The information of the product model are revised and the structural functions of structural members are introduced so that the finite element models can be generated by the system.

Ohtsubo *et al.* (1991, 1992) and Kawamura *et al.* (1993, 1994b) developed an object-oriented finite element modeling system for ship structures. This modeler is based on two new concepts, part object and object-oriented data structure. Ship structure is defined as a collection of part objects and their relations. Each part-object consists of the geometry and analysis conditions such as boundary conditions, loads, material and degree of mechanical importance. All the data necessary for the analysis, as well as the generated finite element, and the relations among the data are treated as the objects. Fully automated mesh generations are implemented by applying frontal method to each part-object.

Further it is shown by Ohtsubo *et al.* (1994, 1995) that different finite element models with varying idealization levels can be automatically generated from a uniquely defined product model by introducing "Part Name" and "Part Type" concepts. Fully automated and flexible finite element modeling systems based on the product model, as described above, can be expected not only to reduce the analysis cost, but also to reduce the subjective uncertainties through the resulting standardization of the analysis models.

### *2.3.4 Adaptive Methods*

Extensive studies have been carried out in the basic problems areas of adaptive finite element analysis. For the posterior error analysis (Kitamura *et al.* 1993; Babuska *et al.* 1993), an objective validation principle of posterior error estimators, using the robustness index is proposed, and numerical validation of various estimators, so far proposed, are carried out for the case of general meshes (Babuska *et al.* 1994). Mathematical investigation of the quality of error estimators based on averaging techniques is presented by Babuska *et al.* (1993). A simple technique suitable to the practical use for estimating a posterior error is proposed by Kitamura *et al.* (1993), based on the local equilibrium of residuals.

As for the mesh refinement, an adaptive global-local refinement strategy, aiming at the analysis of local phenomena, is presented by Fish(1994). Algorithm for adaptive refinement of triangular element (Nambia *et al.* 1993), solution techniques for the p-version analysis method (Papadrakakis 1994), and the important problems of singularities (Gua *et al.* 1994), and boundary stress extraction (Fukuda 1993; Wiberg *et al.* 1994; Niu and Shephard 1994) have also been researched, as well as the extensions to shell element problems (Leino 1994), nonlinear problems (Kitamura *et al.* 1995; Mucke *et al.* 1995), and coupled problems (Demkowics and Oden 1994).

Practical adaptive systems have been implemented for 3-D linear elastic problems (Kocvra 1993; Pressburger and Perruchio 1995). Fully automated adaptive FE analysis system is developed for homogeneous elastic solid, integrating geometric solid modeler, mesh generator, analysis processor, error evaluator and analysis control processor (Pressburger and Perruchio 1995), where the adaptive process is implemented by using h-p multigrid method for linear and quadratic tetrahedron elements (David and Pressburger 1993). In the reference by Kocvra (1993), a newly developed multi-grid technique is applied for an analysis system of 3-D linear elastic problems using quadratic hexahedral elements.

Along with these development of practical tools, the adaptive finite element analysis is expected to become more common in structural design.

## *2.4 Reliability Analysis*

A co-operative research project called Reliability Methods for Ship Structural Design (SHIPREL) funded by BRITE/EURAM was aimed to develop reliability based methods for ship structural design, in particular by calibration of design rules, the results of which were reported in a special issue of Marine Structures (Guedes Soares, 1996). There were ten papers in this special issue and some of them are of interest to this committee report.

The paper by Guedes Soares and Dias (1996) dealt with the probabilistic models of still water loads and it was concluded that these models are ship dependent and mainly governed by the ship length. Furthermore, on a given ship or class of ships, different probabilistic models may be applicable, depending on the routes. The paper by Schellin *et al.* (1996) dealt with the uncertainty in the predictions of linear wave induced load effects, including both two dimensional ship theory and a three-dimensional panel method while a paper by Guedes Soares *et al.* (1996) dealt with the formulation for long-term assessment of non-linear wave induced vertical wave effects. The paper by Soares and Garbatov (1996) dealt with the fatigue limit state of the primary hull structure with time variant formulations and includes the corrosion effect.

Pu *et al.* (1996a) proposed an algorithm for the ultimate strength of analysis of stiffened plates which is calibrated using considerable amount of experimental and numerical data, and it was found that there was a considerable reduction in the model uncertainty factor. Reliability analysis was then carried out using Advanced First Order Second Moment method (AFOSM), the Second Order Reliability method (SORM) and Monte-Carlo simulation to investigate the accuracy of first and second order methods. Structural system reliability method was applied to a built SWATH ship by Pu *et al.* (1996b), where a typical transverse frame in the SWATH was idealized by a two-dimensional model through a series of finite element analyses. The significant failure modes were identified by an extended β-unzipping method, and the system reliability index was then evaluated by Ditlevsen's bounds. It was found that the most critical section is in the haunch area.

The paper by Kaminski *et. al.* (1993) described a method for the probabilistic fatigue design of semi-submersibles which took into account the total life cycle cost consisting of initiations cost, inspection risk, repair risk and failure risk. Application of the system was made to an existing design, and the system can allow systematic parameter variations of the design approach and can investigate the consequences of such variations on the life cycle cost.

The paper by Gierlinski *et al.* (1993) demonstrated a software RASOS for the structural integrity analysis of fixed offshore structures using both deterministic and probabilistic approaches. The assessment includes studies of structural redundancy and damage tolerance, and the system incorporates various computational tools of environmental load modeling, linear elastic analysis, deterministic progressive collapse and system reliability analysis. The paper by Lee *et al.* (1993) described an algorithm for the reliability analysis of structural systems, including offshore structures. The response surface approach is used and investigation of several different forms of response surfaces are presented.

## 3 REVIEW OF STRUCTURAL DESIGN GUIDELINES

### *Ship Structural Analysis Systems Developed by Classification Societies*

To cope with the ever increasing demands of the ship performance and safety issues, several structure analysis procedures have been developed by classification societies to assist designers in assessing the strength and to streamline the classification works. All the systems have been developed or refined in the recent couple of years so that reports of experiences from users are still limited at this moment. Among those developments, the systems issued by ABS, LR, BV, NK, DNV, and GL are described in this section.

**System of ABS**

ABS introduced structural analysis procedure DLA(Dynamic Loading Approach) in 1991. The key part of this approach is the determination of realistic dynamic loads and the load combination effects acting on a ship structure. The procedure is basically a full ship FEM approach. Strength is evaluated at each selected dominant load parameter in an extreme condition in a design life. The principles of the approach can be applied to the most ship types for the evaluation of structural strength in a presumed design life.

Based on the experience of DLA and by directly formulating the dynamic load, ABS launched the SafeHull system as a highly streamlined tool for the design and evaluation of structural strength of tanker and bulk carrier greater than 150 m in length (ABS, 1993b, 1995b). A set of PC-computer software program has been developed and distributed along with the guidance books which elaborate a new set of strength standard for the determination scantlings of structural members based on a systematic considerations of dynamic load effects. As Tsuda (1995) explained, in order to investigate the strength of various structural members by the first principles, responses from dominant dynamic loads should be assessed properly. That is achieved firstly by a simplified engineering method including the assessment of critical loads of each members, response assumption, and acceptance criteria for sizing the scantlings. Secondly, the resulting scantlings will be used as preliminary scantlings for the further analysis which includes several three-hold 3-D FEM analyses and subsequent 2-D refined analyses. Combined effects of full structure under instantaneous dynamic loads are considered. Stress response of each member is subjected to the detailed strength assessment including yielding, buckling, and ultimate strength. Such two-step evaluation procedures are called Phase A and Phase B analyses, respectively.

Also a "net ship" concept has been applied in the SafeHull, where analyses are executed with "net scantlings." Final design scantlings are the sum of the net scantling and the nominal corrosion margin in various positions, which are based on the average corrosion deduced from the data base. Fatigue strength of longitudinal stiffeners with different end attachments can also be investigated in the Phase A process, where a standard design life is assumed as 20 years. A good correlation between predicted lives and service data has been reported by Conlon (1992).

In Phase B, 3-hold models are required, which extend one hold forward and one hold afterward in order to minimize the boundary effects. Only mid-hold results are used for strength assessment. Mesh sizes of the models basically follow the arrangement of supporting girders and transverse web frames. Membrane plate and rod elements which represent the stiffeners are used all over the model. Beam elements are used in the locations where out of plane stiffness is important (ABS, 1993b, 1995b). There are 8 loading conditions for a tanker and 10 loading conditions for a bulk carrier defined as the basic design conditions to be investigated.

Final stresses subject to failure mode check are composed of primary stress, secondary stress and tertiary stress, which include all the possible contributions of stress elements in the linear elastic field.

By the explicit assessment procedure for each structural element, it is possible to achieve a uniform safety throughout a ship. The tanker and bulk carrier versions of the SafeHull guidance have been incorporated in the Rule as mandatory part from May, 1996 (ABS, 1996a). The container ship part of SafeHull guidance has just been issued and can be applied on an optional basis at the time of December, 1996 (ABS, 1996b)

**System of LR**
LR has recently launched SHIPRIGHT procedures, which include a wide range of application. For tankers and bulk carriers greater than 190m in length, structural design assessment (SDA), and fatigue design assessment (FDA) are the two mandatory analyses in the SHIPRIGHT procedures (LR, 1994, 1995, 1996a, b).

SDA is a FEM analysis procedure, which specifies the modeling method, the loading conditions critical to various structural members and corresponding acceptance criteria for the design. Models should in general be extended half hold before and after the mid hold to minimize the boundary effect. Mesh sizes follow the panel-stiffener arrangement (LR, 1996a). Plate, rod and bar elements are used to represent the in-plane stiffness and out-of-plane stiffness of the structure. Responses of the structural elements are subjected to yielding and buckling checks, considering the combined stresses effects from both the local and global hull girder load-effects.

FDA includes three levels of assessment (LR, 1995, 1996b). In most cases, the fatigue strength of stiffener connections needs to be checked by level 2 assessment. A PC-based program is developed in assistance with the investigation in the level 2 stage which is basically a first principles approach. Responses at hot-spot locations of stiffeners in way of end attachments are assessed by simplified FEM beam model or by stress concentration factors from the internal data base (LR, 1995). Rigorous procedures are performed in wave load and motion estimation, voyage simulation, short and long term predictions in order to achieve the stress histogram for assessing the accumulated fatigue damage in various locations. The program runs under MS-Windows with a graphic front end.

**System of BV**
BV launches the VERISTAR system with a set of PC-based software to assist a designer applying the rule strength requirements (BV, 1996). 3-D FEM analyses are also applied to three-hold models. Longitudinal elements of the model can be extruded conveniently from the MARS input data, which is the check routine of the Rule based scantlings. Transverse members and bulkhead structures can be built into the model interactively or manually, and visualized promptly on the screen. The mesh size of the model basically follows the arrangement of major supporting structures. All the nodal forces and boundary forces of the loading conditions specified in the rule can be imposed on the model automatically. Internal check routine is also included providing a convenient evaluation tools to display the insufficiencies of the scantlings so that design improvements can be made easily. Procedure for a tanker and a bulk carrier have been announced at the time of September, 1996.

**System of NK**
The most significant system developed by NK is the PrimeShip-ASSAS, a procedure which starts from wave load calculation to global 3-D full ship model analyses and to the local strength assessment (Tsutsui *et al.*, 1996). Efficient modeler and interface programs (translator) between each steps of analyses are also developed. Strength of structural elements based on yielding, buckling, and fatigue criteria can be assessed in an integrated post-processing program.

**System of DNV**
Computational Ship Analyses(CSA) procedure is the structural analysis system developed by DNV, which is applicable to a tanker, a bulk carrier and a container vessel (DNV, 1995, 1996a). There are two levels of application, CSA-1 and CSA-2. CSA-1 is a refined rule requirement procedure which requires two-hold model subjected to the rule-defined loads and the rule-specified acceptance criteria, while CSA-2 is a sophisticated procedure which requires full ship model applying wave loads and motions derived from direct wave load analysis including more detailed buckling and fatigue strength acceptance criteria. A design life of 20, 25 or 30 years can be selected in the CSA process. Besides, the CSA procedure requires the hull girder bending strength evaluation based on the ultimate collapse strength both under intact and damaged conditions, which is unique among the classification societies. A highly integrated PC-based software tool "NAUTICS HULL" is now under development, which includes the full sets of CSA procedures for assisting industries in applying the rigorous procedures.

**System of GL**
Direct calculation method of GL (Payer and Fricke, 1994, Franc and Fricke, 1995) is an FEM approach which

uses overall or partial hull models and is especially suitable for the structural analysis of modern cargo ships with pronounced line curvature and ships with long hatch openings. Within the approach, an advanced design wave approach is adopted to simulate the ship operating in the most unfavourable wave situations during the selected design life. The results of the finite element analysis are checked with respect to deformations, permissible stresses, buckling strength and fatigue strength at structural details. A new PC-based design tool, named POSEIDON, has been launched by the end of 1996. The first version allows the description of the structural geometry of any ship type and the automatic determination of scantlings based on hull girder bending moments and shear forces as well as local loads. The generation and analysis of partial hold models is under development.

# 4 OFFSHORE STRUCTURES

## *4.1 Introduction*

This chapter reviews the main development within fatigue behavior of tubular joints, local joint flexibility, buckling, and pile-sleeve grouted connections.

## *4.2 Tubular Joints*

### *4.2.1 General*

Today designers can approach fatigue problems in two different ways with varying degree of refinement. The simplest and by far most applied procedure is to use a hot spot stress method in association with S-N curves and a Miner-Palmgren approach. This method represents the industry standard, and has lately been thoroughly updated based on extensive experimental and theoretical work carried out during the last decade.

The more complicated and hitherto less used method is to base the fatigue design on a fracture mechanics analysis. Progress is being done in this field, but mainly on an academic level for investigation of special effects, or in association with advanced inspection planning programs. Due to the inherent uncertainties of the initial flaw size, crack growth and threshold effects, a fracture mechanics analysis normally requires calibration based on tests. Alternatively, it may be used in comparative analyses for which the (normally unknown) initiation conditions are identical.

For both the above methods the development mainly takes place within three fields: 1) simple joints and basic approaches, 2) multiplanar joints, and 3) internally stiffened tubular joints. This section aims at covering major aspects of 1), while 2) and 3) are not covered in detail.

### *4.2.2 Hot Spot Stress Approach*

Since the mid 1960's extensive research has been conducted on large tubular joints. Most of the early work was conducted in the United States, but since the mid 1970's, the majority of the work has been performed elsewhere. In particular the European Coal and Steel Community (ECSC) has placed studies in many European countries, and in parallel similar work has also taken place in other countries (Dimitrakis and Lawrence, 1994).

Soon after the beginning of the North Sea oil and gas exploration and production it became clear that the design principles then used for Gulf of Mexico steel structures did not adequately cover the fatigue problems in the much more hostile North Sea environment. This started a development with a strong focus on fatigue problems, and especially Great Britain and Norway have carried out extensive research and development in order to establish a more reliable basis for the design of welded offshore structures.

In the early days most of the fatigue testing was performed on rather small models, which with hindsight has turned out to be insufficient. A milestone was passed in 1984 when the Department of Energy's code (Offshore Installations, 1984) incorporated the first full-scale test results in the S-N curve for tubular joints, the T curve.

The T curve was a design S-N curve for tubular joints in air to be used together with extrapolated hot-spot stresses. The effects of sea water and cathodic protection was taken into consideration by using a life reduction factor of 2 on air data for unprotected welded joints, while cathodic protection was assumed to restore "in air" behavior. Also, the code gave guidance on size effects by introducing a thickness reduction factor on the stress range for thicker members.

Since 1984 more large scale tests have been conducted, and significant effort has been devoted to screening processes of the test databases for tubular joints, (Stacey and Sharp, 1995a; Dimitrakis *et al.*, 1995; Lalani, 1992). This has established a greatly improved background for reliable fatigue calculations, but many of these results have yet to be implemented in design codes.

According to Stacey and Sharp (1995a) the principal codes for offshore steel structures are now Guidance on the Health and Safety Executive (Offshore Installations, 1990; Offshore Installations, 1995), API RP 2A (API RP2A, 1993), CAN /CSA-S473-92 (CAN /CSA-S473-92, 1992) and the standard being developed by the ISO TC67/SC7/WG3 Joints/Fatigue Technical Core Group. Of these the currently most developed is the Guidance on the Health and Safety Executive document. The detailed background to the HSE guidance is published in Fatigue Background Guidance Document (1995). Because the new HSE fatigue guide represents state of the art, and the fact that the latest US research recommends these results to be implemented in API RP2A, the following will generally be based on Stacey and Sharp (1995a), with comments made to other sources where appropriate.

**S-N Curves**

One curve is defined for all tubular joints, the T' curve. It has been derived from a database with chord diameters D > 150 mm, chord thicknesses T > 16 mm, and from specimen that failed at the chord. The majority of the joints in the database had chord wall thicknesses of 16 mm and 32 mm, and the fatigue life is defined as the number of cycles corresponding to through-wall cracking.

The T' curve includes the local weld notch effect, but not the geometric stress concentration factor, which must be calculated separately in each case. It applies directly to joints with plate thicknesses up to 16 mm, for greater thicknesses a thickness correction factor is used.

Three basic T' S-N curves are defined: 1) fatigue in air, 2) fatigue in sea water with cathodic protection, and 3) fatigue in sea water with free corrosion (see Figures 1 and 2). The S-N curves are given by

$$Log_{10} N = Log_{10} K - m \cdot Log_{10} S_B \tag{4}$$

The slope of the fatigue curves, $m$, has been fixed at the value 3. The T' curves have been found as the mean S-N curve from the data base minus two standard deviations of $Log_{10} N$. For fatigue in air and for fatigue in sea water with cathodic protection there is a change in slope at $(N_o, S_o)$ where $m$ changes from 3 to 5. There is no change in slope for fatigue in sea water with free corrosion. The full range of constants for defining the S-N curves are given in Table 4.

Table 4 Constants for T' design S-N curve (from Stacey and Sharp, 1995a)

| Environment | $S_o$ (MPa) | $N_o$ (Cycles) | $S_B \geq S_o$ | | $S_B \leq S_o$ | |
|---|---|---|---|---|---|---|
| | | | $Log_{10}K$ | m | $Log_{10}K$ | m |
| Air | 67 | $10^7$ | 12.476 | 3 | 16.127 | 5 |
| Seawater (cathodic protection) | 95 | $1.745.10^6$ | 12.175 | 3 | 16.127 | 5 |
| Seawater (free corrosion) | | | 12 | 3 | | |

There is no direct weld shape parameter influence as in API RP2A, but in the HSE 1990 guidance it is stated that an improvement of a factor 2.2 on life can be obtained by controlled machining or grinding of the weld toe. It is specifically recommended not to take this effect into consideration at the design stage.

The current API RP2A (API RP2A, 1993) uses the X and the X' curve where the X' curve is used for tubular joints which do not have weld profiles that smoothly merge with the base metal at the weld toe. These curves were originally obtained from small scale tubular joints and fillet welded plates tested in air under constant amplitude loading. In addition they were intended to be used with a different hot spot stress definition than the one now used. The API curves were based on a Local Hot Spot Stress Range definition where stress (strain) was directly measured as close as practically possible to the weld toe (Dimitrakis and Lawrence, 1994). However, it is now widely recognized that these curves are too conservative (Dimitrakis *et al.*, 1995).

After inclusion of full scale tests and a careful screening where in particular the very thin chord thickness data

(6 mm) have been excluded, recent American research (Dimitrakis *et al.*, 1995) has found similar results to those in the HSE 1990 guidance. They therefore recommend the use of the T'' curve in future API RP2A revisions.

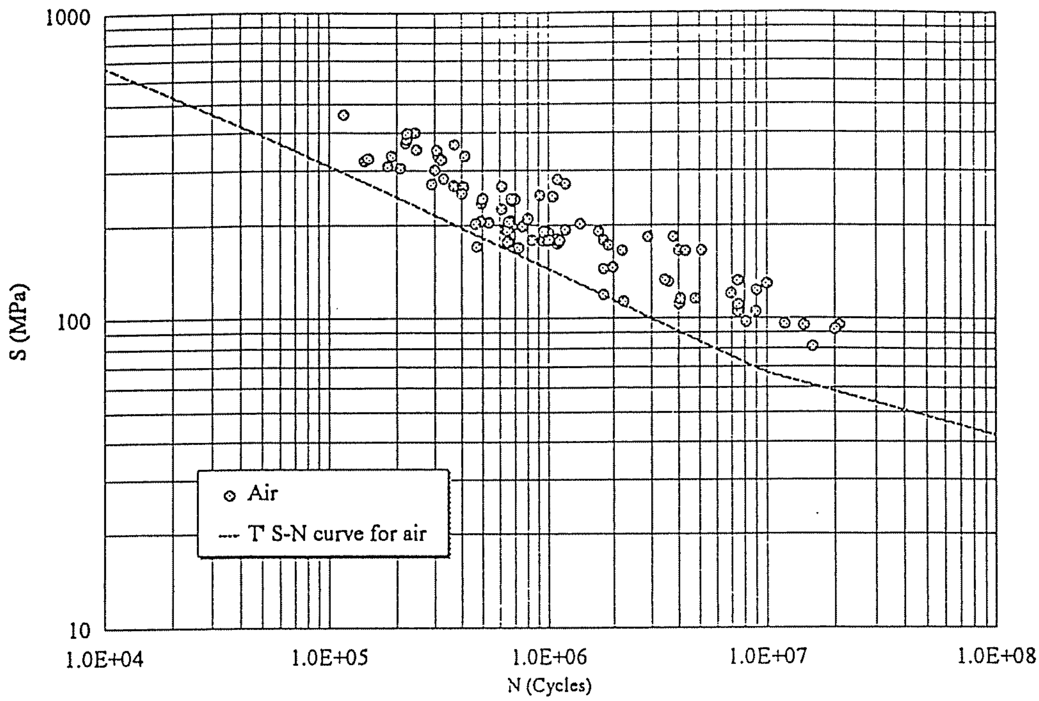



Figure 1 T' S-N curve for air (from Stacey and Sharp, 1995a)

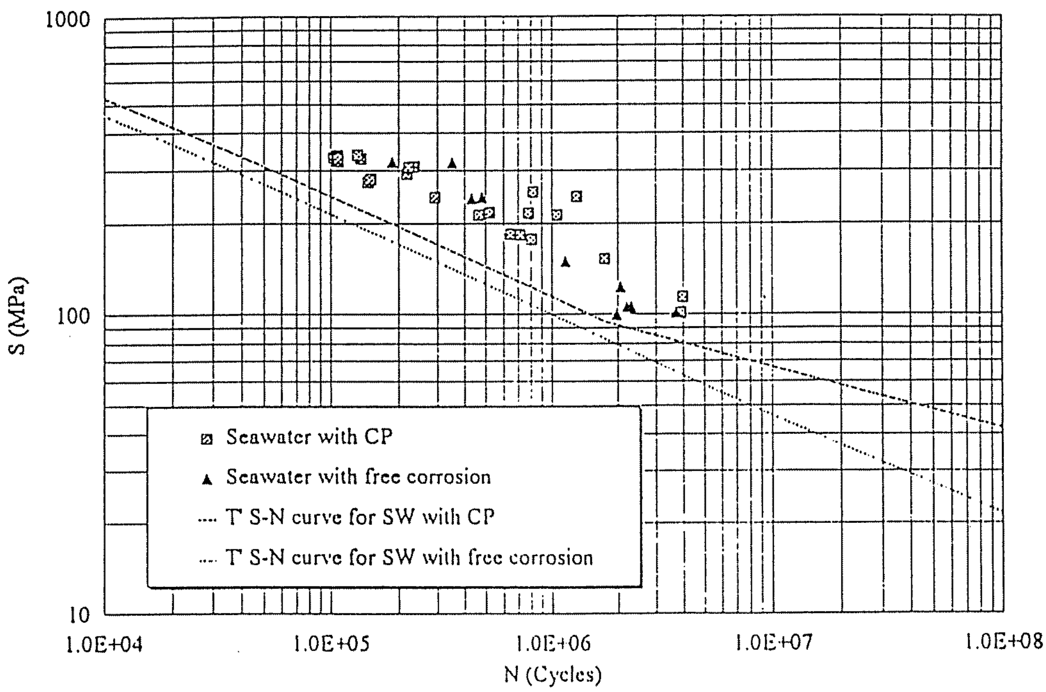



Figure 2 T' S-N curve for sea water (from Stacey and Sharp, 1995a)

**Hot Spot Stress Range**

European research concurrently recommend the use of a linearly extrapolated Geometric Hot Spot Stress Range. The extrapolation is performed with stresses (strains) measured by two strain gages (A and B in Figure 3) located in the region of stress linearity. The local weld notch effect due to the weld shape is included in the S-N curve. Both the HSE guidance and Det Norske Veritas (Fatigue Handbook, 1985) follow these recommendations, but they recommend different values of the parameters a and b. Normally the same extrapolation points are used for experimental values and for calculated (FEA) values.

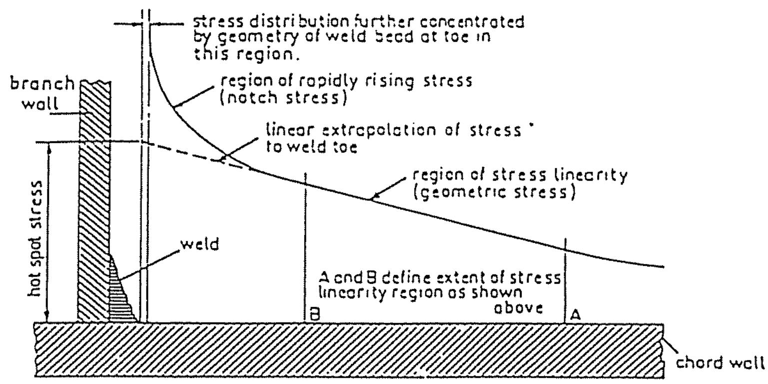



Figure 3 HSE hot spot stress range definition (from Offshore Installations, 1982)

**Thickness Effect**

It is now generally accepted that thicker members have a shorter fatigue life than thinner, i.e. for joints with thicknesses exceeding the basic thickness ($t_B$) a penalty factor should be applied. This is done by modifying the stress range $S_B$ with a thickness correction factor so that the corrected S-N curve is given by

$$Log_{10}N = Log_{10}K - m \cdot Log_{10}\left(\frac{S_B}{\left(\frac{t_B}{t}\right)^q}\right) \tag{5}$$

In this equation $t_B = 16$ mm and $q = 0.3$ should be used.

The thickness correction factor should be used for the member under assessment (chord or brace). The API (API RP2A, 1993) introduces a somewhat similar thickness correction on chord fatigue life based on the brace thickness, but this approach is not recommended. Instead, the above HSE approach is recommended for future API RP2A revisions (Dimitrakis *et al.*, 1995).

The thickness effect has been subject to several test programs in recent years, and it has been demonstrated that the previously used value of $q$ equal to 0.25 is not necessarily conservative for very thick joints. Recent research (Stacey and Sharp, 1995a; Dimitrakis *et al.*, 1995) tend to agree on a value of 0.29 or 0.30, which has been implemented as 0.3 in Offshore Installations (1990). It is known that the thickness effect is not fully described by the above simple equation since it also depends on the joint geometrical parameters and the SCF. However, due to the limited data available a thickness correction factor of 0.3 is recommended unless supported otherwise by experiments or fracture mechanics analyses.

**Effect of Environment**

New data covering the effect of sea water have become available and have been normalized to 16 mm thickness using a thickness exponent of 0.3. The results show that the fatigue lives for tubular joints in sea water are lower than in air. Tubular joints with cathodic protection have been found to have fatigue lives of approximately half of those in air for lives less than $10^6$ cycles, and approximately equal lives for higher lives.

For tubular joints with free corrosion a significant reduction on fatigue lives has been found. A mean reduction factor of 3 on both short and long lives in combination with a constant-N curve slope is recommended (Stacey and Sharp, 1995a; Dimitrakis *et al.*, 1995). These findings have been implemented in the new HSE fatigue guide using the details given in Table 4.

**Stress Concentration Factors**

In practical design work SCF's are normally derived from parametric equations whenever possible, alternatively determined by FE analysis, or quite rarely, determined by experimental methods. It has long been known that the choice of stress concentration factor is crucial in any fatigue analysis, and the benefits of the different SCF formulae have been the topic of many papers. Some pragmatic insight was established when Tolloczko (1991) performed a review of existing SCF formulae and gave clear recommendations in favor of the Underwater Engineering Group (1985) and Efthymiou (1988) equations.

When FE analysis is used to determine SCF's it is often found that the SCF's predicted by parametric formulae are somewhat conservative, but this is unfortunately not always the case. The 1990 HSE fatigue guide has taken the consequence of this, and reviewed the application of a large number of parametric formulae for accuracy and reliability. The result was that only the equations given by Efthymiou (1988) and Lloyd's Register (Smedley and Fisher, 1991) could be recommended. Neither of the SCF sets could be recommended for all situations.

### High Strength Steels

Very limited fatigue data are available for high strength steels, and according to Stacey and Sharp (1995a) the fatigue performance of higher strength steels cannot be confidently predicted. It is thought that the effect of sea water on fatigue performance for high strength steel is more detrimental than for medium strength structural steel due to a greater susceptibility to hydrogen cracking. This susceptibility is known to increase with increasing yield strength and increasingly negative cathodic protection potential. The 1990 HSE guidance therefore only recommends its T' curves to be used for steels with minimum guaranteed yield strength of up to 400 MPa for tubular joints. For higher yield strength steel it recommends data from an approved test program or a fracture mechanics analysis to be applied.

However, indicative results from a small test series on large scale double T-joints in high strength steel have now become available (Agerskov *et al.*, 1994). The investigation compared double T-joints subjected to inplane loading of conventional offshore structural steel with a yield strength of 363-381 MPa to similar joints made from high strength steel with a yield stress of 820-830 MPa. The results showed fatigue lives for the high strength steel joints to be between 0 and 20 % longer than for the joints of conventional steel. It should be emphasized that the number of test results was too limited to allow any significant statistical analysis to be performed.

### Limiting Miner Sums

Most often fatigue damage accumulation in offshore structures is determined from the Miner sum without special consideration to the details of spectrum loading from the underlying wave action. This means that a Miner sum equal to unity is taken to predict the endurance to through-thickness cracking.

Newer research questions this approach. In Agerskov *et al.* (1994) tests with different types of spectrum loading were carried out on both ordinary steel and high strength steel double T-joints subjected to inplane loading. Fatigue tests with variable amplitude loading were carried out for both narrow-banded and broad-banded spectra. The variable amplitude loading was generated by a computer program that simulates a stationary Gaussian stochastic process in real time. The irregularity factor $I$ was defined as the number of positive-going zero-crossings divided by the number of maxima of the stress history. A narrow band loading has a irregularity factor close to unity, while the load spectra considered for the tubular joints were based on the PM wave spectrum and had irregularity factors of $I = 0.82$-$0.84$. The number of tests were too limited to draw firm conclusions, but there was a clear tendency that the Miner sum decreases with the irregularity of the spectrum. For both ordinary steel and high strength steel values of the order $M \sim 0.75$ were found.

Somewhat similar results were found by Tomita *et al.* (1992) for ship structural members. For variable amplitude loading it was found that the fatigue life clearly depended on the load (stress) history. Fatigue crack growth rate, crack opening point and threshold stress intensity factor all depend on the load history in the sense that they are dominated by the time when the larger loads appear in the time history and their pattern of appearance.

### Implications of the HSE 1995 Guidance

In Stacey and Sharp (1995b) the influence of the new guideline on the design of fixed platforms has been investigated. In particular the importance of the three main elements of the analysis; i.e. the S-N curve, the thickness exponent and the selection of the SCF was sought highlighted.

The S-N curves in the HSE 1990 guide and the 1995 edition have been summarizes and compared in Table 5. As can be seen the major changes of the HSE guideline are:

- a revised air S-N curve for tubular joints with a base thickness of 16 mm
- an increased penalty factor for tubular joints in sea water with cathodic protection
- an increased penalty factor for tubular joints in sea water under free corrosion
- a revised thickness effect

A deep water, a shallow water, and a lift deep water platform subjected to North Sea environmental loadings were considered. The assessments concentrated only on fatigue lives using dynamic spectral fatigue analyses for the deep water platforms, and a deterministic fatigue analysis for the shallow water platform.

Table 5 Comparison of S-N curves in the HSE 1990 and 1995 edition

| Property | 1990 edition | 1995 edition |
|---|---|---|
| Basic thickness | 32 mm | 16 mm |
| Thickness correction exponent | 0.25 | 0.3 |
| Penalty on life for free corrosion (relative to air curve) | 2 | 3 |
| S-N curve for sea water (cathodic protection) | Same S-N curve as for air | Penalty factor: • 2 for short lives (< 1.75 x $10^6$ cycles) • 1 for long lives |
| Transition point on S-N curve for sea water with CP | N = $10^7$ | N = 1.74 x $10^6$ |

The study showed that the choice of SCF equation has a larger effect on the fatigue lives than any other parameter. Even when the recommended equations in the revised guidance are used the choice of parametric SCF equation has a large effect on the predicted fatigue lives. An examination of the ten most critical joints for each platform indicated that the choice of SCF equation could cause the fatigue life to vary by a factor of up to 7 for joints in the deep water platforms, and up to 4 for the shallow water platform.

Disregarding SCF effects the revised fatigue guidance is more onerous than the 1990 guidance. The penalty on fatigue life was approximately 1.25 for the majority of the cathodically protected joints, though it could be as high as 1.5. The penalty for freely corroding joints was approximately 3. The modification of the thickness correction exponent from 0.25 to 0.3 was found to have a relatively small influence.

### *4.2.3 Fracture Mechanics Analysis*

A interesting fracture mechanics analysis was performed by Berge *et al.* (1994) who analyzed two joint configurations (a double T-joint and a single T-joint), one of which was also investigated experimentally (the double T-joint). It was found that the fatigue life of tubular joints not solely depends on the hot spot stress, but also is influenced by the through-thickness distribution of stresses. Further, it was claimed that the hot spot stress approach in some cases (low degree of bending) may lead to unconservative predictions of fatigue lives.

Two types of analysis were used: 1) a 3-D shell analysis with Y compliance calibration based on line spring model computations, and 2) a simplified 2-D plate analysis with (and without) a load shedding model. The through-thickness stress distribution was described by a degree of bending parameter, DoB defined as the ratio between the bending stress and the total stress (maximum principal stress).

SN-curves for the two tubular joints were calculated and demonstrated to be in close accordance with the experimental data, which was taken as a proof of the model to adequately represent the fatigue crack behavior. The calculated effect on fatigue life as a function of DoB has been shown in Figure 4. The figure shows both the result of the extensive FE calculations, and the results from the simplified plate solutions with and without load shedding implemented.

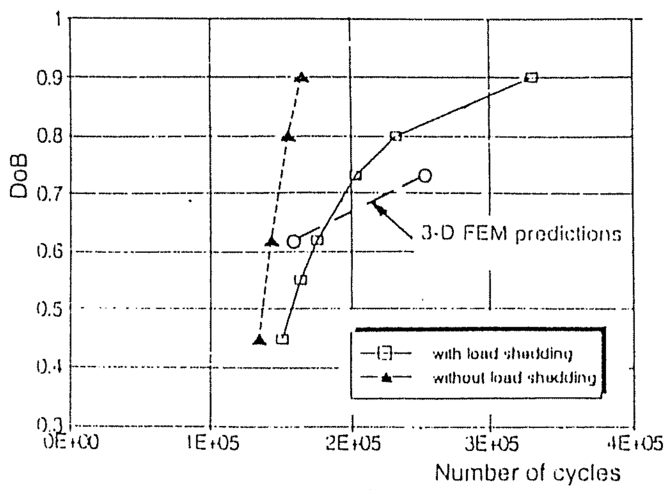



Figure 4 Effect of DoB on fatigue life. 3-D shell model and plate load shedding model shown (from Berge *et al.*, 1994)

It was concluded that the 3-D shell finite element fracture mechanics model is able to predict DoB effects quite accurately, and that simplified 2-D models give results in good agreement with the 3-D analysis. It was further suggested that with the development of parametric equations for DoB similar to the SCF formulae, the DoB effects may be implemented in design codes relatively easily.

### 4.3 Local Joint Flexibility

It has long been recognized that the inclusion of local joint flexibility in a global frame analysis produces results that may be quite different from those from a conventional rigid-joint analysis, see Elnashai and Gho (1992). The local joint flexibility (LJF) of a tubular joint is caused by the chord wall deformation, and it generally tends to relax member-end forces and moments. By considering LJFs in the structural model the structural demands on both the joints and the member-ends can therefore often be reduced as compared to those calculated by the conventional rigid-joint frame analysis.

Especially in re-analysis of existing structures it may be attractive to introduce LJFs as this may lead to less strict inspection schemes, and potentially avoids costly underwater strengthening work.

A thorough review of the analytical and experimental work to determine LJFs can be found in Buitrago *et al.* (1993). It contains a general methodology to calculate LJFs for planar tubular joints based on results from FE analysis, and it gives a new set of parametric formulae for single-brace, cross-brace and K-joints. The methodology has been validated by comparing the analytical LJF results to experimental data.

A discussion of the practical implementation of LJF in conventional frame analysis of offshore structures is also presented. It is argued that LJFs can be modeled by a set of three springs, one axial and another two in the in-plane and out-of-plane rotational directions, at each brace-chord intersection. In this model, the explicit springs are connected to the brace at one end and to the chord centerline at the other end, via a rigid link. The spring stiffness constants are equal to the inverse of the LJF values calculated from the given equations or customized FE analysis. In the case of cross or K joints, the brace cross effects are implicitly accounted for by superimposing LJF influence factors.

A detailed procedure to determine LJF for a general two-brace multiplanar joint is given in Hu *et al.* (1993). The author determines the local flexibility matrix for the joint based on total shell deformations reduced by the beam contribution, and further develops an equivalent element to incorporate the LJF into the analysis. Based on terms from the LJF matrix the stiffness matrix of the equivalent element is derived for inclusion as a general element in a standard space frame analysis program.

In Romeyn *et al.* (1992) the finite element modeling aspects of mulitplanar XX-joints are investigated. This reference shows that the correct modeling of LJF is no trivial job, since the flexibility determined clearly depends on the type of element used, integration scheme, mesh refinement, and whether the weld shape was included or not. The main findings were that the use of 8-node shell elements exhibits a more flexible behavior than the 4-node shell element, and is considered more accurate. The joint flexibility increases with mesh density towards a stable value after a certain mesh refinement, but the flexibility determined depended on the integration scheme used, and whether the weld shape was included or not (stiffer with the weld shape included).

### 4.4 Buckling

Offshore platforms are the most recent large field application for shell structures. The design of such structures is still evolving and involves aspects of both structural (civil) engineering and naval architecture. Since axial loading tends to be dominant, it was natural that aerospace research would also be incorporated (Singer, 1976). Slender shell plating tends to be used (R/t~200-500), similarly to aerospace shells, but ring collapse is designed out using submarine design criteria.

Design requirements were initially based on test data mainly composed of aerospace tests in the elastic range (Miller, 1977). A number of test programs were then set up in order to investigate the buckling of stiffened cylinders under combined axial and pressure load (axial being dominant) in the elasto-plastic range. A major program was led by Conoco Oil and ABS involving experiments carried out at CBI Industries Inc. and at the University of Glasgow (Miller and Grove, 1983; Frieze and Sands, 1984). This experimental work has been used to validate analytical models for further reliability analysis (Das *et al.*, 1984; Das 1987), and has been successfully incorporated into some of the leading offshore design codes. Another major research program was carried out in the U.K., led by the Department of Energy and the Science and Engineering Research Council (now EPSRC) involving the Imperial College, the University College and the Universities of Glasgow, Liverpool and Surrey. The strength of undamaged stiffened shells under complex loading was a leading issue, but also new aspects of shell research were treated such as the residual strength of damaged shells (Walker *et al.*, 1987; Cho and Frieze, 1986; Ronalds and Dowling, 1985; Onofriou and Harding, 1985; Kwok *et al.*, 1987). Analytical and numerical models were validated in view of these experiments.

Having gained more confidence on the strength models, research was initiated on reliability based buckling design. Regression formulae derived from a FE code were used to estimate buckling loads in reliability analysis. Further work was carried out to improve the statistical modeling of the initial imperfections. New research is being carried out on greatly neglected aspects such as ring tripping (Louca and Harding, 1994), and the numerical modeling of material behavior and residual stresses.

Morandi *et al.* (1995) gives guidance on the design of ring frames in orthogonally stiffened cylinders affected by axial loading and hydrostatic pressure. FE analysis of a tension leg platform column, using the eigenvalue and Riks limit point methods in the FE program ABAQUS, give results for bay instability and general collapse. It is shown by a second design that the first model analyzed could be redesigned with smaller ring frames, producing a considerable saving in cost and weight.

The study by Morandi *et al.* (1994) examines the methods of structural design for externally pressurized ring and flange stiffened vessels using the FE analysis program ABAQUS with linear analysis of models of infinite axial length. The results of tests with different geometries and boundary conditions, taking into account modal interaction effects, are discussed with reference to the partial safety factors applied to the reliability analysis and the design of stiffened pressure vessels.

In Louca and Harding (1994) the FE program LUSAS is used to carry out several FE analyses of ring stiffened cylinders for cylinders typically found in offshore steel structures under external pressure. The semi-loof shell element is used and shape imperfections are included in the finite element mesh. Localized torsional ring tripping failures are simulated using a single half bay and symmetrical boundary conditions that simulate a compartment of infinite axial length. A review of current design formulae is included, and comparison is made between these formulae and results obtained by the FE analysis. Emphasis is placed on the reduction of buckling resistance under external pressure caused by "out of circularity" of the cylindrical shell and shape imperfections in both flat bar and tee type ring stiffeners.

Physical model tests of ring stiffened compartments are outlined and compared with FE tests of similar models in Pegg and Bosman (1996). There is an emphasis in this paper on the negative effects from "out of circularity" on the resistance to buckling of submarine structures with secondary structural deck components. The FE program ADINA was used with shape imperfections introduced in the FE mesh, while in the physical model the shape imperfection is intentionally induced. There is a good correlation between physical test results and FE results, and the importance of considering the effects of shape imperfections in this type of design is highlighted.

The magnitudes and distribution of shape imperfections on stringer stiffened and un-stiffened composite ply cylinders are measured and outlined in Chryssanthopoulos and Poggi (1995). These shape imperfections are then introduced in FE meshes. An elastic buckling analysis is carried out on the composite cylinder, and an elastic-plastic buckling analysis is carried out on the stringer stiffened cylinder, both under axial compression. It is shown that probabilistic modeling of shape imperfections allows more accurate partial safety factors to be predicted for use in design of axially loaded cylindrical structures than deterministic methods.

## *4.5 Grouted Connections*

### *4.5.1 General*

In the beginning of the 1980's significant amounts of test data became available for grouted pile sleeve connections, and at the same time both the Department of Energy (DOE) and the American Petroleum Institute (API) issued their revised design guides for grouted pile sleeve connections. They both represented a large step forward for the industry since transfer lengths now could be reduced to a fraction of the lengths then normally used, see Billington and Tebbett (1980).

Both approaches were to be used with an overall safety factor within the frames of a working stress (WS) concept, while today only the DOE approach still uses a global safety factor instead of a load and resistance factor format.

Over the years many more tests have been performed and made available resulting in several different formulae being proposed. It has thus to some extent been possible with the result to refine the approaches of the design formulations currently used in the industry such as those issued by DOE, Det Norske Veritas and API (LRFD and WS).

In Sele and Skjolde (1993) a database of 750 test results has been assessed in order to provide improved and updated design guidance. This source contains valuable information in relation to failure modes, safety levels and practical design implications, and largely forms the basis for what follows.

### 4.5.2 *Failure Modes*

It is generally accepted that there are two failure modes: 1) sliding at the grout pile interface, and 2) shearing through the grout matrix over the top of the shear keys.

In a plain grouted connection without shear keys the failure mode is slipping along the interface between the pile and the grout. It has been found that tensile hoop stresses are set up in the sleeve, and compressive hoop stresses in the pile. This is due to the wedging action caused by the uneven surface of the pile. If the uneven surface is made into an even surface the connection will experience a dramatic reduction in strength. The strength of the connection is therefore dependent on the surface unevenness and the hoop stiffnesses of the pile and sleeve (which can be expressed by the sum of the thickness to diameter ratios of the tubulars). Together they determine the radial stress set up by the wedging action, and the friction force mobilized.

Shear keys (hoop or helical) have been found to increase the strength of grouted connections. Their effect is to increase the surface unevenness and thus the wedging forces. By increasing the number and height of shear keys the strength of the joint can be raised to an upper limit which is reached when the failure pattern shears through the grout over the top of the shear keys of the pile. This is the limiting capacity determined by the grout matrix.

### 4.5.3 *Frequently Used Design Equations*

In Sele and Skjolde (1993) the formulae from DOE (Offshore Installations, 1990), Det Norske Veritas (DNV, 1990) and API LRFD (API RP2A-LRFD, 1989) have been compared for compliance with screened databases. From the 750 test results three increasingly strict levels of screened databases were developed, the strictest screening left 187 tests for statistical treatment. All screened databases showed the "best" mean values and the smallest COV for the DNV, API and DOE in the order given.

Further, a calibration study to derive safety factors to be used with these formulae requiring the same target reliability as for the development of the API LRFD code was performed. The DNV equation was adopted as the failure mode function combined with the same wave force model as used in the API study.

The calibration study consisted in determining required values of a material factor for the DNV rules, a safety factor for the DOE rules, and a resistance factor for the API code.

To arrive at this result 3360 design cases were investigated, and in each case a reliability index was determined. To combine these reliability indexes a penalty function which gave a heavier weighting to the results with lower reliability indexes was used. One extreme approach in determining the final reliability index is to account of the minimum value found only, giving all other values zero weighting, the other extreme is to assign equal weighting to all results.

It turned out that the values obtained by the minimum value approach gave results very close to what was adopted by the DNV rules (a material factor of 3) and the API code (a resistance factor of 0.9). Conversely, the necessary global factor of safety to be used with the DOE formula was determined to 2.7 as opposed to the presently required value of 4.5. It was thus concluded that the DOE code provides a safety margin more than 50 % in excess of what is required to fulfill the same level of reliability as is incorporated in the DNV an API codes.

### 4.5.4 *Other Considerations*

**Fatigue**

The DOE and the API design formulations do not address fatigue specifically, but it has long been industry standard to consider fatigue aspects implicitly covered when the ultimate strength requirements are satisfied. This approach is supported in the standard being developed by the ISO TC67/SC7/WG3 Connections Technical Core Group (scheduled for completion in 1996/97). It states that grouted connections complying with the static strength requirements, and primarily loaded by cyclic wave action, do not require a detailed fatigue assessment. In contrast hereto the DNV approach addresses the fatigue issue specifically.

The available data on fatigue strength are quite limited. An early indication was given in Billington and Tebbett (1980) by a tentative S-N curve with a limiting stress at approximately 40 % of the static ultimate strength below which fatigue damage does not occur. It was concluded that with the current level of safety factors on ultimate load the working stress will be below the limiting range. This implies that a fatigue check is unnecessary in the design.

Newer data can be found in Ingebrigtsen *et al.* (1990) who found that:

- Cycling in pure compression is generally less deteriorating than cycling in compression-tension.
- Plain pipe connections generally seems to have a higher fatigue capacity than connections with shear keys.
- For plain pipe connections cycled in compression or compression-tension loading where the maximum tensile stress is less than 20 % of static load capacity, a fatigue check is unnecessary, provided that the design is carried out according to the formulae for static design.
- For shear key connections specific formulae for cycling in pure compression and compression-tension respectively could be determined. They yielded lower capacities than the static strength, and no cut-off level was found.
- On the other hand, as an example it was also found that with the partial coefficients used in the DNV Rules, the maximum fatigue load will never exceed 64 % of the static capacity. This seem to indicate that in many cases the capacity of the connection will still be governed by static strength considerations.

**Combined Loading**

The referenced formulae determine the axial capacity of the grouted pile sleeve connection, and take this item as the primary one to be checked, irrespective of other coexisting force components (e.g. shear and moment). In Billington and Tebbett (1980) it was found that small moments seem to reduce the axial capacity slightly, while larger moments result in an increase in axial capacity. It was also argued that the relatively large safety factor (4.5) would cover a possible detrimental effect of small coexisting moments. Somewhat similar points of view are presented in the standard being developed by the ISO TC67/SC7/WG3 Connections Technical Core Group that claims, that the axial load capacity is not reduced by coexisting bending and shear. However, this code requires torsion to be combined with the axial load so that the stress resultant of the axial and torsion components of interface stress shall be compared to the allowable interface transfer stress. In addition, the torsional component of interface stress shall be less than the capacity of the connection when considered plain (disregarding shear keys).

**Movements During Grout Setting**

In the ISO TC 67/SC 7/WG 3 Connections Technical Core Group, a new reduction factor on strength has been introduced that considers the effect of movements during (early) grout setting. If the relative pile to sleeve movement during the maximum expected seastate in the 24 hour period after grouting exceeds 0.035 % $D_p$ a reduction in the interface transfer strength shall be made. However, the detailed understanding of this detrimental effect is still subject to discussion.

## 5 SHIP STRUCTURES

### *5.1 Stress Concentration*

Inherent in ship structures are in general many discontinuities due to design and construction requirements. This applies to longitudinals as well as transverse members of all kinds of ships. It needs no further explanation that in view of the mostly high longitudinal stresses in the ship hull these discontinuities have to be analyzed and evaluated as far as stress deflection and stress concentration are concerned. These discontinuities are, e.g. box girder connections, big lightening holes in the side structure and/or superstructures, hatches and other openings.

The possibilities to calculate the global stress had improved considerably by the introduction of the finite element method (FEM). Thus it is now possible to analyze the complex structural geometry and the interaction between the individual members. In general structural details can only be roughly simulated in a FE global model because of the size and the complexity of the ship structure. However, it is also possible to calculate the stresses quite accurately at the notch of a local model. Stress concentration factors (SCFs) of structural details can be calculated by systematic variations of load and geometry. In view of the many different notches it is possible to simplify the practical evaluation.

In view of the evaluation of the load application it is recommended to distinguish between two stress types:
(1) Nominal stresses which, as far as the applied load is concerned, are often divided into global stresses in primary members and into local stresses in secondary members as well as
(2) locally increased structural stresses and notch stresses.
Conventionally, the stress analysis is based on the beam theory which is still of great importance despite the introduction of FEM.

There are various methods for the evaluation of the locally increased structural stresses or notch stresses. The classical method is the use of SCFs which, e.g. are determined by measurements or systematic calculations.

Here, a clear definition of the nominal stress is important.

An alternative to the use of SCFs is the generation of a local finite element model under consideration of increased mesh refinement of the detail. In special cases a zoom-up procedure can be used for one or more potential crack locations of the detail. Normally the nodal displacements are taken from the global model and are applied as prescribed displacements at the boundaries of the local model. Correct results can be expected when the rigidity of the local finite-element model is in accordance with the same part of the global model. This has to be checked by means of the stresses in the transition region. Even better is the direct mesh refinement of the global model at the interesting notch areas. However, this may require a lot of work if the mesh structure has to be changed manually. Here, an interesting alternative is the adaptive mesh technique.

The results are used for the fatigue assessment, also for optimization of newly developed ships. It is expected that this will contribute to avoid early and annoying cracks in the ship structures.

The fatigue strength of novel welded connections, which cannot be classified by the nominal concept, can be calculated by means of three local approaches. In these, the fatigue strength can be found by calculation taking actual loads and geometry of the welded connection into account. The approaches are
(1) the structural stress (hot spot) concept
(2) the notch stress (local) concept
(3) the fracture mechanics based (crack propagation) concept.
These concepts were described by Niemi (1995). The weld itself was originally not shown in the hot spot concept. Today the stress flow close to the actual welded connection is still being linearly extrapolated to the toe of the weld. As in nominal stresses the effects of weld shape, plate thickness and fabrication quality are not included. In this sense hot spot stresses can be considered as modified nominal stresses, which can be used for a fatigue assessment based on a comparable notch case from codes. The structural stress concept is problematic in two respects. The parameters of structural geometry and geometry of the weld cannot often be separated. A concept, which is free from notch case catalogues, therefore, cannot be offered. Secondly, a linear stress distribution does often not exist, especially in case of finer meshes. The hot spot stress depends on the element type and the mesh size.

The parameters of weld geometry are taken into account in the notch stress concept. This is again causing problems because these parameters show a large scatter even along one weld. The parameters of the weld geometry are not known in the design stage and generally are not prescribed for the production presently. This situation has to be taken into account by a safe estimate of the variables in the calculation model. Furthermore, homogeneity and isotropy of the material cannot be expected due to the dimensions of notch radii, grain sizes and depth of plastic zones.

Various approaches have been followed for the estimate of a fatigue effective notch factor $K_f$ from the linear-elastic stress concentration factor $K_t$. Empirical formulae have been presented to calculate $K_f$ from $K_t$ and notch root radius. The experimental assessment of the $K_f$-formulae was, however, not satisfactory. A modified approach has been used with fixed parameters of the weld geometry. For the notch radius, a mean value of about 1 mm was found in several investigations (Niemi, 1995).

The state of art with regard to analysis and evaluation of discontinuities in the ship structure are shown in the following.

A comparative study of a two-phase finite element structural analysis has been undertaken by Technical Committee II.1 of ISSC'94 (Sumi *et al.*, 1995). A side structure of a middle size crude oil carrier was taken as a typical example of orthogonally stiffened panel structures. The first phase of the analysis was the global deformation and stress analysis, while the second phase was the local stress analysis at the intersection of a transverse frame and a longitudinal stiffener. In the global structural analysis, the importance of the structural idealization in the vicinity of the boundary is recognized. In order to calculate the local stresses at the intersection of a transverse frame and a longitudinal stiffener, conventional zooming procedures are used in the majority of the present studies. It should be pointed out that not all of the deformation modes or traction applied on the boundaries of the zoomed-up model could be defined in the previous global analyses, and that sometimes additional new members appeared in the analysis. This means that a variety of hypotheses were introduced in the zoomed-up analyses which lead to different solutions. One way of resolving this problem would be to enlarge the zoomed-up region to extend to a couple of bays both in ship length and in ship depth/breath directions, so that the effects of the specific application of boundary conditions are reduced at the point of stress evaluation. One may also use the super-element (or substructuring) technique, where local models with fine-mesh subdivision are contained in a global model as super-elements. This solution procedure may be effective when a structural designer knows in advance the exact parts of the structure where the local stress analyses should be carried out. Results from this study are believed to be what is expected in current finite element analysis and point to a need for a more well defined unified approach for finite element analysis of ship structures. This is particularly important when the consequences are considered in fatigue analysis where small differences in stress values can lead to large changes in fatigue life estimates.

The local stresses on side longitudinal frames of a bulk carrier in ocean waves are estimated by Kuramoto *et al.* (1993) using the simulation method so called DISAM (DIScrete Analysis Method), which has been developed originally for VLCC. Local stress (5 mm away from weld bead) at the intersection of a side longitudinal and a transverse frame is calculated by using solid elements so that the plate thickness and the weld beads can be modeled in the finite element analyses. It is concluded that the correlation of the stresses induced by several load components is as follows:

(1) Internal pressure and external pressure have negative correlation. The correlation factor $\rho$ is -0.6.

(2) Vertical bending stress and horizontal bending stress of a hull girder have little correlation ($\rho = 0.1$).

(3) Pressure and hull girder bending have positive correlation ($\rho = 0.6$).

A simple calculation procedure of local stress from load assessment is needed especially for fatigue design of ship hull. Although DISAM is one of the solutions to deal with this problem, it takes too much computation time. A simplified fatigue design method for side longitudinals of VLCC is proposed by Watanabe *et al.* (1995). It is based on beam theory incorporating the results of detailed analysis with DISAM. The proposed method is simple and easy to use, and it can cover main factors which contribute the structural response of side longitudinals such as the superposition of multiple load components with phase differences and wave pressure nonlinearity. The basic ideas of the method are listed below.

(1) Fatigue strength is assessed by cumulative damage factor using Palmgren-Miner's rule. Shape of long term distribution of stress ranges is taken into consideration.
(2) Nominal stress due to each load component is calculated by the beam theory.
(3) Stress due to wave pressure, relative deflection of primary supporting member and hull girder bending are, respectively, taken into account.
(4) Wave pressure nonlinearity is considered.
(5) Stress components are obtained by the combination of nominal stress and SCF, where SCF is estimated by zooming analyses using solid finite elements.
(6) The total stress range is calculated as the sum of the stress components with correlation factors which represent the phase difference of each load.

The results are compared with the detailed analysis, and satisfactory agreement has been demonstrated.

A new slot structure to ensure both structural safety and construction easiness has been developed by Kamoi *et al.* (1994). A practical use of the new structure has been studied for the application to 280,000 DWT double hull VLCC. Both static strength analyses by FEM and fatigue strength tests were carried out, and the fatigue strength of the new slot has been evaluated. They conclude that the newly developed slot structure is superior to the conventional one both in static strength and the fatigue strength so that it can be applicable to actual ship construction.

Cheung (1995) demonstrates an actual case study of a tanker double-bottom structure to describe the various degrees of analytical complexity associated with ship structural repairs. 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 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. His paper addresses the field of repair analysis, and the potential benefits to both operators and new construction design.

The mesh refinement in the vicinity of the notch is done empirical-intuitively and involves additional work. The quality of such a mesh can be evaluated by an estimation of errors of the FEM-solution obtained. An adaptive mesh refinement is obtained by this estimation of errors in connection with the concept of mixed interpolated finite-element. Some applications of the automated mesh refinement for FEM-analyses are shown by Reißmann (1996) for typical ship structural elements with locally increased stresses. It becomes apparent that the original mesh only has to be roughly modeled. The even decrease of the total error becomes less after two up to a maximum of five steps of adaptation in such a way that this iterative FE-mesh correction procedure can be stopped. The adaptation region within the global FE-model can easily be located by the adaptive mesh refinement based on the mixed interpolated finite elements where the structure of the original mesh is maintained. This is one good possibility for the local analysis of SCF under the global calculations of the ship strength. In the meantime an application to three-dimensional stiffened panel structures is possible.

With regard to fatigue strength assessment of ship structural details, a hot-spot stress approach is discussed by Kawano *et al.* (1994). A hot-spot stress is commonly defined by the stress at weld toe, which is linearly extrapolated from the stress values at the two points 0.5 and 1.5 thicknesses away from the weld toe, where stresses are usually calculated by using shell elements. They discuss the effective range and limitation of the application of shell elements for the local stress analysis at weld toes. Hot spot stresses are also calculated by Nihei *et al.* (1993) for ship structural details using BEM. Defining the hot-spot stress as the stress at

0.3 thickness away from the weld toe, they found that the proposed hot-spot stress was in good agreement with that obtained by the extrapolation method.

In determining a hot spot stress by finite element analysis, problems will arise when the hot spot is located in the vicinity of single-sided edge gussets welded to a stressed member. In this case there is no self-evident indication for the location of the extrapolation points, such as plate thickness, which is normally taken as basis. Niemi (1994) shows two different methods for the definition of a relevant structural stress:
(1) Three extrapolation points at fixed distances at the plate edge are defined, and quadratic extrapolation to the weld toe is performed, giving a local structural stress.
(2) Two extrapolation points at the plate edge are defined in accordance with conventional hot spot stress determination by linear extrapolation.
In case of quadratic extrapolation the size of the detail is automatically recorded. For the other method the size effect has to be considered by multiplying the hot spot fatigue strength by a conventional size effect factor, considering the plate strip breath, as the apparent thickness.

SCFs for nearly all welded connections - irrespective of design - can be determined by realized numerical procedures (FEM, BEM) with the current computer programs (Anthes *et al.*, 1993; Iida and Uemura, 1994). The influence of individual weld geometry on the SCF can only be concluded from a number of individual investigations. At first the actual weld shape has to be geometrically idealized in such a way that a clear description with only a few weld parameters (notch radius and depth, weld overfill and flank angle) is possible. Data of SCF which is finally described by an appropriate approximation formula can be obtained by systematically varying the weld parameters. This procedure was applied to double symmetrical butt welds and double-T-joints. All SCFs determined relate to nominal stresses of parts of the plate without notch. The investigations showed that the SCF is not only strongly influenced by the relative plate thickness to notch radius ratio, (t/r), but also the flank angle. In connection with the notch stress concept, the fatigue behavior can be calculated by means of the SCFs.

### 5.2 *Buckling*

DREA (Defence Research Establishment Atlantic, Canada) conducted a joint structural testing project with SSC (U.S. Interagency Ship Structures Committee) (Chen *et al.*, 1996). Twelve single stiffened plates were tested at C-FER (Centre For Engineering Research Inc.). Seven specimens were subjected to the combined lateral and in-plane loading, three specimens had part of the stiffener removed to represent corrosion, and two specimens were placed under large lateral loading to create permanent damage followed by in-plane loading. Specimens were simply supported at the ends with symmetric boundary conditions at the sides. Five devices along each side of the specimen were designed to simulate the symmetric boundary conditions. The failure mechanisms included local plate buckling, column flexural buckling and tripping modes. The computer simulations of the test procedure were conducted with the commercial non-linear finite element package ADINA at DREA and ABAQUS at the University of Alberta. The numerical results showed that the non-linear finite element predicted the collapse strength and deformed shape well.

Paik and Pedersen (1995d) developed a theoretical approach to calculate the ultimate load-carrying capacity and the deep-collapse response of plate structures subjected to static/dynamic compressive loads. Two models, an elastic model and a rigid-plastic model were derived theoretically in closed form for analysis of the mean crushing strength. The effect of initial imperfections in the form of initial deflection and welding-induced residual stress were considered in the derivation.

Papadakis and Cho (1995) used an energy methodology to examine the strength of a circular, cylindrical sandwich shell subjected to hydrostatic pressure. They investigated the interactions of the inner and outer shells. The results showed that the local deformations of the core stiffeners significantly reduced the bending strength at the ends of the sandwich shell. Membrane forces substantially altered the axial loading of the inner and outer shells.

Alagusundaramoorthy *et al.* (1995) conducted an experimental study of twelve stiffened panels with four longitudinal stiffeners. Eight panels, four without cut-outs and four with cut-outs extending the full width between stiffeners, were designed for plate failure. Four other panels were designed for stiffener initiated failure. They presented an approximate method based on the strut approach. This method could be used to predict the ultimate strength of simply-supported stiffened panels with initial imperfections and square cut-outs, subjected to uniaxial compression. The proposed method compared well with the experimental results.

Ueda *et al.* (1995) derived buckling, ultimate and full plastic strength interaction relationships for rectangular flat plates and uni-axially stiffened plates subjected to in-plane biaxial and shearing forces. The plates were assumed to have no initial imperfections and residual stresses. Strength functions were expressed in terms of applied forces. The accuracy of these interaction relationships was confirmed through comparison with the results of other analytical methods.

Jang and Seo (1995a) proposed an efficient method to calculate the buckling, post-buckling behavior, and ultimate strength of welded one-sided stiffened plates simply supported along all edges. In this method, the ultimate strength can be obtained as the lowest value of the calculated collapse loads based on several assumed failure modes. This method can provide accurate result with little computational time.

Bedair and Sherbourne (1995) developed a semi-analytical approach for the computation of local buckling of stiffened plates under any combined bi-axial compression, in-plane bending, and shear stress. The plate is treated as partially restrained against rotation and in-plane translation.

Devine *et al.* (1994) summarised the recent experimental and numerical studies in Lehigh University and the Carderock Division Naval Surface Warfare Center, USA (CDNSWC). These studies evaluated instability and collapse behavior of advanced double hull ships with focus on large scale compressive strength testing of multicellular specimens. Test parameters included material type, number of cells, specimen length, plate slenderness, end conditions, lateral loading and innovations including curved plating and localized stiffening. Seven full-scale and two half scale specimens were tested at Lehigh University. Eight small-sized single-cell specimens, three small-sized triple-cell column collapse specimens, and five large-sized specimens were tested at CDNSWC. The tests demonstrated the predominance of local plate instability, the effects of plate imperfections and residual stresses, and quantified buckling and collapse loads. Analytical studies were conducted with the tangent modulus method and the non-linear finite element method. Lehigh and CDNSWC performed the finite element analysis with ADINA and ABAQUS, respectively. The results showed that the tangent modulus method applied to longer column specimens produced reasonable estimates of strength. The non-linear finite element method, on the other hand, could provide a reasonable prediction of buckling and collapse behaviour and could be used to model the full range of parameters affecting collapse strength.

Bai *et al.* (1993) derived a set of finite elements including beam-column, stiffened plate, and shear panel. They used an effective width concept to include the buckling and post-buckling behaviour in the formulation, therefore fewer nodal points were required in the modeling. The analysis, called the Plastic Node Method, included the geometrical and material nonlinearities as well as the initial imperfections. Typical ship collapse examples, such as stiffened plate, upperdeck structure, box and hull girders were analyzed and compared with experimental results and other numerical solutions.

Liang *et al.* (1993) performed a material and geometric non-linear analysis of swedge-stiffened pressure hulls subjected to hydrostatic pressure on three commonly used swedge forms. The load-displacement relations on the responses as well as the distribution of stresses with plastic zone spreading were discussed. Ross and Palmer (1993) conducted a theoretical and experimental investigation of the instability of swedge-stiffened circular cylinders under uniform external pressure. They found these cylinders had similar buckling modes as those of ring stiffened cylinders. The initial out-of-roundness played a significant role in the elastic knock down.

### 5.3 *Full Scale Monitoring*

During the last years there has been an increased attention to full scale monitoring as a way to control both the still water loading and the wave loading. A motivation for installing measurement devises is partly to the accidents related to bulk carriers and partly due to a control or verification of the stresses in fatigue estimation. For about twelve years, the development of an accident recorder arose within IMO from:
- a primary interest in ship safety,
- the need to explain apparently inexplicable losses, and
- the desire to consider the implementation of complete and rational systems for data collection and recording on board ships.

Different data recording prototypes were fitted to ships ("City of Plymouth", 1983, and "Stuttgart Express", 1985). Data were recorded onto bubble memories housed in a floatable and recoverable unit. IMO has arranged a feasibility study on voyage data recorders (VDR), which was successful and resulted in a recommendation for the fitting of stress monitoring to bulk carriers above 20,000 DWT. One can also refer a recent SSC report (Slaughter, Cheung, Sucharski, and Cowper, 1996) for the state of the art review of hull response monitoring systems.

A review of three major classification societies rules shows that there have been developed requirements for measurement systems providing guidance and control related to the installation and use of such equipment. Lloyds Register (LR, 1991) were the first to give such requirements describing two levels of the surveillance system. Level one being a pure onboard surveillance system for display on the bridge and a second level also storing the data for analysis purposes. In 1992, GL presented long term strain measurements on board "Stuttgart Express" to verify the IACS long term bending moment formula. These measurements were based on the aforementioned VDR concept. ABS (ABS, 1995a) has split the hull monitoring system further to divide between the motion and stress monitoring. In addition there are given provisions for monitoring of voyage data. In an extended version the latter could be viewed as a move towards introducing a "black box" voyage recorder for the use in accident investigations as well as normal monitoring, similar to that used in Aviation. DNV (DNV, 1996b) describes two levels of detail, both storing the data, that differentiate on the level of detail for stress monitoring.

Also a hull strength monitoring system has been developed by BMT Sea Tech (Thompson, 1995), where an innovative program which integrates and extends the hull strength monitoring technologies under Windows NT and fully exploits data generated to assist decision-making process during navigation and loading operations. Specifically, it aims to address shortfalls in existing systems by combining stress, motion and radar-based sea state monitoring with the use of on board artificial intelligence tools. The effects of thermal stress on hull stress monitoring are also investigated by comparing the predicted and measured results of a container ship and a bulk carrier (Shi, Thompson, and Hire, 1996).

Table 6 Hull monitoring systems in Classification Societies

| | Accelerations | Global bending stresses | Slam sensors | Separate local stresses | Environmental data | Voyage data (navigation and other equipment) | Recording for storage |
|---|---|---|---|---|---|---|---|
| LR (SEA) | x | x | x | | | | |
| LR (SEA R) | x | x | x | | | | x |
| DNV (HMON1) | x | x | x | | | | x |
| DNV (HMON2) | x | x | x | x | x | | x |
| ABS (HM1) | x | | x | | | | + |
| ABS (HM2) | x | x | x | x | | | + |
| ABS (HM3) | x | x | x | x | x | x | + |

+ means optional.

The applicability of structural monitoring is discussed by Fujimoto *et al.* (1995) for crack-type damage of metal structures. Two types of monitoring methods are examined; detection of existing fatigue crack and prediction of fatigue damage. As the detection of existing crack, the concept of health monitoring is employed. Five types of line sensors; i.e. conductive film-sensor, conductive-paint sensor, plastic optical-fiber sensor, glass optical-fiber sensor, and carbon-fiber sensor are made. Fatigue tests are carried out, and the appearance how the fatigue cracks break the sensors is closely examined. A method to attach sacrificial specimen on a structural member is examined for the prediction of fatigue damage. A sacrificial specimen is designed in such a way that magnified stress is transmitted to the specimen so that earlier crack initiation can be expected. Its geometry is a center notched plate with 60mm-length, 10mm-width and 0.25mm-thickness. The sacrificial specimen is bonded on a smooth specimen by epoxy resin, and fatigue tests are carried out.

Yuasa (1993) discusses the fatigue analyses of deck of a bulk carrier and the web frame of a car carrier using measured stress data, which are obtained through onboard measurements. Spectra of stress ranges are estimated from short term parameters of fluctuating stresses. Both accumulated fatigue damage and fatigue damage during rough seas are calculated. Following relations between stress level and fatigue damage, the relation between wind conditions, sea conditions and fatigue damage are also reviewed. The results show that rough seas fairly affect the accumulated fatigue damage and that fatigue damage during specific rough seas are considerable. It is also found that relatively low level of stress ranges mainly contribute to the fatigue damage of the deck of a bulk carrier, while widely spread stress ranges contribute to the damage in the web frame of a car carrier. As to wind and sea conditions, it is shown that fatigue damage of the deck of a bulk carrier increases significantly as BS exceeds 7 under swell scale 7 and upward.

The characteristics of wave induced pressure fluctuation and local stresses on the side of VLCCs have been investigated by Tozawa *et al.* (1995), because considerable number of cracks were found on side longitudinals of second generation VLCCs. The paper describes summarized results of full scale measurement on a typical second generation VLCC which was carried out aiming at understanding the long-term characteristics of working stresses on longitudinal members and wave pressure acting near the load water line. The actual condition of wave pressure fluctuation and working stresses on the side of the VLCC was revealed through the full scale measurement, and it is confirmed that the prevalent estimation based on the strip method well explained the short-term characteristics of measured wave pressure. The validity of the precise discrete analysis method, DISAM, is also confirmed that the method is effective for the evaluation of local stresses of structural members where various load elements act simultaneously with phase differences.

### *5.4 Cruise Ships: Strength and Stiffness Problems*

#### *5.4.1 Present Trends*

The most popular cruise ship size ranges between 50,000 and 70,000 GRT, with lengths between 180 and 200m and passenger carrying capacity of around 1,500 to 2,000. In the recent years the tendency towards increased capacity has resulted in orders of vessels exceeding 100,000 GRT, 240~250m and number of passengers over 3,000. The structural requirements are similar for all ship sizes but certain aspects concerning longitudinal strength are becoming more difficult to satisfy as ship length increases. At the same time the architects' requirements are becoming more and more original and innovative (Hansa, 1996 a, b; International Cruise & Ferry Review, 1996). Internal open spaces such as big atriums, restaurants and theaters with reduced number of pillars and larger openings on the structural bulkheads so as to give more and more light to the internal public areas are representing a challenge of increased difficulty in the necessity of fulfilling both the static and dynamic structural requirements.

In recent designs, for safety reasons, the lifeboats are stowed between decks 7 and 9 to reduce the distance from the waterline. (Gudmunsen, 1995) This functional concept requires a large recess around mid-depth decreasing the structural continuity and reducing the efficiency of the upper decks in the hull girder bending (Figure 5).

For cruise vessels of medium size mild steel is usually sufficient to meet the longitudinal strength requirements and higher tensile steel is used just to reduce the weight in the upper part and to solve areas of high stress concentration without excessive thickness increase. As the ship dimensions increase, for stability reasons, higher tensile steel is becoming more and more necessary for the longitudinal elements in the upper part and grade D or E steel is used around openings. Light alloy has been used in certain upper areas of some recent cruise ships but its application is normally not considered extremely advantageous as the consequent reduction of weight is reduced by the necessity of fire insulation and particularly for the increase of material and fabrication costs (see ISSC Report of Committee III.3 (ISSC, 1991)).

Theoretical and experimental studies are progressing on the strength of steel corrugated core sandwich panels, on their efficient production through laser welding and on their applicability onboard ships. Structural weight reduction of about 40% is reported (Kujala, 1995). Up to now laser welded sandwich panels have been applied as non-structural and non-critical components in cruise ships (Meyer Werft, 1996). Although some important theoretical and practical problems are still to be solved, the application of sandwich panels seems promising.

### 5.4.2 *Transverse and Local Strength*

Normal deck loadings on cruise ships are generally low compared with cargo ships tweendeck loadings. The racking behavior of cruise ships is generally guaranteed by the fire subdivision bulkheads and by the additional transversal strength given by the main stairways/lift trunks and engine casing. F.E. models of large 3-D central sections of the hull have been prepared to verify the transversal deformability and stress concentrations in critical areas, considering actual deck loadings and rolling conditions according to Rules (Figure 6) (Fincantieri, 1990).

As mentioned before, in certain public areas as theaters, cinemas, lounges and restaurants, some lines of pillars have to be omitted in order to obtain unobstructed views. To support these areas, either longitudinal or transversal bulkheads are generally arranged in the upper tweendecks to transfer the loads to the remaining pillars. These solutions have to be carefully checked with F.E. analyses especially in the after part of the vessels where the main sources of vibration-excitation are present.

In the latest cruise vessels designs the upper decks are extended with its maximum breadth as after as possible being these areas attractive both for cabins and for public spaces. As a consequence an increasing weight of the after part has to be supported by the after skeg when the ship is docked. With the increasing of vessels dimensions and tonnage, difficulties can arise for the available docking facilities due to the limited loading capabilities of dock floors. F.E. calculations have been performed for the bigger size of cruise ships using the 3-D F.E. model of the hull girder and proper springs to take into account the blocks stiffness. From the results of these analyses it appears that the lightship weight is fully balanced at the after end by the reaction forces of the docking blocks only after a length of 3-4 web frame spacings. In this area of the after skeg, the ship has to be supported by closely spaced blocks with a distributed force reaching a peak around 600 t/m.

### 5.4.3 *Hull Girder Global Strength*

Due to a rather uniform distribution of lightship weight and to the concentration of buoyancy toward the midship portion, the cruise vessels are usually experiencing very high still water hogging bending moments. Even with almost unrealistic concentrations of deadweight in the central part, the cruise ships are still in hogging condition. Combination of the rule hogging wave moment and the maximum still water hogging is giving the maximum longitudinal stresses, while the combination of the rule sagging wave moment and the minimum still water hogging can result in buckling problems on the upper decks where the scantlings have to

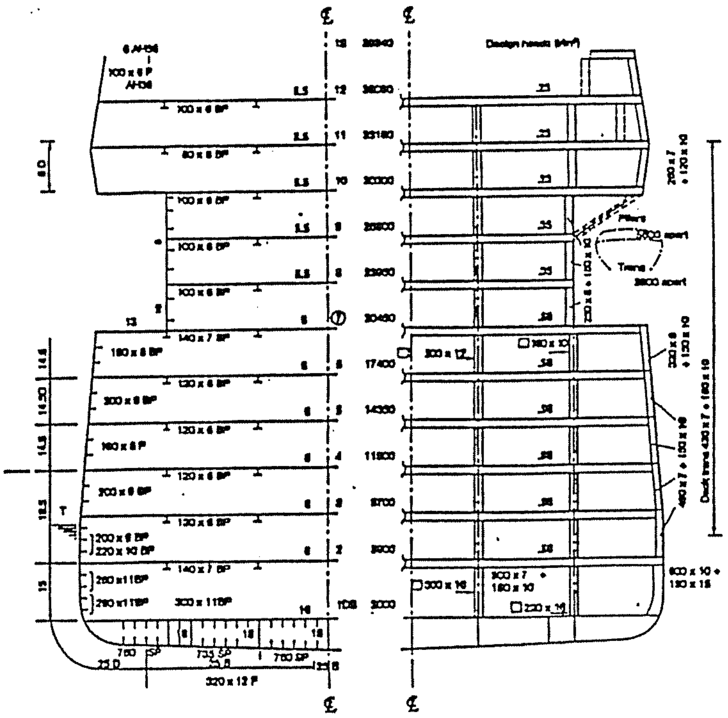


Figure 5 Midship section of a design featuring low level recess

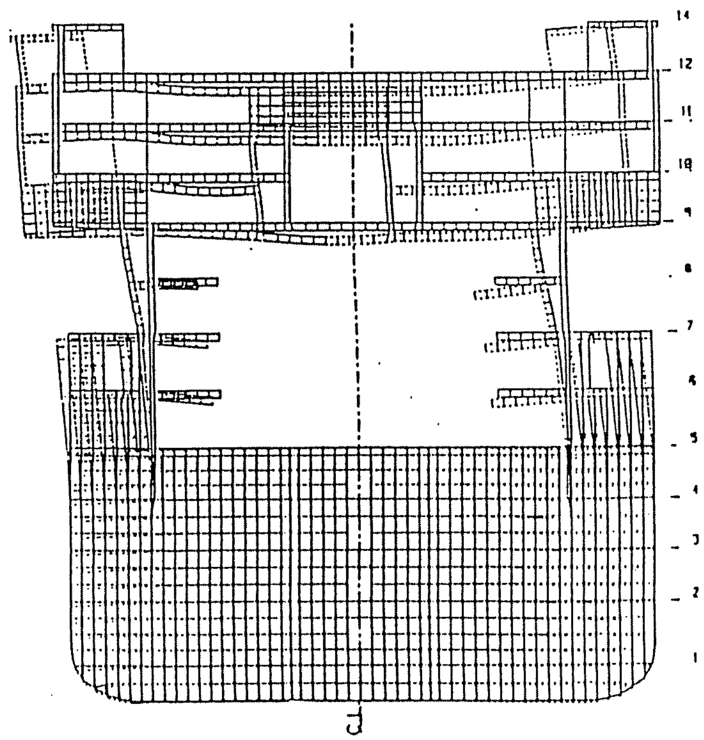


Figure 6 Transverse deformability in the central part of a 3-D racking model

be minimized (Gudmunsen, 1995). According to the recent IACS unified requirements the wave sagging bending moment and the forward knuckle of the wave bending moment distribution are becoming aspects which are critical to be fulfilled especially with the increase of ship size and length (Figure 7). The still water shear forces usually follow a theoretical distribution with peak values at about 0.25 L and 0.75 L from A.P. If positive or negative rule wave shear forces are added, considerable total shear forces have to be supported generally in areas where, on the side shell, openings are needed for embarkation platforms and provisions and baggage handling operations.

To analyze the hull girder of a cruise vessel subject to a bending moment, old theoretical methods are of no practical use. For passenger ships the modeling of a large number of cut-outs in the side plating is a problem when a global finite element analysis is planned at the preliminary stage. Heder *et al.* (1991) attempt to simplify the finite element analysis using 2-D models. The analysis is simplified by using orthotropic panels to represent the side plating and the cut-outs. The panel has the same stiffness as the side plating with cut-outs. The accuracy of the method has been demonstrated by comparison with a comprehensive 3-D FE-model. For practical application, such method seems still to remain in the theoretical field as, for instance, the large recess for the life boats in a passenger ship cannot successfully be modeled. As a matter of fact, the introduction of the lifeboats recess separating a lower hull from an upper superstructure and the presence of internal longitudinal elements contributing to the hull bending, have complicated the behavior of the hull girder. From recent applications the analysis of a complete 3-D F.E. model of the hull seems to be the only practical and effective method. Idealizing each primary structural element and the main openings, these models are reaching 50,000-60,000 degrees of freedom for half ship symmetric in C.L. (Figure 8) (Gudmunsen, 1995; Germanischer Lloyd, 1995; Fincantieri, 1990, 1992). The aims of a 3-D F.E. analysis of the hull girder are the evaluation of:

1) global deformations,
2) effectiveness of upper decks, distribution of longitudinal stresses at each level, buckling of upper decks,
3) transfer of forces between lower hull and upper superstructure, shear stresses in way of the intermediate recess, shear lag at the relevant decks level,
4) stress concentrations around significant openings in way of side shell, longitudinal bulkheads and decks for fatigue considerations,
5) local deformations of important openings to be closed with glass (glass walls, windows, doors), and
6) compression increase or presence of tension forces in pillar lines.

Two loading conditions are usually taken into account considering the combination of maximum still water and wave hogging and of minimum still water hogging and sagging wave. Careful representation of lightship weight, deadweight and buoyancy forces is needed to obtain the desired bending moments. In recent designs the overall hull girder vertical deformation (generally within L/1000) is usually confirming a very good efficiency of the upper part.(Gudmunsen, 1995; Fincantieri, 1992).

The pillaring system is maintaining the transverse sections shape. In the maximum hogging condition the pillars compression loads generally increase in the midship region and relieve at the ends with some pillars even in tension. Comparing the longitudinal stresses at each level as derived from F.E. analysis with the theoretical predictions, reasonable agreement can be found with the conventional beam theory up to embarkation deck where a reduction of stresses may appear (Figure 9), (Gudmunsen, 1995; Fincantieri, 1990, 1992; also see ISSC Report of Committee II.1 (ISSC, 1991) ).

At the level of the shell recess the axial stresses have to be transferred to the recessed longitudinal bulkhead and again out to the shell two decks above, through a shear mechanism (Gudmunsen, 1995). The higher is the number and size of the openings in the recessed bulkhead the less is the efficiency of this shear transfer and the higher is the evidence of the drop of the longitudinal stresses in way of the recess. At the highest levels another drop of stresses could be found as the longitudinal elements in way of the side shell decrease their stiffness and efficiency. In conclusion, depending on the shear stiffness of the longitudinal recessed bulkheads and of the upper external shell, an effectiveness of 60-75% can usually be derived for the upper superstructure (Gudmunsen, 1995; Fincantieri, 1990, 1992).

In way of the end connection of the superstructure sides to the lower hull and in way of narrow deck strips connecting full breadth decks high strains are normally found. (Gudmunsen, 1995) Moreover the shell plating below embarkation deck is usually evidencing significant shear stresses in way of the major openings for shell doors and windows at 1/4 and 3/4 L. At the same time important stress/strains are common in way of windows and doors opened in the recessed longitudinal bulkheads and upper outer shell. Areas of stress concentrations are usually re-idealized with a more detailed mesh for further analysis.

From a detailed analysis of a 100,000 GRT vessel (Fincantieri, 1992), Figure 10 is reproducing fine mesh models of a longitudinal recessed bulkhead with several openings. In a), a midship area is shown with a typical bending deformation. In b) a forward area is clearly giving the evidence of a shear deformation with higher strains and stresses. Where continuous lines of doors and windows are opened in the upper outer shell

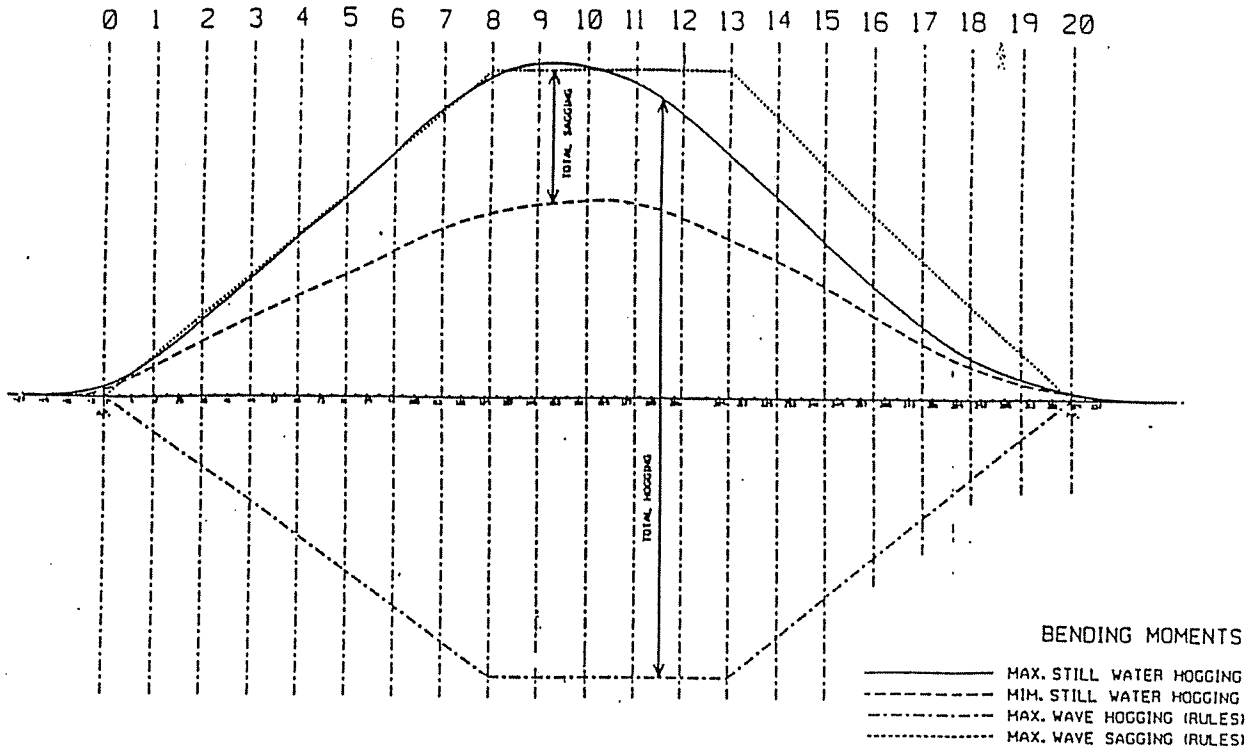



Figure 7 Global hull girder bending moment distributions

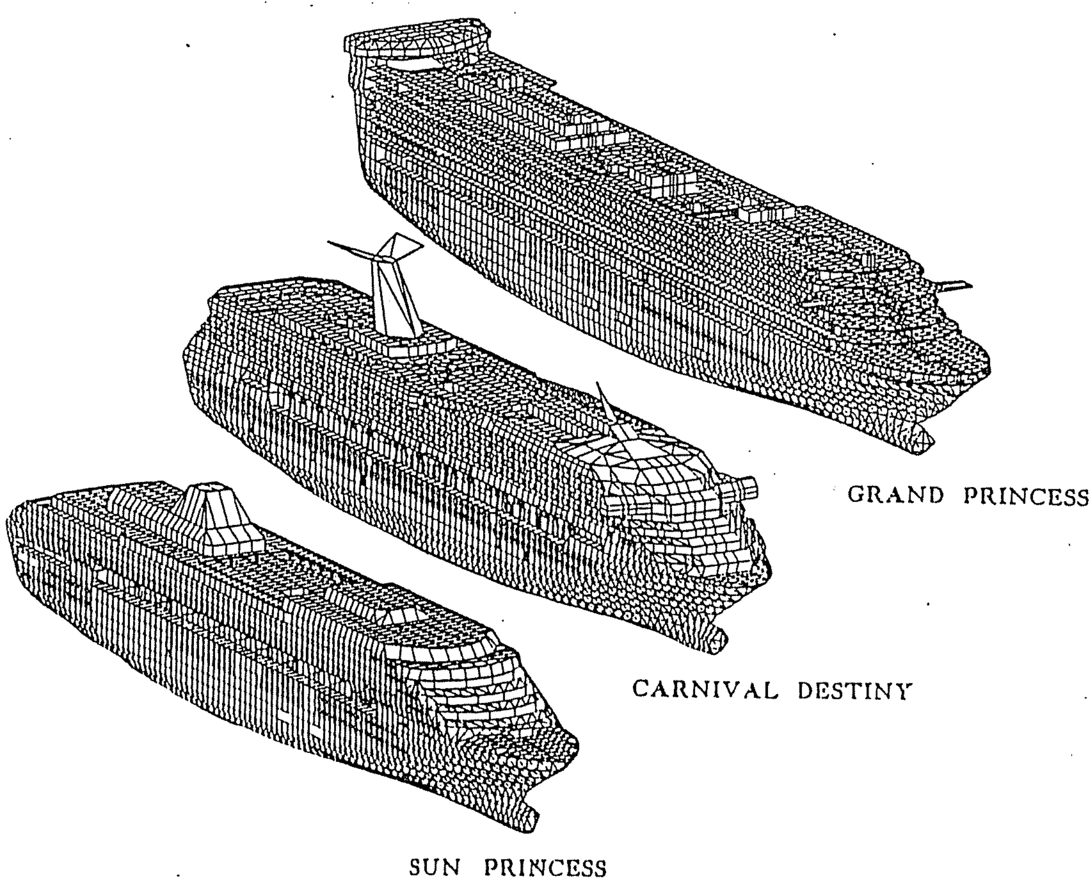



Figure 8 Finite element global models

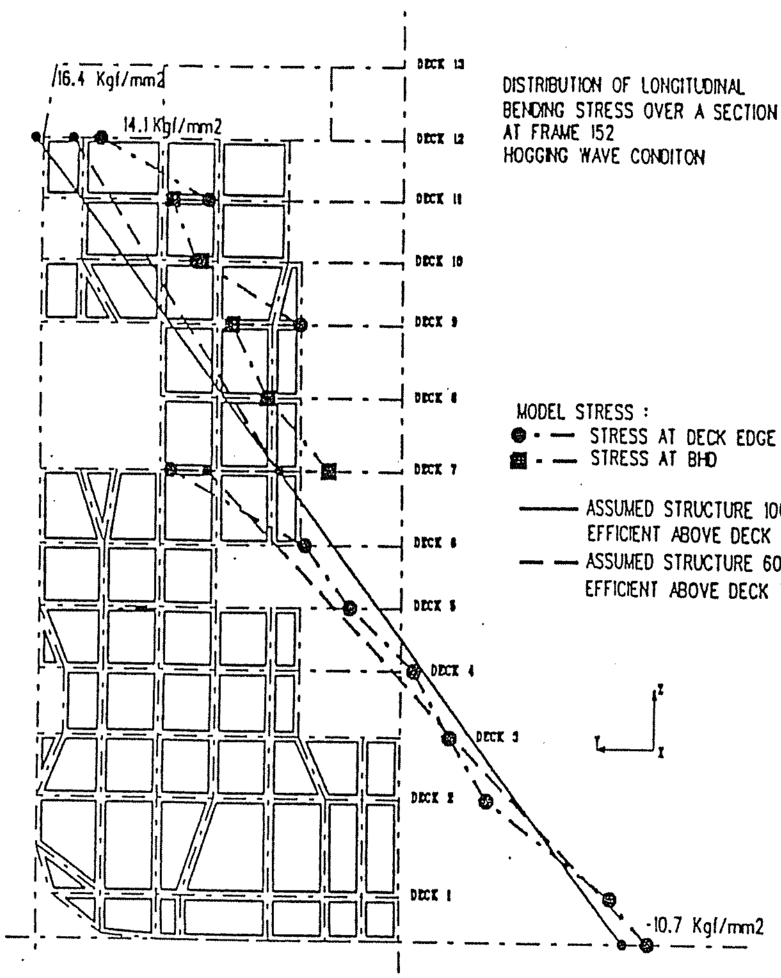



Figure 9 Longitudinal stress distribution

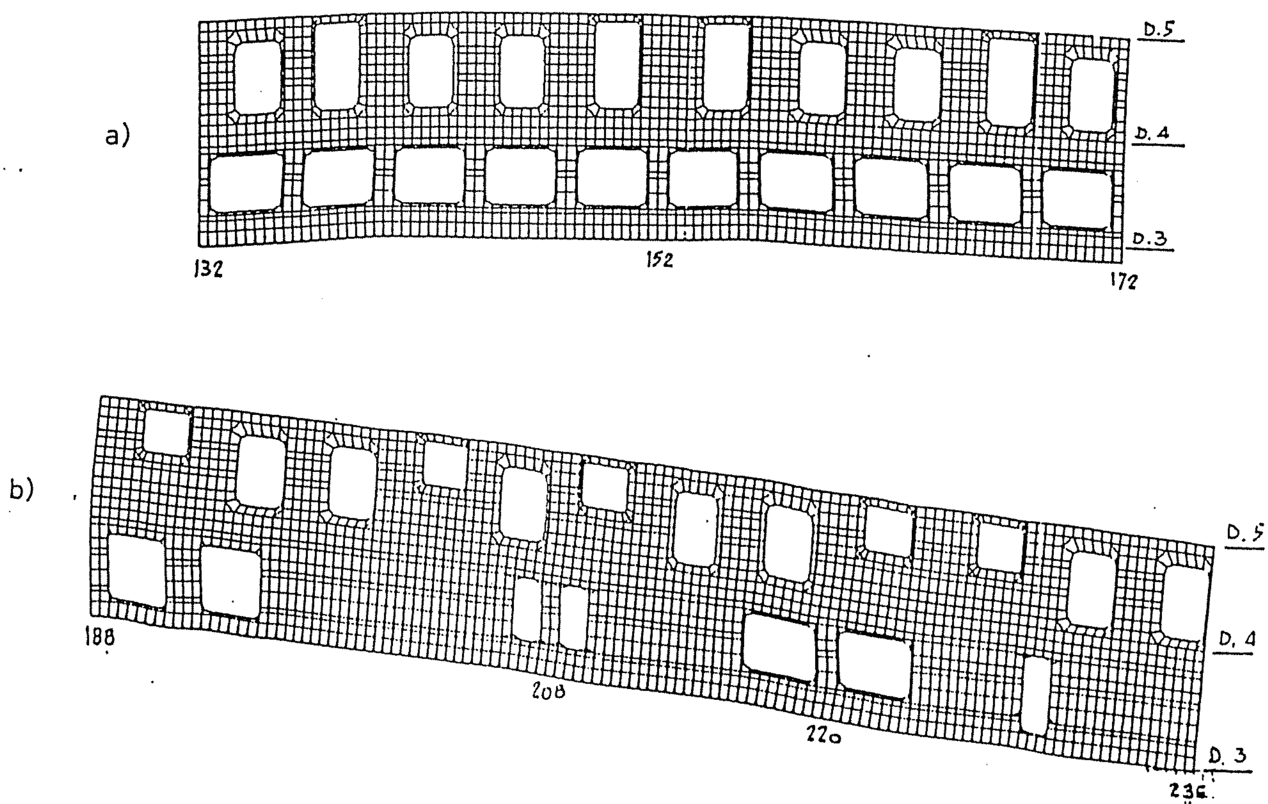



Figure 10 Fine mesh of the longitudinal recessed bulkhead

for access to balconies areas, very fine mesh models are usually provided. Principal stresses exceeding the yield point around the corners of openings can be tolerated on the basis of plasticity considerations and fatigue analysis (Fincantieri, 1992). Higher tensile steel, proper steel grades and grinding in way of critical corners are the typical solutions in these cases.

### 5.5 *Double Hull Ships*

The cost and weight impact of the double hulls required for tankers by the requirements of the U.S. OPA'90 law and the regulations of the International Maritime Organization (IMO) have led to several studies to reduce this impact. One method investigated is the unidirectional double hull. In one such study, several double-hull variants of a 40,000 DWT product tanker were designed considering the use of automated welding processes and modular construction. The baseline design had conventional transverse webs spaced at 2.8 meters within the 2.25 meter deep double hull spaces, and transverse bulkheads spaced 23.5 meters. The first variant had no transverse webs, but reduced bulkhead spacing of 14.78 meters. The second variant had a deep transverse web frame placed at mid-length of the 23.5 meters cargo tanks. The first variant had a total steel weight 9.4 percent greater than the baseline, and the second variant weighed 2.8 percent more than the baseline. However, the estimated cost for producing the cargo section of the first variant was estimated to be 0.73 percent less than the baseline, and the second variant was estimated to cost 5.42 percent less than the baseline design (Sikora *et al.*, 1995).

A design of a 80,000 DWT double hull tanker having no transverse web frames was developed in Korea. A deterministic and probabilistic procedure for evaluating the transverse strength of this hull type was developed and applied to the ship design (Paik *et al.*, 1993).

The strength of an Afra-max double hull tanker without a centerline longitudinal bulkhead or any transverse swash bulkheads was investigated using finite element analysis to determine the effect of the depth of the double bottom, deformations of the double bottom, side shell, bottom girders, and side stringers (Michimoto *et al.*, 1994a). The same authors also investigated the additional stress induced in secondary members because of the large deformation of primary structural members (Michimoto *et al.*, 1994b).

Other studies were conducted on the strength and optimization of double hull tankers. The generalized slope-deflection method combined with the Hooke and Jeeves direct search method was used to obtain a minimum weight design (Jang and Na, 1993). The same authors developed several minimum hull weight designs for a 300,000 DWT double hull tanker, comparing the results to the structure of existing ships (Jang and Na, 1994).

Full scale models of sections of unidirectional double-hull structures were tested for longitudinal compressive strength. The models consisted of 1 to 3 cells of double bottom structure, varying in size from 0.46m to 0.91m deep and wide. The length varied from 1.83m to 10.7m. A series of smaller models, 0.15m square, were also tested. The tested strength was compared to theoretical solutions developed by Frankland, Faulkner, and von Karman, as well as finite element computations (Devine *et al.*, 1995).

During the design of a 300,000 DWT double hull tanker, different crosstie arrangements were investigated (Bong *et al.*, 1995). The Sequential Unconstrained Minimization Technique (SUMT) and the Multiplier Method were used to optimize the structure of a double hull tanker. The objective function of the analysis was the weight of one tank hold length, and the scantlings of the transverse and longitudinal members in the midship segment were used as the design variables (Nobukawa *et al.*, 1993a). In another study, the same authors took the hull construction cost of one tank hold length as the objective function to be minimized. They used the internal penalty method and the multiplier method based on plastic design to emphasize cost minimization compared to the weight minimization and total hull construction cost (Nobukawa *et al.*, 1993b). Numerical calculations of the longitudinal hull girder bending stress distributions in a 140,300 DWT double hull tanker were computed in regular waves of length equal to that of the ship, and a height of 10 meters (Nose *et al.*, 1993).

Sloshing loads are a concern for middle-sized and large double hull tankers because of the large tank size when no longitudinal or transverse swash bulkheads are fitted. In one study of these sloshing loads, a non-linear numerical simulation is made using a new sloshing code based on the SOLA-Surf scheme and a simple two-dimensional sloshing code (Shinkai *et al.*, 1994). Experimental measurements of these loads were made using a ship motion simulator that has four degrees of freedom (quasi 3-D motion) in roll, heave, sway and pitch.

The strength of double hull structures during grounding has been investigated to maximize the benefits of this type of hull design. A series of three quarter-scale models of the double-bottom structure of a 40,000 DWT tanker were tested for grounding strength using a 90 degree cone-shaped simulated rock in a testing machine. The panels were 2.4m wide by 6.1m long by 0.5m deep, and tested for collapse loads up to 3.3 MN (Rodd *et al.*, 1995). In another experimental study, six different structural configurations of double-hull structure were tested for tearing resistance to several wedge-shaped hull penetrators. Specimens were 0.4m wide by 0.56m wide by 42mm deep. The experimental force required to drive the wedge through the length of the section, tearing both the inner- and outer-hull plating was found to agree within 12 percent of the computed force. (Bracco and Wierzbicki, 1995).

Several one-fifth scale models of the double-bottom structure of a 40,000 DWT tanker were tested for the effects of grounding by running them at the speed of 12 knots into a simulated rock. The models were 2m wide by 3.3m wide with a double-bottom depth of 0.37m. The simulated rock was instrumented to determine the force on the structure (Rodd and McCampbell, 1995).

Double hull structures are being investigated for use in naval ships because of possible advantages of producibility and other features. As part of these investigations, a full scale prototype double hull module was fabricated to quantify production costs. Specimens were then cut from the structure to test the axial stability of the box sections formed by the unidirectional stiffening system, and to determine the effects of initial imperfections of the structure on the compressive strength (Pang *et al.*, 1995).

Double hull structures have been suggested as a solution to the problem of structural failures of bulk carriers. One study identified the weakness of the conventional side shell frames because of the sudden change in stiffness from the bottom and deck structure. The use of a double side hull would provide for better structural continuity and reduce the stress concentrations that are the cause of fatigue failures (Chao, 1995).

## *5.6 Bulk Carriers and Container Ships*

### *5.6.1 Bulk Carriers*

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 1990's. In general these were the older ships, serving under third, fourth, or even fifth owners. (Some had their certificates and classification withdrawn because of non-conformance with classification requirements.) Card and Palermo (1995) discussed the scenario of the loss of bulk carriers. They were poorly maintained, exhibited excessive corrosion and generally failed in two principal ways. Section of shell plating severely weakened by corrosion could have carried away under high sea-way loadings, or corroded sections could have permitted sea water to enter the holds resulting in liquefaction and shifting of cargo. Ensuing forces on already weakened, heavily corroded shell plating could then lead to either structural failure or even loss of stability. 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 300m long with beams on the order of 60m. In order to save hull weight, at the same DWT, hulls of the later ships were fabricated of higher strength steel. Thus for a given design loading, thinner hull plating could be used compared with ordinary steel. Therefore the effects of corrosion could be more pronounced.

There is some evidence that a combination of heavy seas, harsh trade routes, the type of cargo carried, and also other factors (such as corrosion and wear) may in some cases lead to an above average incidence of damage. Figure 11 shows known locations of bulk ship failures world wide from 1989 to 1992. This figure, fashioned after an Australian study (BTCE, 1994), indicates the vessels to have suffered failure in areas of known harsh weather. The same study also showed that bulk ships carrying heavy cargo such as iron ore are associated with a large proportion of failures than would be expected based on their at risk voyage exposure.

A common theme running through the majority of damages to bulk and combination carriers has been the wastage of members. Liu and Thayamballi (1995) reported that 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 carriers and conventional bulk carriers. The wastage is due to two causes, a) corrosion, and b) wear from routine vessel operation. The wastage can facilitate cracking and member separation, including by fatigue. Figure 12 shows such a member separation.

DA-WG (1995) investigated the strength reduction of aged bulk carriers, which was a part of the joint research activity for the disaster and aging of ships organized by the Ship Structure Committee, the Society of Naval Architects of Japan. In relation to the recent bulk carrier accidents, new rule amendments have been made by IACS for the side structure of a bulk carrier. The stress analyses have been carried out for the thickness requirements based on both the old and new rules. The thickness reduction due to corrosion is assumed to be 5mm for hold frames and 3mm for side shell plating, respectively (see Figure 13 and Table 7). The calculated results are shown in Table 8, where the finite element model for the detailed analysis is illustrated in Figure 14. In the report, the effects of load re-distribution due to member separation are also discussed. The main conclusions are as follows;
(1) Based on the old rule, the stress level increases 40-50% after corrosion, which could not be covered by the corrosion margin at the design stage,

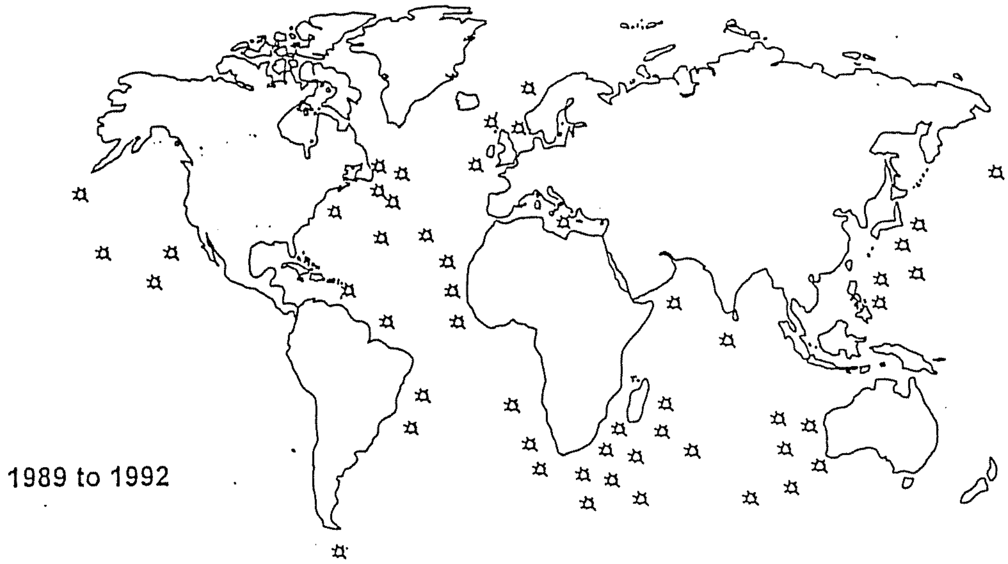



Figure 11 Locations of bulk carrier losses

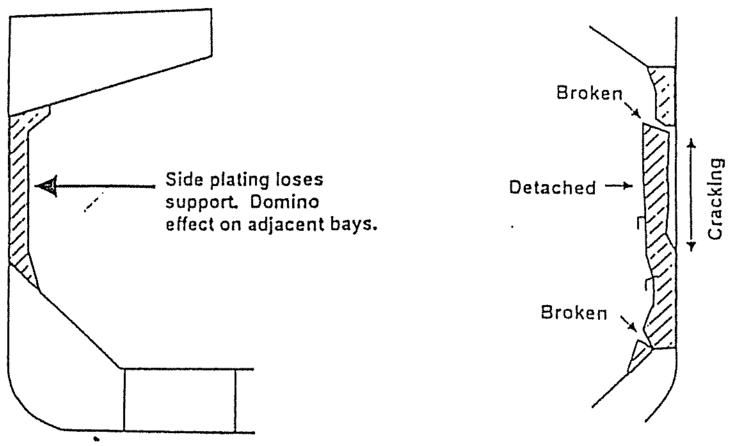



Figure 12 Side frame damage of bulk carriers

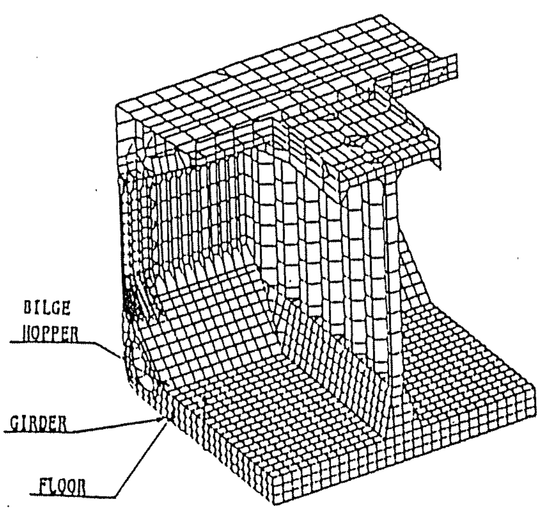



Figure 13 FE model of a bulk carrier

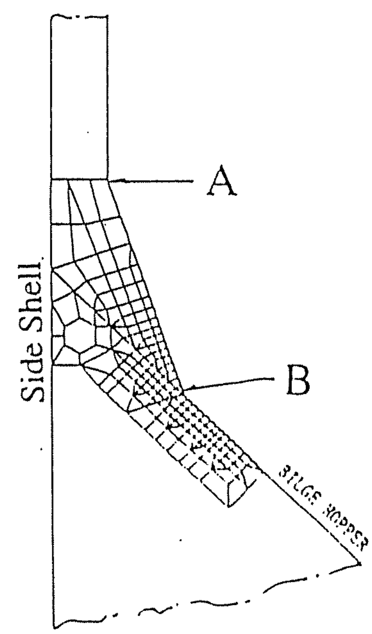



Figure 14 Local FE model

(2) Based on the new IACS requirement, the stress levels of the hold frames are 90% at the parallel part and 80% at the lower end of the hold frame, respectively, in comparison with those based on the old rule,
(3) The new IACS requirement keeps the stresses within allowable level even after the thickness reduction due to corrosion.

Table 7 Plate thickness (unit:mm)

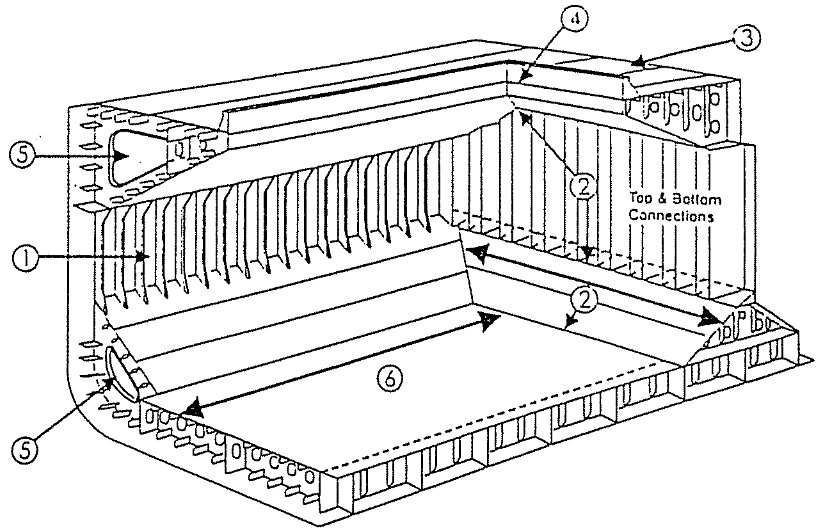
| | location B | | location A | | side |
|---|---|---|---|---|---|
| | WEB | FACE | WEB | FACE | shell |
| old rule | 11 | 150 x 15 | 10 | 150 x 15 | 16.5 |
| old rule with corrosion | 6 | 150 x 10 | 5 | 150 x 10 | 13.5 |
| new rule | 15 | 150 x 15 | 13 | 150 x 15 | 16.5 |
| new rule with corrosion | 10 | 150 x 10 | 8 | 150 x 10 | 13.5 |

Table 8 Relative stress levels

| | location A | | | location B | | |
|---|---|---|---|---|---|---|
| | $\sigma_{max}$ | $\tau$ | $\sigma_e$ | $\sigma_{max}$ | $\tau$ | $\sigma_e$ |
| old rule | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 |
| old rule with corrosion | 1.45 | 1.50 | 1.41 | 1.31 | 1.29 | 1.30 |
| new rule | 0.89 | 0.91 | 0.87 | 0.81 | 0.81 | 0.81 |
| new rule with corrosion | 1.20 | 1.17 | 1.18 | 0.87 | 0.87 | 0.87 |

Liu and Thayamballi (1995) also reported that cracking of bulk carriers typically occurs in the areas indicated in Figure 15. The two most structurally significant cases of cracking are:
(1) At bracket toe regions of hold frame connections to the upper and lower wing ballast tanks,
(2) At transverse corrugated bulkhead intersections with topside tank structures.
The resulting cracks can propagate fore and aft or athwarthship, as the case may be. Other typical types of cracking in bulk carriers occur at:
(3) The intersection of inner bottom and hopper plating/ lower stool of transverse bulkhead,
(4) 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.

System reliability of marine structures have been studied extensively by using framework models. Since it is difficult to idealize a ship structure by a framework model, a new system for plate structures using spatial elements has been developed by Okada *et al.* (1993), who apply the method to the analysis of ship structural system reliability. In the paper, the structural reliability of a large bulk carrier (150,000 DWT) based on the collapse mode analysis are investigated by successively applying the newly developed system. Results are given for the midship part and fore/aft quarter parts of the ship hull under two loading conditions. Main conclusions are as follows;
(1) new formulae for the compressive strength are derived for the longitudinal and transverse frame system panels under combined in-plane compressive stress, shearing stress, and lateral pressure, so that the collapse modes and system reliability under these conditions can be estimated;
(2) the effects of the shearing stress on the dominant collapse modes of the side shell elements are significant, whose collapse probabilities are higher for the fore and aft parts of the ship in comparison with those at the midship part;
(3) the class NK reported the structural damages due to corrosion, which were mainly located at fore and aft parts of aged bulk carriers. This observation is in relatively good accordance with the present results.




## 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 15 Problem areas of bulk carriers (IACS)

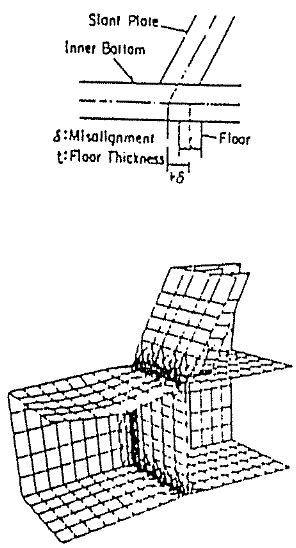



Figure 16 Deformed shape of a lower stool model

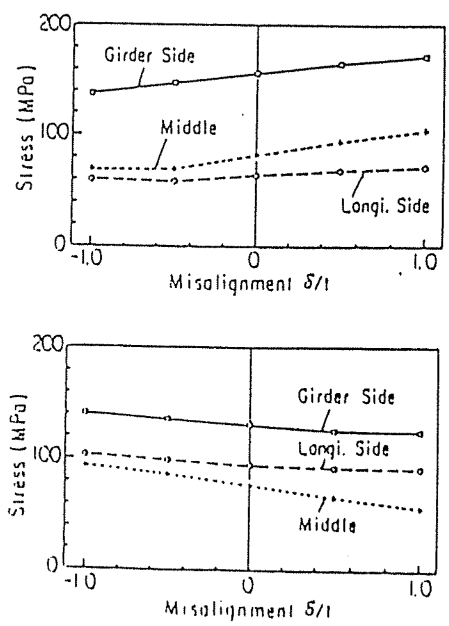



Figure 17 Stress changes with misalignment

Misalignment of structural members causes additional bending moment. The effect of misalignment on behavior and strength has been clarified for simple welded joints such as cruciform joint with fillet weld. However, very few studies have been conducted for complicated structures such as member end connections of ship structural details. Tanigawa *et al.* (1993) have carried out the stress analyses of the joint between bulkhead lower stool and inner bottom of a bulk carrier, where the misalignment of slant plate and the floor is taken into consideration. The definition of the misalignment and the finite element model using shell elements are illustrated in Figure 16. Typical stress increase due to the misalignment is illustrated in Figure 17. They also compare the results with those obtained by using three-dimensional solid elements.

### *5.6.2 Container Ships*

As classified by Rule (1994), the most significant developments of recent container ship design are divided into three types: ships with 11 rows of underdeck containers within a "Panamax" beam, ships with breadths exceeding the Panama Canal limitation, andships with the hatch covers omitted over some or all holds.

The development of the first type of ships are achieved mainly by: omission of deck girders, reduced space between cell guides, and reduced width of side structure - such kind of reduction have been extended further by some yards to accommodate 12 rows in a hold.Over five years of successful service of this type of vessels, 11 rows in a hold become more like a standard arrangement of new "Panamax" beam container ship design.

The development of second type of ships are generally called "Post Panamax" container ships. The beam of the vessels extend from 32.2m to around 37.1m, and further to 40m and 42m for 5,000 TEU and 6,000 TEU size huge container carriers (Min, Kim, and Park, 1994). Torsional strength and combined stress in way of hatch corners are still major issues in the evaluation of strength of main hull structure. Response of multi-cell structure under torsional mode is continuously studied further (Wang *et al.*, 1994, Wu *et al.*, 1994).Torsional moment distribution is sensitive to the increment of ship beam. Studies of wave torsional moment over the ship length of small to large ship beam have been reported by P. Lersberyggen (1995). Significant increase of design moment is found to be necessary, and it will revise the DNV rules. It is obvious that rigorous analysis procedures as proposed by ABS (1993), Han *et al.* (1995), and Payer and Fricke (1994), which include a whole ship FEM model and detailed analysis reflecting the combined effect of wave load, inertia load due to ship motions, and static load components, are necessary to ensure the adequacy of the strength in every detail of those huge and complex ship structures.

The third type of vessels without hatch covers are developed for more efficient handling of container cargoes. As hatch covers are not considered as hull strength members, omission of hatch covers has not imposed any particular effects in the structural design of a main hull structure (Rule, 1994).

## 6 FINITE ELEMENT COMPARATIVE STUDY OF SHIP STRUCTURAL DETAILS

### *6.1 Introduction*

For the evaluation of fatigue strength of ship structures, explicit fatigue analysis is becoming an important part of ship structural design. Combined with the detailed finite element analysis, hot spot stress approach, is expected to be the most practical method. The concept of hot spot stresses, which has been often applied to the tubular joints of offshore structures, is generally understood to be the stresses at weld toe locations taking into consideration all geometrical influences except for the local weld geometry. However, the calculated local stresses around the structural singularities vary depending on structural idealization, used element types and mesh subdivisions. Increasing amount of work is now being carried out for the practical evaluation of hot spot stress of stiffened plate structures. In order to obtain some standards or guidelines for the analysis, ISSC committee II.1 decided to carry out a finite element comparative study focusing attention to the evaluation of hot spot stresses of ship structural details. Experimental model of research project SR219 in Japan was selected as the analysis model of the comparative study so that the calculated results could be compared with the experimental results for the proper validation.

### *6.2 Experimental Model*

The test model of a stiffened panel, shown in Figure 18, is for a part of typical side longitudinal and transverse bulkhead intersection of crude oil tankers. The model consists of three longitudinals and one transverse frame, and has twofold symmetry with respect to longitudinal and transverse directions, respectively. The transverse frame is supported by tripping brackets at the center longitudinal and stiffened by flat bars at the side longitudinals. The model is made of higher strength hull structural steel of class KA32 having the yield strength higher than 32 kgf/mm$^2$ (314 MPa), and constructed applying the same welding procedure as those for actual ship structures.

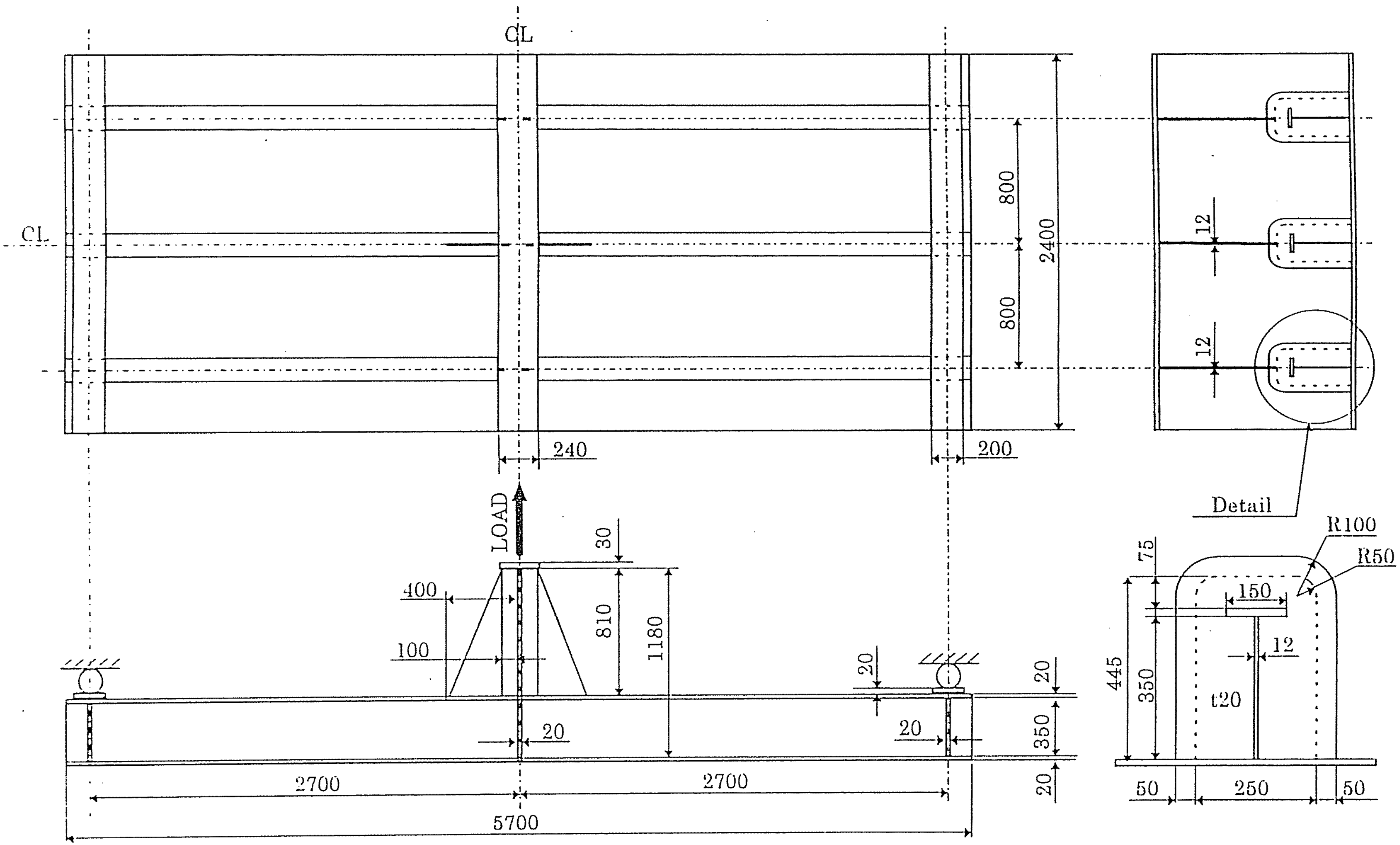



Figure 18 Experimental model

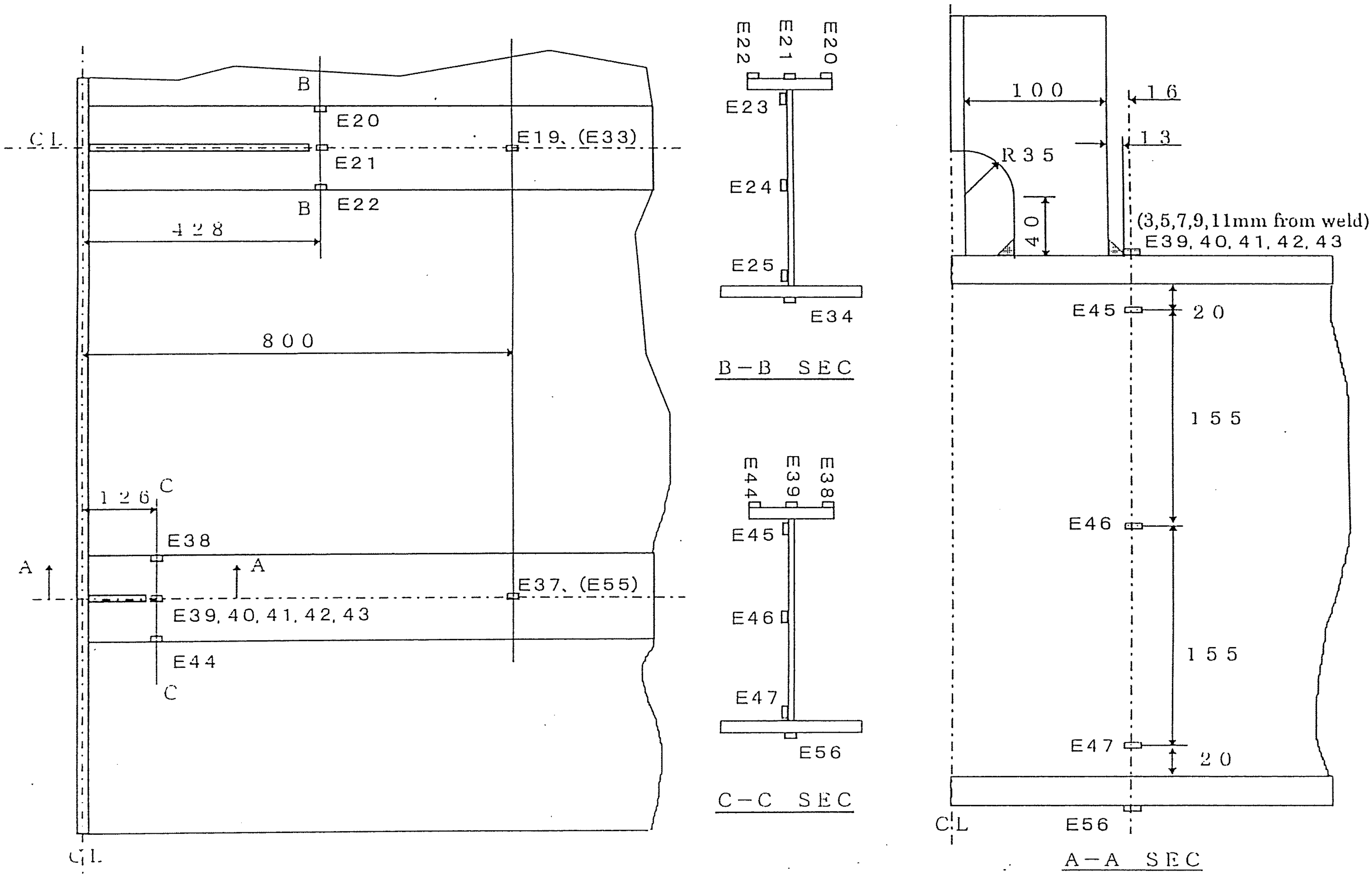



Figure 19 Location of strain measurement

The load is applied in the three point bending configuration in horizontal direction by the hydraulic actuator, through the triangular loading plate of 40 mm thickness attached to the flange plate of the transverse frame. The model was supported at vertical reaction columns through rollers at the ends of the longitudinals. Increasing the loads gradually from zero to the maximum of 392 kN, displacements and strains were measured at several reference points and critical locations. Uni-axial strain gauges were used to measure the longitudinal strain components. At the weld joint of side longitudinal flange plate and flat bar stiffener, which is expected to be the location of first fatigue crack initiation, stress concentration gauges were used to evaluate the hot spot stresses. Measuring points, used for the comparison with the analysis results, are shown in Figure 19.

### *6.3 Analysis Models*

Nine analysis models, shown in Table 9, were collected for this comparative study from seven committee members (Hu, Jang, Nie, Paetzold, Rigo, Sumi, Wu) and one outside collaborator (Kawano). The problem was analyzed independently by each contributor using his own solution procedures, based on the common guidelines,
(1) Solution method is to be linear elastic analysis;
(2) Material properties for the ship structural steel is to be isotropic linear elastic with Young's modulus, E = 206GPa, Poisson's ratio, n = 0.3; and
(3) Magnitude of the applied load is 40 tonf (392 kN).
No information concerning the experimental results was given prior to the analysis.

All the participants made use of the two fold symmetry condition, and the quarter part of the test model was analyzed. Only one model, model (3), was constructed using three dimensional solid elements, while all the other models were idealized using shell elements, where the detailed geometry of the weld was not represented. Graded mesh subdivisions were applied, except for model (8), with the element size of plate thickness near the transverse/longitudinal connections. In model (8), relatively uniform fine mesh subdivisions are used over the zooming model for the 1/12 of the structure on 1,200 mm length.

### *6.4 Comparison of FE-Analyses*

Finite element analysis results are compared in Table 9, together with the experimental results and results by the hand calculation based on the beam theory. Numerical values of measured stresses were obtained multiplying Young's modulus (E= 206 GPa) to the measured strains. Variations of the measured results at symmetrical locations with respect to longitudinal and transverse directions were generally small, several percent in the average and 5% at the most.

Experimental value of the deflection in Table 9, obtained from the measured relative displacement between the center and supporting points of the center longitudinal, is slightly small compared to the analysis results. A likely reason for this difference is the slightly non-linear behavior of load displacement relation observed in the experiment at lower load levels, which could be caused by the effects of the local deformation at the contact points of the supporting rollers and the loading apparatus rigidity.

Table 9 includes the stress results of reference sections at 800 mm from the center of the model, bracket end section of center longitudinal and stiffener end section of side longitudinal. Average of calculated stresses are plotted versus experimental stresses in Figure 20. Excellent correlation between the analysis and experimental results can be observed except for the region close to the weld toe (E21, E40). Coefficient of variation (COV) of the analysis results are shown in Figure 21. It can be said that COV has a tendency to decrease with the increase of stresses except for bracket and stiffener weld toe (E21, E40).

As for the determination of hot spot stresses at the weld toe, several different methods of extrapolation have been proposed. Hence, it was decided that the comparison of the analysis results will be made on the stress distribution near the welded joint, rather than on the hot spot stress itself. The results for the stress distribution on the top surface of the flange of the side longitudinal near the stiffener weld toe are shown in Figure 22. It is interesting to note that the results using shell element have high correlation with each other, but give significantly lower stresses than the experimental results. The results of solid element model (3) agree well with the experimental results at the weld toe.

### *6.5 Discussions*

Since all the shell element models do not include the weld, the difference of the calculation results and the experiment may be attributed to the effects of weld geometry. It is also pointed out by Kawano *et al.*. (1994) that shell element models sometimes give lower estimate for stress concentration due to the effect of plate thickness effect. Although the number of experimental measurement and type of the structure investigated is limited in the present comparative study, it can be concluded that the solid element model including weld geometry is recommended when the correlation with experimental results is to be pursued.

Table 9 Comparison of analysis results

| MODEL | EXP | BEAM | (1) | (2) | (3) | (4) | (5) | (6) | (7) | (8) | (9) | AVERAGE | COV |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| element | | | shell | shell | solid | shell | shell | shell | shell | shell | shell | | |
| Num of DOF | | | 1, 904×6 | 2, 109×6 | 30, 400 | 1, 294×6 | | 1, 231×6 | | 2, 640×6 | | | |
| solver | | | ALGOR | NASTRAN | MARC | NASTRAN | | ANSYS | NASTRAN | MSLIICN | | | |
| pre-post | | | ALGOR | PATRAN | MENTAT | I-DEAS | | ANSYS | PATRAN | | | | |
| DEFLECTION (mm) | 4. 63 | 4. 89 | 4. 91 | 4. 99 | 5. 02 | 5. 62 | 5. 64 | 4. 59 | 5. 22 | 4. 06 | 4. 81 | 5. 0 | 0. 099 |
| E19 | 80. 2 | 83. 4 | 86. 8 | 83. 9 | 88. 2 | 89. 8 | 83 | 77. 7 | 88. 4 | 81. 4 | 89. 9 | 85. 5 | 0. 049 |
| E33 | -26. 3 | -24. 2 | -28 | -20. 4 | -28. 4 | -26. 2 | -21. 8 | -24 | -24. 8 | -28. 3 | -28. 8 | -25. 6 | 0. 120 |
| E20 | 95. 3 | 99. 8 | 94 | 97. 2 | 102. 3 | 102. 3 | 101. 2 | 107. 1 | 97. 6 | 94. 5 | 112. 5 | 101. 0 | 0. 060 |
| E21 | 140. 2 | 99. 8 | 108. 1 | 105. 5 | 165. 2 | 114. 7 | 109. 6 | 119. 7 | 112 | 109 | 122 | 118. 4 | 0. 155 |
| E22 | 95. 3 | 99. 8 | 94 | 97. 2 | 102. 3 | 102. 3 | 101. 2 | 107. 1 | 97. 6 | 94. 5 | 112. 5 | 101. 0 | 0. 060 |
| E23 | 78. 2 | 86. 5 | 85. 3 | 90 | 97. 2 | 102. 1 | 104. 1 | 104 | | 87 | 95. 4 | 95. 6 | 0. 079 |
| E24 | 26. 7 | 34. 7 | 31. 6 | 38. 6 | 31. 2 | 33. 2 | 35. 4 | 39. 3 | 31. 4 | 25 | 27. 3 | 32. 6 | 0. 145 |
| E25 | -19. 3 | -17. 2 | -22 | -17. 6 | -26. 7 | -24. 5 | -22. 1 | -14. 7 | -28 | -17. 5 | -23. 8 | -21. 9 | 0. 205 |
| E34 | -33. 7 | -30. 5 | -36. 8 | -25. 4 | -36. 1 | -30. 8 | -29. 9 | -29. 7 | -34. 4 | -34 | -36. 5 | -32. 6 | 0. 119 |
| E37 | 79. 4 | 83. 4 | 78. 4 | 77. 6 | 80. 8 | 83 | 82. 8 | 69. 9 | 80. 8 | 78. 4 | 74. 9 | 78. 5 | 0. 053 |
| E55 | -24. 6 | -25. 5 | -24. 5 | -18 | -25. 4 | -23. 9 | -18. 3 | -21. 2 | -22. 6 | -27. 3 | | -22. 7 | 0. 146 |
| E38 | 111. 3 | 113. 0 | 104. 9 | 106. 9 | 110. 6 | 116. 5 | 115. 8 | 98. 8 | 101. 4 | 108 | 98. 6 | 106. 8 | 0. 062 |
| E40 | 180. 7 | 112. 9 | 114. 1 | 128. 6 | 185. 9 | 131. 6 | 117. 8 | 116. 7 | 118. 1 | 117 | 115. 7 | 127. 3 | 0. 179 |
| E44 | 107. 0 | 113. 0 | 104. 5 | 106. 6 | 109. 9 | 116. 2 | 98. 6 | 78. 4 | 101. 1 | 108 | 98. 8 | 102. 5 | 0. 104 |
| E45 | 81. 8 | 97. 9 | 90. 2 | 90. 2 | 99. 1 | 111 | 122. 9 | 88. 7 | | 91 | 80. 4 | 96. 7 | 0. 143 |
| E46 | 30. 1 | 39. 2 | 33 | 37. 6 | 33. 6 | 36. 5 | 46. 4 | 28. 2 | 31. 4 | 30 | 28. 9 | 34. 0 | 0. 167 |
| E47 | -21. 4 | -19. 4 | -23. 7 | -22. 9 | -29. 9 | -27. 6 | -22. 4 | -22. 1 | -28 | -22 | -23. 8 | -24. 7 | 0. 120 |
| E56 | -36. 5 | -34. 6 | -36. 5 | -28. 2 | -38. 3 | -37. 7 | -28. 9 | -33 | -34. 4 | -37. 1 | -33. 6 | -34. 2 | 0. 108 |

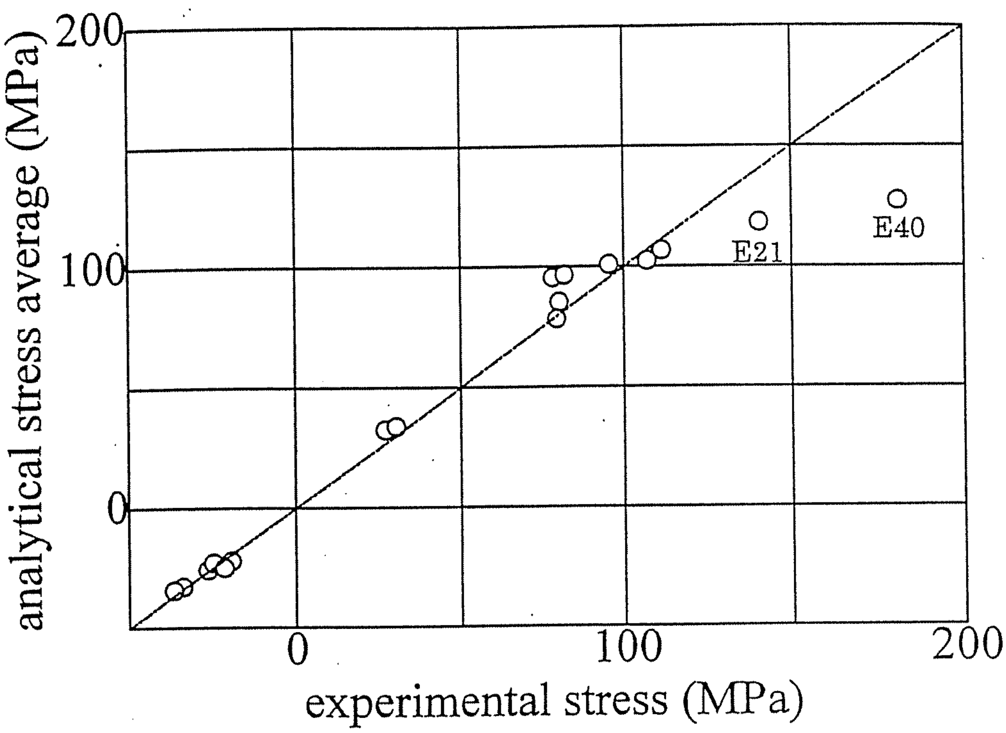



Figure 20 Comparison of analyses with experiment

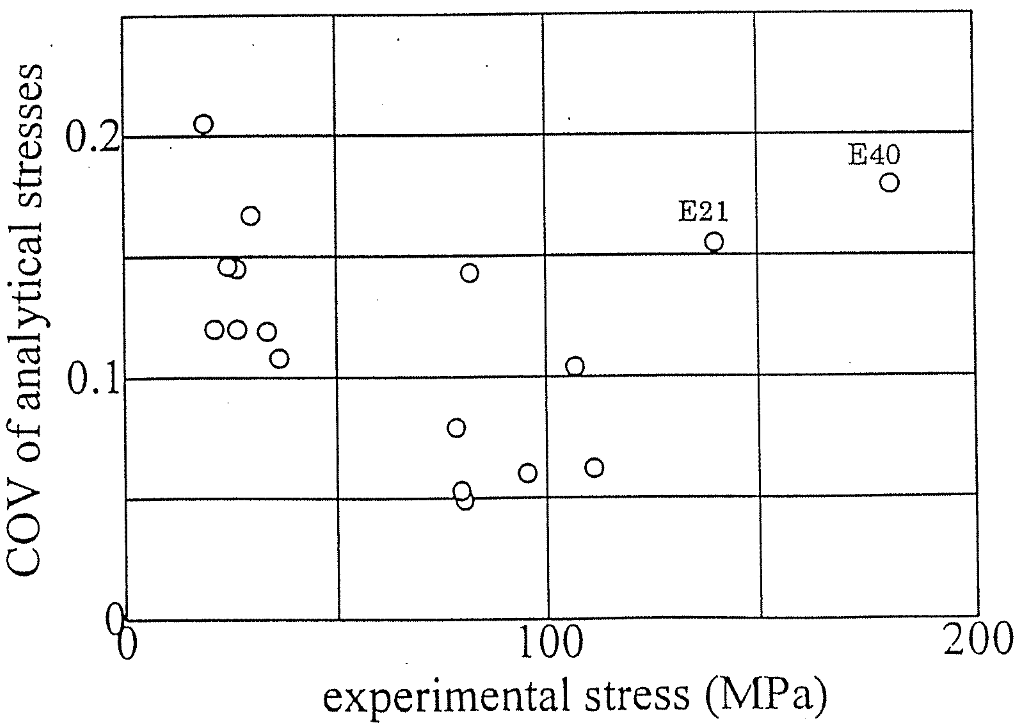



Figure 21 Variation of analysis results

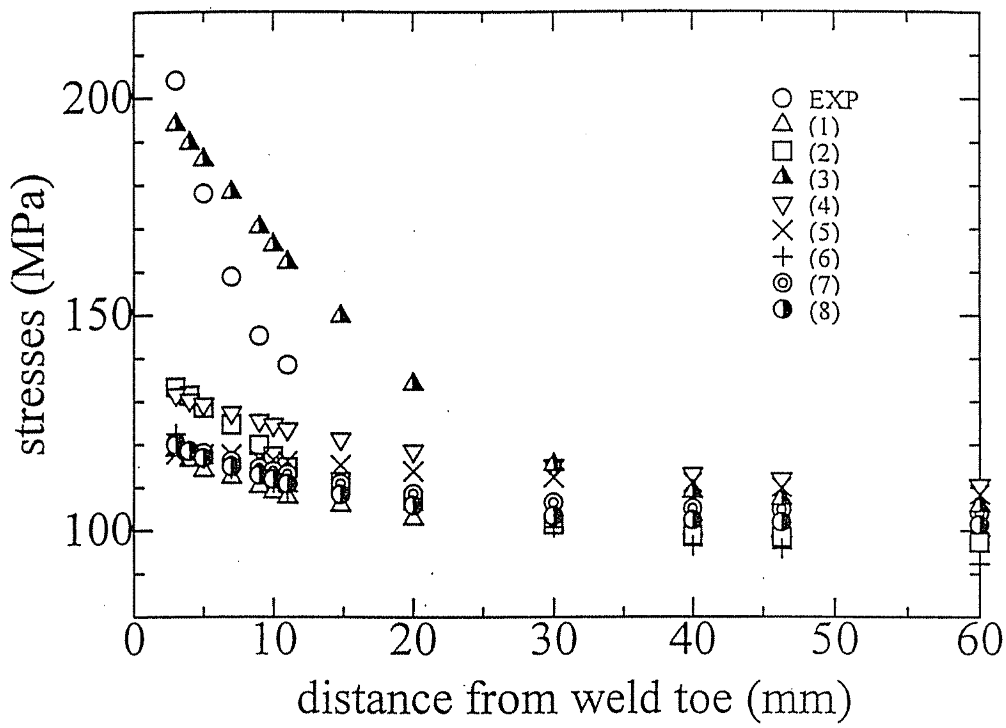



Figure 22 Stress distribution at the weld toe of the stiffener end

However, considering the expensive analysis cost of the solid element model, it is also important to establish reliable and practical procedure based on shell element idealization for hot spot stress estimation. It should be noted that some of the contributors with shell element models used extremely fine meshes at the area of stress concentration, but the variations of the results remained sufficiently small. This indicates that the shell element model could be used as a reliable tool in the design analyses. In order to avoid underestimate of stresses concentration by the shell element models, however, some modification or interpretation have to be introduced. Machida *et al.* (1992) proposed to include geometrical effects of weld connections into shell element idealization by increasing the element plate thickness by about half the weld leg length. In the procedure of hot spot stresses evaluation by DNV (Cramer *et al.*, 1995), hot spot for shell element models, which do not include weld, is located not at the weld toe, but at the element intersection lines and the stresses at a distance t/2 and 3t/2 from this intersection line are used for the extrapolation. Similar procedure to shift the stress distribution is used in NK Guidance (1996). These practical procedures of hot spot stress evaluation, validated by experimental and analytical studies, are important for the accurate fatigue life estimation of ship structural details.

## 7 CONCLUSIONS AND RECOMMENDATIONS

Having reviewed the current research efforts on the topics within our scope of the committee mandate, we shall conclude this chapter with some future research recommendations.

There is little work available addressing simplified calculation procedures of structural response for use in reliability methods, so that the materials covered in section 2.1 are slightly shifted to the simplified strength evaluation procedures which could be applicable to reliability assessments. The reliability analyses applied to ship structures are reviewed in section 2, and the statistical modeling of initial imperfection are discussed in section 4 in relation to buckling strength of offshore structures.

Presently majority of response calculations are carried out by using complex finite element analyses interfaced with modern pre- and post-processors, while probabilistic methods could be, in principle, applicable for the evaluation of both ultimate strength and fatigue strength. As was indicated in our previous committee report (ISSC, 1994), the missing link in the application of these methods is still the adequate definition of suitable load models, especially for fatigue strength evaluation, in which the magnitude and frequencies of local stress should be precisely determined from the long term prediction. Considerations should be made for the phase difference of each load components at certain locations of typical ship structures in longitudinal direction (fore, middle, and aft) and transverse section such as deck, side, bulkhead and bottom parts. It may also be desirable to have the information on stress history. Questions then arise as to how to proceed this design process, how should we determine the long term distribution of the local stress range; by using a stress response function applied to short term prediction, or by using certain design waves which correspond to a certain level of exceedance-probabilty of wave-induced load-effects? Closer cooperation between Committees I.2 and II.1 or perhaps a Special Task Force to look at the load application to structural models would be worthwhile.

In the direct calculation of fatigue analysis, sophisticated finite element models can only predict accurate local stress response but yield typically to hierarchical multi-level modeling, where lower level global model is used for calculating loads for the higher level local model. The approach of hot spot stress in which the joint classification is replaced by the structural (geometrical) stress concentration has commonly used in these days. There exist several problems associated with the application of the hot spot stress approach. Accurate calculation of the stress concentration requires very fine mesh, and the variation of calculation model may cause large scatter in results. It seems that marine community needs not only good finite element education and training for students and structural designers, but also good analysis guidance for the problems specifically oriented to ship and offshore structures.

In order to obtain some standards or guidelines for the analysis, Committee II.1 has carried out a finite element comparative study focusing attention to the evaluation of hot spot stresses of ship structural details. Experimental model of research project SR219 in Japan was selected as the analysis model of the comparative study so that the calculated results could be compared with the experimental results for the proper validation. Committee II.1 should continue to work on uncertainties associated with the finite element analysis. Also closer cooperation between Committees III.2 and II.1 or perhaps a Special Task Force to look at the hierarchical finite-element modeling for the analysis of hot spot stress of ship structural details would be worthwhile. With this respect, it should also be noted that the collaboration with the experts in offshore structures could be worthwhile.

Other topics which could be included in future Committee work are:

- a review of full scale monitoring for verification of calculation methods,
- a review of reliability codes,
- maintenance strategies using structural monitoring,
- the assessment of strength of corroded and/or damaged bulk carriers, tankers, and marine structures,
- PC-based structural analysis code oriented to marine structures,
- education and training system of marine structural analysis by FEM,
- adaptive finite element analysis for marine structures,
- reliability of buckling strength in terms of initial imperfection, materials, and residual stresses,
- continued study of large cruise vessel, and
- continued study of large container vessel.

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